Stream Jet Ejectors for the Process Industries (Mcgraw-Hill Chemical Engineering)

Power, Robert B.

Steam Jet Ejectors for the Process Industries

Other Chemical & Mechanical Engineering Books of Interest

========= (McGraw-Hill publisher unless otherwise noted) ===============

BASTA + *Shreve's Chemical Process Industries Handbook*
BAUMEISTER ET AL + *Mark's Standard Handbook for Mechanical Engineers*
CHOPEY + *Environmental Engineering in the Process Plant*
CHOPEY + *Handbook of Chemical Engineering Calculations*
COOK & DUMONT + *Process Drying Practice*
DILLON + *Materials Selection for the Chemical Process Industries*
FARAGALLAH + *Liquid Ring Vacuum Pumps and Compresessors* - GULF
FREEMAN + *Standard Handbook of Hazardous Waste Treatment and Disposal*
GRANT & GRANT + *Grant & Hackh's Chemical Dictionary*
HARRIS + *Modern Vacuum Practice*
KISTER + *Distillation Design*
KISTER + *Distillation Operation*
MANSFIELD + *Engineering Design for Process Facilities*
MANGNALL + *Vacuum/Pressure Producing Machines and Associated Equipment* - HICK HARGREAVES & CO
MCGEE + *Molecular Engineering*
MILLER + *Flow Measurement Handbook*
PALLUZI + *Pilot Plant Design, Construction, and Operation*
PERRY & GREEN + *Perry's Chemical Engineers' Handbook*
REID ET AL + *Properties of Gases and Liquids*
REIST + *Aerosol Science and Technology*
RYANS & ROPER + *Process Vacuum System Design and Operation*
SANDLER & LUCKIEWICZ + *Practical Process Engineering*
SATTERFIELD + *Heterogeneous Catalysis in Industrial Practice*
SHINSKEY + *Process Control Systems*
SHUGAR & BALLINGER + *The Chemical Technicians' Ready Reference Handbook*
SHUGAR & DEAN + *The Chemist's Ready Reference Handbook*
SMITH & VAN LAAN + *Piping and Pipe Support Systems*
STOCK + *AI in Process Control*
TATTERSON + *Fluid Mixing and Gas Dispersion in Agitated Tanks*
TATTERSON + *Scale-up of Industrial Mixing Processes*
WILLIG + *Environmental TQM*

Steam Jet Ejectors for the Process Industries

Robert B. Power

AUTHOR-PUBLISHER
381 Baker Ln
Charleston, WV 25302 USA
Tel / Fax (304) 344-2887
Website: www.jetwords.com

Advance Book Information (ABI) data

Robert B. Power
Title: Steam Jet Ejectors for the Process Industries / Second Edition [3rd Printing (electronic) 2012]
ISBN : 0-9755706-4-1
ABI Number : 1249355

Second Edition, First Printing

First Edition, Two Printings

The author converted the 2d edition of this book to digital format, reset it in Century Schoolbook, and designed the new cover.

Printed and bound by C & M Press

Book Specifications:

522 pages
175 illustrations
    147 line drawings
    28 half-tones
17 tables

sewn binding
50# White Finch Opaque

with love
to my wife
***JOAN MERING POWER** with*
whom I share
my bed and my planet. She
makes it all worthwhile.

P.S. We celebrated our 50^{th}
wedding anniversary in 2002

Contents

Unadvertised Extra: POP QUIZ

A series of technical problems has been introduced to stimulate and entertain those readers who enjoy solving puzzles. Located on pages that were blank in the first edition, ten problems are placed randomly throughout the book. Each problem statement includes the page number on which the answer is given.

Emphasis is on methods which yield useful accuracy, rather than great precision. Turn this page to start with problem number 1.

If you want an overview of these quizzes, see Pop Quiz Table of Contents on page 484.

Welcome to POP QUIZ!

Getting down to business,

You are participating in the test of an ejector. You are asked to supply two pieces of information for the test. Five percent accuracy is acceptable. You have a calculator, and could look up the equations within 20 minutes, but you might score some points if you can give the answers right now -- off the top of your head, if you are right!
.

1. How much D&S 75 psig steam will flow through a 0.375 inch rounded-entrance nozzle into a vacuum??

2. What is the diameter in inches of a rounded-entrance orifice that will admit 40 pph of dry air from standard atmospheric conditions into a vacuum?

See page 56 for the answer

Preface

This is the book I looked for and couldn't find in 1960 when I began the task of learning how my employer could use steam jet ejectors better. The book is written to help people in the process industries who are responsible for specifying, designing, installing, operating, and maintaining steam jet air ejector vacuum systems and similar ejector equipment. Specifically, it was written with the needs of these busy people in mind

- The production person who wants to just get the ejector problem fixed and get back into normal production
- The operating person who wants to "tune" an ejector so it will start up quickly and run smoothly for a long time
- The maintenance person who is looking at a disassembled ejector and wondering whether a "sort of" worn part should be replaced, and whether the nozzles of two stages lying near each other on the bench might have been swapped
- The person who has become the designated ejector tester for the plant and wants to acquire the necessary hardware and testing skills to do the job properly
- The design engineer who wants to produce a high-quality, cost-effective ejector installation, wants to know what adds to reliability, and wants to avoid buying a misdesigned ejector
- The startup engineer who wants to make sure the ejector equipment has the required capacity at the design pressure, and wants to save the test results as a benchmark reference for future maintenance and troubleshooting
- The manager who welcomes a quality-improvement program to plug into the organization by handing a book to a competent employee and saying, "See if this program makes sense, and what it takes to implement it!"

In attempting to answer those needs, I have limited the scope of the bulk of the book to this domain:

- Rough vacuum-process pressure is between 1 mmHg Abs and atmospheric pressure.
- Steam is the motive fluid.
- Air leakage is usually the basic load, typically accompanied by a saturation quantity of condensable process vapors.
- Initial drawdown and steady-state loads are handled by one or more ejectors.
- Ejector-related activities include specification, installation, operation, maintenance, and troubleshooting.

Now, having imposed the above restrictions, I realize that if you work with steam jet air ejectors, you will have appropriate opportunities to work with other types of ejectors. So, I have added more information for those of you (especially engineers), who wish to work outside the above domain for steam jet air ejectors and other types of ejectors. The added information will help you do the following:

- Prepare estimates of size, utility usage, cost, and performance.

- Perform other engineering calculations for design, maintenance, and testing of ejector systems.

- Estimate performance, size, and other factors for other types of ejectors, such as steam-jet refrigeration, thermocompressors and gas-jet ejectors, water-jet ejectors, and fluid-jet solids conveyors.

Upon completion of this last category, in Chap. 11, I was pleasantly surprised by the convenience of a simple, generalized method for estimating the performance of thermocompressors. This is potentially the most valuable portion of the book for users who are willing to survey their existing and future processes for energy-saving opportunities.

Ejector vacuum systems can be made quite reliable. After I had gained some experience with ejectors, I acted as an in-house ejector consultant to several process engineers who were designing more than 20 ejector systems for a grass-roots plant. A few years later I was pleased to hear the general manager of that plant observe that the steam jet ejectors were the only class of major equipment that was trouble-free during a difficult startup of new units. By contrast, he said, they had routinely been among the worst problem areas on previous startups at other locations in his experience.

Ejectors have some unique performance characteristics that aren't intuitively predictable from experience with other equipment common to the process industry. However, the behavior of one stage is easy to learn from a description or performance of a simple standard test procedure. Learning how one stage behaves is your key to understanding how multistage ejectors behave, why and how an ejector responds to changing process conditions, what causes ejectors to not work properly, and how to troubleshoot an ejector that is not working properly.

In explaining the operation of ejectors and condensors, I have begun with basic fundamentals and developed the topics slowly, avoiding burdensome mathematics. Most of the major equations are accompanied by example calculations. I assume that the reader can use a scientific calculator to perform exponentiation.

To focus my presentation, I have characterized my audience as technicians and mechanical or chemical engineers. Some of the basics may develop slowly for the engineers. I personally prefer to read slowly and confirm what I think I just learned. Whenever appropriate, I have offered simple recommendations to answer the common question, what should I do? Then, I give details to answer the followup question, why?

The location of my presentation varies: sometimes it is at a blackboard, sometimes in the field, and sometimes just sitting and talking. For some points that need special emphasis, I have included stories that have helped teach and focus me. Because I believe that fear of change and lack of motivation limit us as much as ignorance does, I have included words of encouragement to persuade you to strike out and use this information, even though you may not have precedents near you for comfort. My reason for doing so is that I perceive a big gap between the way many people are using ejectors and the way they would use them if they understood them. I am impatient to get on with it.

If you are familiar with the basic concepts of fluid statics and dynamics, thermodynamics, vapor-liquid equilibrium, and heat transfer, you will be able to use the information in this book to work comfortably with all aspects of ejectors.

My first words of advice:

1. Do not spend too little on ejectors. Buy an ejector that has enough stages and large enough condensers to use steam effectively. Install it carefully, with sufficient features to permit proper operation and field testing and troubleshooting. While writing the last few pages in this book, I was called by a client who was spending $1500 per hour in excess operating costs because he could not identify the source of his vacuum problem while operating, and was going into an unscheduled shutdown to look for the problem. A few small improvements in the installation would have permitted informative tests without interrupting operations.

2. Conduct effective field tests. The results will help you predict failure, tell whether your ejector is working properly, and get out of trouble when it does happen.

My recommendation that you use detailed specifications and always check bids for reasonableness before purchasing ejectors may appear to be encouraging the development of an adversarial relationship with ejector manufactures. Far from it! The detailed specifications test your understanding of your application before you request bids. Your checking of the design adequacy of bids further tests whether your specifications contain adequate information for design. Your relationship with ejector manufacturers will be conducted on a higher level, with fewer misunderstandings that lead to distrust and anger.

You will spend more time working toward the objectives of improving operating reliability and reducing long-term costs, rather than becoming preoccupied with the false economy of buying always from the low bidder.

Here I give you one caution. I don't wish to encourage you to get into "do-it-yourself" mechanical design and fabrication of ejector stages. The information in this book is simply not adequate, and I believe the practice is not cost-effective for most users. If you have a special need to get into this phase, I recommend that you read the appropriate section of Chap. 11.

Finally, I share with you my apprehensions about being far less qualified to write this book than I would wish. Those feelings were followed immediately by the test, “Would I have wanted this book when I first started working with ejectors?” My answer was “Yes”. So, like the Little Red Hen in the children's story, I did it.

I invite your feedback on errors, topics you would like to have seen covered, and topics you may have found especially helpful. I will attempt to respond to all communications. See the website for errata sheets for this edition and the two printings of the previous edition. The website will also have other data and forms and documents for sale.

See the last 4 pages in this book for more details about the website.

Robert B. Power
915 Alynwood Circle
Charleston, WV 25314, USA
Web: www.jetwords.com

Acknowledgments

So many people have contributed to this book that I hardly know where to begin. I will do so chronologically, and ask for tolerance from those many who have helped me who are not mentioned here by name because of limited space and my faulty memory. The many tabulated references are testimony to my debt.

I thank my previous employer, Union Carbide, for an enjoyable and varied engineering career. My ejector assignment in 1960 started out as a brief problem-solving task, and evolved into a major specialty for a few years. I was given the time and latitude to develop the technology and procedures that seemed appropriate to the needs.

Bob Hagen (retired) of the Graham Manufacturing Company was my personal mentor in my ejector education; he taught me patiently. I regard him as a Super Salesman, in the best sense of the term, both as an engineer and as a business associate.

George Schwarz inherited my files when I left the ejector area to work on a machinery assignment. Recently, when I began thinking about writing this book, he returned to me all of my files. *All* of them! It was like being born again.

All the ejector manufacturers contributed generously. They responded with encouragement to my plans to write the book; furnished me with catalogs, reprints, "how to" documents, drawings, and photographs and permission to use them, and spent hours on the telephone guiding me through many problems and feeding helpful suggestions through me to my readers. Don Ruck of Graham Manufacturing manned a hot line to give me special help, and provided useful estimating data to fill some holes in my data.

Valued reviewers for technical correctness, pragmatic usefulness, and readability include four people who read every word. Jim Ryans of Tennessee Eastman represented the user engineer. He strongly encouraged me and shared much valuable information with me, including generous quantities of material from *Process Vacuum System*

Design and Operation, the book he coauthored with Dan Roper. Nels Hammond (retired) from Schutte & Koerting and Dick Richenberg of Graham Manufacturing represented ejector manufacturers. Bill Griffith of Union Carbide represented ejector technicians.

George Schwarz of Union Carbide gave special attention to a review of Chap. 5, "Condensers," and helped me think through other portions of the book. Vic Fondrk, ejector consultant, and Dick O'Conner of Nash-Kinema gave special help on my problems with Chap. 6, "Pressure Control" Nash-Kinema generously allowed me to run a special test to generate data I could not find in the literature or estimate with any confidence.

I thank all of you for helping me, and I enthusiastically introduce my readers to this friendly, responsible ejector community.

Second Edition

I thank my daughter, Deborah Power Carter, for her encouragement and many hours of help in cleaning up the scanned text, then merging the text and graphics to meet strict constraints. Then she gave the whole “finished” draft a ruthless Proofreader’s Scan. I was pleasantly embarrassed at the number of corrections she marked. You and I will both profit from those improvements. Finally, she converted my cover design into the CMYK color mode preferred by the printer.

Jack Burns of Burns Engineering Services suggested several improvements, some of which are described in the Postscript at the end of this edition.

Shirley Grose, S. E. Grose & Assoc. Inc. Printing Consultant, helped me find my way through the complexities of computer and printing graphics and PDF.

And finally, I thank my wife, Joan, for tolerating the many hours I spent at the computer in this project that became much larger than I had expected. I have spent many of our relationship points, not without notice. I realize that I am deeply indebted for this indulgence.

Steam Jet Ejectors for the Process Industries

Chapter

1

How to Use This Book

1.1 In a Hurry

If you are in a rush to "get in, solve the problem, and get out," I suggest you go to the index or the table of contents to find your topic.

1.2 For Orientation Only

If you are interested in a one-time reading to give you a minimum orientation that is appropriate for your job assignment, see Table 1.1 and read the chapters recommended for the job assignments that resemble yours.

1.3 Organized to Match Your Priorities

This book is organized differently from my earlier writings on ejectors [1, 2]. Then I wrote from the viewpoint of a person who was primarily involved in specifying and buying ejectors, and who was also involved with other aspects of ejectors. My writings were organized to match the chronological life of ejectors. That was adequate for those who had experience with ejectors, especially if they had a similar job assignment.

I decided to arrange the topics in this book in order of my perception of your priorities. Basic behavior and theory are covered first, followed by operation, testing, troubleshooting, and maintenance. Then specification and purchase follow, based on my conviction that someone who specifies and buys equipment should first know how it works and what people will do with it in the field. Also, I believe there are

TABLE 1.1 Recommended Reading Priorities by Chapter*

Job assignment	Highly recommended	Suggested	Useful background	As needed
Process engineering vacuum/ ejector specialist	All, in order			
Process design engineer				
With ejector specialist	2, 3, 6, 9, 10	4,5,7,8	12	11
Without ejector specialist	2-10	12		11
Project designer, detailed installation	2	3,7	4,5,8	
Witness for shop performance tests	2,3,8,	10	4	
Construction engineer	2,7,	4,6,8		
Startup:				
Engineer	2,3,6-8	4,5,10		11
Technician	2,3,8	12	4-7	11
Plant ejector specialist (tests, troubleshooting):				
Engineer	2,3,7,8,11,12	4-6,9,10		
Technician	2,3	7,8,11,12	4-6,9,10	
Production				
Engineer (with above)	2-4,6-8	5	9-11	
(without above)	2-8	9-11	12	
Technician	2-4	5	6-9,11	
Maintenance (inspection, repair)				
Engineer	2,3,7,8	4,5,12	6,9,10	
Technician	2,3,8	5	4	
Managers				
Design engineering	2,12	3	9,10	
Production	2,12	3	9	
Maintenance	2,12	3	8	

*Chapter Headings for Reference
2 What Is an Ejector?
3 How Ejector Stages Behave
4 Stages, Engineering Calculations
5 Condensers, Engineering Calculations
6 Presure Control
7 Installation Procedures and Hardware
8 Operation, Testing, Troubleshooting, and Maintenance
9 Selecting a Vacuum Producer: Ejectors and/or Mechanical Pumps
10 Specifying and Buying Ejectors
11 Other Ejector Applications
12 Upgrading Existing Ejector Procedures and Hardware

more people struggling to make ejectors work well than there are people specifying and buying them.

Part 1 (Chap. 2) is an introduction to ejectors, describing in simple terms how they work, the different kinds used in industry and homes, what they are used for, and some of the basic nomenclature used in the industry. An understanding of the basic behavior of ejectors and related equipment leads to better performance and increased enjoyment and learning as you operate, troubleshoot, and maintain ejectors. It is much less complicated for you and me than trying to memorize or write long lists of do's and don'ts for working with ejectors.

Part 2 (Chaps. 3 to 8) helps you get the best performance out of existing steam-jet air ejectors; it gives detailed descriptions of ejector stages and condensers, describes how an ejector system behaves with different methods of controlling system pressure, and covers installation, operation, testing, troubleshooting, and maintenance.

Part 3 (Chaps. 9 and 10) tells how to select, specify, and buy to obtain the best new ejector systems.

Part 4 deals with special topics and how to upgrade ejectors. Chapter 11 is a collection of information about several specialized "cousins" of the steam-jet air ejector, some low-cost stock-design "utility" ejectors, and some uncommon situations requiring special design attention. Chapter 12 offers suggestions and guidelines for upgrading your procedures and hardware for better overall ejector performance.

Although most topics are written to be understood by readers who are not interested in all the technical details, many also include information for engineers and technicians who need to calculate numerical answers. In writing for both sets of readers, I realize that I may irritate both. Please tolerate the material which goes slowly for you and read lightly or skip over the detailed technical areas which do not concern you.

1.4 Where to Invest Your Reading Time

As Table 1.1 shows, I recommend that all persons working with any aspect of ejectors read at least Chap. 2 to learn enough of the basic concepts and nomenclature to communicate and work effectively with other people and understand the literature they use.

I recommend that ejector/vacuum specialists and process engineers who often design ejector systems read the whole book. At the other extreme, their managers may wish to read only the introductory subjects plus the suggestions I offer for upgrading working procedures and ejector hardware to make ejectors work better. Between those extremes are many engineers and technicians who need a different mix related to their specific job assignments. Even if your job assignment is different from any in the table, you may still find the table useful.

The reading priorities given are my judgment about what will be the most productive use of your time.

1.5 Do It, Don't Just Read It!

To benefit from your investment of dollars and valuable reading time, it is necessary for you to act differently after reading this book than you did before you began reading it. My objective in writing this book and your objective in reading it are probably the same: to give you the ideas and tools to enable you to be more productive with your working time, and to get more satisfaction in the process. If you simply read it, agree and disagree with my ideas, then put it aside and resume life as usual, we both fail.

I am hoping that I can give you the tools and the courage to do some things differently than before--better! You will have to take some risks, such as spending more money and time on certain aspects of ejectors or deciding to save some money by buying a *smaller* ejector on the next job than you used on the last identical one. Until you sweat and worry a little, you are not changing as much as you probably want to.

Performance-test an ejector, read and interpret performance curves, order missing data from the manufacturer, discuss a problem with the manufacturer, use the specification forms adapted to your needs, meet with someone who has a related interest in ejectors, and discuss ways in which you might more effectively procure and use ejectors. Mark up the book; it is simply another tool you own.

In looking back from a semiretired position, I realize I was more timid than I would like to have been. With this book I am releasing you from the excuse of ignorance. Have courage and enjoy!

1.6 Nomenclature

The nomenclature section given in most chapters defines abbreviations, jargon, and equation variable names for that chapter. Some symbols are defined and used differently in different chapters.

1.7 References

1. R. B. Power, "Steam-Jet Air Ejectors: Specification, Evaluation, and Operation," ASME Paper 63-WA-143. Reprinted by *Hydrocarbon Processing and Petroleum Refiner,* 43 (2): 121ff and 43 (3): 138ff, 1964.
2. R. B. Power, "Steam-Jet Air Ejectors," 10-part series, *Oil and Gas Equipment,* October 1965-July 1966.

Part

1

Introduction to Ejectors

Chapter

2

What Is an Ejector?

An ejector is a device in which a high-velocity jet of fluid mixes with a second fluid stream. The mixture is discharged into a region at a pressure higher than the source of the second fluid. The ejector pumps the second fluid from one place to another. Because the ejector has no moving parts, it is difficult to understand how it operates. Therefore, it is timely now to consider some related experiences common to many people.

2.1 Behavior of a Water Hose Jet

The high-velocity jet of water emerging from a hose nozzle has the power to sweep away water, mud, sand, and small objects along with it. The mixture may have enough velocity to flow up and over a small curb or wall at the edge of a driveway, street, or concrete slab.

If the jet is directed into a bucket partially filled with water, it will penetrate into the water, carrying air along with it and mixing the contents vigorously. If the hose nozzle is adjusted to spread out the jet, it may carry more air into the bucket. Indeed, many people who have done this have been surprised to have the mixture of air and water spray back out of the bucket onto their faces and clothing. The confined space within the bucket concentrates and directs the energy of the jet.

2.2 Adding a Mixing Tube

Improving on the principle of concentrating and directing the energy of a jet, we may spray the water through a cylindrical device such as a 4-in-diameter section of stovepipe 2 ft long. It may be necessary to adjust the nozzle to spread out the spray to "fill" the stovepipe and

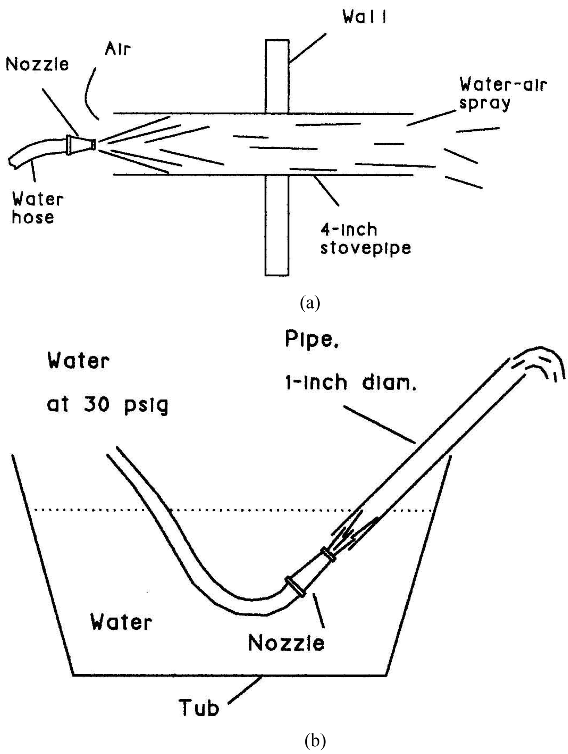


Figure 2.1 *(a)* Crude exhaust air pump, water-jet air ejector; *(b)* crude water pump, water-jet water ejector.

get the maximum air-pumping effect. The mixture of air and water will emerge as a high-velocity directed spray. If we place the stovepipe through a hole in the wall of a room, we will have a crude but working exhaust air pump, as shown in Fig. 2.1*a*.

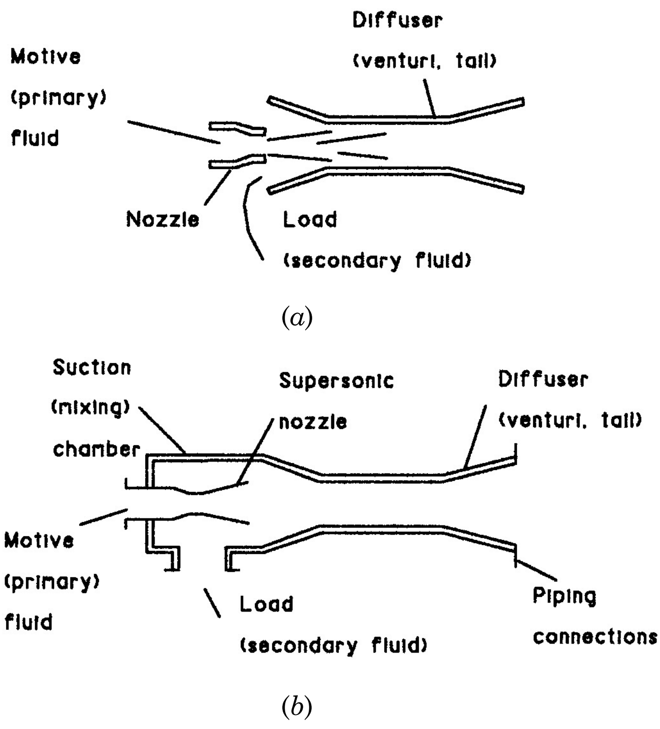
As a variation, we may spray the water through a 1-in-diameter pipe 2 ft long, adjusting the nozzle for a narrower spray pattern. If we submerge the hose nozzle and the lower end of the pipe in a tub of water and place the nozzle near the end of the pipe, so that the jet sprays into the pipe, as shown in Fig. 2.1*b,* we will see a steady flow of water emerge from the top end of the tube and notice that the water level goes down in the tub. We have created a crude water-jet water ejector. By experimenting with the nozzle adjustment and position, we can maximize the pump-out rate.

2.3 Improving the Mixing Tube

Two improvements can be made in the shape of the tube used to confine and direct the jet action: The inlet can be tapered to provide a smoother path for the load fluid to enter and mix with the jet fluid, and the outlet can be tapered to reduce the velocity of the mixture in a manner which converts velocity energy into pressure energy. This improved mixing tube is shown in Fig. 2.2*a*. It now has the shape which is the dominant feature of an ejector. It is commonly named the *diffuser* because it spreads out (diffuses) the stream in a manner which increases the pressure. This is the name used in the *Heat Exchange Institute Standards for Steam Jet Ejectors* [1] and the one I shall use. The diffuser is sometimes called a venturi because it resembles the well-known flow measurement device with that name. Sometimes it may be called the tail of the stage.

2.4 Basic Ejector Stage

A few more basic features are added to the stage hardware to arrive at the typical basic ejector stage shown in Fig. 2.2*b*. The suction chamber has been added to keep the nozzle properly positioned with respect to the diffuser and to direct the flow of the load (suction) fluid.



(*a*)

(*b*)

Figure 2.2 *(a)* Improved diffuser; *(b)* basic ejector stage.

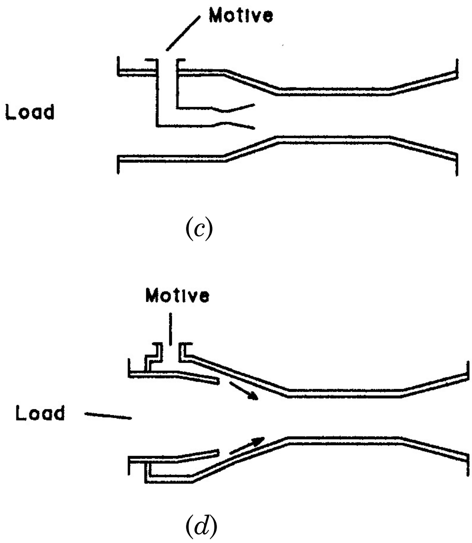


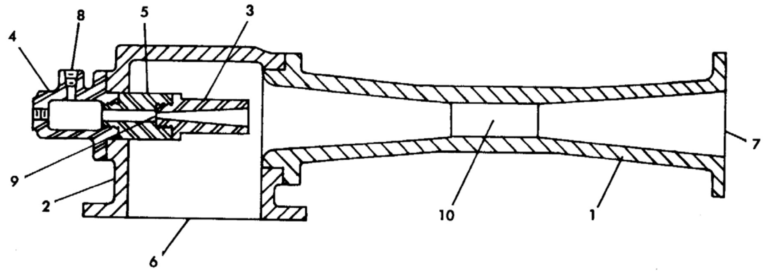
Figure 2.2 *(Continued)* (c) Inline ejector; (d) annular-jet ejector.

so that it mixes efficiently with the jet of motive fluid. The term suction is commonly used to describe the load fluid and connection, even if the pressure there is above atmospheric pressure. The nozzle is refined in this drawing to a converging/diverging (deLaval) shape required for gases to expand to velocities greater than the speed of sound (supersonic). Finally, provisions are made for connections to external piping because usually the motive and suction (load) fluids are brought to the ejector in process piping and the mixture is discharged into process piping.

A variation from the basic configuration is the in-line configuration shown in Fig. 2.2*c*. Here the external connections for the motive and suction fluids are switched, although the nozzle exit is still positioned to discharge along the diffuser centerline. Another variation is to introduce the motive fluid through an annular nozzle, as shown in Fig. 2.2*d*. Other variations which will be seen later include multiple nozzles for more efficient mixing and shorter stages, suction chambers which are an integral part of a process vessel, open-structure suction for mud pumps, hopper suction chambers for conveying solids, sanitary connections for easy cleaning when handling foods or pharmaceuticals, and screwed, welded, or glued connections.

2.5 Typical Steam-Jet Air Ejector Stage

A typical steam-jet air ejector stage is shown in Fig. 2.3 with its component parts labeled. The diffuser in this stage is a one-piece casting



1. Diffuser
2. Suction Chamber
3. Steam Nozzle
4. Steam Chest
5. Extension (if used)
6. Suction
7. Discharge
8. Steam Inlet
9. Nozzle Throat
10. Diffuser Throat

Figure 2.3 Typical steam-jet ejector stage assembly. *(Courtesy Heat Exchange Institute)*

with a flanged connection to the suction chamber. Two-piece diffusers are common, joined at the diffuser throat. Smaller sizes of corrosion resistant materials may be machined from bar stock. Larger sizes are typically fabricated by rolling and welding plates. Externally visible features that are often present are steam jackets on diffusers and test pressure taps in the suction chambers and steam chest. Internal features which may vary are the type of gaskets used for the steam nozzle and extension, steam jacketing of some steam nozzles, the method of mounting and removing the steam nozzles, and steam strainers built into the steam chest.

2.6 Variation of Velocity and Pressure in a Stage

Figure 2.4 shows how velocity and pressure vary for the motive and suction materials through an ejector stage. Both streams flow toward the lowest-pressure spot in the stage and mix together there in a violent and rapid manner. Then the mixture slows down and the pressure rises before the mixture emerges at low velocity at the discharge. The discharge pressure is usually somewhere between the motive and suction pressures.

2.7 Some "Cousins" in the Ejector Family

Ejectors are extremely versatile from two viewpoints: They can be made of almost any solid material, and they can use a wide variety of

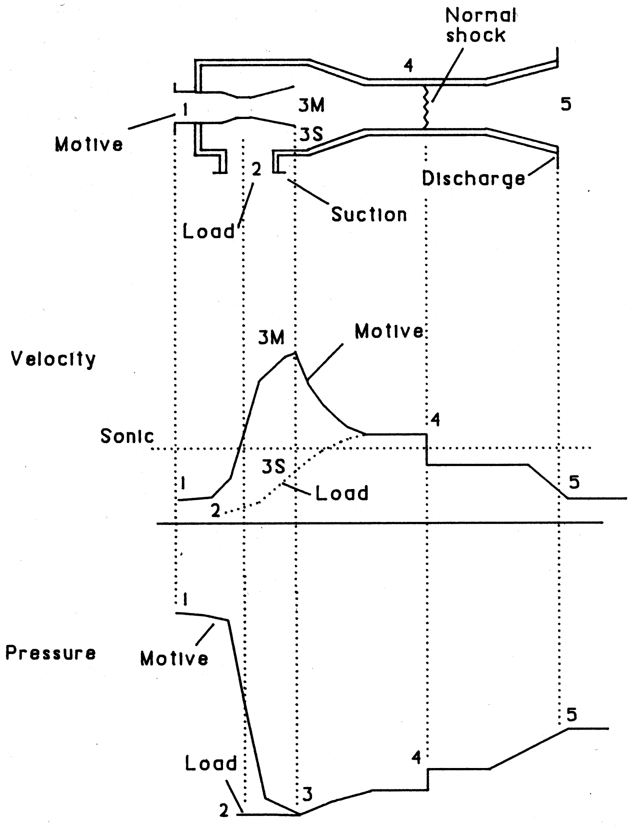


Figure 2.4 Velocity and pressure profiles in an ejector stage.

pressurized gases or liquids to pump gases, liquids, and even granular solids. Ejector manufacturers portray their product lines in different manners: arranged by application function (what is being pumped), by motive fluid (what is doing the pumping), and by their hardware lines (where the job gets done). The names they assign to hardware systems and components also vary: They may use variations on a trademark product line name, use functional names for each application, or sometimes simply use product line model numbers.

Table 2.1 is my overview of the ejector family. It arranges ejectors by motive fluid and load (suction) material. This is simply one generic display of selected applications, which you may find listed by manufacturers in other manners using other, more specific descriptions of the applications and hardware names [2, 3].

Here I will briefly describe a few of what I regard as the most important combinations of specific motive and load materials, then group some of the others into sets of similar applications. More

TABLE 2.1 Ejector Applications

These Fluids ↓ Do These Jobs

	Load Materials				
	Water vapor Steam	Air	Gas, vapor	Liquid	Solids
Steam	Refrigeration, stripping, drying, compressor	Vacuum, compressor	Vacuum, compressor	Pump, heater, injector	Refrigeration, stripping, drying, compressor
Air		Vacuum, compressor	Vacuum, compressor	Sampling, mixing	Conveyor
Gas, vapor		"BTU controller," vacuum	Vacuum, compressor	Sampling, mixing	Conveyor
Liquid	Vacuum, condenser	Vacuum, pump priming	Vacuum	Pump, mixing	Conveyor, mixing

detailed descriptions of these applications will be found in Chaps. 11 and 12.

Steam and air are handled separately because of their status as common utility fluids. Water vapor is lumped with steam for convenience. It is thought of here only as the subatmospheric pressure occurrence of the vapor phase of water. Gases and vapors are also grouped together because their use as motive fluids is less common and requires special design attention. Gases are commonly thought of as materials which are not condensable under normal operating conditions, and vapors are thought of as potentially condensable under some normal conditions.

Steam-jet ejector family

This category includes air ejectors, the vacuum pump application that most of this book is about. Steam has special properties, including its heating ability, and is often available at one or more pressure levels throughout a processing plant. Thus, steam is the most versatile of the motive fluids and is used to pump all types of gases, liquids, and granular solids.

Vacuum refrigeration. Vacuum refrigeration is the commercial application of a common science demonstration. In the demonstra-

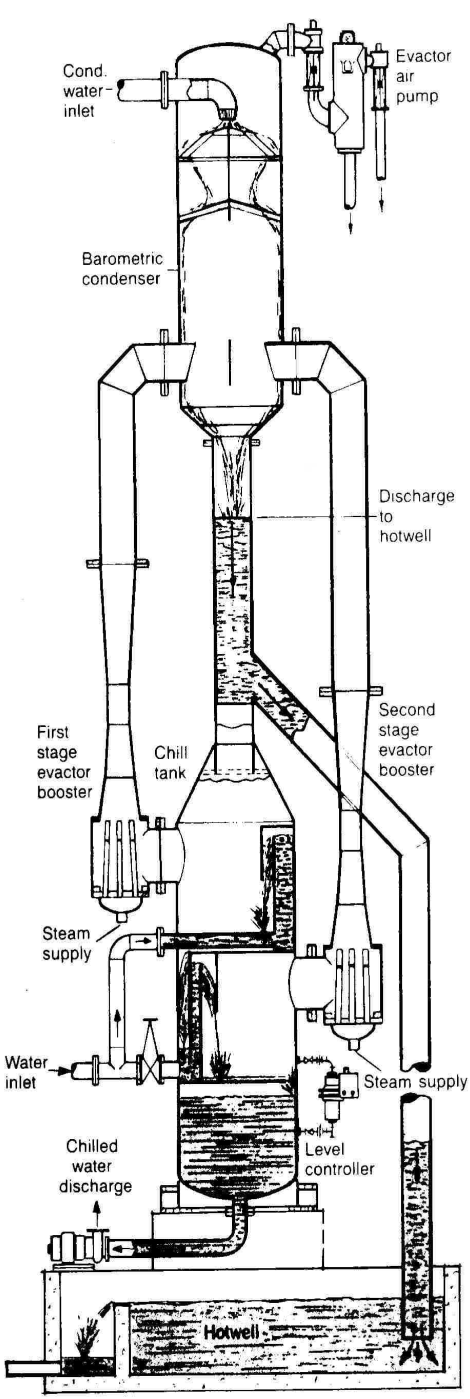


Figure 2.5 Vacuum refrigeration. *(Courtesy Croll Reynolds Co.)*

tion, a bowl of water is placed under a glass bell jar, and a vacuum pump lowers the pressure inside the bell jar until the water boils at room temperature. Similarly, as warm water is sprayed into a vacuum tank, some of the water will evaporate, cooling the remaining liquid.

Large ejector stages (Fig. 2.5) chill a spray of water under vacuum by removing water vapor from the spray and compressing the vapor to a higher pressure so that it can be condensed along with the motive steam in a condenser. The condenser may be a direct contact device in which the vapors condense into a stream of cooling water, or it may be a water- or air-cooled surface condenser. Advantages are simplicity, low first cost, small real estate requirements, and reliability.

Vacuum stripping, drying. These are similar to refrigeration except that the water vapor is extracted from a liquid, slurry, or solids mixture containing material other than water. These ejector stages are commonly called "boosters"; they boost the water vapor to a pressure at which it may be condensed. Such a booster application is shown in Fig. 2.6 with the booster stage followed by a large condenser and two small air-handling stages.

Steam compressor (thermocompressor). This device often operates at pressures above atmospheric pressure. It uses high-pressure steam to pick up low-pressure steam and deliver it at a pressure sufficiently high that it condenses and yields useful heat to some process. The device recovers heat which might otherwise be lost, thus reducing the

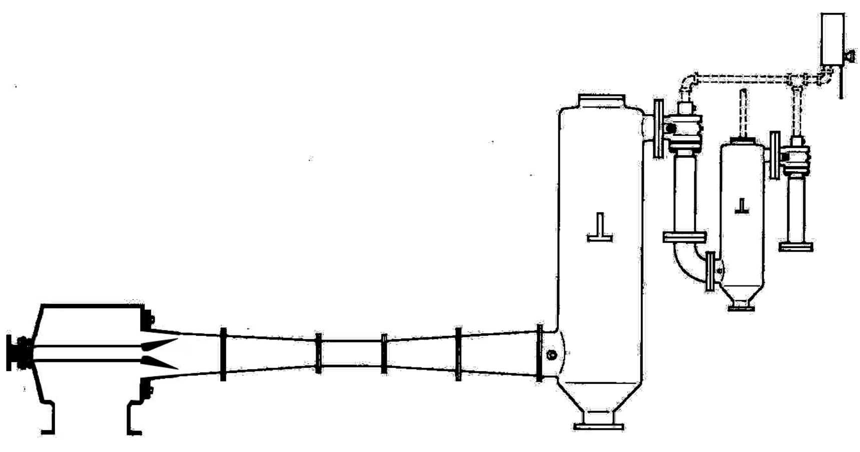


Figure 2.6 Booster stage, three-stage ejector. *(Courtesy KETEMA, Schutte & Koerting Division)*

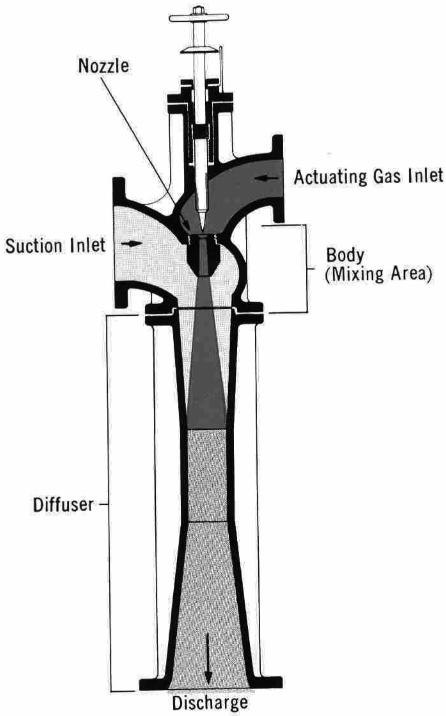


Figure 2.7 Jet compressor. *(Courtesy KETEMA, Schutte & Koerting Division)*

quantity of expensive high-pressure steam consumed. Figure 2.7 shows a typical jet compressor with a spindle nozzle which permits manual adjustments to change the operating characteristics.

Steam-jet air ejector. The air which leaks into all commercial process systems operating at subatmospheric pressure must be removed continuously to maintain proper operation. If the process pressure is much below atmospheric pressure, then two or more ejector stages in series may be required, as shown in Fig. 2.6. Typically, condensers are used between stages to reduce steam consumption. In the process industries this is a demanding application because of the variety of chemicals encountered and the complex considerations of safety, corrosion, contamination of products and the environment, economics, automation, reliability, operation, and maintenance.

Steam-jet air compressor. High -pressure steam may be used to compress atmosphere air to a modest pressure, supplying 20 psig instrument air, for example [11], or backing up a motor-driven instrument air compressor. I once developed the design of a two-stage unit that would deliver air at 40 psig, using high-pressure motive steam and water-cooled condenser-coolers.

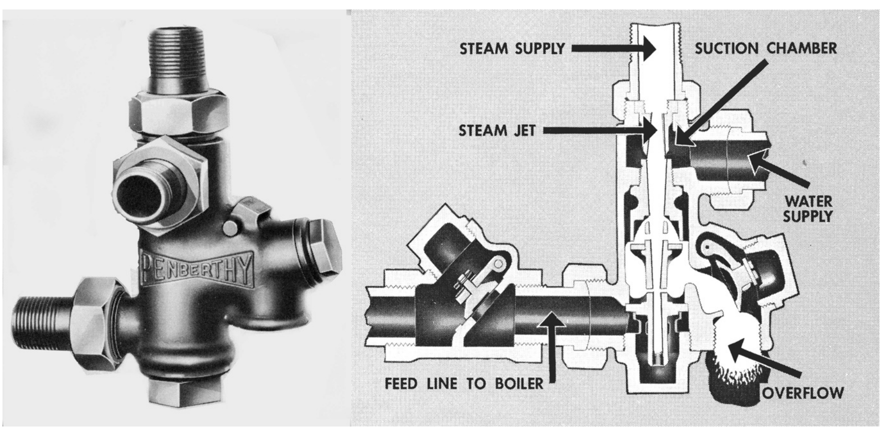
Steam-jet liquid ejector, injector. When steam pumps a liquid, it typically condenses and heats the liquid. Depending on the effect that is most desired, it may be described as a pump (steam-jet exhauster, jet syphon, sump drainer, jet pump) or as a heater (steam-jet heater, in-line heater).

The injector is an unusual application that seems at first glance to defy some laws of nature. It is typically used to pump feedwater into a boiler, using steam from the boiler. As an example, 100 psig steam can lift water at a temperature of 74°F as much as 20 ft from a supply source, mix with it and condense into it, and provide enough energy to pump the mixture back into the boiler against the boiler pressure-with no moving parts! Figures 2.8*a* and *b* show the external and internal appearance of this device.

Air-, gas-, and liquid-jet ejectors

Compressed air, like steam, is often available throughout plants and may be used for a variety of ejector applications. Because it has less energy than steam and does not condense to a liquid under ordinary conditions, air is usually more costly than steam for continuous operations. Other pressurized gases are often available and are proper for some ejector applications. Pressurized water and process liquids are also used to run ejectors. Here are some of those applications.

Air-jet vacuum pump. Typically, air is used when the load is small and can be accomplished by one stage, or when the convenience of not having condensers justifies the cost of the air used.



(a) (b)

Figure 2.8 Injector: (*a*) external appearance; (*b*) working parts. *(Courtesy Penberthy Division of Stanwich Industries, Inc.)*

Organic-fluid/process-vapor-jet vacuum pump. In recent years many people have adopted the practice of using pressurized vapors other than steam to drive their vacuum ejector systems. Manufacturers have accumulated experience in developing and designing such systems, typically including feed pumps, vaporizers, and controls in a fabricated package. Benefits are: complete separation of water from the product, elimination of water contamination, more freedom in choosing materials of construction, and sometimes a reduction in total energy usage.

Hot gas ejectors for space simulations. The space program has developed applications for some of the largest ejectors ever built. Visualize a two-stage ejector with individual stages 100 ft long, with a suction connection 10 ft in diameter, and using over a million pounds per hour of hot, high-pressure gas to drive it. Then imagine the terrible noise it would make when operating!

Gas-jet compressors. Identical in appearance to thermocompressors, these differ only in that the pumping job does not primarily involve thermal benefits. They may use the excess pressure in a feed gas stream to circulate gases, or to entrain a second gas and raise the pressure of the mixture. One interesting application is the "BTU controller" shown in Fig. 2.9. A battery of appropriately sized ejectors uses high-pressure propane gas to entrain atmospheric air and deliver a controlled-BTU mixture at a usable pressure. Note that the adjustments here are made by turning individual units on and off instead of using spindle nozzles.

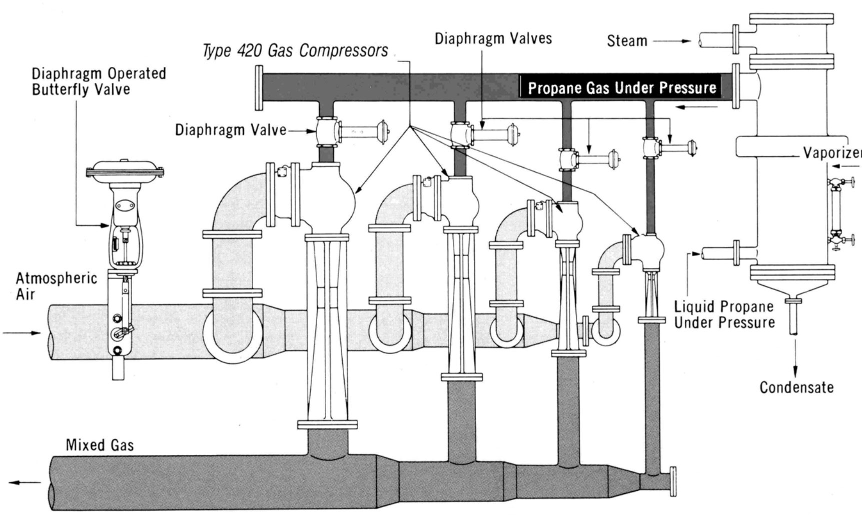


Figure 2.9 BTU controller, gas-jet ejector.
(Courtesy KETEMA, Schutte & Koerting Division)

Utility ejectors. Many small, standard-design ejectors are kept in stock by some ejector manufacturers. Low first cost, simplicity, availability, and convenience are their virtues. They are simplified versions of their special-purpose cousins. Typically, the diffuser and suction chamber are cast as one piece and combined with a screw-in nozzle which provides modest adjustments for high and low motive fluid pressure and for either liquids or gases as the motive fluid.

Some generalizations about these may be useful. Steam pumps anything well, gas pumps gas, and liquid pumps liquid. Gas is inefficient at pumping liquid if no condensation occurs, and liquid does not pump gas well if no condensation occurs. When efficiency and speed are not very important, any combination may work fine. Safety and convenience are usually the major considerations.

Figures 2.10*a*, *b*, and *c* show the construction of one such unit and two typical applications. Figures 2.11*a*, *b*, and *c* show another variation in the design of the small units, plus the addition of a removable diffuser throat piece in the larger units. The combination of removable nozzle and throat pieces permits much more adjustability in get-

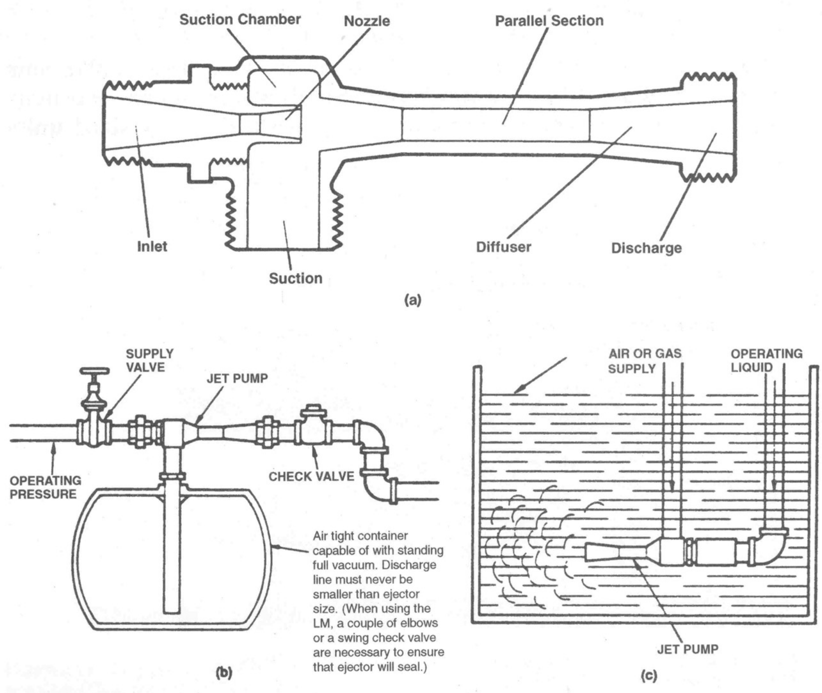


Figure 2.10 Utility ejector, two-piece, male fittings: *(a)* construction; *(b)* producing a vacuum; *(c)* aeration or agitation. *(Courtesy Penberthy Division of Stanwick Industries, Inc.)*

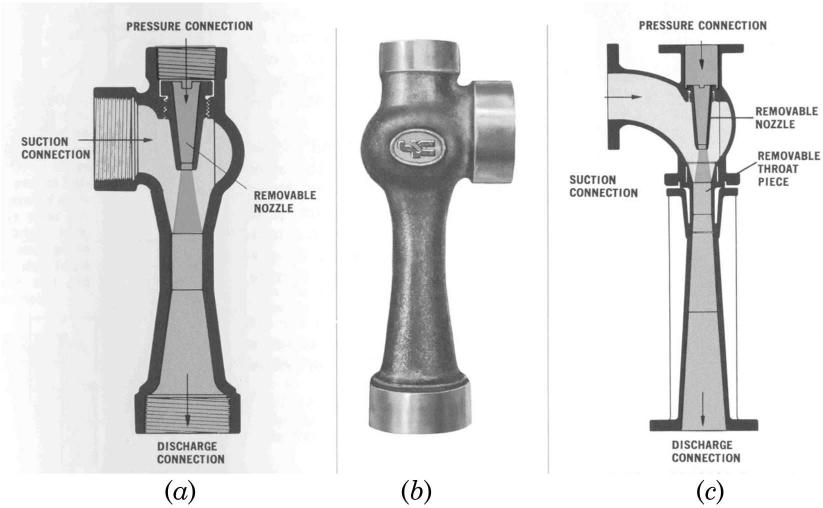


Figure 2.11 Utility ejectors: (*a*, *b*) two-piece, female fittings; (*c*) four-piece, flanged fittings. *(Courtesy KETEMA, Schutte & Koerting Division)*

ting the best performance for a given application. Figure 2.12 shows a utility ejector used to prime a pump that has a suction lift.

Very small air-driven units have been used to draw continuous samples from an effluent water flume. Small water jets are routinely used to create modest vacuums in laboratories. Properly sized units

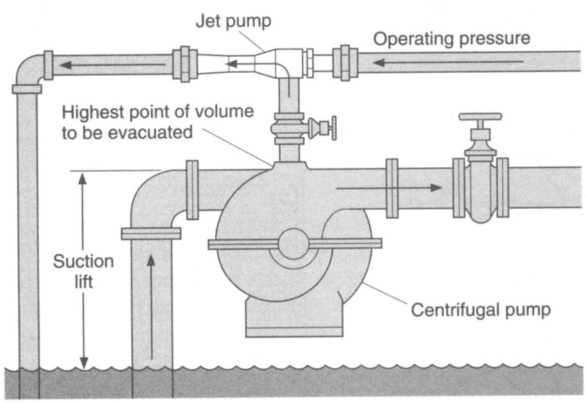


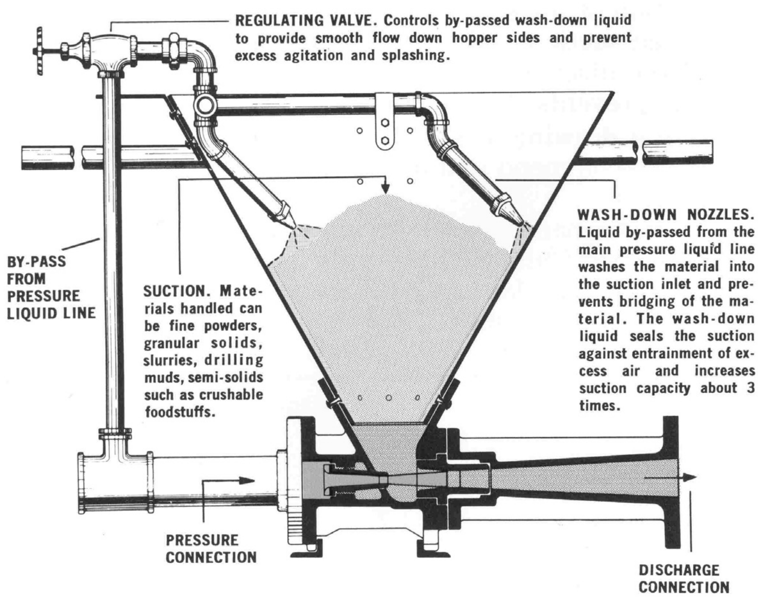
Figure 2.12 Utility ejector priming a centrifugal pump. *(Courtesy Penberthy Division of Stanwich Industries, Inc.)*

use the excess pressure in one liquid stream to assure rapid mixing with a second stream.

Jet-type solids conveyor. Liquid- or gas-powered ejectors are used to convey granular solids. A hopper is used to deliver the solids to the ejector suction, as shown in Fig. 2.13. In this liquid-driven application, some of the liquid is diverted to wash the solids down and partially seal the suction. Variations on this design have used dry nitrogen gas to lift dry catalyst into a reactor in a batch operation mode. The hopper was fitted with a cover and purged with nitrogen to flush out air before the transfer was begun.

Variations on this have been used to pump ashes, sand, and other granular solids.

Jet-type garbage pump. A tough application is handling liquid streams containing debris that would tend to plug up the suction chamber of the normal, right-angle inlet ejector. For this application, the in-line arrangement eliminates the troublesome corner. Arranging multiple jets in an annular configuration leaves the center open and sweeps the walls vigorously at the inlet to the diffuser, as shown in Fig. 2.14. This is called a garbage accelerator. (I love that name!)



(*a*)

Figure 2.13 Hopper-inlet solids conveyor: *(a)* large unit with wash-down nozzles. *(Courtesy KETEMA, Schutte & Koerting Division)*

Figure 2.13 *(Continued)* (*b*)Hopper inlet solids conveyor, small unit with standard-size suction. *(Courtesy KETEMA, Schutte & Koerting Division)*

(*b*)

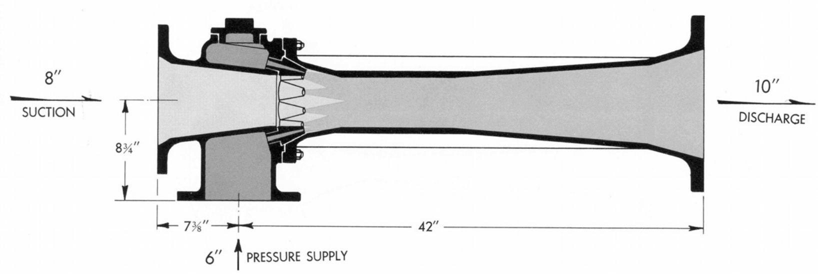


Figure 2.14 Garbage pump. Peripheral arrangement of jets helps minimize plugging. *(Courtesy Derbyshire Machine & Tool, Inc.)*

Jet-type sludge pump . An adaptation for pumping really messy liquid-solid materials is the liquid-jet ejector with an open-structure suction geometry to permit the load mixture to flow easily toward the high-velocity jet. Some of the motive liquid is sprayed back into the suction area to help break up and liquefy dry or hard lumps in the mud or sludge. The ejector is at the end of a pair of hoses (motive and discharge streams) and is carried to the work site manually or by a crane or boom. That sounds like a better way to do it!

Ejector-venturi scrubber. The pumping and intimate mixing in ejectors are used to advantage in the venturi scrubber to clean contaminated gases in an energetic liquid spray, as shown in Fig. 2.15.

Jet-type vacuum condenser. In a direct-contact condenser, water flows downward, condensing steam and other vapors, then flows out of the drain connection, carrying with it some of the noncondensable gases. In the jet condenser design shown in Fig. 2.16, the water is directed downward in the form of several high-energy jets aimed carefully at the drain connection, which functions as a diffuser. The jets entrain and remove such a large amount of noncondensable gas that it is sometimes practical to omit a vacuum pump, which usually is required for air removal. This benefit is gained at the expense of greater than normal water usage.

2.8 Ejectors versus Other Pumping Devices

As the previous review of ejectors tends to indicate, they can do many pumping jobs---why consider any other pumping devices? The answer is that although ejectors have some "natural" applications which they handle better than any other pumping device does, there are other applications where they are clearly the wrong choice, and a lot of "fuzzy" areas where a careful comparison of the various pumping devices doesn't clearly favor any one of them.

Simple geometry and absence of moving parts makes the ejector sturdy. It can handle two-phase flow and is undamaged by slugging; it can handle solids; it is easily jacketed or insulated for extreme-temperature service; and it can be designed for high pressures or hazardous services.

Briefly, the ejector's primary advantages are low first cost, simplicity, reliability, and availability in almost any material of construction. Its primary disadvantage, low efficiency, results directly from its unique feature---no moving parts. Consider one of the alternative pumping devices, a centrifugal pump. The pump impeller can gently engage the pumped fluid and accelerate it to high velocity so that the velocity can be converted to pressure. Lacking such a part, the ejector must mix the high-energy device (the jet) directly with the load fluid to accelerate it to a high velocity. This mixing is the primary source of the low efficiency of most ejectors.

Low ejector efficiency has often been overstressed and distorted

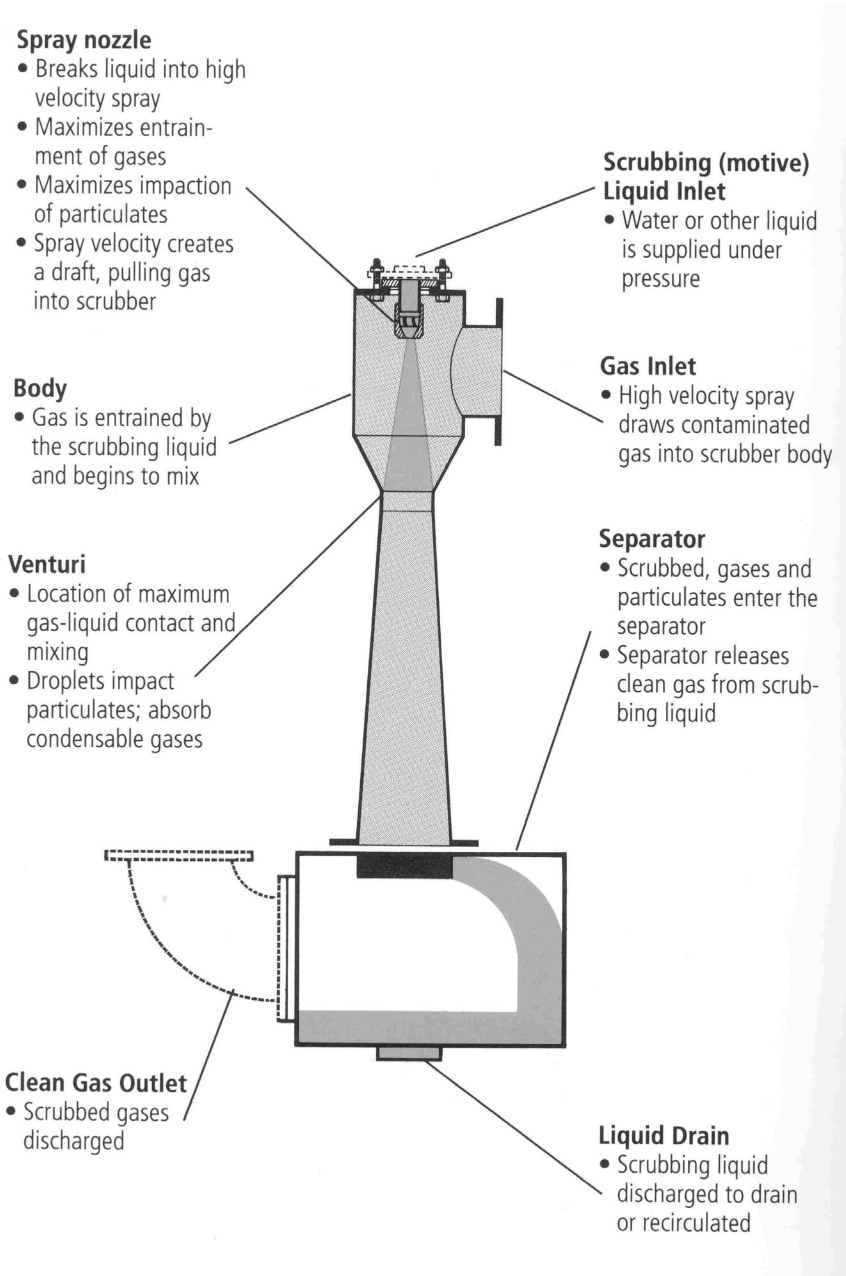


Figure 2.15 Venturi scrubber. (Courtesy KETEMA, Schutte & Koerting Division)

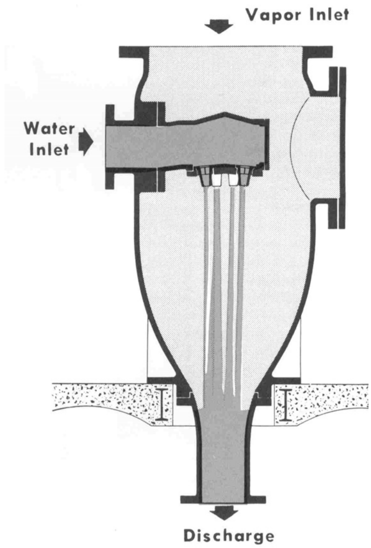


Figure 2.16 Jet condenser. *(Courtesy KETEMA, Schutte & Koerting Division)*

by people promoting a variety of vacuum pumping devices. I believe a realistic, practical approach to any comparison of alternatives for any pumping application should go well beyond simply looking at those factors to which cost values can be easily assigned. It is often easy to estimate installed costs and operating costs, then develop an informed guess about maintenance costs for each altemative.

Plant designers really begin to earn their pay when they start on the harder task of developing accurate measures of the many very important qualitative factors. Among them are reliability, equipment life, safety, operation and maintenance skill requirements, spare parts stocking, logistics of equipment repair, product and environment contamination, and the requirements for technical and management support of the installation. In this book I will share with you what I know about ejectors and comment on some other vacuum devices with which I am less familiar.

2.9 Pressure Scales and Measurement

If you want to get the best performance out of vacuum ejector systems, you must be able to measure subatmospheric pressures with adequate accuracy.

Almost everyone has had occasion to read some kind of pressure measurement device, whether it be a barometer, an automobile oil pressure gauge (years ago), a steam or air pressure gauge, a manometer, or an instrument designed specifically for vacuum measurements. Which brings us to the question, "What is a vacuum?" Strictly speaking, a vacuum is the absence of anything---nothing. Commonly, it is used to describe a pressure that is lower than atmospheric pressure, or a system at such a pressure. If you make a hole in the wall of the system, air leaks into the system.

A common method of measuring the pressure in a vacuum is to connect a device such as a manometer or a pressure gauge between the inside of the system and the atmosphere. The measurement obtained is the difference between the pressures inside and outside the system. That measurement is then subtracted from the atmospheric pressure to obtain the absolute pressure. Because atmospheric pressure is measured with a barometer, it is often called barometric pressure.

The above procedure is often sufficiently accurate when measuring small differential pressures, but it becomes awkward when measuring the low pressures encountered working with steam-jet air ejectors. For example, a mercury manometer might measure a vacuum as 29.9 inHg *below* a standard barometric pressure of 30.0 inHg. Notice that the reference barometric pressure must be given or implied to give the full meaning of the measurement. An easier way to describe the same pressure is to call it 0.1 inHg *absolute*. Absolute pressure is more useful than gauge pressure for much of the theory and calculations involving ejectors.

Absolute pressure is pressure measured relative to the pressure that exists inside a perfect vacuum --- zero. Although that perfect reference condition can never be created, it can be approached closely enough for many practical purposes. The specific methods for creating that reference condition will be described in Chap. 8. In this chapter I will finish this topic by describing the relationships between absolute pressure, barometric pressure, and vacuum and gauge pressures. Figure 2.17 shows the relationships graphically and in equation form.

In this book I will use torr, after Torricelli, as the unit of pressure measurement for vacuum systems. This is an internationally accepted standard. One torr is equal to 1.0 mmHg absolute. The standard atmospheric pressure at sea level is 760 torr. In other units, it is approximately 30 inHg, 14.7 psia, or 34 ft of water. Another common standard is the bar, equal to 750 torr, 100,000 pascals, and 1000 mbar.

Rough vacuum

When talking about extremely low pressures, it becomes convenient to use scientific notation. One hundredth of a torr thus becomes 1.0E-02 torr. Vacuum technologists have found it useful to refer to four

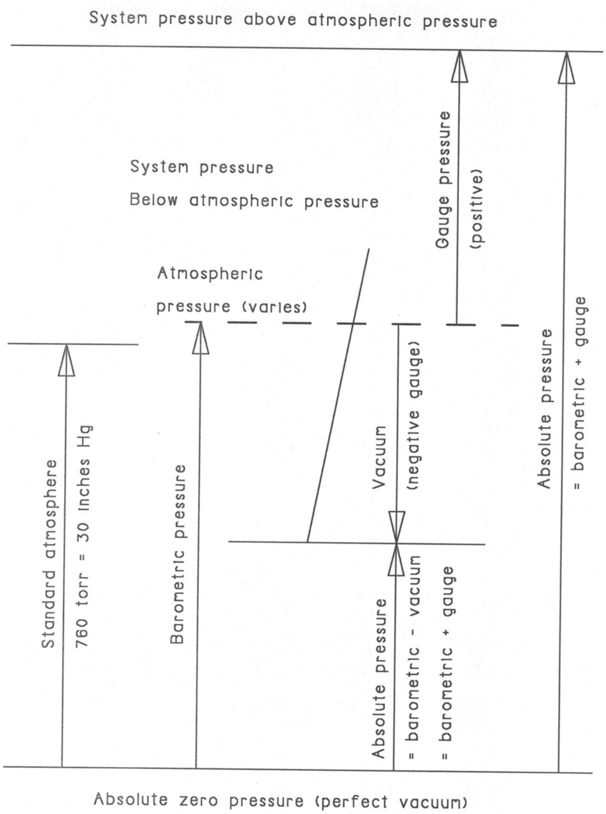


Figure 2.17 Comparison of pressures: absolute, atmospheric, vacuum, and gauge.

separate ranges of subatmospheric pressures [4]. Note carefully that this convention refers to low absolute pressures as high vacuums.

Rough vacuum	760 to 1 torr
Medium vacuum	1 to 1 E-3 torr
High vacuum	1 E-3 to 1 E-7 torr
Ultra-high vacuum	1 E-7 torr and below

New and exotic fluid behaviors occur at the very low pressures. Molecular flow, adsorption of gases onto solid surfaces, and diffusion out of and through metal walls become important. Fortunately for us, most of the major processing vacuum operations are conducted in the range of 0.1 to 760 torr [5].

2.10 Temperature Scales

Temperature measurement is also important to a person working with ejectors, and fortunately temperature is relatively easy to measure with adequate accuracy. Two minor complications are encountered and must be handled carefully. Common scales used to measure temperature are Fahrenheit (F) and Celsius (C). For calibration convenience, these scales have zeros at temperatures that are easy to create in laboratories. However, theory and calculation methods make it more convenient to use absolute temperatures. Like absolute pressure, absolute temperature is measured relative to an ideal zero condition: no molecular motion. The absolute temperature scales, Rankine (R) and Kelvin (K), have zeros at absolute zero. Zero on the Rankine scale corresponds to -460°F, and a 1°R change exactly equals a 1°F change. In a similar manner, 0°K equals -273°C, etc.

2.11 Thermodynamics

The subject of thermodynamics deals with the properties of materials and how they are affected by processes which work or heat and cause the material to change from one solid, liquid, or vapor state to another. For example, gas laws describe the relationships between pressure and temperature and the volume occupied by a quantity of gas. They also describe how a gas cools and reaches high velocity as it expands through a nozzle, how it mixes with a suction gas stream, and how the mixture slows and warms as the pressure in the diffuser rises.

For condensers, we use thermodynamics to describe the amount of heat released as steam and vapors condense, and how much vapor accompanies gases that emerge from the vent of a condenser and enter the next ejector stage or leave into the atmosphere.

I am introducing these concepts of thermodynamics here to introduce the nonengineer readers to some basic concepts and technical terms they will soon encounter, even though more detailed theory and equations will occur only in later chapters. Terms such as enthalpy, specific heat, and entropy are used to define the nature of certain processes, such as the flow through a nozzle. Ideal laws are simple equations that describe approximately how materials behave under a limited range of conditions. The ideal laws are often accurate enough for engineering design or rough estimates. Sometimes corrections must be made for the nonideal behavior of gases or liquid mixtures to get the needed accuracy.

A few important terms are encountered early in a study of gas-jet ejectors. Expansion ratio is the ratio of the motive gas absolute pressure to the absolute suction pressure. If the motive pressure is kept constant and the suction pressure gradually lowered (expansion ratio increased), the mass flow will continue to increase until the expansion ratio reaches about 2. At that condition, the maximum flow through the nozzle is reached, and the gas will emerge from the nozzle at the velocity of sound for that gas. If the expansion ratio is greater than about 2, the gas emerging from the nozzle will have a velocity greater than the velocity of sound for that gas. That condition is called *critical flow*. Further reduction of the suction pressure will not change the motive flow. Thus, critical-flow nozzles are constant flow devices when the upstream conditions remain constant.

Compression ratio is the absolute discharge pressure divided by the absolute suction pressure. Thus, an ejector stage that accepts air at 10 torr and delivers it at 80 torr has a compression ratio of 8. Just as with the nozzle, the overall ejector operation is defined as critical if the design compression ratio is greater than 2. If the stage was designed to have a full stable range, the resultant effect is that reducing the discharge pressure will have no effect on the suction conditions.

2.12 Fluid Statics and Dynamics

Another area of engineering study that is quite useful in working with ejectors is fluid statics and dynamics. Statics deals with the way pressure acts on fluids and the way pressure increases with depth in a liquid or gas, and the fluid property of most interest here is density. Dynamics deals with the behavior of fluids as they move around. Terms such as viscosity, velocity, momentum, efficiencies, flow coefficients, and kinetic energy are commonly used. These subjects will be discussed in detail in later chapters.

2.13 Condensers

The function of a condenser is to reduce the total load going to an ejector stage or to the atmosphere by cooling and condensing out most of the water vapor and condensable process vapor. The simplest condenser is the contact type, in which the condensing water sprays downward and cools a rising gas/vapor stream. Some of the noncondensable gas will be washed down with the water, but typically most will pass out the vent connection at the top. The cooled vent gas then passes on to the next stage or to the atmosphere. Usually it is desirable to avoid contaminating the cooling water, so a surface condenser transfers heat between the two streams. Chapter 5 treats this subject in depth.

2.14 Stage and Configuration Numbering and Naming Conventions

Because many ejector systems have more than one stage and may have one or more condensers, it is important to have a method of referring to them in a manner that clearly identifies what we are talking about. There is a clear authority for this subject, but there are many exceptions in current and past practice. I will review them here and recommend that you remain alert to possible errors in interpretation as you read about and discuss ejectors.

HEI [1] is the dominant standard in the United States. Stage 1 is the first stage into which process vapor flows, and stage numbers after that follow the process flow sequence. HEI recommends naming the stages alphabetically, so that the last stage is Z, the prior stage is Y, etc.

Precondensers (condensers ahead of the first stage) are assigned the letter P, aftercondensers (condensers after the last stage) are assigned the letter A, and intercondensers are assigned a two-letter label identifying the preceding and following stages. A drawing annotated this way is easy to refer to.

Written references are more complicated. A single-stage ejector with an aftercondenser would be named ZA. Multistaged systems are easier to visualize if there is a hyphen between each two components. Thus, a three-stage system with two intercondensers would be named X-XY-Y-YZ-Z. If a precondenser and aftercondenser were present, it would become P-X-XY-Y-YZ-Z-A. A six-stage system with two intercondensers and an aftercondenser would be U-V-W-X-XY-Y-YZ-Z-A.

Some manufacturers use other naming conventions, which may have changed over the years. Graham began by using A, B, and C for three-stage systems. As technology extended to lower absolute pressures, they added AA stages, then AAA, etc. Schutte & Koerting picked another logical, but different convention. Single-stage ejectors were given the name S. The first stage of a two-stage system was T, and the first of a three-stage system was Th. This extended to lower pressures as F, Fv, etc. Other manufacturers simply number the stages 1, 2, 3 in the order of process flow.

A short description of the configuration is often useful, even if it may be ambiguous. An example is 42c or 4-2c, which indicates four stages and two condensers. Usually the condensers would be assumed to follow stages 2 and 3. However, it might also be interpreted as one intercondenser following stage 2 and an aftercondenser. Another uncertainty is that the condenser types are not yet identified and may be both surface and direct contact in a system.

I recommend the Hicks Hargreaves practice of using J (jet) for contact condensers and V (vertical) or H (horizontal) for surface condensers. This is an unambiguous naming convention which avoids conflict with the above systems and with words such as Contact/Condenser and Surface/Spray. Then you may use any of the stage-naming conventions and let the pre- and aftercondensers be identified functionally by their position in the system. Thus, a three-stage system with a vertical surface pre condenser and horizontal first intercondenser with contact final intercondenser and aftercondenser could be described as V-X-H-Y-J-Z-J, V-A-H-B-J-C-J, V-1-H-2-J-3-J, or V-Th-H-T-J-S-J. There remains a tiny ambiguity between the V stage and the vertical condenser, but that should not be a problem.

My personal choice of conventions depends upon the circumstances. In the field I tend to use the 1, 2, 3 convention for stage naming unless the people I am working with show another preference. Only in written communications is ambiguity likely to be a concern.

Fun, huh!?

2.15 Evacuation/Steady Load, Single-Point vs. Extended Curve

Although ejectors are often specified and designed to handle one specific combination of load and suction pressure, that is only a simplified representation of the overall duty they must perform. First, the ejector usually has to start up with the process system at atmospheric pressure. It pulls gases and vapors out of the process system and after a time it reaches the desired system pressure, assuming no problem is present. Typically, the load coming out of the process system is less than the design load and has a different composition and temperature. Often the operator desires to operate the system at a pressure above the design pressure. During any field performance tests, a test technician wants to measure the stage suction pressures when the ejector is handling zero load.

For all the above reasons, there is more to ejector performance than the design load and suction pressure. Here again, I am merely introducing some ideas to condition your thinking for the details which will come in later chapters.

2.16 Common Questions and Answers about Ejectors

Q. Aren't ejectors basically constant mass-flow devices, whereas mechanical pumps are basically constant volumetric flow devices?

A. Yes.

Q. Aren't zero load tests tough to run because ejectors backfire at zero load?

A. Ejector stages may be specified and designed to be stable at zero load by using a little more steam. Then a backfiring stage indicates a problem.

Q Is there an art to "tuning" an ejector system to get it working best?

A. A properly designed and installed ejector system can usually be turned on and off without fancy adjustments. Proper instrumentation and clear operating procedures will enable any competent operator to run one. A "sick" ejector should be diagnosed and cured. "Tuning" an ejector system in cool weather to save steam and/or cooling water may create trouble in warm weather or if the air leakage increases. See Chap. 8.

Q. Must the last stage be discharged directly to air because ejectors don't work against much back pressure?

A. Ejectors may be easily designed to work against any reasonable discharge pressure, or modified later to do so. Unless asked to do otherwise, the manufacturer will design the last stage to discharge against 0.5 or 1.0 psig. If such a default design is connected to a long discharge line, submerged deep in a hotwell, or piped to a remote flare stack or scrubber, then it may not work at all.

Q. Can I use the next-to-last stage from one ejector system as a replacement for the next-to-last stage in another system?

A. It will seldom work out right. Most process ejector systems are custom designed for the specific application, and the combination of load, motive steam pressure, and suction and discharge pressures seldom work out right. Often a last stage may be substituted if the design steam pressures match, but the result is typically an oversized last stage or a reduced capacity system.

2.17 Important Phases in the Life of an Ejector

Ejector systems can be made to operate effectively throughout their life if proper care is taken by the people who work on them. They can also be defeated by uninformed treatment in many ways. Here are some of the activities I regard as important, described as an existing state of excellence.

In the process design phase, realistic estimates are made of cost, performance, and feasibility. During the procurement phase, the written specifications are complete and accurate, bids are evaluated realistically, and constructive decisions are made about shop tests and witnessing them.

In the detailed design phase, the installation permits the ejector system to operate as intended, with convenience, safety, and maintainability provided for the people who will work with the ejector.

Any vacuum pressure control system is integrated with the ejector design, and all reasonable operating conditions are anticipated. Construction, inspection, and startup procedures assure that the installed system does operate as intended.

Operation, testing, and maintenance people are coordinated and informed so that long-term reliability is achieved, with no serious failures that may be anticipated by modest monitoring and inspection. Troubleshooting is done by trained people in accessible work areas, using quality instruments, and informed by good records.

Modifications and repairs are done in an informed manner to preserve the performance and capacity of the ejector system. Records are sufficiently complete that the plant operation improves with time.

2.18 General References to Ejector Systems

Concise overviews of steam jet ejectors are available from manufacturers of ejectors [3, 6, 7, 8, 9]. Each approaches the subject in a different manner, so that you may find some match your individual needs better than others. If you plan to spend much time with ejectors, you may wish to read several of them.

Contained in some of the above references and in other manufacturers' literature are guidelines for installing, operating, troubleshooting, and maintaining ejectors. I have borrowed ideas from all of them for this book. I found one small booklet to be especially rich in suggestions for troubleshooting an existing ejector system [10].

2.19 References

1. *Standards for Steam Jet Vacuum Systems*, 4th ed, Heat Exchange Institute, Cleveland, OH: 1988.
2. Penberthy Bulletin 1100, Section 1000, Penberthy Div. of Stanwich Industries, Inc., May 1987.
3 "Ejectors and Their Applications," AMETEK, Schutte & Koerting Division, 1981.
4 "Vacuum Technology, Its Foundations, Formulae and Tables," PN 99.800.004, Leybold-Heraeus Vacuum Products, Inc., p. 34.
5 J. L. Ryans and Daniel L. Roper, *Process Vacuum System Design & Operation,* McGraw-Hill, New York, 1986.
6 "Jet Ejectors," Bulletin 70-B, Graham Manufacturing Co.
7 "Steam Jet Ejectors," Bulletin 5E-H, AMETEK, Schutte & Koerting Division.
8 "Vacuum Systems," Bulletin E68A, Croll-Reynolds Co., Inc.
9 "Vacuum / Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd, England.
10 Victor V. Fondrk, "How to Keep Your Ejectors Up to Snuff," distributed by Unique Systems, Inc.

11 F. Duncan Berkeley, "Ejectors Have a Wide Range of Uses," *Petroleum Refiner,* 37: 95-100, December 1958.

Answer to Problem Number 5

The unusually low pressure achieved by the last stage invites attention. As treated briefly on pages 99 and 176, the zero-load suction pressure varies directly with the motive steam pressure. Thus, the stage was acting as though its motive steam pressure was 50 psia instead of the 200 psia in the supply header. Disassembly of the steam line to that stage revealed a sintered metal element instead of the perforate basket specified in the strainer for that stage.

The discharge line from the last stage was sealed in the hotwell. A water spray in the discharge line washed down enough air to maintain a very low discharge pressure near zero-load (about 200 torr!)

Problem Number 6

A test was performed on an ejector with the design air load. The suction pressure was 24.7 inches Hg vacuum, and the barometric pressure was 14.5 psia at an elevation 500 feet above the test site. Convert the suction pressure to Torr.

Answer on page 230

Part

2

Steam Jet Air Ejector Performance

Chapter

3

How Ejector Stages Behave

Knowing how a typical ejector stage behaves in response to changes in load, discharge pressure, and steam pressure will improve your ability to understand and remember many of the explanations and recommendations in this book. A performance curve is a graph that shows how the stage suction pressure changes as the flow of load gases changes at the suction. This chapter describes how ejector-stage performance curves are created and how the stage curves may be used to predict the behavior of single-stage and multistage ejector systems.

It is well worth your time to read and reread this description. Become familiar with this simplified model of stage behavior, then verify it by testing an ejector to fix the ideas in your mind. Later, you will see that some stage behavior may differ from this simplified model under special conditions. See Chap. 2 for definitions of technical terms and stage hardware nomenclature.

3.1 Testing an Ejector Stage

Imagine yourself participating in a performance test of a single ejector stage, typically designed for a discharge pressure more than twice the suction pressure (compression ratio is greater than 2.0). For example, if the design suction pressure is 10 torr, the design discharge pressure will be greater than 20 torr. Most multistage steam jet stages are designed for compression ratios greater than 2.

The test objective is to collect data from which a useful set of performance curves may be created. The performance curves consist of a suction curve and a discharge curve. The suction curve will show the relationship between suction pressure and suction load over the operating range. The discharge curve will show the maximum discharge

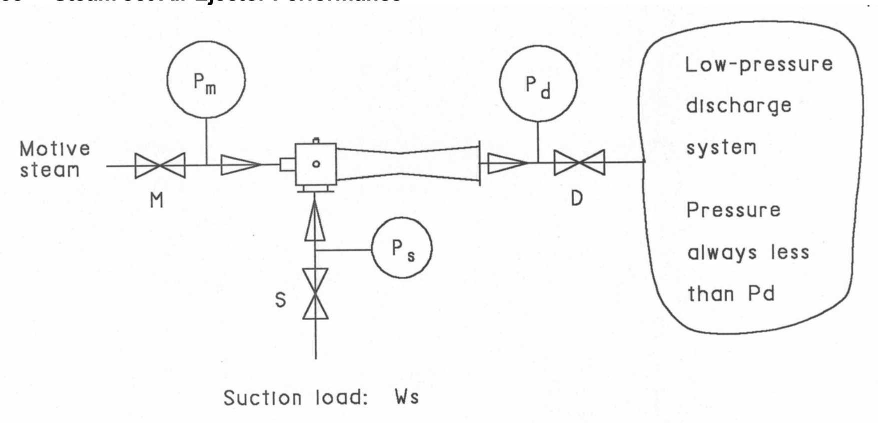


Figure 3.1 Test setup for an ejector stage.

pressure that the stage can operate against without disrupting the suction curve. With a set of performance curves like this, we can assemble a multistage ejector from a set of appropriate individual stages.

Figure 3.1 shows a hardware setup for such a performance test of an ejector stage. Note the similar arrangements of pressure gauges and flow control valves at the suction, motive, and discharge connections. To keep this simple, the suction pressure is shown as being measured in the suction line. In practice it is usually measured at a pressure tap located for that purpose in the suction chamber.

Normally the test data are recorded on tabular data sheets from which the performance curves are later drawn. Example data sheets are shown in Chap. 8 and in the Appendix. For simplicity here, let us record the test results directly on the curves (see Fig. 3.2*a* and *b*). If you have read ejector performance curves before, you will probably be more familiar with the rotated-scale convention used in Fig. 3.2*b*. Although that convention permits recording the discharge pressure data with greater precision, I have not found the extra precision to be very useful. I prefer the vertical pressure scale convention in Fig. 3.2*a* and will use it here. It is quite useful for describing how several stages combine to form a multistage ejector. I recommend it to you.

Let the test begin! Open valve D completely and make sure that the line is open to the low-pressure discharge system. If the stage is designed to discharge below atmospheric pressure, the low pressure is usually maintained by a vacuum pump or large ejector. Open the motive steam valve M until the pressure Pm equals the design pressure. Use 100 psia as an example and assume that the steam is dry and saturated. Because the pressure inside the ejector is less than half the pressure of the steam, the flow of steam will not be affected by changes in the suction pressure Ps or the discharge pressure Pd. The ejector stage is now in its "normal" stable operating condition.

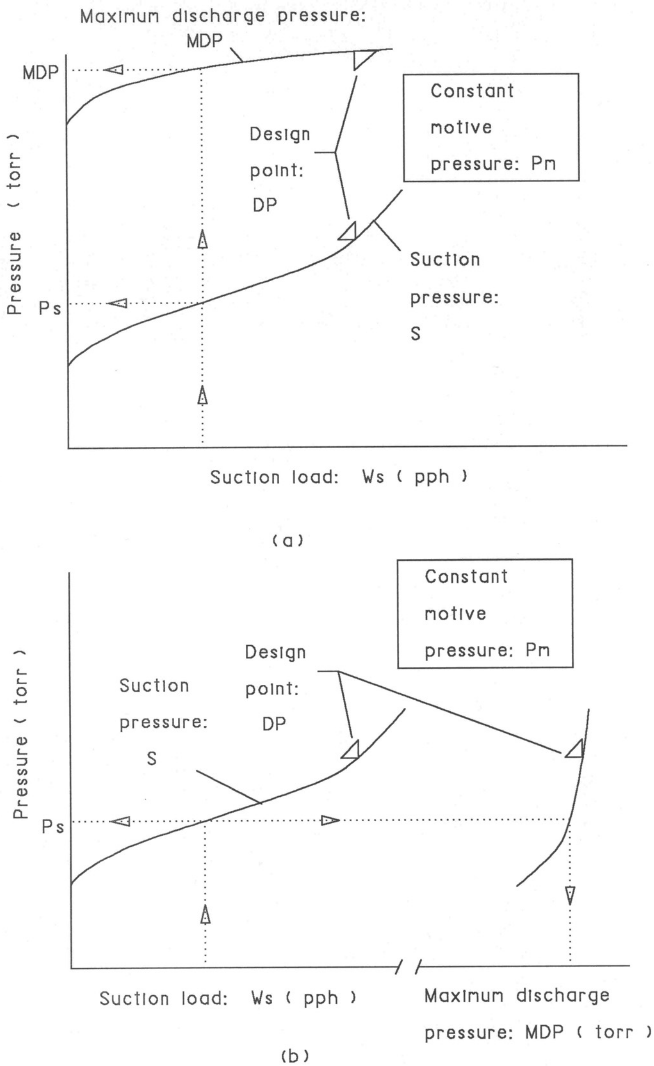


Figure 3.2 Single-stage performance curve from test: (*a*) vertical pressure scales; (*b*) rotated pressure scale.

Suction pressure curve

Now, find how the suction pressure P_s varies with the suction load W_s. We are not yet going to be concerned about the discharge pressure. Figure 3.1 shows valve S as the means for controlling the suction load W_s. It is necessary to measure the suction load W_s in some manner. One such method is briefly mentioned here.

A simple method of measuring W_s is to connect the ejector suction line to a chamber that has one or more flow orifices which may be opened to permit atmospheric air to enter the vacuum system. Some details of such an arrangement are described in Chap. 8. If the pressure inside the chamber is less than half the atmospheric pressure, then the flow through each such orifice will not be affected by changes in the suction pressure. The orifices may be plugged or covered with caps to activate only the combination that is desired. For example, if we wish to create a suction load W_s of 73 pph, we might open a combination of 50-, 20-, and 3-pph orifices. In Chap. 4 and the Appendix we will see how the orifices are sized. In Chap. 8 we will see how a commercially available "flute" or "piccolo" is quite convenient for such testing.

The pressure-indicating device is shown only schematically here. It might be a mercury manometer, an electronic device, or some other type of instrument. We observe the pressure P_s that corresponds to the suction load W_s and plot one point on the plot in Fig. 3.2*a*. Repeating the procedure with several loads including zero load and connecting the plotted points with a smooth curve yields the suction curve S. This "stable operation" curve is a very basic property of this ejector stage. It will not change if we lower the discharge pressure P_d and it is not very sensitive to modest changes in motive steam pressure P_m.

Maximum discharge pressure curve

The next curve we will create is the maximum discharge pressure, MDP, curve. Its meaning is best understood by going through the test procedure. We still keep the steam pressure at the original value, 100 psia here. Establish a suction load, say 73 pph, and verify that the original value of P_s is repeated. Now slowly close the discharge valve D and note that although P_d begins to increase, P_s remains constant. At some value of P_d it will be found that P_s has increased, indicating that we have exceeded the MDP somewhat and "broken" the operation to an "unstable" operation described below.

Note that the terms "broken" and "unstable" are used interchangeably here and refer to a fluid flow condition that is off-design, but not necessarily creating undesirable effects.

When an ejector begins to operate in a broken mode, usually we will also hear a characteristic change in the sound of the ejector to a rough "hiss." Then we slowly open D until P_s returns to its basic value ("recovers") and the ejector sounds normal. Plot that discharge pressure P_d point on Fig. 3.2*a*. Repeat with various loads, connect the points with a smooth curve, and you have the MDP curve.

The significance of the MDP curve is that if the ejector stage "sees" a discharge pressure that is at or below this curve, then the suction pressure depends only on the suction load W_s. Conversely, if the discharge pressure exceeds the MDP by a small amount, then the stage will be broken (unstable) and the suction pressure will be above that of the suction curve. It is difficult to predict the suction pressure in a broken (unstable) condition because it now depends in a complicated way upon the suction load, the discharge pressure, and the motive steam pressure.

Effect of changing motive steam pressure

This effect is easiest to understand by considering an atmospheric stage, one designed to discharge to the atmosphere. In Fig. 3.3*a*, note that this ejector has a MDP curve that "droops" at loads less than 10 pph to a pressure below atmospheric pressure. Thus, the ejector will operate in a broken state when the load is less than 10 pph. The broken state is indicated by the dashed lines, which might be reproducible during a test, but which are not easily predicted with the information available to users.

If we raise the steam pressure to 105 psia, we will see very little change in the suction curve S, but the MDP curve will be uniformly raised, so that this stage will now be stable down to zero load. On the other hand, if we lower the steam pressure to 95 psia, we see that the MDP curve is below that for 100 psia. The result is that the stage is broken below a load of 20 pph.

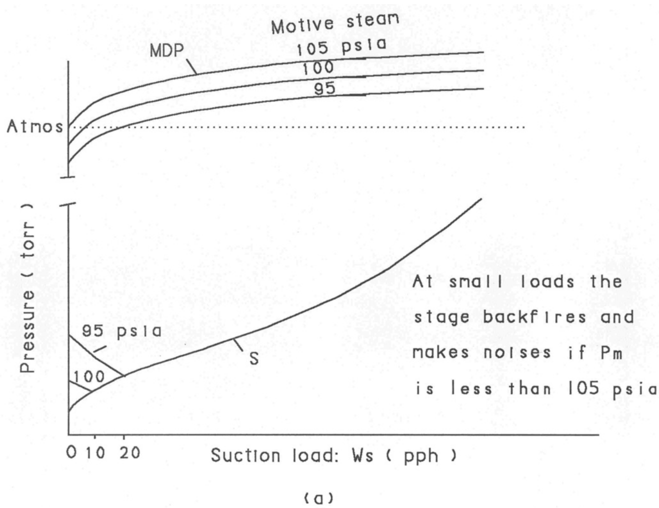
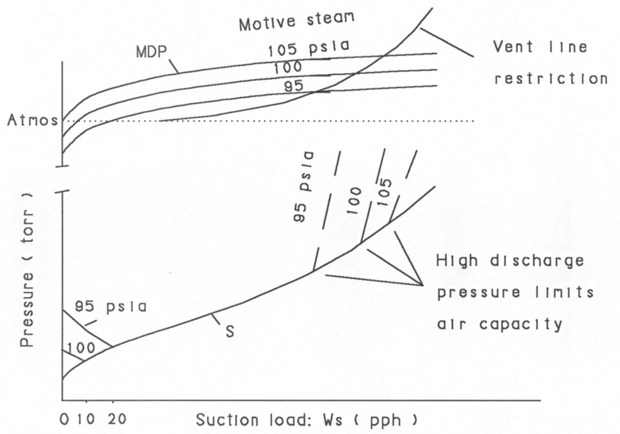


Figure 3.3 Effects of changing steam pressure and discharge pressure: (*a*)motive steam pressure.



(b)

Figure 3.3 (Continued) Effects of changing steam pressure and discharge pressure: (*b*) discharge pressure.

Unstable operation characteristics

The "stable" condition is sometimes defined as the absence of violent fluctuations in the suction pressure. More specifically, it is the condition in which the discharge pressure for a critical stage is less than the MDP associated with the load, thus allowing the suction pressure to be determined only by the load. When the MDP is exceeded, the condition is defined as unstable (broken), and the suction pressure now depends not only upon the load but also upon the discharge pressure and the motive steam pressure.

Let's look at how the stage behaves when unstable (broken). If we slowly increase the severity of operation of a stage by increasing the discharge pressure, lowering the steam pressure, or decreasing the load, we will detect the broken condition by a rise in suction pressure above the stable curve. Typically, we will also hear a hissing or popping sound and may notice that the suction chamber and suction piping become too hot to touch. The noise effects are more noticeable at low loads. The hot spots are described in Chap. 8.

In order to "recover" to stable operation, it is necessary to return to a condition less severe than when the stage became broken. Thus, at a given load and discharge pressure, we might find that the stage breaks at a steam pressure of 100 psia and recovers at a pressure of 105 psia. Or, we might find that at a given load and steam pressure, the stage breaks at a discharge pressure of 790 torr and recovers at a pressure of 770 torr.

When the load is small and the ejector breaks, a flow reversal occurs. Load gas and motive steam flow backward into the suction piping. This is called backfiring and explains why the system becomes hot to the touch. The backward flow raises the pressure in the suction system until it is high enough to resume stable operation. Then the forward flow begins, and abruptly the stage again becomes stable. If the load flow remains too small, the ejector will continue to cycle, pulling down to the lowest pressure it can maintain, breaking with backflow until the pressure rises, and resuming stable operation briefly. Soon we will see how this condition may be corrected.

Depending on the volume and geometry of the suction system and the load and characteristics of the ejector stage, the cycling may occur over a period of many minutes or it may occur many times per second. Anyone who has watched the pulsations of an unstable atmospheric stage fill up a mercury manometer with water will not forget what backfiring means.

Effects of high discharge pressure

Examine Fig. 3.3*b* to see how the stage behaves when it sees a system discharge pressure curve that may represent a restricted vent line for an atmospheric stage. When the load increases, the pressure drop in the vent line increases until the discharge pressure exceeds the MDP curve for the motive steam pressure. The dashed lines indicate the regions of broken operation, showing that increasing the steam pressure does increase the stable operation capacity a little.

3.2 Some Last Stages Are Unstable at Low Loads

Some atmospheric stages are deliberately designed for lowest possible steam usage at the expense of low-load stability. This is usually the last stage in a multistage condensing ejector. In a properly designed system, the "lap" between the last-stage suction curve and the preceding-stage MDP curve is large enough that a modest "bobble" in the suction pressure will not upset the operation of the preceding stage. It does complicate testing and troubleshooting.

3.3 Making a Stage Intrinsically Unstable

It is possible to degrade the performance of an ejector stage so severely that it is unstable at low loads regardless of the discharge pressure [1]. This severely off-design condition shown in Fig. 3.4 results from pulling the steam nozzle too far back. This condition does not represent any properly designed ejector, and there is no reason to duplicate

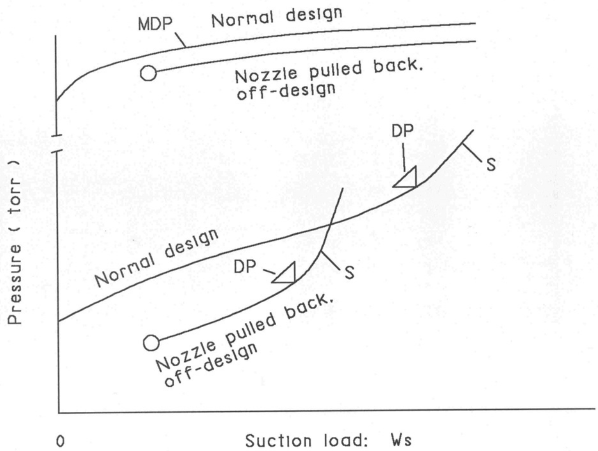


Figure 3.4 Inherently unstable stage, steam nozzle pulled back to severely off-design position.

the experiment. I mention it here only to correct an erroneous interpretation on my part that I propagated in my previous published articles [2, 3]. If I misled you earlier, I apologize. If you didn't see the articles, no harm done.

3.4 Low-Compression-Ratio Stages

An ejector designed for a compression ratio less than 2.0 has no single stable suction curve and corresponding MDP curve, and thus no broken mode of operation. Its suction pressure depends upon the load, the motive steam pressure, and the current discharge pressure. A set of curves is required to show the many possible combinations of suction pressure, discharge pressure, motive steam pressure, and load. More on this in Chaps. 6 and 11.

3.5 Multistage Ejectors

When the compression task can't be done by one stage, or when the potential steam savings justifies the extra cost of additional stages, then two or more stages may be connected in series to perform the overall task. Let us first look at how a two-stage ejector may be designed, then look at multistage ejectors in general.

Two-stage noncondensing

The simplest multistage ejector is one in which the load gases and motive steam emerging from the first stage pass directly into the sec

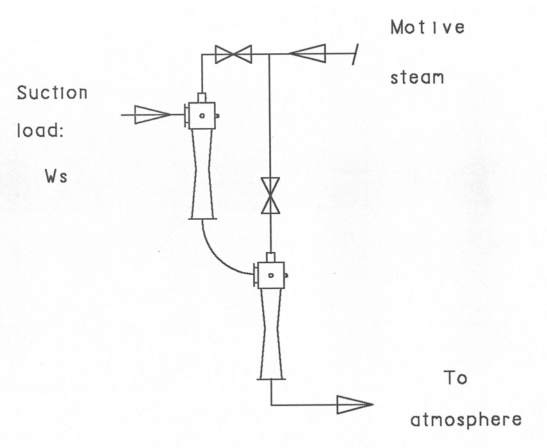


Figure 3.5 Two-stage noncondensing configuration.

ond stage, which compresses the mixture and discharges it to the atmosphere. Figure 3.5 shows the arrangement of such an ejector.

As an example, let the load to the first stage be 20 pph dry air at 30 torr and 70°F, the steam pressure be 150 psig, and the design discharge be 813 torr. Let the design interstage pressure be 200 torr. The first stage will require about 50 pph motive steam. The hot mixture of air and steam from the first stage will become the load entering the second stage. The second stage will compress the mixture from 200 torr to 813 torr, using about 225 pph motive steam. The total steam usage is 275 pph. Selecting another interstage pressure may change the total steam usage.

The performance curves for both stages are shown in Fig. 3.6*a*. Examine the curves and note where the above specifications appear in the data displayed. Note also that because the mixture from the discharge of stage 1 contains hot steam, the 70 pph mixture is actually equivalent to about 90 pph of dry air (DAE). This concept and the conversion calculation methods are explained in Chap. 4.

This design has a first stage that is stable down to zero load. The second stage is unstable at loads below about 50 pph, but relies on motive steam from the first stage to keep it on the stable portion of its operating curve. If the user prefers that the last stage be stable by itself at zero load, then an increase in the steam usage and minor hardware changes will result.

Logarithmic scales for performance curves

Note that Fig. 3.6*a* becomes crowded near the origin and that the plotting precision is reduced for the first stage because the single

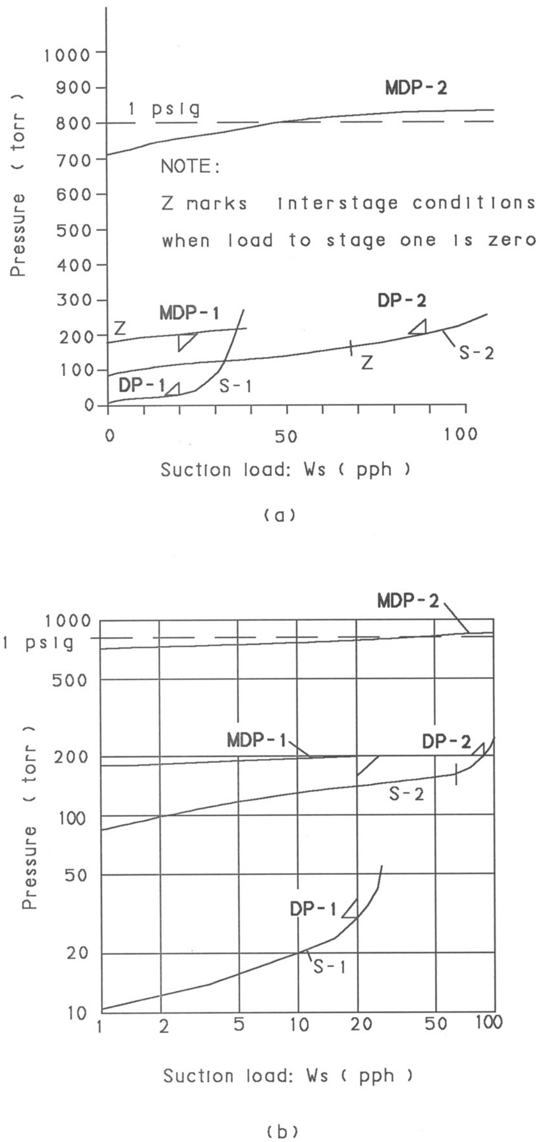


Figure 3.6 Two-stage noncondensing curves: (*a*) linear scales; (*b*) logarithmic scales.

pressure scale is linear. A linear scale becomes useless for more than two stages, explaining the common practice of preparing separate performance curve sheets for each stage.

Now examine how the same data are displayed on the logarithmic scale in Fig. 3.6*b*. A useful property of a logarithmic scale is that equal distances on the scale connect numbers having a constant ratio.

Thus, we find the distance between 2 and 10 to be the same as the distance between 50 and 250. Each stage occupies about the same vertical space on the logarithmic pressure scale because they all have about the same compression ratio. All points are plotted with the same precision, about 1 to 5 percent for common scales. That is usually quite adequate for most users.

Thus the logarithmic pressure scale permits us to display several stages, with equal space and importance assigned to each stage. It also avoids the deceptive appearance of getting "close to zero" in pressure. The unlimited vertical scale thus encourages thinking about pressure in a manner that better matches engineering theory about gas compression.

A problem arises with a logarithmic load scale when we attempt to plot the zero load pressure from a test, because there is no zero on the scale! A simple solution is to observe that for most commercial ejector applications, 1 pph is adequately close to zero load. In fact, true zero load does not exist any more than true zero pressure does. Some leakage exists through closed valves, plugged test connections, and around slip blank gaskets. To avoid possible confusion about zero load pressures, I have sometimes modified the logarithmic load scale as follows: Change the 1 to zero, place a 1 near the 1.5 location, and split the scale between 1.5 and 2.

Why so much emphasis on unified logarithmic scale curves? Seeing the curves of all ejector stages on one simple chart greatly improves a user's ability to understand how the ejector works and allows the user to make better decisions in stressful design and troubleshooting situations. Lack of this understanding is the biggest problem faced by most ejector users.

Although this plotting convention departs from many years of convention by ejector manufacturers, the effort needed to replot their data in this form is trivial for them and well worth any modest extra cost to the user. The user works with one sheet rather than several, avoids the awkwardness of interpreting the rotated scale for MDP, and uses torr for all pressures instead of a mixture of psig, inches of mercury vacuum, inches of mercury absolute, and torr. A blank graph for this purpose is included in the Appendix.

I value highly having all the stage performance curves together on one sheet. When working with old curves on existing ejectors, I often replot the manufacturer's curve data on one logarithmic plot for my convenience. Plotting the data yourself gives you a better understanding of the data than if you simply refer to several curves. Although a full-time ejector specialist employed by an ejector manufacturer can do good field work with scant data, a user is typically struggling to understand the situation and needs all the help he or she can get.

For several reasons, ejector troubleshooting is stressful, and anything that reduces mental effort is welcomed by a person fatigued from walking up and down the stairs that lead to the ejectors on the third to fifth levels in a plant structure exposed to the weather. I have worked on ejector problems in all kinds of weather, including high winds, rain, and blizzards at 4 a.m., and it is frustrating trying to keep several paper sheets from getting wet or blown away. In those circumstances the benefit of working with one simple sheet versus several complicated ones is priceless.

In training and consulting situations, I have found the one-sheet performance curve format to be an effective learning and communicating tool. Imagine the convenience I enjoyed in working with a client on two five-stage ejectors, each of which had a second element ahead of the first condenser. Two sheets instead of twelve!

Two-stage condensing

A condenser between ejector stages will reduce the load to the next stage by condensing out part of the load if the process conditions permit it. The pressure must be high enough and the cooling water cool enough that adequate condensation will occur. Condensers will be discussed in detail in Chap. 5.

Consider the previous example ejector specification: 20 pph dry air load at 30 torr and 70°F, with 150 psig steam and a discharge pressure of 813 torr. Let's see what happens when we place a condenser between the stages.

Assume the first stage still compresses the load from 30 to 200 torr, using 50 pph steam. Use a direct contact condenser in which the cooling water sprays onto the hot vapor mixture from the first stage to condense out most of the steam. The cooled vapor load will emerge from the condenser at a lower pressure, say 190 torr. If the load going to the second stage is cooled to 100°F, then the 20 pph air load will have about 5 pph water vapor with it. The second stage will use about 100 pph steam. This two-stage ejector uses a total of 150 pph steam and perhaps 10 gpm cooling water. Selecting a different interstage pressure will alter the utilities usage and may result in lower total steam and water usage.

The installed cost is greater than for the noncondensing design. The condenser and cooling water supply were added, the unit will typically be located in an elevated portion of the structure, and a drain pipe and hotwell must be installed for gravity to remove the mixture of warm cooling water and condensate from the condenser drain connection.

The configuration sketch and performance curves for this ejector system are shown in Fig. 3.7*a* and *b*. Here the second stage is designed to be stable at zero load because the designer cannot rely on

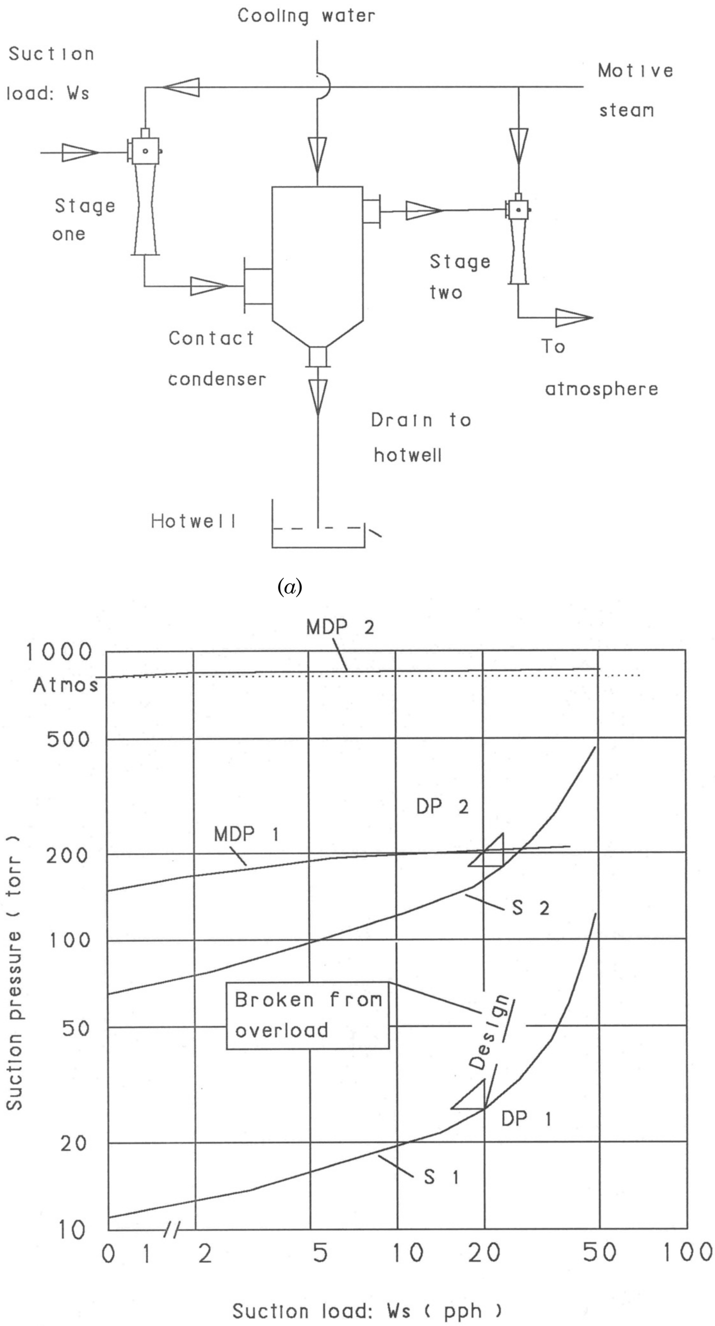


Figure 3.7 Two-stage condensing ejector: (*a*) configuration; (*b*). curves.

a fixed minimum load from the first stage. Also, the system has become sufficiently more complicated that zero-load stability of the last stage is desirable for troubleshooting convenience.

Two-stage condensing with off-design last stage

Because the last stage is usually the smallest and because it must operate against the ever-present atmospheric pressure, it is typically the first part to fail in a multistage ejector system. Sometimes a resourceful person will find a used or new atmospheric stage, install it and conduct a successful zero load test, then pronounce the problem solved. As Fig. 3.8 shows us, that is not the complete story. The performance curves for the previous two-stage condensing ejector are drawn as before, but see what happens with a half-size last stage! Although the ejector is stable down to zero load, the useful air-compressing capacity has been reduced from 20 pph to about 10 pph. That capacity might be acceptable for system operation, but the user should be aware of the limitation. Also, the investment and operating cost of the first stage and condenser are only half utilized because of the limited capacity of the last stage.

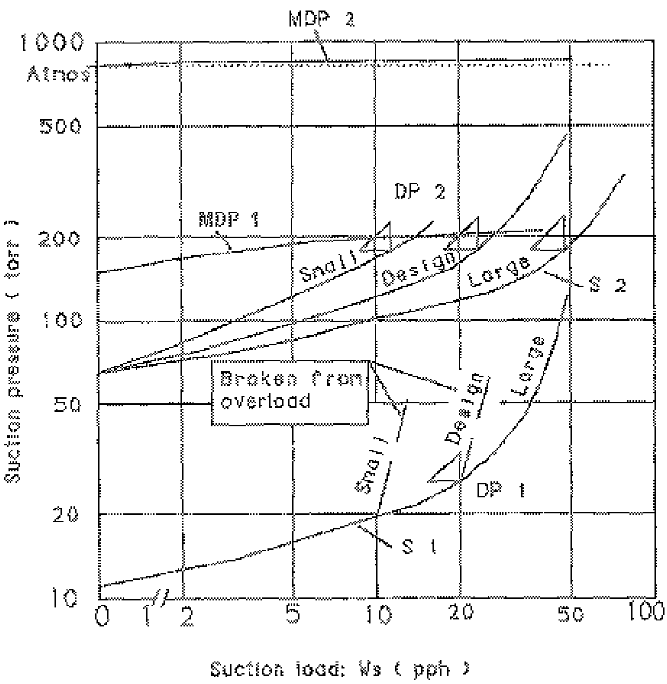


Figure 3.8 Two-stage condensing ejector curves with undersized and oversized second stages.

Now suppose the replacement atmospheric stage had twice the capacity of the failed stage. The ejector would pass the zero-load test, but its air-compression capacity at the design pressure would still be only 20 pph. The stable range would be extended beyond 20 pph, but the first-stage curve rises so sharply that the extra air capacity occurs only at high suction pressures.

This simple example illustrates that although ejector systems can often be kept working by cannibalizing off-design parts, the practice will result in underutilized investment and operating costs. The useful performance of a multistage ejector system is limited by the smallest-capacity stage in the series. Because operating costs for the steam and cooling water are typically much more important to the plant than the installed cost of the hardware, it is false economy to avoid buying and installing the proper replacement parts.

Four-stage condensing

Figure 3.9*a* and *b* shows the configuration and performance curves for a four-stage ejector with two intercondensers of the direct contact type. Steam and cooling water piping are omitted from the sketch for

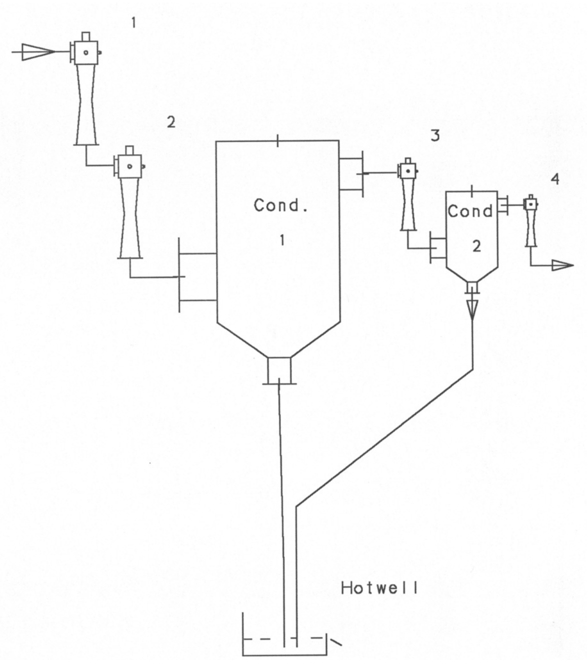


Figure 3.9 Four-stage condensing ejector: (a) configuration.

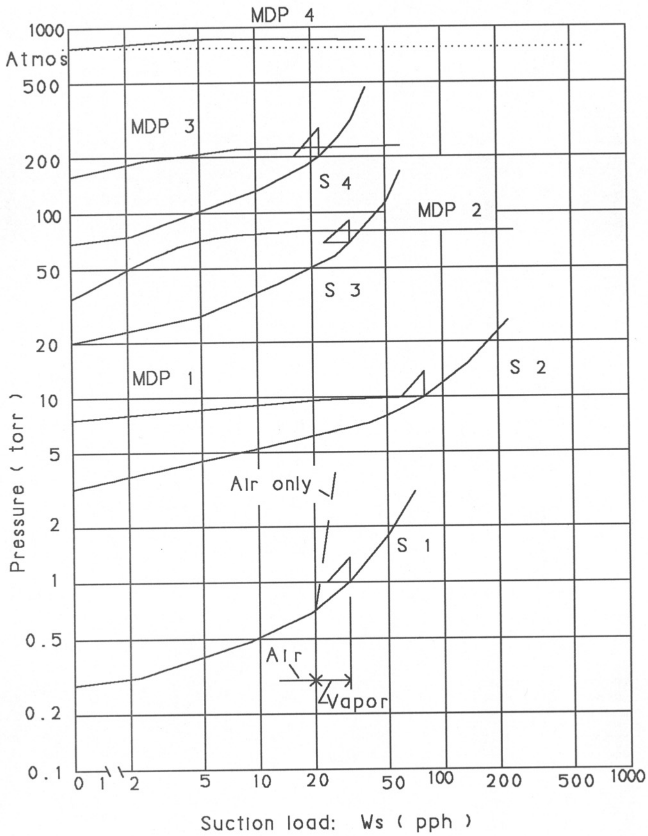


Figure 3.9 (Continued) Four-stage condensing ejector: (b) curves.

simplicity. The system is designed to handle 20 pph air plus 10 pph of condensable vapor at a pressure of 1 torr.

Note that there is no condenser between the first two stages because the pressure there is too low to permit the water vapor or condensable vapor to condense at temperatures attainable with normal cooling water. Thus, the first two stages form a noncondensing series sometimes called "booster" stages. Booster stages are those which have suction pressures below a practical condensing pressure. They may be visualized as boosting the vapors up to the condenser pressure. A second condenser is placed between the third and fourth stages.

Examining the performance curves, we observe the design load to the first stage as being about 30 pph at 1 torr. The pressure between stages 1 and 2 is 10 torr, and stage 2 is seen to have a limited stable operating range, relying upon the motive steam from stage 1 to keep it on the stable portion of its suction curve.

The first intercondenser pressure is somewhere between 50 and 100 torr, depending upon the cooling water temperature, the type of condenser, the properties and quantity of the condensable vapor, the suction pressure, the steam pressure, and the economic factors present. In this application, the condensable vapor dropped out in the first condenser, and the third stage sees only the air load plus about 10 lb of water vapor per pound of air. In the second condenser, almost all the water vapor is removed, so the last stage sees only the air load plus about 2 pph water vapor.

All stages except stage 2 are designed to be stable at zero load, making it easy to test this ejector. Stage 2 will also be stable with the first stage turned on. However, if stage 1 is turned off, stage 2 may not be stable at zero load if the first intercondenser is too warm. The third stage no-load suction pressure is about 20 torr, apparently providing a comfortable "lap." However, if the cooling water is warm, the water vapor pressure at the third stage may be above 30 torr, loading up the third stage and causing the second stage to become unstable near zero load. Thus, saving steam by turning off the first stage in a booster series may create unstable operation and backfiring at low loads. The behavior of a stage operating in an unstable (broken) mode, including the backfiring, is described in detail in Chap. 6.

3.6 Unstable First Stage in a Series of Booster Stages

Occasionally, the first stage of a series of booster stages may not be stable to shutoff if there is too small a lap between its design-point discharge pressure and the suction pressure of the next stage, perhaps as a result of designing for lowest possible steam usage. Figure 3.10 shows performance curves for such a condition. This is generally undesirable unless a user is willing to give up the convenience of testing at shutoff and permit backfiring of steam into the process system in exchange for reduced operating cost.

3.7 Summary

In this chapter you have learned about the characteristics of a typical steam jet ejector stage: how a performance test is conducted, the significance of the suction and maximum discharge pressure curves, stable and broken operation, and the effects resulting from changes in the motive steam pressure and discharge pressure. You have learned two different ways of plotting the curves for a stage. Multistage ejectors are a series of stages: noncondensing, condensing, or a combination. You have been encouraged to use the single-sheet method of

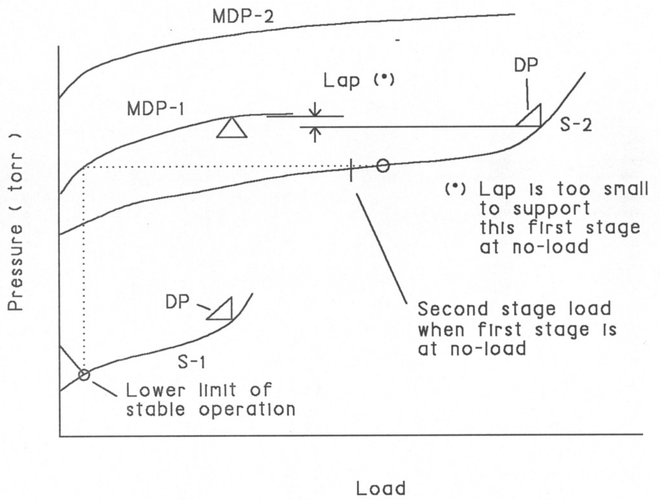


Figure 3.10 First stage unstable at no-load because booster second stage is underdesigned.

plotting performance curves on logarithmic scales for multistage ejectors and have seen the error of using cannibalized parts as a longterm replacement for the proper replacement parts. The subjects of booster stages and the factors affecting the selection of condenser pressures have been introduced.

3.8 Nomenclature

D	Discharge flow valve
DP	Design point for a stage
M	Motive steam valve
MDP	Maximum discharge pressure curve
P	Pressure
pph	mass flow units, pounds per hour
torr	absolute pressure unit, 1 mmHg
	760 torr = standard atmospheric pressure
S	Suction flow valve, suction pressure curve
W	Mass flow, pph

Subscripts and Suffixes

d	discharge
m	motive
s	suction
-1,-2,-3,-4	stages 1 through 4

3.9 References

1. L. S. Harris and A. S. Fischer, "Characteristics of the Steam-Jet Vacuum Pump," ASME Paper 63-WA-132, 1963.
2. R. B. Power, "Steam-Jet Air Ejectors: Specification, Evaluation, and Operation," ASME Paper 63-WA-143, 1963. Reprinted by *Hydrocarbon Processing and Petroleum Refiner,* 43 (2): 121ff and 43 (3): 138ff, 1964.
3. R. B. Power, "Steam-Jet Air Ejectors," 10-part series, *Oil and Gas Equipment,* October 1965-July 1966.

POP QUIZ answers are intentionally brief, using shortcut methods and minimum annotations. This requires the reader to identify the basis for each equation, and the use of proper engineering units.

ANSWERS for problem 1, page xii

Steam nozzle flow: $W = 50\ D^2\ P^{0.96}$ [P72]

= 50 (0.375**2) ((75+ 15)**0.96)

= 529 pph (within 1 %)

or, base on 4152 pph of 100 psia D&S steam through a 1-inch nozzle, and flow proportional to pressure for modest changes in pressure. [P73]

W = 4152 [(75+ 15)/(100)] [(0.375)(1.0)]**2

= 525 pph

Air orifice sizing: 956 pph std air in a 1-inch orifice [P70]

D = (40/956)**0.5

= 0.205 inch

or, an easier approximation to remember:

10 pph through a 1/10 inch orifice (0.1")

from which,

D = 0.1 [40 /10]**0.5

= 0.20 inch

Problem Number 2

Your maintenance shop calls you with a problem. They removed the nozzles from a 2-stage ejector for cleaning, and mixed them up. Now they don't know which nozzle goes into which stage. There are no useful markings on the nozzles. What do you do?

Answer Page 350

Chapter 4

Stages:

Engineering Calculations

4.1 Overview of Ejector-Stage Calculations

I feel really good about writing this chapter. I would have given my eyeteeth to have these data when I first started to work with ejectors by making estimates, reviewing quotations, and troubleshooting ejectors in some operating plants.

This chapter shows how you can make many calculations and estimates related to ejector stages. I am assuming here that you have some engineering training or experience. I assume you have a basic familiarity with fluid statics and dynamics and thermodynamics. Without that background, you may have some difficulty reading this chapter. If so, I suggest you either read lightly for general information, or prepare for some extra study effort and get help from your engineering texts or your engineer friends.

In Chap. 3 I described the behavior of typical ejector stages having a compression ratio greater than 2:1. That was mostly a qualitative discussion. Now I will describe how to estimate the performance of an ejector at its design point. The design point is usually defined as a specific combination of pressures, temperatures, material, and flow rates that represents the maximum-efficiency point. Efficiency is lower at any other set of conditions. Usually the shape of the performance curve away from the design point is of secondary interest.

However, sometimes the shape of the performance curve is important, especially if the ejector is to handle two or more design conditions. Then the designer must give it special attention. Some "stock" design ejectors have published tables or curves of performance, which the buyer or the manufacturer may consult to select an ejector. Such a stock ejector will usually have a lower efficiency and a lower price than one custom-designed for the specific application.

4.2 A Step-by-Step Approach to Stage Calculations

Stage calculations can quickly become so complex that some basic relationships are obscured by the computational details. To avoid that, I am going to approach the subject in steps. The first step is to consider a liquid-jet ejector, in which the motive and load fluids are both incompressible. Further, I will begin by using simplifying assumptions which lead to simple equations that describe the important performance relationships. From that analysis we can review (or learn about) the basic behavior of nozzles and diffusers, and review relevant fluid dynamic concepts of kinetic energy, momentum, flow coefficients, and efficiencies. The results will show how the design-point performance is affected by the densities of the motive and load fluids. The equations derived here for liquid-jet ejectors will be used in portions of Chap. 11, where liquid-jet ejector applications are discussed.

The next step is to consider the ejector that handles compressible fluids. This brings in the subject of thermodynamics, and the possibility of easily getting lost in the complexity of calculations. Ideal gas laws are useful for predicting the performance of low-compression ratio ejectors (see gas-jet compressors/thermocompressors, Chap. 11), but have limited value in predicting the behavior of steam ejectors at low absolute pressures. The presence of moisture or ice as steam expands to low pressures leads to a departure from ideal gas behavior. Ideal gas laws do describe approximately the flow in nozzles and diffusers, as well as the general nature of supersonic flow and shock phenomena.

4.3 Selecting an Approach to Ejector Steam Usage Calculations

Before selecting a simplified strategy, I will briefly review some alternative models for analyzing ejector stage performance. Then I will describe a simplified approach that I am selecting for users who want approximate answers to common design situations. The approach uses a single page of curves to describe the steam usage of moderate-compression-ratio ejector stages using motive steam at 165 psia, suction pressures down to 0.3 torr, and discharge pressures to 1000 torr. For other steam pressures there are two steam usage multipliers. Finally, a size correction factor and an adjustment for optional stability at no load complete the estimate of steam usage of one ejector stage. A user who plans to make many such estimates at a steam pressure other than 165 psia may wish to prepare a similar set of curves for his or her steam pressure.

This set of design curves will match ejector manufacturers' data within about 20 percent, usually within 10 percent for steam pressures above 65 psia and compression ratios less than 10:1.

The user who is satisfied with using the simplified design curves will benefit by reading a little about the phenomena that occur in ejector stages. This explains some of the unusual characteristics of stages and some of the calculation methods. Also, the user will be less bewildered when he or she occasionally hears ejector manufacturers talk about various topics such as shock waves, compression shock, compressibility, supersaturation, nonequilibrium, overexpanded and underexpanded nozzles, etc.

4.4 Derivation and Application of Calculation Methods

Most equations and data used here may be found in any introductory text on fluid dynamics and thermodynamics. Other sources will be identified. The simplified equations are very useful for doing preliminary estimates, such as when you are away from your office. The more precise methods become tedious, and are seldom needed by a user to answer the most common questions. I have attempted to shortcut the procedures whenever the time saved does not introduce too much error. My general guideline is to regard 10 percent uncertainty as generally acceptable for estimates of size and steam usage. The relationship between nozzle flow and size is known much more precisely, usually within 2 or 3 percent. Other size estimates are offered for general reference and may be off by as much as 30 percent.

The differences in sizes and shapes of designs prepared by different manufacturers reflect their different approaches to designing ejectors, and become a practical limit to the accuracy of these estimating methods.

When you combine this with the condenser information in Chap. 5, you have a powerful tool. You can examine a manufacturer's quotation stage by stage to see whether it contains a gross error or is questionably "tight" in its estimate of steam usage. You can perform your own semioptimization to anticipate the interstage pressures and steam and cooling water rates that will be quoted by the most competitive bidders. As you will see in Chap. 11, you can use these data to estimate the performance of steam-jet refrigeration systems. Or, you can use similar methods to estimate the performance of gas-jet ejectors or thermocompressors.

If you are considering purchasing a preengineered single-stage or multiple-stage ejector maintained in stock for quick delivery or low price at the expense of lower efficiency, these estimating methods can give you a useful estimate of the performance penalty involved. You can estimate the performance attainable by a custom-designed unit designed for maximum efficiency at a single design point and see how much more expensive the stock unit will be to operate.

4.5 How a Liquid-Jet Liquid Ejector Works

Consider the ejector stage in Fig. 4.1. The numbered locations in the stage do not correspond to any standard practice; thus you must be careful if you compare drawings and equations in this book to drawings and equations in another reference. The ejector in Fig. 4.1 is designed for handling incompressible fluids, here water pumping water. Assume that the motive water is supplied at a pressure of 60 psig, that it mixes with an equal part by weight of water entering at the suction connection at atmospheric pressure, and that the two mix and emerge at the discharge connection at a higher pressure. What will that pressure be if the ejector is designed for this condition?

To simplify the analysis, assume there are no losses due to friction inside the ejector, and that the nozzle and diffuser both have an efficiency of 100 percent. The water will emerge from the nozzle at atmospheric pressure with a velocity of about 94 ft/s. Inside the mixing chamber it mixes with the suction water, still at atmospheric pressure, and the mixture has a uniform velocity of about 47 ft/s. As the mixture flows through the diffuser, it emerges from the diffuser discharge at low velocity and a pressure of about 15 psig.

Now, let us see how we reached that answer. For this I borrow from the excellent concise derivation by Jumpeter [1].

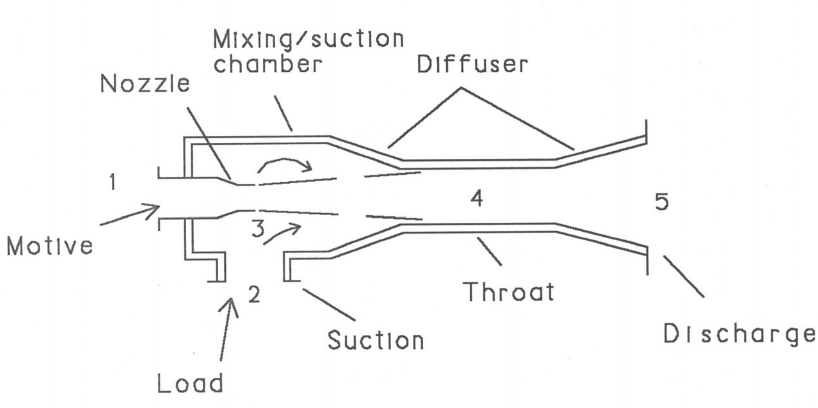


Figure 4.1 Ejector stage, incompressible fluids.

Nozzle

First, use the energy equation to see how the nozzle converts the pressure drop into velocity energy. The numbered subscripts refer to the numbered locations in Fig. 4.1.

$$\frac{P_1}{w_1}+\frac{V_1^2}{2g_c}=\frac{P_3}{w_3}+\frac{V_3^2}{2g_c} \tag{4.1}$$

To simplify this analysis, we use the common assumption that the entering and exiting velocities are negligibly small, and omit the terms containing them. We also note that the densities at 1 and 3 are the same and that the pressure at 3 is the same as the pressure at 2. Rearranging the above equation, we get

$$\frac{V_3^2}{2g_c}=\frac{P_1-P_2}{w_1} \tag{4.2}$$

These terms are typically called the "motive head" or "operating head." They represent the energy available to do the pumping work.

Rearranging again and solving for the nozzle exit velocity,

$$V_3=\left[(2g_c)\frac{P_1-P_2}{w_1}\right]^{1/2} \tag{4.3}$$

Consider the example ejector discussed above. P_1= 60 psig, P_2 = 0 psig, and w_1 = 62.4 lbm/ft^3. The nozzle exit velocity is

$$V_3=\left[(2)(32.2)\frac{(60-0)(144)}{62.4}\right]^{1/2}=94 \text{ ft/s}$$

Mixing

In the idealized description of how the two fluids mix together, the mixing occurs at the suction pressure, P_2. Further, it is assumed that no friction losses occur at the walls of the ejector as the fluid passes through the mixing chamber. With these conditions, it is possible to use the momentum equation to determine the velocity after mixing. All velocities are defined as parallel to the axis of the ejector diffuser.

$$\text{Momentum in} = \text{momentum out}$$

$$W_1V_3+W_2V_2=(W_1+W_2)V_4 \tag{4.4}$$

Previously we noted that V_2 is assumed to be zero. So we can rearrange the above and solve for V_4:

$$V_4 = V_3 \frac{W_1}{W_1 + W_2} = \frac{V_3}{1 + (W_2 / W_1)} = \frac{V_3}{1 + R_e} \tag{4.5}$$

where R_e is the entrainment ratio. Inserting the numbers for the above example, we calculate the mixture velocity at 4:

$$V_4 = \frac{94}{1 + 1/1} = 47 \text{ ft/s}$$

At this point the careful reader may notice that some energy has been lost in the mixing process. The mixture mass flow is twice the nozzle mass flow, but the mixture has only one-fourth the velocity head, and thus only one-half the original energy. The lost energy was dissipated through fluid friction within the fluid stream and is present as an increase in the internal energy of the water. It is a little warmer. Because internal energy of liquids is not a valuable form of energy in these calculations, it is commonly left out of the energy equation. This energy "loss" into liquid warming is roughly similar to head loss in pipe flow calculations.

In summary, for the ideal mixing process, momentum is conserved, but useful forms of energy are not.

Pressure recovery

Now we consider the activity the ejector was chosen for: The velocity energy is converted to pressure energy, raising the pressure higher than the suction pressure. In this action the diffuser acts much like a nozzle in reverse. Because we have idealized the performance for this first look, it is exactly like a nozzle in reverse. Therefore, we again use the energy equation to compare conditions at the diffuser throat and the discharge.

$$\frac{P_4}{w_4} + \frac{V_4^2}{2g_c} = \frac{P_5}{w_5} + \frac{V_5^2}{2g_c} \tag{4.6}$$

Here we note that $P_4 = P_2$, $w_4 = w_5 = w_2 + w_1$ and V_5 is assumed to be zero. Substituting and rearranging,

$$\frac{V_4^2}{2g_c} = \frac{P_5 - P_2}{w_1} \tag{4.7}$$

These terms are typically called the discharge head. Rearranging to solve for the discharge pressure rise,

$$P_5 - P_2 = \frac{V_4^2}{2g_c} w_1 \tag{4.8}$$

Substituting the numbers for our example ejector above,

Discharge pressure rise $= P_5 - P_2 = \dfrac{(47)^2}{2(32.2)} \dfrac{62.4}{144} = 15 \text{ psi}$

Generalized equations for pressure recovery

Using the above procedures, but generalizing the analysis to the situation where the load fluid density is different from the motive fluid density and where the entrainment ratio *Re* remains a variable,

$$R_h = \frac{(P_1 - P_2)w_2}{(P_5 - P_2)w_1} = \frac{(P_1 - P_2)}{(P_5 - P_2)} \frac{SG_2}{SG_1} = \frac{H_1 - H_2}{H_5 - H_2} \tag{4.9}$$

Jumpeter shows that

$$\equiv = \frac{\text{motive head, ft-lbf/lbm}}{\text{discharge head, ft-lbf/lbm}}$$

and

$$R_e = R_h^{1/2} - 1 \tag{4.10}$$

Inspection of these equations reveals that it is easier to pump a high-density liquid than a low-density one. Stated differently, if the liquid being pumped has a higher density, then it can be delivered against a higher pressure, or by use of a lower motive pressure, or with a higher entrainment ratio. "Water pumps mercury easier." Actually, those two liquids are not miscible, and thus the mixture might not behave quite as these equations imply.

Practical efficiencies and sizing

The above relationships show the maximum possible performance, with frictionless nozzle, mixing, and diffuser. An efficiency factor of $e = 0.9$ is typically used in the *Re* equation,

$$R_e = e\left(R_h^{1/2}\right) - 1 \tag{4.11}$$

Remember that this equation describes the relationship at the design point only-it does not describe the performance curve of an actual ejector! Once the dimensions of an ejector have been fixed, the performance curve for the ejector touches the above curve only at the design point. At any other entrainment ratio, the efficiency factor will be lower.

This equation answers the question typically asked by a person who is considering getting an ejector to perform a given pumping task, "How many pounds of load fluid will be pumped per pound of motive fluid, given the suction and discharge pressures and the motive fluid pressure?" It answers that question well, but more is needed to answer the next question, "How big is it, approximately?"

To answer that question, it is first necessary to consider briefly the implications in the efficiency factor introduced above. It is an overall factor, useful in predicting overall performance, and representing the losses in the nozzle, mixing area, and diffuser. It turns out that if we describe all the efficiencies in terms of energy ratios, then

$$e = \left(E_n E_m E_d\right)^2 \tag{4.12}$$

_The individual efficiencies have been investigated by the ejector manufacturers and other researchers. Nozzle efficiencies are so close to 1.0 that they are commonly assumed to be 1.0. The mixing efficiency is also high, leaving most of the energy loss for the diffuser. So e is roughly equal to E_d^2

A second consideration after determining fluid velocities is to determine the flow coefficients for the nozzle and diffuser. Nozzles have flow coefficients of 0.95 or higher for smoothly rounded entrances and sizes larger than about 0.1 in inside diameter. A rough estimating approach for diffusers is to assume that the mixture velocity equals the ideal velocity, and use a flow coefficient of 0.9. This is crude, but it is simple, and my objective is not to attempt to design the mechanical details of ejectors. My objective is to answer user questions such as, "Will that opening pass solid particles of 0.2 in diameter?"

For both nozzles and diffusers, use the equations below, with the appropriate flow coefficients.

$$Q = 27.8 C_f D^2 \left(\frac{\Delta P}{SG}\right)^{1/2} \tag{4.13}$$

from which

$$D = \left[\frac{Q}{27.8 C_f \left(\Delta P / SG\right)^{1/2}}\right]^{1/2} \tag{4.14}$$

Q = flow through nozzle or diffuser, gpm
D = diameter of throat of nozzle or diffuser, in
C_f = flow coefficient
ΔP = pressure difference across nozzle or diffuser ; psi

As an example calculation, consider our same ejector above and add the information that it is to use 20 gpm motive water to pump 20 gpm of load. Estimate the nozzle and diffuser throat diameters.

$$D_3 = \left[\frac{20}{27.8(0.95)(60/1.0)^{1/2}}\right]^{1/2} = 0.31 in$$

$$D_4 = \left[\frac{40}{27.8(0.90)(15/1.0)^{1/2}}\right]^{1/2} = 0.64 in$$

The important internal sizes have been determined, and the external connection sizes may be estimated to be three or four times larger. Thus, this ejector would have a motive liquid connection size of about 1 in and a discharge connection of about 2 in. Usually the suction connection size will match the discharge size.

Limiting operating condition, NPSH

A practical limitation of liquid jet ejectors is that if the net positive suction head available (NPSHA) is less than 28 ft (better: ft-lbf/lbm), then the efficiency will decrease. I would expect that critical value to vary in proportion to the motive head, as with mechanical pumps, but the manufacturers' literature does not indicate such. I suggest you remain skeptical of this feature and discuss it with the manufacturer if predictable performance is extremely important.

4.6 How a Compressible Fluid Ejector Works

When the motive and load fluids are gases such as steam and air, the ejector performance becomes more complicated than when the fluids are liquids. The gas densities change as pressures and temperatures change within the ejector, and other new phenomena appear. Typically, the velocity of gas emerging from the nozzle is supersonic, faster than the velocity of sound in the motive gas at that temperature. Typically, the velocity is still supersonic after mixing, and compression shock occurs in the diffuser. Let's examine these phenomena a little before selecting simplified procedures for estimating ejector performance and size.

Ideal gas laws

Several equations describe the behavior of gases with enough accuracy to be useful for estimating and designing ejectors. These may be found in any introductory text on thermodynamics. Some equations resemble the equations used with liquid ejectors, and some are new.

Equation of state

The equation of state is an accurate description of the relationships among pressure, volume, and temperature for gases at low pressures and moderate temperatures.

$$Pv = nR_g T \tag{4.15}$$

P = absolute pressure, psf
v = volume, ft^3 in the system
n = lbm-moles of gas in the system
Rg = universal gas constant, 1546 ft-lbf/lbm-mole R
T = absolute temperature, R

As an example of the use of this equation, rearrange it and solve for the specific volume (cubic feet occupied by 1 lbm) of saturated steam at atmospheric pressure. The boiling point is 212°F (672°R), standard atmospheric pressure is 14.696 psia, and the molecular weight of water is 18.

$$v = \left(\frac{nR_g T}{P}\right) = \left(\frac{(1/18)(1546)(672)}{(14.696)(144)}\right)$$

$= 27.3$ ft^3 which is close to the steam table value of 26.8

The 2 percent error in the above estimate is usually acceptable. It becomes larger as the pressure increases and the temperature decreases. When greater precision is required, the equation of state is often modified by introducing a compressibility factor, Z.

$$Pv = ZnR_g T \tag{4.16}$$

where $$Z = \frac{Pv}{nR_g T}$$

Z has values starting at 1.0 for gases at low pressure and high temperature, and dropping to about 0.2 at the triple point for each gas.

Thermodynamic texts describe how to estimate Z in terms of reduced temperature and reduced pressure, which brings us briefly to the subject of the critical point.

We may describe some of the characteristics of the critical point by considering a transparent container filled with a pure material whose liquid phase occupies the bottom of the chamber in equilibrium with its vapor in the top of the chamber. As the temperature is increased, the pressure will increase, and the physical properties of the gas and liquid phases will become more and more similar. At the critical point, the differences between phases will just vanish and the visible interface will disappear. Above that temperature, no liquid phase exists. The critical temperature and pressure are basic properties of each material. Reduced temperature and pressure are defined below in terms of absolute temperatures and absolute pressures.

$$T_r = \frac{T}{T_c}$$

$$P_r = \frac{P}{P_c}$$

The equation of state will be used frequently in this book, not necessarily identified by name. The compressibility factor will not be used in this book for vacuum ejector calculations.

Reversible expansion and compression

The most concise term for these processes is isentropic, often loosely called adiabatic, but more precisely described as reversible and adiabatic (not gaining or losing heat). The equations describing these processes are very useful in calculating the conditions in nozzles and diffusers. A useful basic relationship between pressure and specific volume in such a process is

$$Pv^k = \quad \text{constant} \qquad = P_1 v_1^k = P_2 v_2^k \tag{4.17}$$

where k = c_p/c_v (1.4 for air, 1.3 for superheated steam; obtain this value from a thermodynamics text)

c_p = specific heat at constant pressure

c_v = specific heat at constant volume

As an example of the use of this equation, find the specific volume of air in the throat of a critical-flow nozzle metering atmospheric air into a vacuum, given that the specific volume at atmospheric conditions is 13.3 ft^3/lbm, and that the throat pressure is 53 percent of the upstream pressure. Rearranging and solving,

$$v_{throat} = v_{atm}\left(\frac{P_{atm}}{P_{throat}}\right)^{1/k} = 13.3\left(\frac{1}{0.53}\right)^{1/1.4}$$

$$= 20.9 \text{ ft}^3/\text{lbm}$$

By combining Eqs. (4.15) and (4.1), we obtain

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{4.18}$$

As an example of the use of this equation, consider the air nozzle above, with an atmospheric temperature of 70°F (530°R). Letting 1 represent the outside location and 2 represent the throat, the throat temperature is

$$T_{throat} = 530(0.53)^{0.4/1.4} = 442°R = -18°F$$

Gas specific heat. The specific heat at constant pressure is a gas property which is used often. Any thermodynamics text will show that for an ideal gas it is

$$c_p = \frac{k}{k-1}\frac{R_g}{MW} \tag{4.19}$$

As an example, find c_p for air, given its molecular weight of 29.

$$c_p = \frac{1.4}{0.4}\frac{1546}{29}\frac{BTU}{778 ft \bullet lbf}$$

= 0.24 BTU/lbm°R (agrees closely with normal value)

Nozzles

Nozzle calculations are the easiest and most precise of the ejector calculations. Nozzles convert pressure energy into kinetic (velocity) energy with almost no friction losses. The flow coefficient for a large, well-designed nozzle also approaches 100 percent. The basic energy equation (4.1) is here modified by adding the internal energy term U and noting that $v = 1/w$.

$$(Pv)_1 + U_1 + \frac{V_1^2}{2g_C} = (Pv)_3 + U_3 + \frac{V_3^2}{2g_c} \tag{4.20}$$

For calculations involving gases, it is extremely useful to note that both Pv and U are determined only by T for ideal gases. See your thermodynamics text for this. Thus, the two terms may be replaced by enthalpy, H:

$$H = Pv + U$$

and our energy equation as it is used for nozzles now becomes

$$H_1 + \frac{V_1^2}{2g_c} = H_3 + \frac{V_3^2}{2g_c} \tag{4.21}$$

As we did with the liquid-jet ejector, we consider V_1 to be negligibly small and rearrange to solve for V_3. Note here that a correction factor involving the supply pipe diameter and fluid velocity is appropriate for any flow nozzle if the supply pipe diameter is less than twice the nozzle throat diameter. Consult any text on thermodynamics, fluid flow, or flow measurement.

$$V_3 = \left[2g_c\left(H_1 - H_3\right)\right]^{0.5} \tag{4.22}$$

The enthalpy data must now be found. For an ideal gas the enthalpy change is

$$\left(H_1 - H_3\right) = c_p\left(T_1 - T_3\right)$$

and

$$c_p = \frac{k}{k-1}\frac{R_g}{MW} \qquad \text{from Eq. (4.19)}$$

and

$$T_3 = T_1\left(\frac{P_3}{P_1}\right)^{(k-1)/k} \qquad \text{adapted from Eq. (4.18)}$$

Combining all into Eq. (4.23),

$$V_3 = \left\{2g_c\left(\frac{k}{k-1}\frac{R_g}{MW}\right)T_1\left[1 - \left(\frac{P_3}{P_1}\right)^{(k-1)/k}\right]\right\}^{1/2} \tag{4.23}$$

Critical flow rates. Industrial ejectors designed for high efficiency will have high nozzle flow coefficients. The HEI standards [2] shows the variation of nozzle flow coefficient with the throat Reynolds number, and should be consulted if accuracy is important. I will generally use a flow coefficient of 0.97 for both air nozzles and steam nozzles; this is adequate for most user calculations of flow rates and nozzle sizes. Nozzles with a throat diameter smaller than 0.10 in may have a slightly lower flow coefficient.

As an example calculation, consider the previous orifice metering atmospheric air into a vacuum. The throat pressure is 53 percent of the atmospheric pressure. Substituting and solving,

$$V_3 = \left\{ 2(32.2)\frac{(1.4)(1546)}{(0.4)(29)}(530)\left[1-(0.53)^{0.4/1.4}\right] \right\}^{1/2}$$

$$= 1027 \text{ ft/s}$$

Now we are prepared to calculate the mass flow rate through a rounded entrance nozzle with a 1.0-in diameter throat and a flow coefficient of 0.97, using the continuity equation and the values of v_3 and V_3 above,

$$W_3 = C_f \frac{A_3 V_3}{v_3} = 0.97 \frac{(3.1416/4)(1.0/12)^2(1027)}{20.9}$$

$$= 0.260 \text{ lbm/s} = 937 \text{ pph (vs. 956 per HEI tables) [2]}$$

This is a handy fact to remember or to write down on a set of notes for quick reference when working with ejectors: a 1-in diameter rounded entrance nozzle will pass 956 pph of air from standard atmospheric conditions into a vacuum system in which the pressure is less than half of atmospheric pressure.

If you wish to make corrections for actual atmospheric temperature and pressure, use this equation:

$$W_{1in} = 956 \frac{P_{atmos}}{760 torr}\left(\frac{T_{atmos}+460}{530}\right)^{1/2} \tag{4.24}$$

For some more information on orifice sizing for metering test loads, see the Appendix.

Subcritical flow. If the pressure downstream of the air nozzle is greater than 53 percent of the upstream pressure, then sonic velocity will not be attained in the nozzle. This subcritical flow condition requires a different calculation. You may choose to use Eq. (4.23), letting P_3 equal the downstream pressure, or you may use the Fig. 4.2 correction factor based on the HEI method.

As an example of the use of this correction factor, consider a test in which the atmospheric pressure and temperature are standard and atmospheric air is being introduced through a rounded entrance flow orifice into a system at 600 torr pressure.

$$\frac{\Delta P}{P_1} = \frac{760-600}{760} = 0.2105$$

Msc = 0.83 from Fig. 4.2

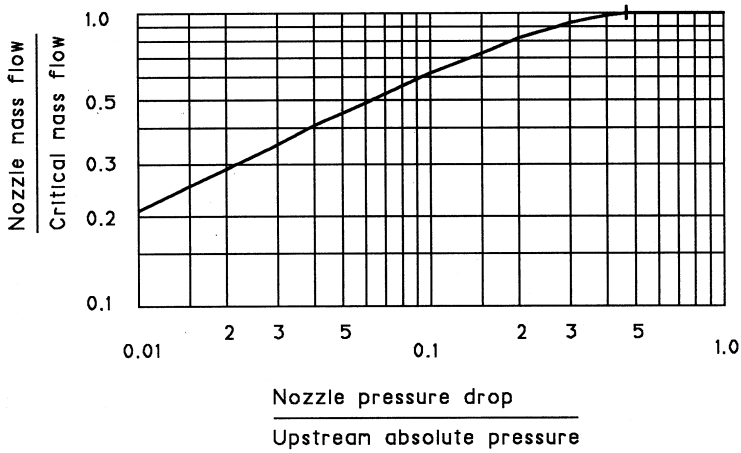


Figure 4.2 Subcritical nozzle flow correction factor.

and the flow through a 1.0- in nozzle is

$$W_{1\ in} = 0.83(956) = 793 \text{ pph}$$

Nozzle outlet size. If the pressure downstream of the nozzle is less than half the upstream pressure, then supersonic velocity is developed, and a properly designed nozzle must have an outlet diameter larger than the throat diameter. Although the inlet may be smoothly curved in almost any pattern which pleases the eye, the diverging area must follow a smooth transition from the throat and usually has an included angle of about 20 degrees or less. Although the most efficient nozzles have curved walls which become parallel at the discharge, an ejector nozzle is usually a straight taper at the outlet. Ideal gas equations above may be used to estimate the nozzle outlet size, based upon the outlet pressure rather than the throat pressure.

At this point the ejector manufacturer's art enters into the design, and the opening may be made larger (overexpanded) or smaller (underexpanded) in accordance with experience and special performance requirements. For example, an overexpanded nozzle will lead to better performance at suction pressures below the design point. An underexpanded nozzle will perform better at pressures above the design point. One reference [5] tends to minimize the importance of the efficiency loss from underexpansion, noting a momentum loss of about 5 percent at a Mach number of 2. That represents an energy loss of about 10 percent, a significant loss in a high efficiency design.

Figure 4.3 shows the approximate ratio of the outlet diameter to the throat for steam nozzles as a function of the nozzle expansion. It

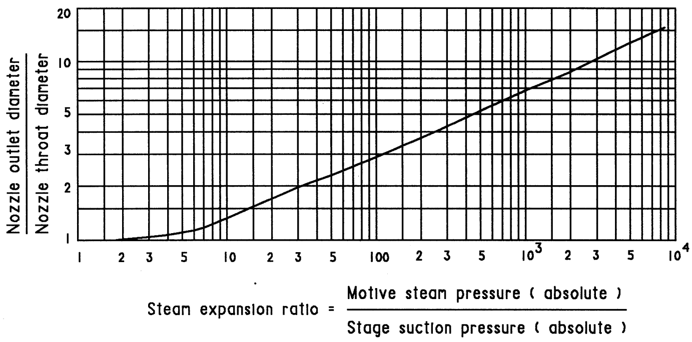


Figure 4.3 Steam nozzle outlet diameter. [Bases: 165 psia dry and saturated steam, ideal gas, $k = 1.3$, fully expanded.]

is based on 165 psia steam dry and saturated, ideal gas laws, and a specific heat ratio k of 1.3. Actual ratios will probably differ as much as 10 to 30 percent from this. It will help identify the nozzles in a multistage ejector if they get mixed up during a maintenance task. Actually, once you use the curve, you will remember the general relationship and seldom need to make precise calculations.

Steam nozzles. Steam behavior is not described accurately by the ideal gas laws for the operating conditions in ejectors. Accordingly, many calculations are best done by using steam tables or charts or simplified equations for specific tasks. To obtain the enthalpy change for calculating nozzle velocity, one may find H_1 on a chart or table at the steam supply conditions, then drop vertically (constant entropy process) on a Mollier chart to find the H_2 value at the nozzle throat pressure (about 0.56 times the supply pressure). The velocity may then be calculated with Eq. (4.22). The specific volume is calculated by noting the moisture fraction on the Mollier diagram at the throat condition and summing the specific volume contributions of the liquid and vapor phases at that pressure. Finally, the mass flow may be calculated using the continuity equation.

Another approach is to use the steam flow curves in the HEI Standards [2], together with an appropriate flow coefficient.

I recommend a third approach as being adequate for almost all needs of ejector users. The equation below, with the adjustment factors, yields estimates within 1 percent of the HEI curves over the range of 15- to 300-psig steam. It is based on dry and saturated steam and includes a flow coefficient of 0.97.

$$W_{\text{steam,pph}} = 50D^2{}_{in}\left(P_{\text{m,psia}}\right)^{0.96} \tag{4.25}$$

For 100°F superheat, deduct 8 percent.

For 200°F superheat, deduct 13 percent.

For 300°F superheat, deduct 17 percent.

As an example calculation, determine the critical flow of 100 psia dry and saturated steam through a 1.0-in-diameter nozzle.

$$W_s = 50(1.0)^2(100)^{0.96} = 4159 \text{ pph} \quad \text{(vs. 4152 from HEI)}$$

This is another data item to add to your quick reference sheet. The flow rate can be scaled up or down in direct proportion to the absolute pressure for quick estimates. For example, the flow for 65 psia will be about 0.65 times 4159, or 2703 pph.

Often the need arises to estimate the throat diameter of a steam nozzle. For that, simply rearrange Eq. (4.25):

$$D_m = \left[\frac{W_m}{50(P_m)^{0.96}}\right]^{1/2} \tag{4.26}$$

As an example calculation, find the throat diameter of a steam nozzle sized for 400 pph of140 psia dry and saturated steam.

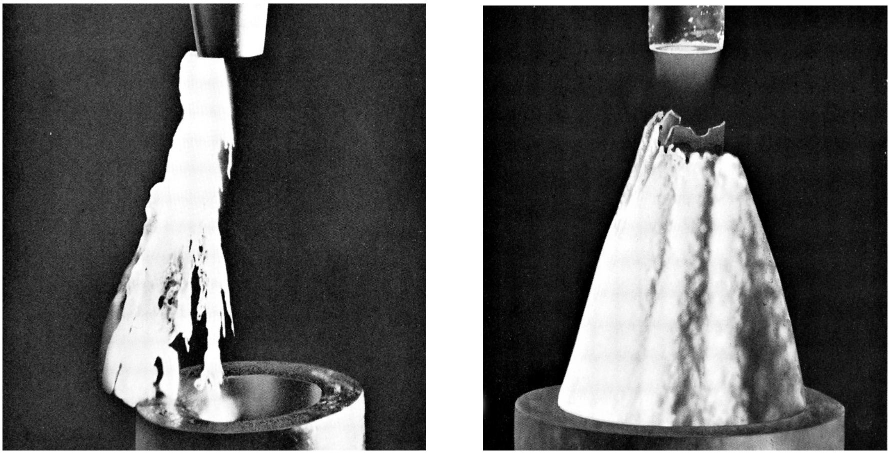
$$D_m = \left[\frac{400}{50(140)^{0.96}}\right]^{1/2} = 0.264 \text{ in}$$

If the steam were superheated 100°F, the corrected size would be

$$D_m = \frac{0.264}{(1.00 - 0.08)^{1/2}} = 0.275 \text{ in}$$

Ice formation. As the example calculation for Eq. (4.18) demonstrated, the gas cools significantly as enthalpy is converted into kinetic energy. For the air nozzle with a throat temperature of -18°F, any water vapor present might become snow or ice.

The same is true for steam nozzles: the temperature of the gas/liquid mixture emerging from the nozzle at supersonic velocity corresponds to the water saturation temperature at the suction pressure. For example, if the suction pressure is 20 torr, the temperature will be about 72°F and the mixture will be about 20 percent liquid by weight. The liquid will be present as a finely dispersed fog that remains mixed with the cool water vapor. The liquid phase may be slightly delayed in its appearance, especially for short, smooth nozzles. This two-phase condition complicates the design of steam nozzles. For more on this subject, read about the Wilson Line in your thermodynamics text.



(*a*) (*b*)

Figure 4.4 Ice formation on steam nozzles below 5 torr: (*a*) without heating; (*b*) nozzle heated. (*Courtesy NItech, Inc.*)

When the suction pressure is below the triple point of water, 4.6 torr, the liquid phase freezes and ice formation becomes a consideration. Although it may be hard to believe until you have thought about it for a while, Fig. 4.4 has convincing photographs of ice formation in an ejector stage. Within the space of only a few inches, and within a tiny fraction of a second, the steam cools from as much as 300°F to below the temperature of ice.

As shown in Fig. 4.4, heating the nozzle prevents ice from forming there. However, ice is seen to form on the diffuser and grow toward the nozzle, interfering with the flow of the load gas. I assume from the pattern that the suction flow was at or near zero. Diffusers on stages below 5 torr typically have steam jackets on the diffuser inlet to prevent ice from forming there. Ice is present as dispersed crystals in the gas flow, and seems to behave less like a gas than does water fog. Whereas the water fog seems to vaporize back to gas as it emerges from the diffuser, the ice crystals seem to take longer to absorb heat from the superheated gas. Thus, the discharge from a stage below 5 torr may be a high-velocity mixture of ice crystals and superheated gas, scouring or eroding any object they encounter. For years I kept a severely eroded graphite steam nozzle on my office bookshelf as a reminder of that phenomenon.

Mixing chamber and diffuser

I will review here the events inside the mixing chamber and diffuser in more detail than before, especially the new phenomena associated with compressible fluid flow. Then I will briefly mention several analytical approaches to detailed design. Finally, I will select an empirical correlation and rough estimate methods which represent a balance between accuracy and simplicity that I have found to best answer a user's needs.

Flow phenomena

The supersonic flow of steam emerges from the nozzle and encounters the low-pressure, low-velocity load gas stream. As with the water-jet ejector described previously, the two streams mix, and the medium-velocity mixture enters the diffuser. There, most of the remaining kinetic energy is converted into pressure energy. The mixture entering the diffuser is still supersonic if the stage is designed for a compression ratio P_d/P_s greater than 2:1.

The diffuser is not able to behave exactly as a reversed nozzle when handling supersonic flow. In supersonic flow situations, it is convenient to represent the velocity by its Mach number, defined as the actual gas velocity divided by the velocity of sound in the gas. In this diffuser action, as described by Shapiro [3], the mixture enters at a Mach number greater than 1.0. Within the diffuser it experiences a normal compression shock, after which the Mach number is less than 1.0 and the pressure and temperature are higher. At the diffuser exit section, most of the remaining velocity energy is converted into an additional pressure rise.

The normal shock occurs in a very narrow plane which usually is somewhere in the throat of the diffuser and oriented normal to the direction of flow. Normal shock is roughly similar to the hydraulic jump phenomenon experienced with incompressible liquids and observed sometimes at dam spillways or when hosing off a driveway. The process is not reversible, so the efficiency is lower than in a subsonic diffuser.

4.7 Surveying Analysis Methods

The actual flow situation in the mixing chamber is a little more complicated than the idealized model used previously for the water jet ejector. The mixing chamber does not have a constant pressure everywhere. The entering load gas flows in the direction of decreasing pressure and typically must negotiate a 90° turn before mixing with the

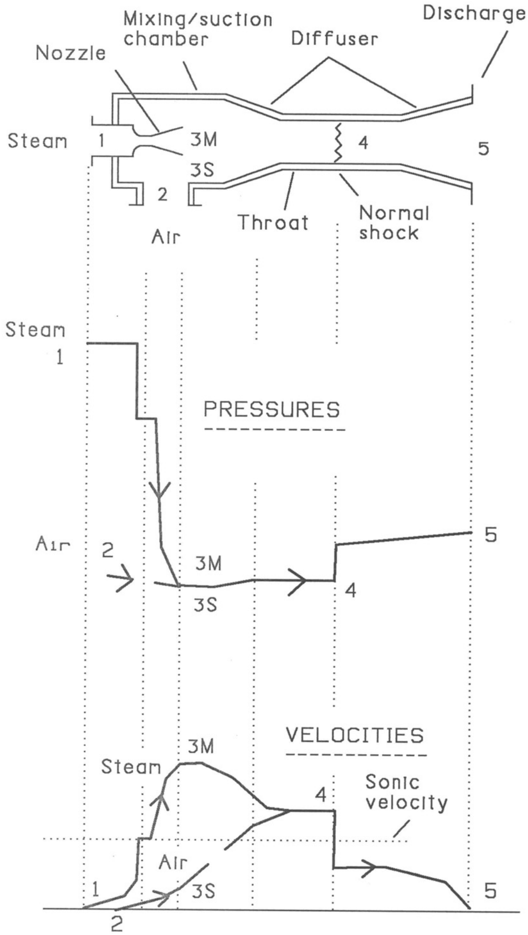


Figure 4.5 (*a*) Critical ejector pressures and velocities.

motive fluid. The profiles of velocity, pressure, enthalpy, and entropy for the two entering fluid streams and for the mixture are shown in Fig. 4.5*a* and 4.5*b*, which borrow heavily from Hick Hargreaves [4].

The actual local conditions inside ejector stages have been examined by inserting slender pressure probes. A typical result is that the suction fluid may enter through location 2 at a pressure of 10 torr, drop to 9 torr at location 3s, rise to 11 torr just before location 4, suddenly rise to 55 torr at location 4, and rise to 65 torr at the discharge.

The technical literature contains several different approaches for analyzing the behavior of critical compressible-flow ejectors. Most use the constant-pressure mixing model I have used here, some use constant-area mixing because it eliminates the uncertainty about the

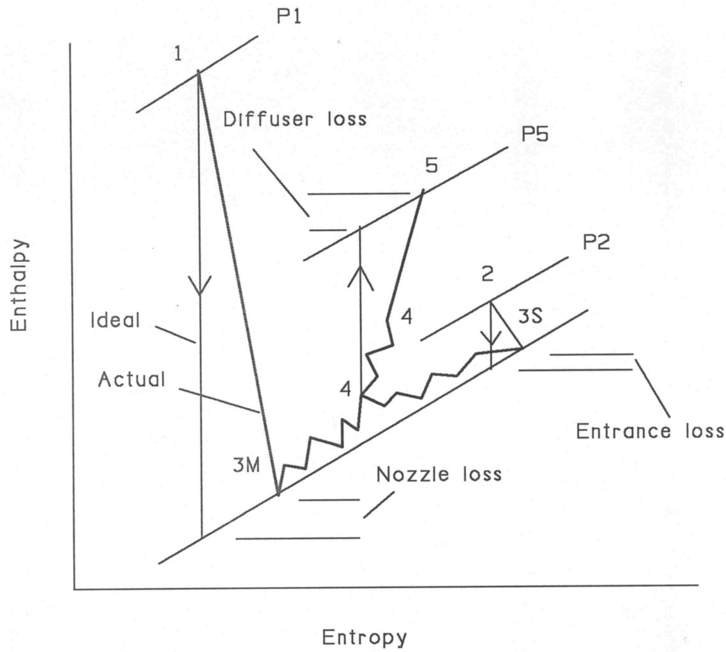


Figure 4.5 (*Continued*) (*b*) Critical ejector Mollier chart of process paths.

effects of geometry in the mixing area, and an isothermal model is even used for simplicity. In Chap. 11 I refer to some of the attempts to devise an orderly method of predicting performance and finding the geometry necessary to achieve it. However, I strongly recommend against attempting to design and fabricate your own ejectors. Except for the trivial situations where almost any performance will be acceptable, or for the unusual situations where industrial or national security justify the high learning costs, it is more cost-effective to buy commercial products. They are supported by people who have invested in years of research and testing, and who are prepared to offer competent aftermarket support.

Here I will mention only Kroll [6], who describes how to design single-stage ejectors for the unusual situations where a user feels strongly compelled to do so. I note that Kroll wrote his paper at a time when energy efficiency was not a significant factor in chemical plant design, and industrial secrecy was very important. His article is a concise treatment of many performance and geometry factors. Use only with care!

Chapter 11 offers a combination of semiempirical and simplified calculation methods for thermocompressors and gas-jet compressors, whose performance is well predicted by theory.

4.8 Empirical Performance Correlations

In reviewing the literature for methods that were accurate and convenient, I found none that were superior to some simple methods I have used for years. I improved them a bit and offer them as my best current offering. Computer advances and the concurrent emergence of powerful fluid dynamics analysis software may permit someone to devise and disseminate a technique for predicting performance and designing the hardware, but this method will do until then.

The basic procedure is to first use the HEI procedures and curves to compute the dry air equivalent (DAE) of the vapors entering the suction of an ejector stage. The next step is to estimate the motive steam required to compress that air load from the suction pressure to the discharge pressure. Then you may make estimates of the sizes of the nozzle throat and outlet, the diffuser throat, and the suction and discharge connections.

Motive steam is estimated by determining how much 165 psia steam would be required. Then correction factors are added for the actual motive steam pressure and for the approach ratio if the discharge pressure is more than 2 percent of the motive pressure. The steam flow estimated at this point describes a stage that is designed for maximum efficiency, and is conditionally stable to no-load. The condition for stability is that the discharge pressure seen by the stage at no-load must be less than its maximum discharge pressure MDP at that load. This requirement is not met by the last stage, and may not be met for the first stage that is followed immediately by a booster stage. An ejector stage followed immediately by a condenser is inherently stable to no-load.

If this stage is the last stage or discharges directly into the next stage, you may want it to be stable to no-load for ease of testing or to reduce the probability that water vapor will backfire into the process system. If the stage is the final stage in an ejector system, you may also want it to be able to operate against the full discharge pressure at no-load as well as at the design point. That will also add to the steam usage. Finally, a last check is made to see whether the stage has such a low flow of steam that a size correction factor is required or whether the steam nozzle size must be increased to meet a user-specified minimum.

If this sounds a little tedious or complicated, please do not be distressed. You will find the method easy to follow, and the calculation steps for one stage in a multistage ejector are often identical or similar to those for other stages. If you do much of this work, you may organize it for ease of calculation, and you will see that it goes quickly.

4.9 Dry Air Equivalent Gas Load

It is convenient to be able to test any ejector with loads of air and water vapor, then calculate its performance handling other gases. It is often impractical to test ejectors in the manufacturer's test stands using the actual gases which the ejector is designed to compress. Often the gases are flammable, toxic, highly corrosive, or very expensive. To resolve this common problem, HEI sponsored tests to predict the gas-handling capacity of stages from their air-handling capacity at the specified suction and discharge pressure. Results from those tests were published in 1951 [7, 8] and incorporated in summary form into the HEI Standards for Steam Jet Ejectors as entrainment ratio curves (Figs. 4.6 and 4.7). Entrainment ratio is defined as the weight of gas divided by the weight of air handled at the same suction and discharge pressure.

The curves quantify a finding from the analysis of liquid-jet ejectors earlier in this chapter: It is easier to pump dense fluids through a given pressure differential than it is to pump less dense fluids. As the ideal gas laws demonstrated earlier, gas density relative to 70°F air at the same pressure depends upon the gas molecular weight and temperature. The molecular weight curve is qualified as being applicable above 10 torr, but it is commonly applied at all pressures.

The HEI procedure for using the curves does simplify and standardize load calculations for steam-jet ejectors. Any weakness such as the

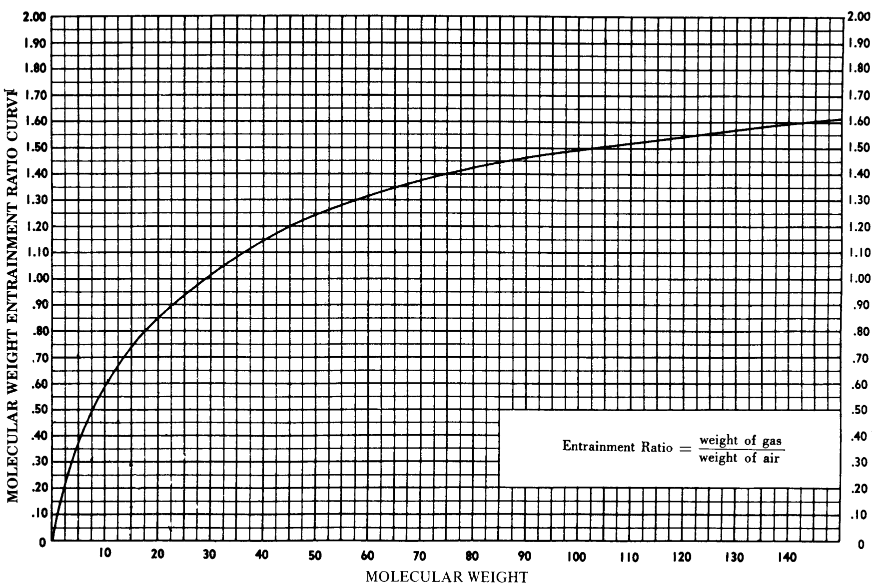


Figure 4.6 Molecular weight entrainment ratio curve. (Courtesy Heat Exchange Institute)

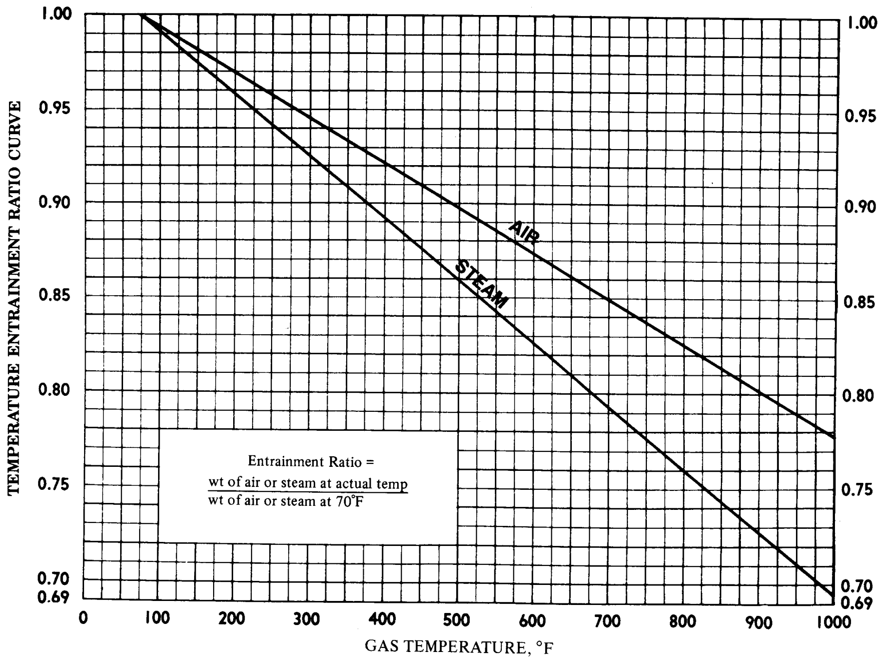


Figure 4.7 Temperature entrainment ratio curve. (Courtesy Heat Exchange Institute)

absence of gas specific heat and specific heat ratio from the correlations is usually secondary to the virtue of simplicity. In the uncommon event that you require precise prediction of performance on some unusual gas load, I suggest that you discuss the subject with several ejector manufacturers and be prepared to pay for prototype testing in their facilities or yours.

Converting loads to dry air equivalent

The HEI procedure for calculating the DAE is to consider any water vapor in the load separately from all other gases, then to treat all other gases in accordance with the molecular weight of the mixture.

As an example, suppose the load consists of 100 pph of water vapor, 58 pph of air, and 132 pph of carbon dioxide at a temperature of 200°F. From Fig. 4.6 find the molecular weight entrainment ratio (MWER) for water (MW = 18) to be 0.80. From Fig. 4.7 find the temperature entrainment ratio (TER) for steam to be 0.96. The DAE of the water vapor portion of the load is

$$\text{Water vapor DAE} = \frac{100}{(0.80)(0.96)} = 130 \text{ pph DAE}$$

The balance of the load is a mixture of gases for which we must find the average molecular weight, using this procedure and noting that the quantity in lbm-moles equals the quantity in lbm divided by the molecular weight.

Component	pph	MW	lbm-moles/h
Air	58	29	2.0
Carbon dioxide	132	44	3.0
Totals	190		5.0

from which the average molecular weight is 190/5.0 = 38. The corresponding MWER is 1.12, and the TER for "air" is 0.97.

$$\text{Other gases DAE} = \frac{190}{(1.12)(0.97)} = 175 \text{ pph DAE}$$

and thus the total standardized load is

$$\text{Total load DAE} = 130 + 175 = 305 \text{ pph DAE}$$

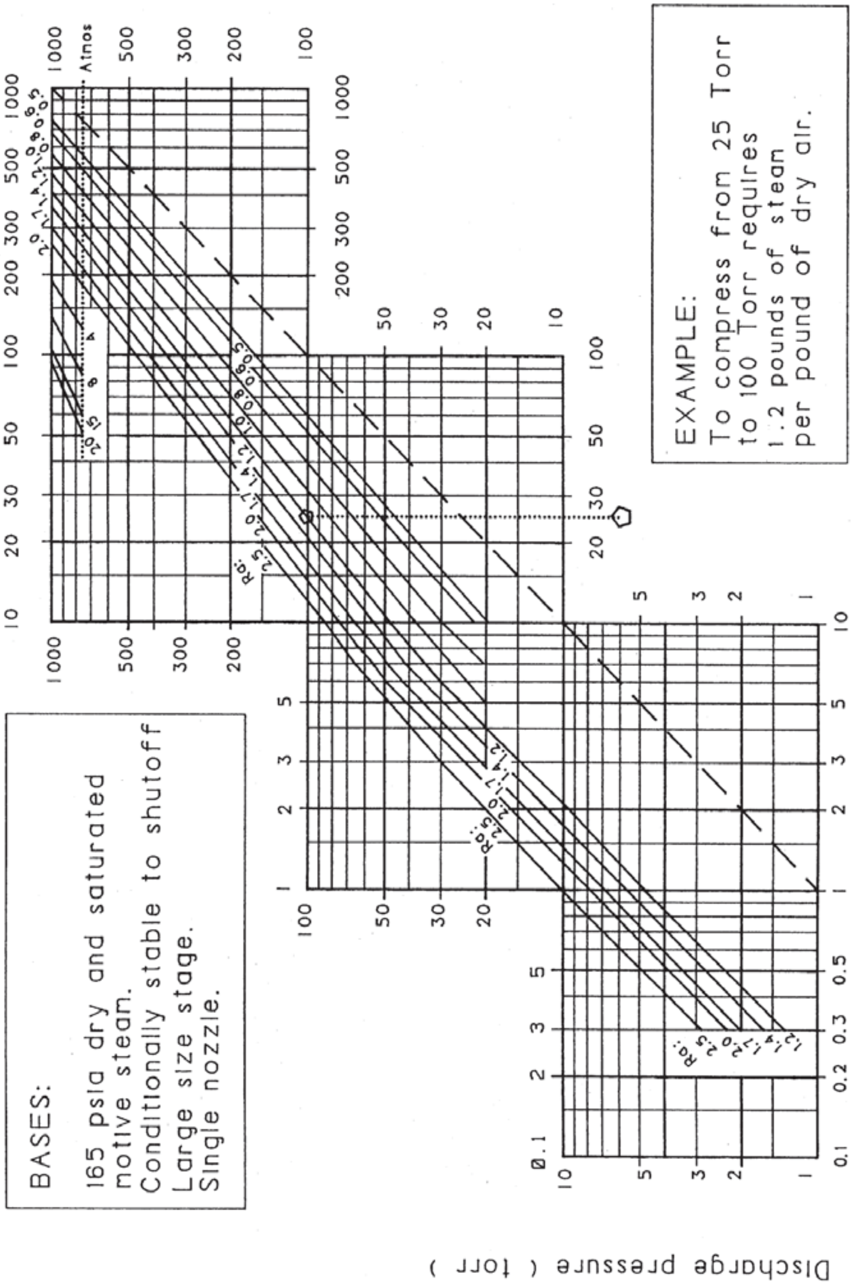
Air and water vapor as test loads

The useful result is that this stage can be tested at the specified pressures for motive steam, suction, and discharge and with a load of 305 pph air at 70°F. Or it may be convenient or appropriate to test the stage with the equivalent flow of water vapor, or with some combination of air and water vapor. The same procedure is used to calculate the proper test load combination if it is not simply air at about 70°F.

4.10 Stage Steam Usage

To determine the steam requirements of an ejector stage, we must know the load gas DAE, the suction and discharge pressures at the design point, the motive steam pressure, whether the stage is to be stable to no-load, and whether the no-load discharge pressure is equal to the design discharge pressure. If the total steam usage is small, then a size adjustment must be added. Finally, we must know whether there are any user-defined minimum nozzle throat sizes. Combining most of these considerations into one equation,

$$R_a = (R_{a,165}\ M_{Pm}\ M_{Pm/Pd}\ M_{stab}\ M_{pd\text{-}0})\ M_{size} \qquad (4.27)$$


BASES:
165 psia dry and saturated motive steam.
Conditionally stable to shutoff
Large size stage.
Single nozzle.
EXAMPLE:
To compress from 25 Torr to 100 Torr requires 1.2 pounds of steam per pound of dry air.
Suction pressure (torr)
Discharge pressure (torr)
Atmos
Ra:

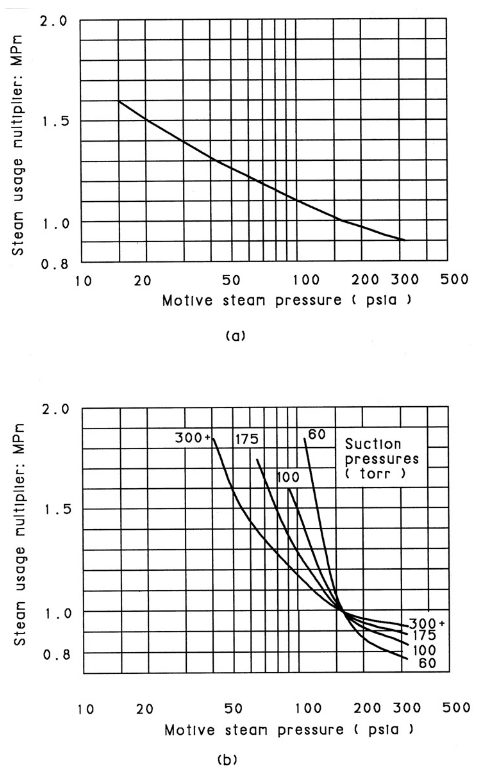


Figure 4.9 Steam rate motive pressure multipliers for 150 psig base: (*a*)stages other than atmospheric, Z-stage; (b) atmospheric, (Z) stage.

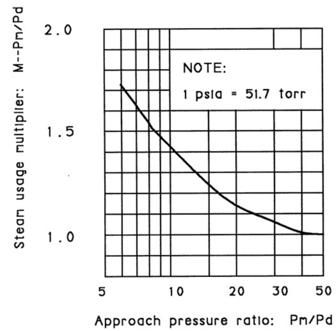


Figure 4.10 Steam rate approach pressure multiplier

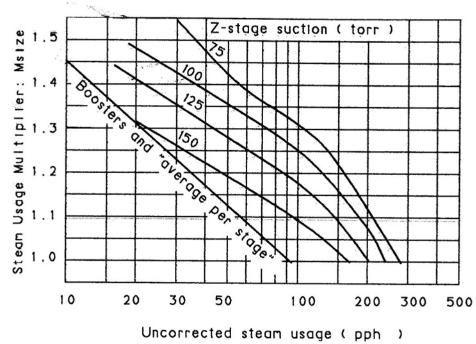


Figure 4.11 Steam rate size multiplier

where R_a = lbm motive steam/lbm load, adjusted

$R_{a,165}$ = lbm motive steam/lbm load from Fig. 4.8 based on single nozzle designs

M_{Pm} = multiplier for motive steam pressure, Fig. 4.9

$M_{Pm/Pd}$ = multiplier for closeness to motive pressure, Fig. 4.10

M_{stab} = multiplier for stability to no load, 1.15 for stable last stage. May be required for the first stage of a booster stage series

$M_{Pd\text{-}0}$ = multiplier for design discharge pressure at no-load, 1.10

M_{size}=multiplier for steam mass flow in a stage

Example stage steam usage calculation

As an example calculation, assume that the 305 pph DAE is to be compressed from 15 to 75 torr using 100 psig steam in the first stage of a multistage condensing ejector. Also, let us require that the stages be stable to no-load, that the motive steam nozzles prior to the atmospheric (Z) stage have a throat at least 3/32 (0.0938) in diameter, and that the Z-stage nozzle throat be at least 1/8 (0.125) in diameter.

Because this first stage will have a generous "lap" between its MDP curve and the second-stage suction curve at no load, no extra steam is required for no-load stability (M_{stab} = 1.0). In Fig. 4.8 we find $R_{a,165}$ = 1.37; in Fig. 4.9 we find M_{Pm} = 1.05; and in Fig. 4.10 we find $M_{Pm/Pd}$ = 1.0. Solving for R_a

$$R_a = 1.37(1.05)(1.0)(1.0)(1.0) = 1.44 \text{ lbm/lbm DAE}$$

and the uncorrected (raw) steam flow,

$$W_m = 1.44\ (305) = 439 \text{ pph motive steam}$$

Now we have only two size considerations to review. We find from Fig. 4.11 that this stage is large enough that the size multiplier makes no adjustment (M_{size} = 1). The last step is to see whether the nozzle size meets the minimum specified. As a useful test for any stage, we may calculate the steam flow through a 0.0938-in nozzle and see whether we have met that requirement. From Eq. (4.25),

$$W_{m,0.0938\text{ in}} = 50\ (0.0938)^2\ (115)^{0.96} = 42 \text{ pph}$$

The flow exceeds the minimum standard by a large margin; thus no correction is needed. Any raw steam usage smaller than 42 pph would have to be adjusted upwards, leading to a larger capacity for that stage or perhaps to an adjustment of the stage pressure interval to use the steam most efficiently.

Accuracy of steam usage estimates

These estimating methods are most accurate for motive steam pressures above 75 psig and low compression ratios associated with R_a values less than 4.0. The data are most useful in the region corresponding to high-energy-efficiency ejector systems. The curves represent smoothed data from several sources. I expect them to agree with manufacturers' data within about 10 percent over the best-fit range. Beyond the stated limits the uncertainty will be greater, but the general trends and feasibility will be useful. For example, although the feasibility of using low-pressure steam for booster stages can be verified, the uncertainty in steam usage becomes greater than 10 percent. Also, high-compression stages achieving a 20:1 compression ratio are beyond the scope of these data.

The steam usage of larger stages using multiple nozzles may be as much as 10 percent lower than the single-nozzle size represented here. When the stage suction connection is 10 in or larger, then the multiple-nozzle option may be desirable. The nozzles are arranged in a ring, usually aimed to converge at the throat of the diffuser. The smaller nozzles have a correspondingly shorter mixing length, permitting adequate mixing to occur in a shorter diffuser, with a more uniform velocity profile after mixing. The reduced wall area and the reduction of jet impingement on the inlet cone may reduce friction losses enough to explain some of the increased efficiency. I have encountered an observation that multiple-nozzle ejectors are noisier than single-nozzle ejectors.

4.10 Stage Size Estimates

At this point in the calculations, enough information is available to roughly size the diffuser. I continue to emphasize that these estimates are only approximate, and are not sufficiently accurate or complete enough to design an ejector stage. Missing information essential to stage mechanical design includes the shape of the inlet cone and the precise positioning of the nozzle relative to the diffuser. Ejector manufacturers determine this information by expensive testing.

Diffuser throat size

For this calculation I will assume that the compression ratio is greater than 2:1 and use a simplified model which treats the diffuser as a "reverse nozzle" in which the gas mixture obeys the ideal gas laws with $k = 1.3$ and with throat velocity 90 percent of sonic velocity. The basis is 165 psia dry and saturated motive steam, $R_a = 2.0$, stage suction temperature of 100°F, and a discharge pressure of 100 torr.

The size for other discharge pressures is ratioed directly from that:

$$D_{4,in} = 0.9\left(\frac{W_{5,se-pph}}{P_{5,torr}}\right)^{1/2} \tag{4.28}$$

where $W_{5\text{-se}}$ = the "steam equivalent" of the total flow at the discharge, equal to the motive steam for that stage plus 0.8 times the DAE load to that stage.

The length of the diffuser throat is quite variable. Maximum efficiency does not seem to be very sensitive to the throat length, but Nash-Kinema [9] prefers a minimum of 5 diameters for adjustments in nozzle position to attain the design efficiency and for an extended range of acceptable performance under changing loads and motive conditions.

As an example calculation, consider the previous example stage calculation where $W_{5\text{-}se}$ = 503 + 0.8(305) = 747 pph and P_5 = 75 torr.

$$D_4 = 0.9\left(\frac{747}{75}\right)^{1/2} = 2.8 \text{ in, with a length of 14 in}$$

Stage external dimensions

The most useful value of these estimates is for preliminary layout planning, and occasionally for discerning "which stage is the first one" when the assembly sequence is not completely clear.

A little thinking about the mass flow and pressure conditions in a multistage ejector shows that successive stages in series get smaller. In energy-efficient ejectors the compression ratio in each stage will usually be somewhere between 5:1 and 8:1. Examination of Fig. 4.8 shows that the corresponding R_a values at a suction pressure of 10 torr are about 1.3 and 2.2 lbm of steam per lbm DAE. Correcting for the steam molecular weight and temperature makes those 1.8 and 3.0 DAE, respectively, which when combined with the load gas become 2.8 and 4.0 DAE total flow at the discharge connection. Thus, the increasing mass flow is more than offset by the increasing pressure. Because the same velocity guideline is typically used for sizing the discharge connection as the suction connection, we can arrive at the following approximate relationship between the discharge and suction connection sizes:

$$\frac{D_5}{D_2} = \left(\frac{W_5 P_2}{W_2 P_5}\right)^{1/2} \tag{4.29}$$

Now substituting the 5:1 compression ratio data,

$$\frac{D_5}{D_2} = \left[\frac{(2.8)(10)}{(1.0)(50)}\right]^{1/2} = 0.75$$

and substituting the 8:1 compression ratio data,

$$\frac{D_5}{D_2} = \left[\frac{(4.0)(10)}{(1.0)(80)}\right]^{1/2} = 0.71$$

This leads to the generalization that the discharge connection size will tend to be about 75 percent of the suction connection size. Often the manufacturer makes them the same size, especially for the smaller stages. The overall length of a single-nozzle stage is about 10 times the suction connection diameter. A multiple-nozzle stage may be as much as 30 percent shorter (7 diameters long). The diameter corresponding to a typical design velocity of 200 ft/s is

$$D_{in} = 1.6\left(\frac{W_{DAE}}{P_{torr}}\right)^{1/2} \tag{4.30}$$

This "size factor" is an extremely useful measure of the size of an ejector. It leads to a good approximation of the size of the first stage and, as will be found in Chap. 10, is the key to preliminary estimates of purchase cost. One of my rules of thumb which you might consider adding to your quick estimating sheet is that an ejector handling 1 pph DAE at 1 torr will have a theoretical suction connection of 1.6 in and will be 16 in long. Of course, that is not a standard size, which may make it easier to remember!

Manufacturers tend to use higher velocities for larger units at lower pressures, up to 300 ft/s. Conversely, velocities as low as 150 ft/s may be used for smaller sizes at pressures at or above atmospheric pressure. For very high pressure gas-jet ejectors, or for very low compression ratio ejectors such as fume scrubbers and ventilators, it may be as low as 50 ft/s.

As an example, the size required to handle 300 pph DAE at 10 torr is

$$D_2 = \left(\frac{300}{10}\right)^{1/2} = 8.8\,\text{in}, \quad \text{say } 8 \text{ or } 10 \text{ inches}$$

4.11 Multistage-Ejector Steam and Water Usage

When a gas compression task involves an overall compression ratio that cannot be economically reached by one ejector stage, two or more stages may be connected in series to accomplish that task. The load and the motive gases from the first stage combine to form the load into the second stage. The second stage may discharge into a third stage, and so on, until the mixture pressure is high enough to discharge into the atmosphere.

Usually condensers are located between some of the stages to condense out much of the motive steam and the condensable portion of the load vapors. The condensed liquids formed in a condenser are removed by gravity or a pump, so that only the remaining gas mixture continues as a load to the ejector stage connected to the condenser vent. The task of estimating the steam usage is thus more complicated than for an individual stage.

The most direct, but most time consuming method is to do a stage-by-stage design, estimating the interstage pressures and the conditions within each condenser. The second method is quicker and usually adequate. It is based on a chart that gives the overall Ra (lbm steam per lbm DAE) versus suction pressure for different ejector configurations. A third method is a variation of the second method. It is applicable to two- and three-stage ejectors handling mixtures of air and condensable vapor. All three methods will be described in this chapter.

The cooling water used in the condensers represents additional installation cost and operating cost that must be considered in the design economics. A simple rule for estimating the cooling water usage of a multistage ejector with condensers is that the water usage in gpm is equal to 0.15 times the total steam usage in pph [10]. Thus, an ejector that uses 2000 pph steam will require about 300 gpm cooling water. The rule corresponds to an assumption that all the steam is condensed and the heat is absorbed by a 15°F rise in the water temperature. This is a rough estimate, adequate for most situations because the water cost is usually much less significant than the steam cost.

Stage-by-stage calculations

This method of estimating ejector performance is similar to the procedure used by an ejector manufacturer and may be adopted by you for several different reasons. You may be checking an ejector quotation to detect and correct apparent errors that might otherwise result in purchasing an inadequate ejector.

Or you may be doing a preliminary study to anticipate the optimum design of an unusual or very important ejector. The results of the study provide a ready reference that allows you to communicate quickly and easily with the ejector manufacturers if questions arise during bid preparation and review. As described in more detail in Chap. 10, such a detailed estimate provides an objective technical reference during discussions with ejector manufacturers. It permits you to avoid an unethical discussion of one manufacturer's design details with one of the competing manufacturers.

A third reason for doing a detailed estimate, described in Chap. 11, is the rare situation where you may wish to become involved in some manner in the actual stage-by-stage design. The ejector may be integrated in some unusual manner into a process system; thus you may have a special reason (as I once did) for purchasing special corrosion-resistant stages and condensers from different sources, or you may have some unusual secrecy requirements which require that you arrive at a functional design without divulging critical process information.

Ejectors having more than three stages usually have two or more stages in series before the first condenser. Because this is a common situation, it might seem reasonable to use design curves for such two or three-stage noncondensing series. The reason I did not add such curves to Fig. 4.8 is that too many adjustments must be considered for each stage; thus any such design curve would be limited to one steam pressure and large sizes. Anyone who plans to do very much estimating may very well invest an hour or two to make up his or her own chart similar to Fig. 4.8 for the specific motive steam pressure, size, etc., appropriate to his or her needs.

Lap between stages

It is common practice for ejector designers to allow 5 to 10 percent of absolute pressure as a "lap" between stages in a multistage ejector. This allows for uncertainties in predicting performance, variations in motive steam and load conditions, and a tolerance for minor fouling. If a condenser is placed between stages, the typical allowance for pressure drop is about 5 percent for contact condensers and 10 percent for surface condensers, included in the lap. When two consecutive stages are separated by a condenser, the lap will generally permit the leading stage to be stable to no-load.

Example three-stage calculation

As a simple example of a multistage ejector, consider a three-stage ejector with a design suction pressure of 15 torr and a load of 40 pph air and 60 pph water vapor at 70°F. Motive steam pressure is 165 psia, and the ejector is to have all stages stable to shutoff and a discharge pressure of 813 torr at design load and at no-load. Contact intercondensers are specified. Interstage pressures are given as 90/85 torr at the first condenser and 200/190 torr at the second condenser. Condenser vent temperatures are 90°F, and the water vapor components of the loads to the second and third stage are 20 and 6.4 pph, respectively. Minimum nozzle throat is 1/8 inch for the last stage, 3/32 inch for all others.

Following the procedures described previously,

$$\text{Stage 1 load} = \frac{40}{(1.0)(1.0)} + \frac{60}{(0.8)(1.0)} = 115 \text{ pph DAE}$$

Stage 1 R_a = 1.65(1.0)(1.0)(1.0)(1.0) = 1.65 lbm steam/lbm DAE

Stage 1 steam = 1.65(115) = 190 pph   no size correction

$$\text{Stage 2 load} = \frac{40}{(1.0)(0.994)} + \frac{20}{(0.8)(0.993)} = 65.4 \text{ pph DAE}$$

Stage 2 R_a = 0.85(1.0)(1.0)(1.0)(1.0) = 0.85 lbm steam/lbm DAE

Stage 2 steam = 0.85(65.4) = 56 pph   size correction is needed

From Fig. 4.11, adjust 56 to 62 pph.

This steam flow exceeds the 3/32-in minimum nozzle flow of 59 pph; thus no adjustment is required.

$$\text{Stage 3 load} = \frac{40}{(1.0)(0.994)} + \frac{6.4}{(0.8)(0.993)} = 48.3 \text{ pph DAE}$$

Stage 3 R_a = 2.3(1.0)(1.0)(1.15)(1.1) = 2.9 lbm steam/lbm DAE

Stage 3 steam = 2.9(48.3) = 140 pph,   no size correction

This steam flow exceeds the 1/8-in flow of 105 pph. Therefore, no adjustment is required.

Total estimated steam usage is 190 + 62 + 140 = 392 pph

Total estimated water usage is 0.15(392) = 59 gpm

Optimization of a multistage ejector design. Now, if this were a competitive bidding situation and I were preparing a bid which I expected to be evaluated with a significant cost assigned to each pound of steam required, I would slightly redesign the system to make each unit of steam work for me. I would adjust the two interstage pressures to find the lowest steam usage.

I took a few minutes to check this out and found the best interstage pressures for this job to be about 74/70 torr for the first intercondenser and 250/240 torr for the second, requiring about 350 pph steam. It is appropriate here to state that the objective in optimization is not to seek the ideal design with nit-picking accuracy, but rather to avoid the design extremes of excessive operating cost and excessive purchase price. Usually the most energy-efficient design among competing bids will have the most stages and intercondensers, a low first intercondenser pressure, and the largest first intercondenser if it is a surface condenser. It often will also have the largest purchase price. Refer to the Appendix for a detailed discussion and example of ejector economics and design optimization.

If the condensers were surface condensers, then I could also experiment by adjusting the water flow rate and total surface area as well, possibly adding a more expensive gas-cooling arrangement at the vent location. Of course I would have this capability on my computer, so I would simply load all the job specifications, including steam and water costs and minimum nozzle sizes, and let the program find the best offering I can prepare from my selection of stage and condenser designs. If the program cannot handle all the specifications of the job, it will at least get close to the best design. A few hand calculations or "guided" computer designs will then finish the job.

Another possible way in which this ejector system design may be improved is to add another stage and intercondenser. Because Europe has experienced higher energy costs and a scarcity of water for years, they have already been using more stages and intercondensers than have U.S. manufacturers. Factors which make it attractive to add more stages and intercondensers are low steam pressure, low cooling water temperature, large load, large condensable vapor in load, high costs for steam and cooling water, continuous operation, long project life, and a user who is willing to invest the extra money to build a process plant which will be cost-effective over the project life.

Because condensers and vapor-liquid equilibrium calculations are not discussed until Chap. 5, this is as far as we can currently go in exploring design optimization. In Chap. 10 the subject will be explored again, after all the involved unit operations and practical operating considerations have been discussed.

Where stability is important. The previous example three-stage ejector was specified by the user to have each stage stable from the design point down to zero load (shutoff). That feature reduces the probability of backfiring steam from the first stage back into the process system. It also makes it much easier to troubleshoot and test the individual stages. If the last stage discharges to a vent header or is sealed by submergence in a hotwell, then the high back pressure specification is essential. (Chapter 6 describes backfiring in detail.)

In a sequence of noncondensing stages, the stages after the first need not be stable to shutoff because the motive steam from the previous stages provides a minimum load to keep them in their stable operating range. Other reasons for not requiring stability always include considerations of economy and a willingness to accept some backfiring and instability. Steam-jet refrigeration and water-stripping "booster" stages are seldom required to operate much below the design suction pressure, and a little backfiring causes no problem. For a very large ejector or a battery of ejectors operated and maintained by ejector technicians who have the training and instruments to work comfortably with unstable ejectors, it may be desirable to reduce the operating costs by accepting some inconveniences.

Overall multistage estimates

When rough estimates are completely adequate, time can be saved by using overall estimating methods. When only noncondensable gases are present in the load, the air-only method is sufficient. If a portion of the load vapors is water or is expected to condense along with water, then the air-water method is better. The air-water method may be useful with ejectors serving turbine condensers or water stripping booster condensers.

Dry air load only. For this method you need to know the load DAE, the steam pressure, the water temperature, and the condenser type. For a single-stage ejector, the steam R_a (r) value may be obtained from Fig. 4.8 and adjusted as described for that figure. For a single-stage ejector, the condenser type does not affect the steam consumption.

For multistage ejectors, obtain R_a from Fig. 4.12 for the suction pressure, ejector configuration, and water temperature for your situation. If your steam pressure is not 100 psig, then adjust the R_a value using the steam pressure multiplier from Fig. 4.13. Note that this multiplier is *not* the same as the ones from Fig. 4.9, which are for correcting 165 psia steam data.

Figure 4.12 is from Ryans and Roper [11] and represents a family of optimized ejector designs using surface condensers and a discharge pressure of 800 torr, and equivalent fixed investment values of $333/pph steam and $250/gpm cooling water. This is a more carefully prepared version of similar curves that I published earlier [12,13]. See Chap. 10 and the Appendix for discussions of the role of economics in design and bid evaluation.

I wish to emphasize that these curves are not performance curves of actual ejectors. Rather, they represent the envelopes of such curves, created by connecting the maximum-efficiency points of a large number of individual curves. Examination of the general slope of these curves leads to the rough rule that lowering the design suction pressure by 10 percent raises the steam usage by about 5 percent.

As an example of their use, consider the three-stage ejector in the previous stage-by-stage example. The steam pressure was 165 psia,

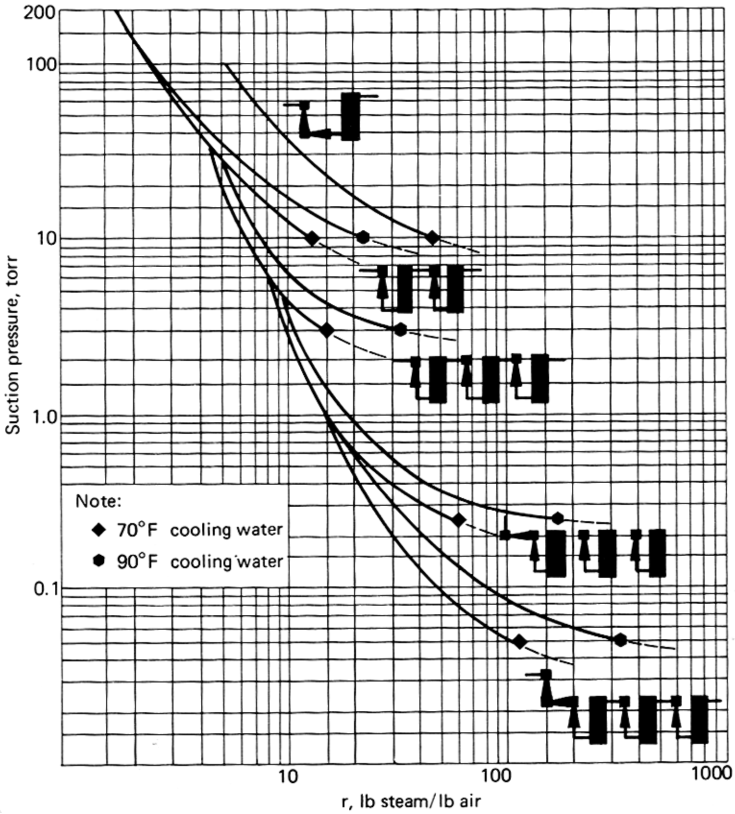


Figure 4.12 100 psig motive steam usage for multistage ejectors. *(From Ryans and Roper,* Process Vacuum System Design & Operation, *McGraw-Hill, New York, 1986)*

*t*he total load was 115 pph DAE, and the vent temperatures of 90°F suggest that the cooling water was about 85°F. From Fig. 4.12 we obtain R_a (r) = 6.0; from Fig. 4.13 we obtain $M_{Pm} = 0.91$.

$$\text{Total steam} = 6.0(0.91)(115) = 628 \text{ pph} \qquad \text{no size correction}$$

But we. know that the water vapor in the load drops out completely in the first condenser and is not seen by the last two stages. Thus, we may make a simple correction to this estimate by subtracting the effect of the load not seen by the last two stages. By inspection of the stage-by-stage calculations, we note the condensable portion of the load to by

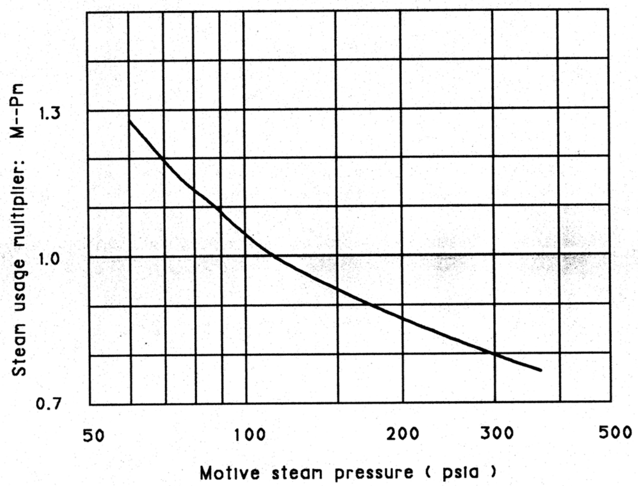


Figure 4.13 Steam rate motive pressure multiplier, for 100 psig base. *(From Ryans and Roper*, Process Vacuum System Design & Operation, *McGraw-Hill, New York, 1986)*

75 pph DAE, and the second-stage suction pressure for my "optimized" design is 70 torr. From Fig. 4.12 we find R_a (r) = 3.0;

$$\text{Steam reduction} = 3.0(0.91)(75) = 205 \text{ pph}$$

$$\text{Corrected steam} = 628 - 205 = 423 \text{ pph}$$

Because Fig. 4.12 is based on surface condensers, the steam flow should be reduced by about 15 percent for the contact condensers present in the example. However, the three-stage example requires each stage to be stable to shutoff, has a discharge pressure of 813 torr at shutoff, and specifies minimum nozzle sizes. The extra steam for these criteria roughly cancels out the reduction for contact condensers, and we may conclude that this quick method yields good accuracy if one makes a reasonable estimate of the first intercondenser pressure.

For condensable vapor other than water, the method will be less accurate.

Air and water vapor load, two-stage and three-stage. A significant number of ejectors are designed to handle a load of air and water vapor only. Many others have a condensable vapor that is known to condense nearly completely in the first condenser and thus can be treated as though it were water. Figures 4.14 and 4.15 show Fondrk's data for estimating the steam consumption of two- and three-stage ejectors with surface condensers [14]. His data give the 100 psig steam rate per pound of load vapor as a function of suction pressure and weight

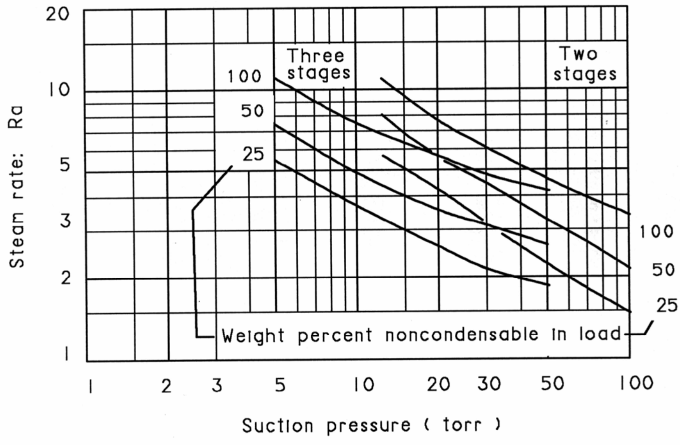


Figure 4.14 100 psig motive steam requirements for ejectors handling condensable vapors. *(Reproduced with permission from* Petroleum Refiner, *December 1958)*

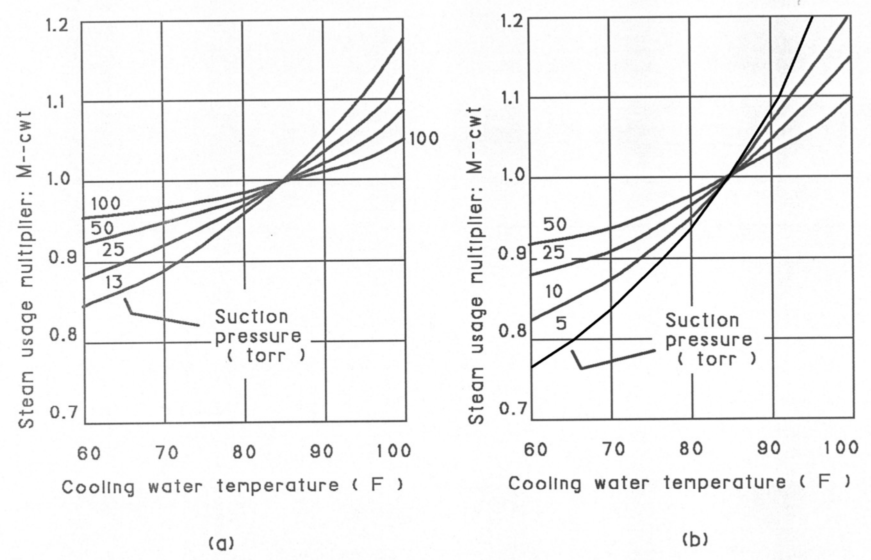


Figure 4.15 Steam rate cooling water temperature multipliers: (*a*) two-stage; (*b*) three-stage. *(Reproduced with permission from* Petroleum Refiner, *December 1958)*

percent noncondensable in the load. He also provides correction factors for cooling water temperature. Steam pressure correction is the same as that given in Fig. 4.13 for 100 psig steam. For contact condensers, reduce the two-stage steam by 10 percent and the three-stage steam by 15 percent.

As an example, consider again the three-stage ejector previously used. The air is 40 percent of the load by weight, and the cooling water is estimated to be about 85°F. From Figs. 4.14, 4.15, and 4.13,

$$R_{1oad} = 3.8 \qquad M_{cwt} = 1.0 \qquad M_p = 0.91 \qquad M_{cont.\ cond.} = 0.85$$

$$\text{Steam usage} = 3.8(1.0)(0.91)(0.85)(115) = 338 \text{ pph}$$

For all stages stable to shutoff and discharge of 813 torr at shutoff, add 5 percent and 3 percent (15 and 10 percent in the last stage):

$$\text{Adjusted steam usage} = 1.05(1.03)338 = 366 \text{ pph}$$

This is acceptably close to the other two estimating methods.

4.12 Priming/Hogging/Initial Evacuation

How do we start up a vacuum system in the first place? Usually the process system is at atmospheric pressure, filled with air or with an inert gas such as nitrogen. A 1000-ft^3 system initially contains 75 lb of air. Before we can operate that system at, say, 5 torr, we must first remove over 99 percent of the air. This startup process is variously called priming, hogging, or initial evacuation.

As a general rule, the fastest way to evacuate a system with an ejector is to first apply steam to the last stage (Z) only. If the performance curves are available, we may examine them to find the first "crossover" pressure, below which the air-handling capacity will be greater with the last two stages (Y and Z) operating. Of course, any condenser between those stages must have its water turned on. As the pressure continues to drop past the next crossover pressure, we turn on the next previous stage (X), and so forth until the desired operating pressure is reached. If the crossover pressures are not known, we may use typical values such as 200 torr and 75 torr, below which all stages and condensers are turned on.

By contrast, the simplest way to evacuate a system is to turn on the cooling water and steam to all stages and let the system take a longer time to reach the desired operating pressure.

In choosing between these two methods, we encounter some more questions. What about the air that begins to leak in as the pressure drops, what is the effect of evaporation and flashing of process liquids in the system during the evacuation, and what evacuation time is required? In my experience, the measured air leakage is typically a small fraction of the ejector design air capacity. The effect of evapora tion is usually to flush the air toward the ejector, speeding the air removal. And the required evacuation time depends upon the preferences of the system designers and operators, influenced strongly by whether the system is a batch system with many startups or continuous operation for months at a time.

Rough estimate of multistage ejector evacuation time

My preferred approach to this question is to make a rough estimate of the evacuation time by assuming that the effective average air removal rate is twice the design noncondensable gas capacity. I divide this into the initial air in the system and ask the designer or operator if that time is acceptable.

As an example, assume that a 1000-ft^3 system is to be evacuated by an ejector which has a design noncondensable gas capacity of 25 pph. The average evacuation rate is assumed to be twice that (50 pph), requiring 1.5 h to remove the 75 lb of air.

Modifications to shorten evacuation time

If this time is not acceptable, then several options are possible, as discussed in Chap. 10. Sometimes the last stage may be made a little larger. Sometimes it is preferable to dedicate a noncondensing "hogging" ejector system to this task, switching over to the more efficient condensing ejector system when the operating pressure has been reached. The hogging ejector can also serve as an emergency spare to one or more condensing ejectors, as discussed in Chap. 10.

Sizing noncondensing evacuation ejectors

I find a lot of scatter in the published evacuation data, probably because the subject is not of much economic importance and the manufacturers select the cheapest combination of standard stages that will meet the user specifications. Figure 4.16 shows the steam requirements of one-stage and two-stage noncondensing ejectors evacuating a theoretical 1-ft^3 system from atmospheric pressure to the evacuation pressure in 1 min. The two curves represent upper and lower bounds on the scattered data. The lower curve represents the most efficient units, and the upper curve represents standard units which may be stocked for quick delivery and low cost.

As an example, find the 100 psig steam usage of a noncondensing ejector which will evacuate a 2000-ft^3 system from atmospheric pressure to 20 torr in 60 min. From Fig. 4.16, select 16 pph as the base

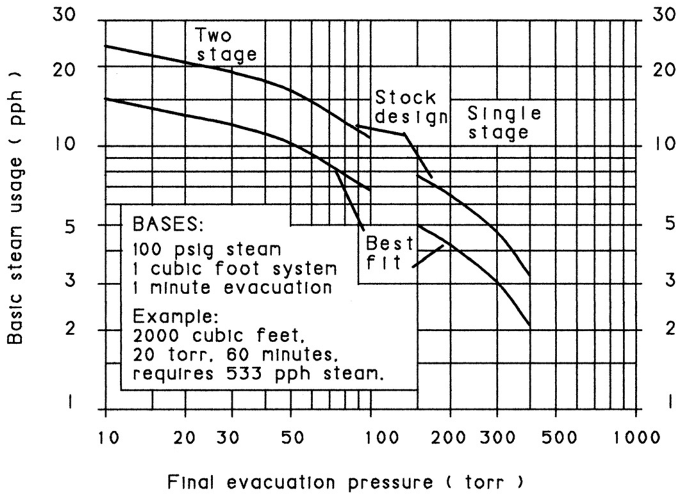


Figure 4.16 100 psig motive steam requirements of evacuation ejectors.

usage, which is midway between the curves. The estimated steam usage is

$$\text{Steam usage} = \frac{16(2000)}{(60)} = 533 \text{ pph}$$ with an uncertainty of about 30 percent

For evacuation to pressures above 400 torr, you may interpolate linearly to 760 torr assuming that a single-stage unit designed for 150 torr has a constant capacity above 400 torr. The "midway" base rate is 3.0 pph steam at 400 torr. If we wish to evacuate a 2000-ft^3 system to 500 torr in 20 min, the estimated steam usage is

$$\text{Steam usage} = 3.0\,\frac{2000(760-500)}{20(760-400)} = 212 \text{ pph}$$

For other steam pressures, use the steam pressure multiplier for 100 psig steam.

4.13 Performance Curve Shape

An ejector-stage performance curve has some interesting information away from the design point. If the stage is not stable to shutoff, it is useful to know the load below which it is unstable. Figure 4.17 illustrates several characteristics of stage performance curves for a stage which is stable to shutoff. The development of these curves

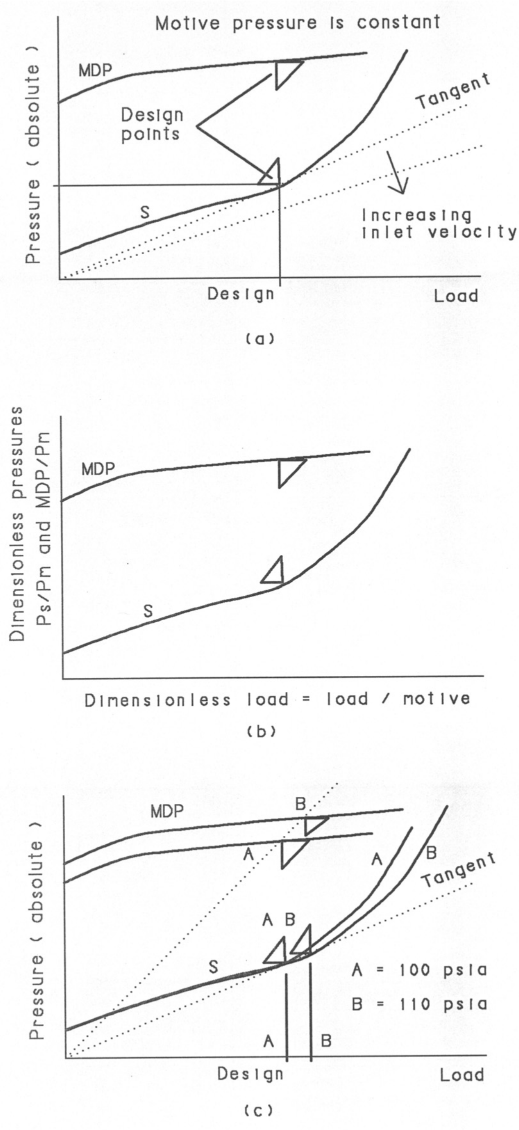


Figure 4.17 Stage performance curve characteristics: (a) conventional scales; (b) dimensionless curves; (c) effect of increased steam pressure.

was described in Chap. 3. If the stage is stable to shutoff, the shutoff suction pressure is typically from 20 to 30 percent of the design pressure. Above the design pressure, the curve turns upward, reaching a load maximum which may be two or more times the design load.

Design point

As seen in Fig. 4.17*a*, the design point is the maximum-efficiency point on the suction curve and corresponds to the maximum inlet velocity condition. On a linear plot of suction pressure versus load, this corresponds to a tangent line drawn from the origin. On logarithmic graphs (log-log), this tangent line is a 45° line. Harris [10] describes how variations in nozzle position and steam flow will alter the performance curve. This information is qualitative only, neither intended nor adequate for designing or modifying ejector stages. Manufacturers' literature and articles contain example performance curves. Manufacturers rely on test data to predict off-design performance curves, but this is seldom a precise estimate.

Given the above, we could sketch the performance curves for the example three-stage ejector, starting with the design points for each stage, adding the approximate shutoff pressures, and extending the curves upward crudely. As Chap. 3 showed us, we could note the maximum air-handling capacity with a dashed line marking the commencement of overloading. We could also add the maximum discharge pressure curves for each stage, noting that in general the curves droop to their lowest pressure at shutoff.

Similitude and generalized dimensionless curves

The ideal gas laws and the laws of similitude suggest some generalizations that tell us what to expect of ejectors operating under changed conditions. One important factor is the effect of changes in the motive steam pressure. If we have curves for an ejector stage supplied with 100 psia motive steam, we might ask how they would change If we were to increase the steam pressure to 110 psia. If we neglect the slight increase in steam saturation temperature, and consider the steam to behave approximately as an ideal gas, we can make some predictions. The motive flow will increase by 10 percent, the motive velocity and velocity at all locations in the stage will remain unchanged, the load/motive mass ratio will remain constant at the design point, the load mass flow will increase by 10 percent, the shutoff pressure will increase by 10 percent, and the discharge pressure curve will increase by 10 percent.

In fact, if we were to use the concepts of similitude to redraw the original performance curve with the pressures replaced by their ratio to the motive pressure and load flow replaced by the ratio of the load to the motive flow, then the same set of curves would be characteristic of the stage hardware and would describe both ejector steam pressure conditions. Figure 4.17*b* is such a dimensionless curve. Note that the curve now describes a family of stages, all having the same geometry ratios and varying only in size. Thus we see how a small model can be tested to verify the design geometry of a large ejector, allowing for size effects.

Ejectors, like fluid machines, become more efficient as the size increases. A test on a small model is a conservative basis for predicting the performance of a large ejector. In smaller ejectors the wall friction effect reduces the efficiency, requiring more pounds of motive fluid per pound of load for a given pumping task.

Each curve would have application for a range of steam pressures and the corresponding range of suction and discharge pressures. Changing the nozzle geometry and position or changing the diffuser geometry would expand the information into the data banks maintained by ejector manufacturers.

How steam pressure rise changes performance curves

This set of dimensionless curves then tells us a little about the effect of raising the steam pressure. If we inspect Fig. 4.17*a* and *b* and contemplate the previous paragraph, we can see that the design points on the suction and discharge curves will shift along straight lines from the origin through those points, shown in Fig. 4.17*c*. Then we recall that the suction pressure curve is tangent to such a line, leading to the conclusion that a small increase in the steam pressure does not change the suction pressure curve in the vicinity of the design point! It will reduce the capacity below the design pressure and increase it above the design pressure. Because of the nonideality of steam, full credit should not be taken for the increase in the discharge pressure curve.

A detailed example application of the laws of similitude is given in Chap. 6.

A field application of similitude laws

The insight given by the similitude laws was vividly illustrated for me when I participated in a troubleshooting session with a "sick" five stage ejector. It had very little air-handling capacity, and when we tested the last stage at shutoff we measured 15 torr! Wow! The ejector engineer announced that we had to check the steam to that stage. The nozzle was not fouled, and it was the right size. A pressure gauge indicated about 60 psig instead of the 175 psig everywhere else! Working back upstream, we disassembled the steam piping until we found that the strainer in that line had a sintered metal element instead of the low-pressure-drop perforate basket element specified.

How could it have worked at all against atmospheric pressure? It did not! The last stage discharged through a pipe sealed in the hotwell, and a flow of water was injected to condense the steam. This combination entrained enough air that it acted as a low-efficiency water-driven ejector, maintaining a low discharge pressure for the last stage only while the air load was nearly zero. The construction people had red faces, the ejector engineer was a hero, and I learned another tidbit about these fascinating ejectors.

4.14 Nomenclature

A	flow cross section area, ft^2
C_f	flow coefficient
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
D	diameter, in
e	overall ejector efficiency factor
E_d	diffuser efficiency
E_m	mixing efficiency
E_n	nozzle efficiency
g_c	units conversion factor, 32.2 (ft·lbm)/(lbf·s^2)
gpm	gallons (U.S.) per minute
H	energy head, or enthalpy: ft·lbf/lbm (or BTU/lbm)
k	specific heat ratio = c_p / c_v
M	multiplier for steam usage, identified by subscript
M_{sc}	multiplier for subcritical flow in a nozzle
n	mass molar quantity, lbm/molecular weight Example: 36 lbm water is 36/18 = 2.0 lbm·moles
P	pressure, psf
P_c	critical pressure, psf
P_r	reduced pressure = P / P_c
psi	pressure unit, pounds force per square inch pressure
psf	unit, pounds force per square foot
Q	volumetric flow, gpm
pph	lbm/hr
R_a	lbm motive steam per lbm load air
R_g	universal gas constant = 1546 (ft·lbf)/(lbm·mole·°R)
R_e	mass flow entrainment ratio: load/motive
R_h	operating head ratio: motive/discharge

SG	specific gravity, water = 1.0
T	temperature, °R (Rankine)
T_c	critical temperature, °R (Rankine)
T_r	reduced temperature = T/T_c
U	internal energy, ft□lbf/lbm (or BTU/lbm)
v	specific volume = $1/w$, ft^3/lbm
V	velocity, fps
w	mass density, lbm/ft^3; water = 62.4
W	mass flow, lbm/s
Z	compressibility factor for equation of state, dimensionless

Subscripts

d	discharge
m	motive
$P_{d\text{-}0}$	discharge pressure at no-load same as at design
P_m/P_d	measure of closeness to motive pressure
s	suction
size	stage size, as steam mass flow
stab	stability at no load
1	motive inlet
2	suction inlet
3	nozzle exit
4	diffuser throat
5	diffuser discharge

4.15 References

1. Alex M. Jumpeter, "Jet Pumps," Sec. 4 in Igor J. Karassik, et al. (eds.), *Pump Handbook,* McGraw-Hill, New York, 1976.
2. *Standards for Steam Jet Vacuum Systems,* 4th ed., Heat Exchange Institute, Inc., Cleveland, Ohio, 1988.
3. Ascher H. Shapiro, *The Dynamics and Thermodynamics of Compressible Fluid Flow*, vol. 1, Ronald Press, New York, 1953.
4. "Vacuum/Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd, Bolton, England, Jan. 1989.
5. Siegfried H. Hasinger, "Performance Characteristics of Ejector Devices," Wright Patterson Aerospace Research Laboratories, ARL TR 75-0205, June 1975.
6. A. Edgar Kroll, "The Design of Jet Pumps," *Chemical Engineering Progress*, 1 (2): 21-24, 1947.
7. W. C. Holton, "Effect of Molecular Weight of Entrained Fluid on the Performance of Steam Jet Ejectors," *ASME Transactions,* 73 (7): 905-910, 1951.
8. W. C. Holton and E. J. Schulz, "Effect of Temperature of Entrained Fluid on the Performance of Steam Jet Ejectors," *ASME Transactions*, 73 (7): 911-913, 1951.
9. "*Steam Jet Ejector Systems,*" Bulletin No. 771-A, Nash-Kinema, August 1990.
10. L. S. Harris and A. S. Fischer, "Characteristics of the Steam-Jet Vacuum Pump," ASME paper 63-WA-132, and *Transactions of the ASME, Journal of Engineering for Industry,* November 1964, pp. 358-364.

10. L. S. Harris and A. S. Fischer, "Characteristics of the Steam-Jet Vacuum Pump," ASME paper 63-WA-132, and *Transactions of the ASME, Journal of Engineering for Industry,* November 1964, pp. 358-364.
11. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation,* McGraw-Hill, New York, 1986.
12. R. B. Power, "Steam-Jet Air Ejectors: Specification, Evaluation, and Operation," ASME Paper 63-WA-143. Reprinted by *Hydrocarbon Processing and Petroleum Refiner,* 43 (2): 121ff and 43 (3): 138ff, 1964.
13. R. B. Power, "Steam-Jet Air Ejectors," 10-part series, *Oil and Gas Equipment,* October 1965 through July 1966.
14. V. V. Fondrk, "Figure What an Ejector Will Cost," *Petroleum Refiner,* December 1958, pp 101-105.

Chapter

5

Condensers:

Engineering Calculations

5.1 Overview of Condenser Calculations

A condenser is a heat-transfer device which cools a vapor stream to cause all or part of the vapors to change to the liquid state, or condense. When we first learn about condensers, we usually think of that primary task and do not concern ourselves with the presence of noncondensable gases. We tend to think of the device as a "total" condenser. However, we are now looking at the necessity of using a vacuum pumping device to remove the noncondensable gases from the condenser vent and discard them to the atmosphere. Therefore, we must look again at the function of a condenser, first to define more carefully just what its task is, and then to look more carefully at how it performs that task.

The first objective of this chapter is to review some of the basics of condenser theory to give you insight into how condensers work and how they behave. The next step is to give you enough quantitative information to estimate total vapor loads to ejector stages and approximate the size and water consumption of condensers. This sizing information is not adequate for selecting or designing condensers. Detailed thermal, fluid flow, and mechanical design of condensers is far beyond the scope of this book.

If the objectives of this chapter are met, you will be in a position to specify, install, and operate ejector condensers with the same care you apply to large process heat exchangers. You will also be aware of several of the many considerations which may become important in a given application. Much of the manufacturer's design work, however, uses relatively simple correlations based on experience and adapted to the practical requirements of commercial designs.

Condensers are used with ejectors to reduce the vapor load handled by ejector stages and to reduce the vapors discharged to the atmos phere. Condensers come in two types, contact and surface. It is appropriate to think of ejector condensers as partial condensers because the noncondensable gases present are an important design consideration.

To calculate the vapor load leaving each condenser vent, it is necessary to calculate the amount of condensable vapor in equilibrium with

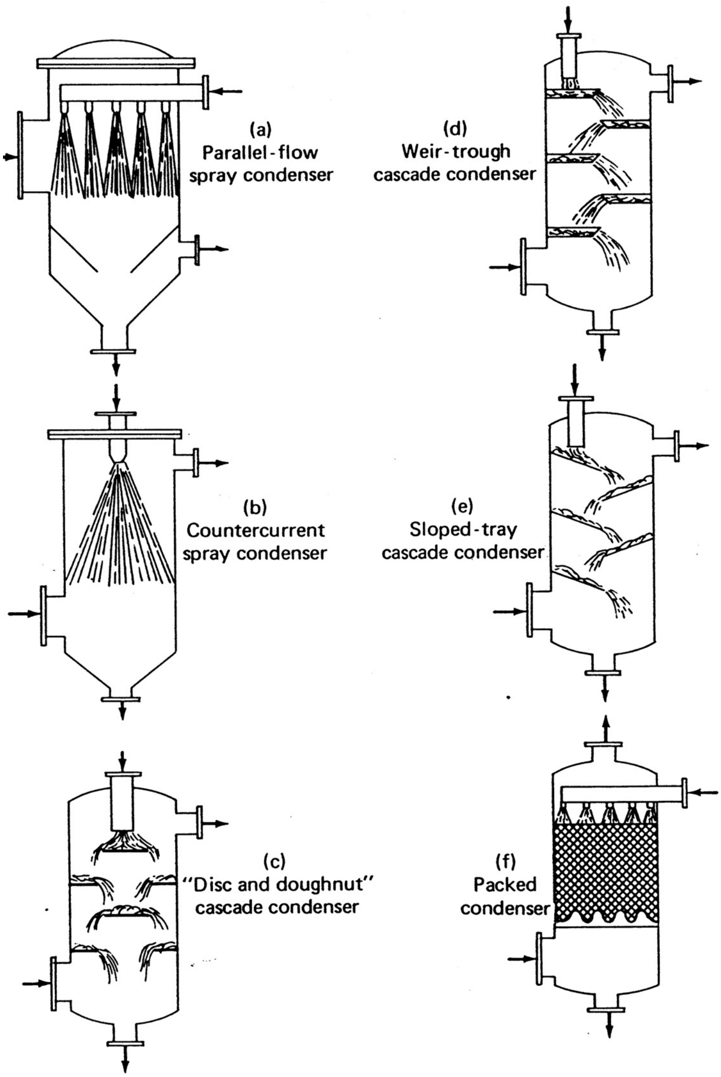


Figure 5.1 Direct contact condensers: (a) parallel-flow spray condenser; (b) countercurrent spray condenser; (c) "disc and doughnut" cascade condenser; (d) weir-trough cascade condenser; (e) sloped-tray cascade condenser; (f) packed condenser.) *(From J. L. Ryans and D. L. Roper, Process Vacuum System Design and Operation, McGraw-Hill, New York, 1986*

the liquid at the vent location in the presence of the noncondensable gases. The subject of vapor-liquid equilibrium is so important to an understanding of how condensing ejector systems work that I will discuss it before looking in detail at the condenser hardware. To conclude this overview, I will briefly describe the two classes of condensers.

A simple contact condenser is shown in Fig. 5.1*b*. It is a vertical cylindrical vessel with a single downward-pointing cooling-water full-cone spray nozzle at the top center and a vapor inlet nozzle at the lower side. The rising vapors are cooled by the falling water, and some of the vapors condense to the liquid state. Liquid condensate from the vapor and steam mixes with the cooling water and exits through a drain. Cooled air and some uncondensed vapors exit through a small vent on the side near the top. The other configurations will be discussed later.

Surface condensers are usually of the shell-and-tube design, typically oriented horizontally with condensing vapor in the shell and cooling water in the tubes, as .shown in Fig. 5.2. Many design variations exist to deal with the important considerations of first cost, vent gas cooling, corrosion resistance, ease of maintenance, water economy, space limitations, operating convenience, etc.

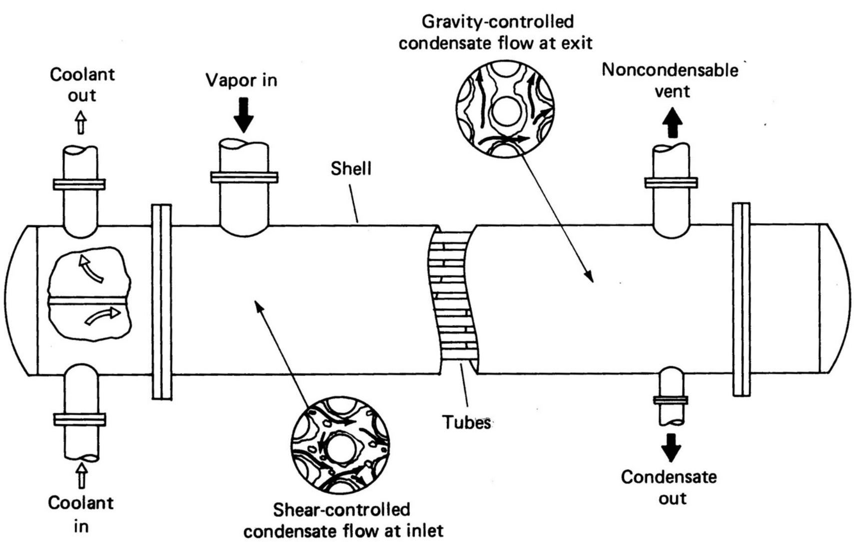


Figure 5.2 Surface condenser. *(From J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986)*

5.2 Vapor-Liquid Equilibrium

This discussion of vapor-liquid equilibrium as it relates to ejectors begins with an extension of the ideal gas laws reviewed in Chap. 4. It continues with a review of the subjects of vapor pressure, ideal solutions, nonideal solutions, and techniques for performing the calculations or substituting simplified conservative methods for very complex situations. Remember that the ideal gas laws and ideal solution concepts are never found exactly in nature. They are useful in many circumstances and often lead to comfortably accurate predictions about how systems will behave. It is also true that they are occasionally misapplied in circumstances where the user has no basis for believing them to be adequate.

An attitude of optimism will not change nature's laws.

Partial pressure

The concept of "partial pressure" is helpful in analyzing the behavior of vapors in the presence of liquids. In a container filled with air and water vapor at 100 torr, we would state that the total pressure is 100 torr. If 30 percent of the molecules were water vapor, we would state that the partial pressure of water vapor is 30 torr. By subtraction, we find the air partial pressure to be 70 torr. The ideal gas laws tell us that at low pressures and moderate temperatures and with gases which do not tend to react with each other, the total system pressure is equal to the sum of the partial pressures of the component gases (Dalton's law). The partial pressure of each component is equal to the total pressure (as measured by a pressure gauge) multiplied by the mole fraction of that component in the gas mixture:

$$p_i = y_i P \tag{5.1}$$

where p_i = partial pressure of ith component
y_i = mole fraction of ith component in gas mixture
P = total system pressure, usually in torr

As an example of how we calculate the mole fractions, consider a situation in which 10.8 lb of water vapor (MW = 18) is mixed with 40.6 lb of air (MW = 29):

Moles water vapor = 10.8/18 = 0.6 lbm • moles

Moles air = 40.6/29 = 1.4 lbm • moles

Total moles = 2.0 lbm • moles

Mole fraction water vapor = 0.6/2.0 = 0.3

Mole fraction air = 1.4/2.0 = 0.7

We now wish to relate this to the conditions in the liquid phase, for which we start with the ideal solution relationship (Raoult's law):

$$p_i = x_i P_i^o \tag{5.2}$$

where x_i = mole fraction of *i*th component in liquid phase
P_i^o = vapor pressure of pure *i*th component

The liquid-phase mole fraction is defined and calculated the same way as the gas-phase fraction. We now must define the vapor pressure for a pure component.

Vapor pressure

Consider a container partially filled with liquid and supplied with an absolute-pressure gauge and thermometer. Heat the liquid until it boils and drives out all the air from the container. Remove the heat and close off the vent to prevent gases from reentering the container. If the container is well insulated, the temperature in it will become everywhere equal and the liquid and vapor phases will be in equilibrium. The pressure gauge in the vapor space will indicate the vapor pressure (absolute) at the corresponding temperature. If the system is heated to a higher temperature and allowed to stabilize at the new condition, a second set of vapor-pressure/temperature data is obtained. By repeating the test over a range of temperatures, a complete set of vapor-pressure data can be obtained for that pure material. The vapor-pressure data can be organized in the form of tables, plotted curves, or equations.

Vapor-pressure curves. Figure 5.3 gives the vapor pressure of several liquids over a range of temperatures and pressures of interest in the design of ejector condensers. It is plotted as log-pressure versus reciprocal absolute temperature, although for convenience the scales are labeled as torr for pressure and degrees Fahrenheit for temperature. This method of plotting produces nearly straight lines for many materials over a useful range and plot is useful for interpolating and extrapolating (carefully!) using two data points in the range of interest for condenser design, say 20 to 50°C (68 to 122°F). If you wish to prepare your own plots, you may add them to Fig. 5.3 or the blank form in the Appendix, or you may wish to expand a portion to cover the

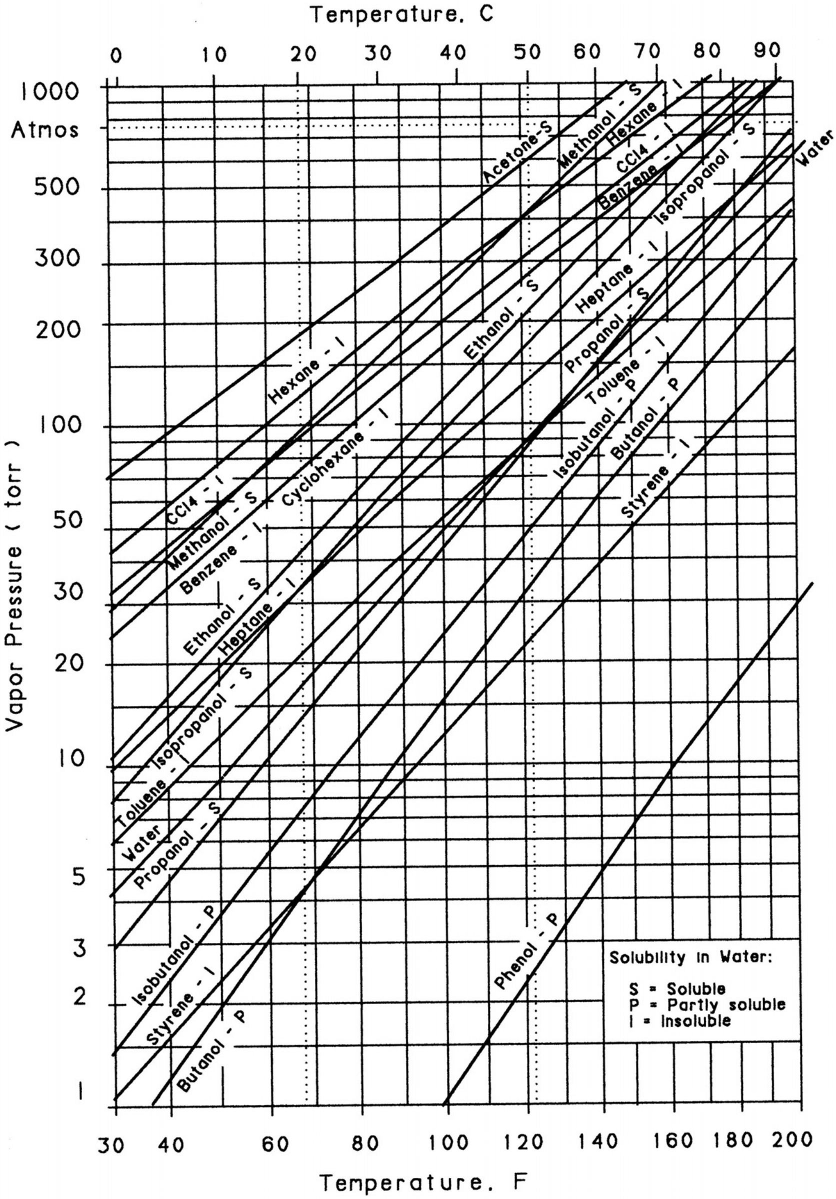


Figure 5.3 Vapor pressure vs. temperature.

pressure/temperature range of greatest interest to you. Note that if the temperature scale is labeled with temperatures increasing to the right (the "natural" direction), then the reciprocal values will be increasing to the left. A little thinking about this will clear up possible confusion.

Vapor-pressure tables. Vapor-pressure data are often tabulated along with other data on physical and thermodynamic properties for frequently used process liquids, the steam tables [1] being a common example. Many thermodynamics texts and engineering handbooks have such tables. Vapor-pressure data are commonly required for the process design of industrial plants; thus the data are usually available when required for estimating or specifying ejector requirements.

Here are a few low-vapor-pressure liquids commonly used at the low end of the rough vacuum range, with the approximate vapor pressure and specific gravity of each at 25°C (77°F):

	Vapor pressure, torr	Specific gravity
Mercury	2×10^{-3}	13.6
Butyl phthalate	4×10^{-5}	1.050
Convoil 20	5×10^{-7}	0.86
Octoil-S	4×10^{-8}	0.9122

Vapor pressure, equations. Although vapor-pressure curves or tables are often adequate when doing hand calculations, it is more convenient to use equations for computer calculations. A straight line on the log-pressure versus reciprocal temperature diagram can be represented by a two-parameter empirical equation. A three-parameter fit extends the accurate range when representing a curve. The Antoine model is such an equation, having the form

$$\log_{10} P^o = A - \frac{B}{C+T} \tag{5.3}$$

where three pairs of po and T may be used to define A, B, and C for that material over that range and for that specific set of pressure and temperature scales. There is considerable uncertainty in a set of Antoine constants unless all the defining conditions are specified. It is not prudent to assume that the reader will know the defining conditions because no such standard is universally recognized.

Air-water mixtures

Now that the vapor pressure of pure compounds has been discussed, we are ready to continue with a review of the vapor-liquid equilibrium conditions for an ideal solution. Let us start with the common situation in which air and water are the only two materials involved. For our immediate purpose, we can neglect the small solubility of air in liquid water. Thus water is the only material in the liquid phase, so its mole fraction there is 1.0. The partial pressure of water vapor is thus equal to the vapor pressure of water. We now have the methods for calculating the vapor load emerging from a condenser vent, given the vent temperature, the total pressure, and the airflow rate.

As an example, assume that the basic air load is 20 pph, and the condenser vent pressure and temperature are 60 torr and 100°F. Consulting a vapor-pressure curve or table, we find a value of about 49 torr at 100°F. This, then, is the partial pressure of water vapor over the water phase, leaving 11 torr as the resultant partial pressure of air. As a learning exercise we may first calculate the quantity of water vapor accompanying 1 lb of air. The water vapor and air mole fractions will be in proportion to their partial pressures, leading to

$$\begin{aligned} R_{wv/air} &= \frac{\text{(water vapor pressure)(water MW)}}{\text{(air partial pressure)(air MW)}} \\ &= \frac{(49)(18)}{(11)(29)} = 2.76 \text{ lbm water vapor/lbm air} \end{aligned} \tag{5.4}$$

Results of this type of calculation are shown in Fig. 5.4 for the range of pressures and temperatures of interest for ejectors. Note how

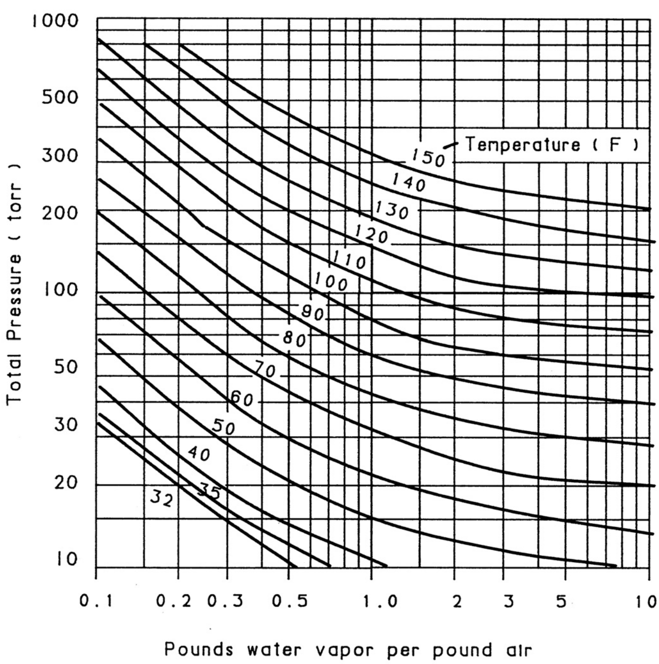


Figure 5.4 Pounds of water vapor per pound of air.

sensitive the water/air ratio is to small changes in temperature and total pressure when it is greater than 1.0. For example, if we decrease the total pressure by 1 torr to 59 torr and raise the vent temperature by 1° to 101°F, the water/air ratio increases more than 30 percent:

$$R_{wv/air} = \frac{(50.6)(18)}{(8.4)(29)} = 3.74 \text{ lbm water vapor/lbm air}$$

Ideal multicomponent solutions

When more than one material is present in the liquid phase, the vapor-liquid equilibrium calculations become more complicated, and an iterative method must be used. See Fig. 5.5 for an example calculation using one such iterative method. A blank worksheet form can be found in the Appendix. The example has four components: air, water, and two labeled arbitrarily as A and B. The mass units are converted to molar units for the iteration calculations, and the results are converted back to mass units for calculating the load to the next ejector stage.

The starting input L/V (liquid/vapor) value is arbitrary; you simply follow the example and calculate an output L/V value. The example converges faster if you double the indicated change. For example, an input L/V of 1.0 yielded an output of 2.2, with an indicated change of plus 1.2. I doubled the change and used a rounded value of 3.0 as the next input. Note that if I had followed my own rule, I would have used 5.0 as my next input and would have saved one iteration. Plotting output versus input L/V may help you pick a better third input value to speed convergence. Note also that if I had used my experience better, I would have looked at the data and observed that at 200 torr and 100°F almost all the steam will condense, along with a lot of soluble vapors. Following that logic, I probably would have picked 5.0 as a starting value and saved some time.

This does not take much time for occasional estimates, but is a little tedious for repeated calculations for estimates, quotation reviews, or design optimizations. A person who does a lot of this may wish to use a computer spreadsheet program, obtain an existing program which does it, or write his or her own computer program.

Of course, it is also true that a person who performs these tasks repeatedly begins to make much better estimates of starting values and lays the work out for efficiency. For example, in optimizing the interstage pressures for a multistage ejector with two intercondensers, I would fill in part of a master worksheet for the job, labeling components and giving their molecular weights, plus adding job data. I would then make several copies of the master sheet and proceed. If the vent

VAPOR-LIQUID EQUILIBRIUM CALCULATIONS

BASES: Ideal solution (Raoult's law), ideal gas (Dalton's law)

$$(p_i = x_i P_i^o , \quad P = p_1 + p_2 + \ldots)$$

Total pressure, P 200 Torr

Temperature 100 F ~~+ C~~

Steam, TERs 0.99

Other gases, TERg 0.99

Component	w PPH	MW	n	P^o Torr	$K = P^o/P$	* v1	v2	v3	v4	v5	Vapor PPH
input (L/V)						1.0	3.0	6.0	5.0	5.1	
Water Vapor	90	18	5.01	49	0.245	0.984	0.378	0.196	0.234	0.229	Ws 4.1
Other Gases and Vapors											
Air + ~~NC~~	20	29	0.69	Inf	Inf	0.69	0.69	0.69	0.69	0.69	20
A	30	60	0.50	300	1.50	0.30	0.167	0.10	0.115	0.114	6.8
B	5	50	0.10	20	0.10	0.009	0.003	0.0016	0.002	0.0019	0.1
Totals	W 145		N 6.29		V	1.98	1.24	0.99	1.04	1.035	Wg 26.9
					N − V = L	4.31	5.05	5.30	5.25	5.26	Vg 0.806
					output (L/V)	2.2	4.1	5.37	5.05	5.08 OK	MWg ____
											MWERg ____

* calculation of vapor moles: $v = \dfrac{n}{\dfrac{(L/V)}{K} + 1}$

Example, $v_A = \dfrac{0.50}{\dfrac{1.0}{1.50}+1} = 0.30$

Dry Air Equivalent of vapor load

Water vapor: $\dfrac{4.1 \text{ PPH}}{0.8 \text{ MWERs } (0.99 \text{ TERs})} = 5.2$ PPH DAE

Other gases: $\dfrac{26.9 \text{ PPH}}{1.05 \text{ MWERg } (0.99 \text{ TERg})} = 25.9$ PPH DAE

Total 31.1 PPH DAE

Figure 5.5 Worksheet calculation of vapor-liquid equilibrium for ideal solutions.

temperature for several calculations is the same, then the vapor pressures will be the same. Also, all calculations for the first intercondenser will have all mass flows except water unchanged. If you infer that I am trying to persuade you to do these calculations, you are correct.

Nonideal solutions

An improvement on the ideal solution is Henry's law, which observes that for many materials the partial pressure over a dilute solution is equal to a constant (Henry's) multiplied by the mole concentration. Thus, Henry's constant may be substituted for the vapor pressure of the pure material in the calculations. Another approach to describing that nonideality is to describe the material as being more (or less) active than if it were ideal, introducing an activity coefficient, γ. Note that for an ideal solution, γ is 1.0. In summary,

$$p_i = x_i \gamma_i P_i \quad \text{(Henry's constant.)} \quad (5.5)$$

γ varies with the kinds and concentrations of materials in solution, and with temperature. For a single material mixed with water, you may describe its behavior by plotting or tabulating it for a range of temperatures and concentrations. Another approach is to plot the partial pressure versus temperature curves at constant concentration for a range of concentrations useful in designing ejector condensers. One factor which may simplify this task a little is that the mole concentration is often well below 50 percent. A quick estimate of the relative quantities of water present for any specific situation may simplify this task.

Perry [4] and Ryans and Roper [7] offer more extensive help for phase equilibrium calculations with nonideal solutions.

Some materials have such large activity coefficients that very little of the material will condense with water. A conservative way to handle this is to treat the material as immiscible with water, as described in the next topic, or to treat it as a noncondensable gas.

Immiscible liquid mixtures. Oil and water do not mix. That phrase is used in a variety of situations and is a phenomenon experienced by us all. These materials are said to be immiscible. When we shake a bottle of oil and vinegar and watch it start to separate into distinctly different liquid phases, we might expect that the vapor-liquid equilibrium calculations would be a little different than for "nicer" mixtures. They are.

Pause briefly to note that immiscibility is another ideal condition which never exists. Everything will dissolve a little bit in anything else. The concept is useful because it leads us to accurate results using simple calculation methods. For ejectors it usually produces slightly conservative results, erring in the direction of slightly larger ejector stages.

The behavior of an "oil and water" mixture can be understood more easily if we mentally violate some laws of nature and visualize the two liquid phases as existing side by side in a container with a vertical partition which separates the two liquid phases and extends only partly into the vapor space. Each liquid phase will interact with the vapor phase as if the other liquid phase were not present. This is true regardless of the relative quantity in each liquid phase, as long as there is at least one drop of liquid in each phase.

An example calculation will show how this is handled. Let the load mixture be 90 pph water vapor, 20 pph air, and 40 pph of vapor *A* with an MW of 80. The pressure is 200 torr, and the temperature 100°F. The water vapor pressure is 49 torr, and *A* has a vapor pressure of 60 torr. Assume that both liquid phases exist and check the assumption later. The sum of the partial pressures over the two liquid phases is 49 + 60 = 109 torr, leaving the air partial pressure as 91 torr. The vapor composition is

$$\text{Vapor } A = 20\left(\frac{80}{29}\right)\left(\frac{60}{91}\right) = 36.4 \text{ pph} \quad \text{therefore some condensed}$$

$$\text{Water vapor} = 20\left(\frac{18}{29}\right)\left(\frac{49}{91}\right) = 6.7 \text{ pph} \quad \text{therefore some condensed}$$

Checking the original assumption, the quantity of each vapor is less than the total quantity of that material present, indicating that some of each material is present as a liquid phase.

As a demonstration of one of the occasional errors made in designing ejector systems, consider the effect of changing the condensing pressure to 150 torr with other factors unchanged. We would calculate the air partial pressure to be only 41 torr. Testing our assumption that two liquid phases are present,

$$\text{Vapor } A = 20\left(\frac{80}{29}\right)\left(\frac{60}{41}\right) = 80.7 \text{ pph}$$

which exceeds the total quantity present. Therefore, no *A* liquid phase can exist here!

Now what do we do? In this situation *A* behaves like a noncondensable gas, so we combine it with the air to obtain 20/29 + 40/80 = 1.19 moles total noncondensables with a combined partial pressure of 150 - 49 = 101 torr. The corresponding water vapor flow is

$$\text{Water vapor} = 1.19\,(18)\left(\frac{49}{101}\right) = 10.4 \text{ pph} \quad \text{therefore some condensed}$$

The total load to the next ejector stage is 20 + 40 + 10.4 = 70.4 pph.

Two types of errors can be made easily in this situation. First, if the user fails to identify the immiscibility of material *A*, it will be treated as an ideal solution and the calculations will show most of it condensing with the water vapor. The load to the next stage will be calculated as 20 pph air and less than 10 pph combined water vapor and *A*, for a total load of about 30 pph. This ejector stage will have less than half the required capacity.

Second, the user could simply describe material *A* as "condensable," leaving the interpretation to the ejector designer. The designer might interpret "condensable" to mean soluble in 'water and with very low vapor pressure, so that it dropped out completely with water in the first condenser. The next ejector stage would be designed for about 26 pph total load, making it even more undersized than in the previous example.

Another way an ejector designer can handle the ambiguity in the word "condensable" is to be protective of the user and prepare a conservative design in which subsequent stages are sized by treating the material as a noncondensable gas. The ejector-system design will use more steam and will probably have a higher purchase price. The user who buys and operates it will be committed to a higher operating cost for the life of the project. He or she might benefit slightly from an improved ability to handle air leaks, depending upon other design factors in the system.

The user looking at several competing bids for this ejector wants to know which one to accept. With an eye on steam economy, he or she will not buy the conservative design. What about the bid that requires less steam to do the job? Does it look good because it contains one of the errors described above? Only a calculation to check the reasonableness will answer that question.

Chapter 10 deals at length with specifications and bid evaluations, but it is appropriate for me to alert you here to one problem you may face. The general solution is to think through your specifications carefully, communicate them accurately and completely to the ejector manufacturer, and critically review the quotations you receive to avoid buying an error that will hurt both you and the manufacturer. Nobody wins in a bad job. The manufacturer will lose money, and your plant losses or extra costs may easily be 10 or 100 times as large.

Multicomponent immiscible liquid mixtures. What if a mixture of several immiscibles is present in the load? If the liquid phase which is immiscible with water behaves within itself as an ideal mixture or one in which vapor pressures can be replaced by Henry's law constants for *V-L* calculations, the worksheet of Fig. 5.5 may be used with a slightly modified procedure. First, determine the water vapor pressure and subtract it from the total pressure to arrive at the partial pressure representing the sum of the partial pressures of all other components. To the immiscible phase and noncondensables, that is the effective total pressure they see.

VAPOR-LIQUID EQUILIBRIUM CALCULATIONS FOR IMMISCIBLES + WATER

BASES: Ideal solution (Raoult's law), ideal gas (Dalton's law)

$$(p_i = x_i P_i^o , \quad P = p_1 + p_2 + \dots)$$

IMMISCIBLE Total pressure, P 281 Torr

P_{TOTAL} = 330 TORR
P^o_{WV} = 49
IMM.PP = 281

Temperature 100 F/~~C~~
Steam, TERs 0.99
Other gases, TERg 0.99

input (L/V) FOR IMMISCIBLES: 1.0 | 1.3 | 1.38

Component	w PPH	MW	n	P^o Torr	K = P^o/P	* v1	v2	v3	v4	v5	Vapor PPH
Water Vapor		18									Ws
Other Gases and Vapors											
Air + ~~NC~~	15	29	0.517	Inf	Inf	0.517	0.517	0.517			15
A	120	100	1.2	15	0.0534	0.0608	0.0473	0.0447			4.5
B	30	120	0.25	25	0.0890	0.0204	0.0160	0.0151			1.8
C	12	120	0.10	10	0.0356	0.0034	0.0027	0.0025			0.3
D	120	60	2.0	500	1.779	1.2803	1.1556	1.1263			67.6

Totals W 297 N 4.07

	v1	v2	v3	v4	v5	
V	1.88	1.74	1.706			Wg 89.2
N − V = L	2.19	2.33	2.364			Vg 1.706
output (L/V)	1.16	1.34	1.387			MWg 52
			OK			MWERg 1.26

* calculation of vapor moles: $v = \dfrac{n}{\dfrac{(L/V)}{K} + 1}$

AIR PP = $\frac{0.517}{1.706}(281)$ = 85 TORR
WATR VAPOR, BASE ON AIR
$W_s \approx 15\left(\frac{18}{29}\right)\left(\frac{49}{85}\right) = 5.4$ PPH

Dry Air Equivalent of vapor load

Water vapor: $\dfrac{5.4 \text{ PPH}}{0.8 \text{ MWERs } (0.99 \text{ TERs})}$ = 6.8 PPH DAE

Other gases: $\dfrac{89.2 \text{ PPH}}{1.26 \text{ MWERg } (0.99 \text{ TERg})}$ = 71.5 PPH DAE

Total 78.3 PPH DAE

Figure 5.6 Worksheet calculation of vapor-liquid equilibrium for immiscible solutions.

As a worked example containing some odd data, consider the worksheet calculations in Fig. 5.6. Remember, the results are valid only to the extent that the solution is ideal or is one in which Henry's law constants effectively replace vapor pressures. Note that only about half of the component with high vapor pressure condensed. It would have been overly conservative to have treated it as a noncondensable gas for the purpose of sizing the next stage.

Condenser vent gas must be cooled

The temperature of the gases emerging from a condenser vent must be lower than the dew point of the gases entering the condenser. Otherwise, nothing will have condensed. As an example of inadequate vent gas cooling, consider a steam condenser with a vent pressure of 50 torr and a vent temperature of 100°F. The water vapor at that temperature is 49 torr, leaving only 1 torr for the air partial pressure. As a result, each mole of air leaving the vent will be accompanied by 49 moles of water vapor. That translates to 30 lb of water vapor with each pound of air.

If the condenser is redesigned or supplied with cooler water to cool the vent gas to 95°F, then the water vapor pressure will drop to 42 torr, the air partial pressure will rise to 8 torr, and now only 3.3 lb of water vapor accompany each pound of air. This example of the benefit of gas cooling shows why this is such an important subject in the design of condensers for ejector systems.

When the vent from a process vacuum condenser contains a large quantity of condensable vapors with the air, it may be economical to place another condenser between the two systems to reduce the load to the ejector as described below.

Vent condensers and precondensers

These two terms are used interchangeably to describe a partial condenser which is placed between a process condenser and an ejector to reduce the vapor load to the ejector. The two terms seem to reflect two different points of view. If the person responsible for the process condenser design decides to add a vent condenser as an effective way to reduce vapor losses, then it is simply considered an extra condenser added to the vent. If the condenser is added in connection with the design and procurement of the ejector system, then it is considered a precondenser to that system. If you are responsible for both systems, you cannot go wrong.

Here are several reasons for considering the use of a precondenser. It reduces the amount of vapor which the ejector must compress, thus saving steam, cooling water, and ejector purchase price. Valuable vapor that is condensed and recovered before it reaches the ejector and contacts steam may be recovered at less cost than if it had to be separated from the steam condensate in the ejector system. If it is not economical to separate the process condensate from water, then it may have to be sent to a waste treatment system, where every extra pound of process material incurs an extra processing cost. All of these cost considerations may be present in a design situation.

When the process vapors freeze and form solids at temperatures in the ejector system, special precondensers called sublimers are sometimes used to minimize fouling of the ejectors. These and other special precondensers are discussed in Chap. 11.

When a process condenser has an entering dew point only slightly higher than the cooling-water supply temperature, the vent gas temperature may be only a few degrees below the dew point, and the vent may contain a large condensable vapor load. A small vent condenser may be added to reduce the vapor load, using cooling water or a refrigerant. The smaller condenser is better at cooling the gas and subcooling the liquid condensate.

Occasionally it is desirable to use a vent condenser following an ejector aftercondenser to reduce product loss and air contamination at that location. Such a vent condenser is likely to use a refrigerant for cooling. Conditions which favor such a vent condenser are a large vapor load of high-vapor-pressure material or a condensable vapor which has a high activity coefficient or is immiscible with water. High recovery value and/or an obnoxious odor or other environmental penalties add extra incentives.

Condensing temperatures and vent temperatures

Warm vapor mixtures enter a condenser and are cooled as they encounter the cooling-water spray or the water-cooled tubes. Condensation begins when the temperature of the water spray or the tube wall surface is below the dew-point temperature. Once condensation begins, additional cooling will cause more liquid to condense with very little reduction in temperature if the concentration of noncondensable gases is low. Finally, the condensation will be nearly complete and the remaining mixture will have a higher concentration of noncondensable gases.

The temperature may drop significantly in this final gas cooling process. This final cooling process is designed to cool the gas mixture to remove all the vapor that it is economical to condense. The Appendix has a detailed description of the effect of vent gas temperature on the total operating cost of an ejector. A more detailed, quantitative treatment of the vapor-temperature profile will be given in the description of surface condensers later in this chapter.

Terminal (condensing) temperature difference. The temperature at which most of the condensation occurs (the dew point) is called the *condensing temperature* and is slightly below the saturation temperature of the dominant material (usually steam) being condensed. The difference between the condensing temperature and the cooling-water exit temperature is called the *terminal temperature difference* [5]. To avoid ambiguity, it is best to call this the *terminal condensing temperature difference.*

Most of the discussion in this chapter is applicable primarily to precondensers and intercondensers, where the vent conditions strongly affect the steam usage. Aftercondensers are relatively simple, having no effect on the steam usage. In aftercondensers there is typically a large difference between the dew point and the cooling-water temperature, and design objectives are to limit atmospheric pollution and avoid fouling on the water side from high metal temperatures.

The terminal condensing temperature difference in large, well-designed contact condensers with low concentrations of noncondensable gases is typically assumed to be 3 to 5°F above the outlet water temperature. One reference [2] splits this into 3, 4, and 5°F, respectively, for negligible, normal, and large amounts of noncondensable gases. Another [7] splits it into 3, 5, and 10°F based upon condenser diameter: greater than 21 in, 12 to 21 in, and less than 12 in, respectively.

For surface condensers the terminal condensing temperature difference is often higher, in part because the heat-transfer path now includes the tube and its two fouling layers, and in part because the heat-transfer area here is more expensive than the spray in a contact condenser. At least 6°F is common for surface condensers with low noncondensables [3]. The Appendix shows, however, that lower differences may be appropriate where steam cost is high.

Vent (approach) temperature difference. A variety of methods are used to estimate vent temperatures. The best method is to examine the design of each specific condenser or consult the condenser designer to determine how closely the vent temperature will approach the inlet or outlet temperature of the cooling water. To supplement that method and offer some general guidance, here are some rules of thumb.

The "approach" temperature difference between the vent and the cooling-water inlet may be as low as 3°F and is usually not greater than 10°F [10]. The simplest rule is to allow an approach temperature difference of 5°F above the water inlet temperature for large contact condensers in which the noncondensable load is less than 2 percent. Thus, if condensing water is supplied at 80°F, the design vent temperature will be 85°F. The corresponding difference for surface condensers will typically be 5 to 20°F, strongly affected by the noncondensable load and by economic considerations. See the Appendix example.

Reference 7 gives the normal vent temperature as equal to the water outlet temperature for contact condensers less than 12 in diameter. For condensers 12 to 21 in, the normal vent temperature is 7°F above the water inlet temperature. For larger condensers, it is 5°F above the water inlet temperature.

Another rule is to allow an arbitrary amount of gas cooling below the condensing temperature (dew point). Chilton and Perry [4] offer 10°F for that rule. HEI [5] takes a different approach for vacuum equipment serving power plant condensers. As a reference temperature, it uses the saturation temperature of water at the design inlet pressure of the ejector and assumes that the temperature is 7.5°F lower. In effect, the ejector load is assumed to be cooled 7.5°F below the condensing temperature. That rule typically results in 2 to 3 lb of water vapor for each pound of air load.

I am not endorsing the guidelines in the preceding paragraph, I am merely reporting them and adding my opinion that they are not a substitute for an informed estimate based on knowledge of the specific condensers involved. Better performance (lower vent temperatures) can often be expected from well-designed condensers.

Note that the vent temperature from the aftercondenser determines the amount of product loss to the atmosphere per pound of air and other noncondensable gas vented. If this temperature is not specified by the user, the manufacturer will tend to use a high temperature to minimize the price, because it does not affect steam usage.

Pressure drops: approximate rules for contact and surface condensers

As a rough generalization, the pressure drop across a contact condenser is 5 percent of the absolute pressure. For surface condensers, it is 10 percent for process condensers and precondensers, 5 percent for intercondensers, and less than 5 percent for aftercondensers.

5.3 Contact Condensers

Contact condensers are sized based upon vapor velocities, water flow rates, noncondensable vapor load, absolute pressure, allowable pressure drop, and considerations of terminal and approach temperatures at the water outlet and inlet, respectively.

Figure 5.1 shows six variations in contact condenser designs. Figure 5.1*b* consists of a single spray nozzle in a vertical cylindrical vessel. It presents the water to the vapor as a full-cone spray through which the vapors must pass before the uncondensed gases exit through the vent. Some portions of the spray pattern are less effective than others, and a partial blockage of the nozzle may permit a significant amount of uncondensed vapor to bypass the condenser and overload the next ejector stage.

More complex contact condenser designs may have multiple spray nozzles to create a more uniform spray pattern, or they may have internal baffles which cause the water to cascade back and forth as a series of water curtains through which the rising vapors must pass. Nozzles or weirs at the top distribute the water. Properly designed and installed, these complex condensers are more effective heat exchangers than the smaller simple condensers. The reduced terminal condensing temperature difference and vent approach temperature differences may be accompanied by a higher pressure drop.

The cocurrent flow pattern in the Figure 5.1*a* condenser reduces its heat-transfer effectiveness, so the vent temperature will be higher and more uncondensed vapor will be present in the vent flow. Some special designs, variations of Fig. 5.1*a*, use larger quantities of water supplied at higher pressures to jet downward and entrain small amounts of noncondensable gases, eliminating the need for a subsequent vacuum pump if the noncondensable gas load is very small.

An air load source which must be considered in the design of contact condensers is dissolved air that is present in the cooling water. The air is released as the water is warmed and subjected to vacuum. See Fig. 5.7 for the amount of air dissolved in normal cooling water.

Properly designed contact condensers must be installed, operated, and maintained adequately to assure the proper flow of water to accomplish the heat transfer. A tilted cascade-type unit will have an off-center flow pattern which may allow bypassing, especially if the initial flow distribution is made by a weir. Low flows may create breaks in the water curtains, and high flows may create large pressure drops. Fouling of the nozzles and cascade geometry will also degrade the performance.

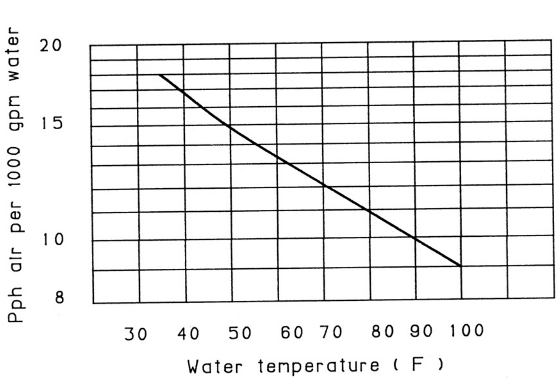


Figure 5.7 Air dissolved in cooling water. (*From* Mark's Standard Handbook for Mechanical Engineers, 8th ed., *McGraw-Hill, New York*)

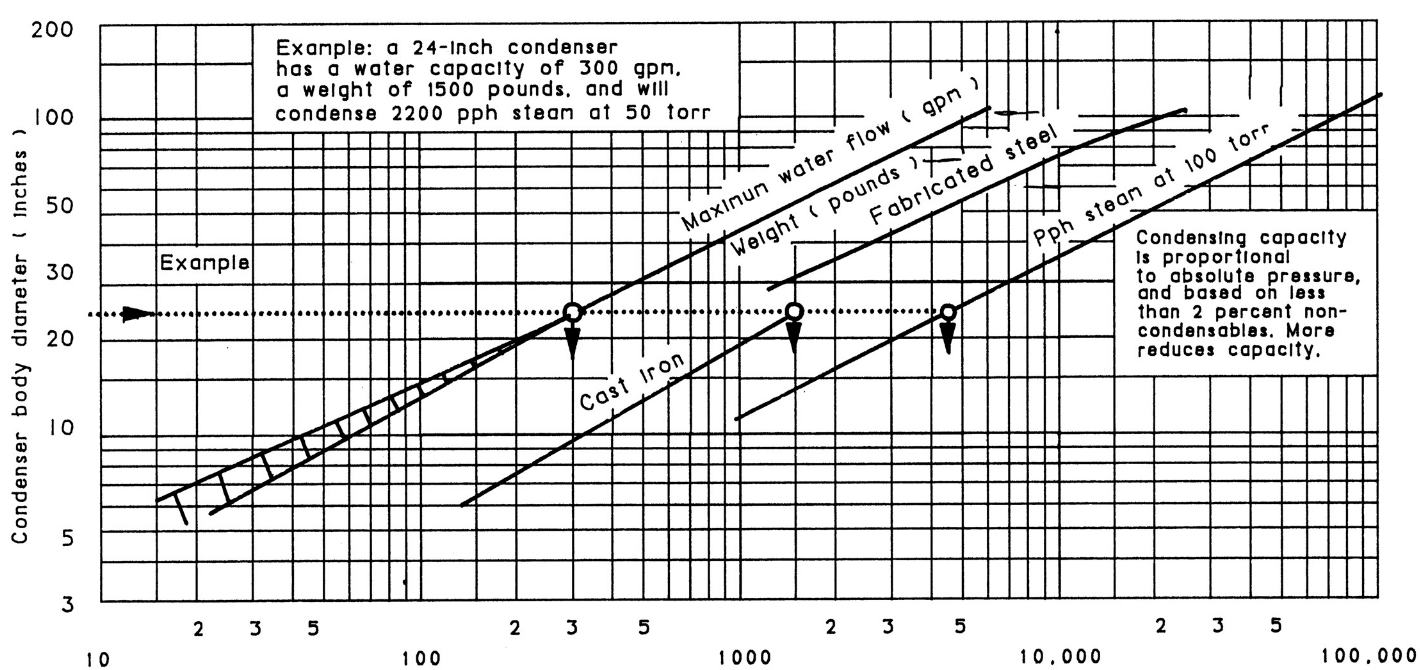


Figure 5-8 Contact condenser sizing

Sizing

The simple countercurrent spray condenser is sized based on water loading or vapor velocity, whichever controls. The water loading is about 80 gpm per square foot of cross section. For example, a 24-in-diameter condenser has a cross section of 3.14 ft^2, permitting a water flow of 3.14(80) = about 250 gpm. The allowable vapor velocity is about one-third the velocity of the discharge of an ejector stage, say 70 ft/s. The diameter will be roughly twice the diameter of the ejector discharging into it.

Figure 5.8 has curves for estimating size and weight. Note the qualification that the condensing capacity is based on less than 2 percent noncondensables in the load. Typical intercondensers are sized based on test data and will generally be larger than the size in this figure. If more than 2 percent of noncondensable gas is present, the diameter may have to be increased to provide more contact time and reduce the high vent velocities which would create excessive liquid entrainment in the vent.

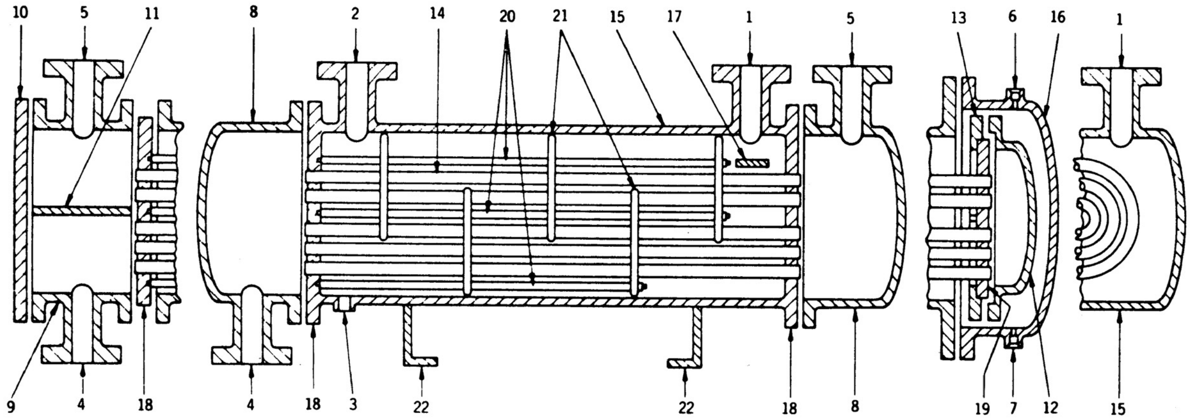
More complex condensers have internal flow control devices which control the flows of water and vapor. The space occupied by these devices must be compensated for by increasing the shell diameter.

5.4 Surface Condensers

The shell-and-tube type surface condenser commonly used with ejectors is shown in Fig. 5.9, which gives the nomenclature for the basic hardware features. Many users forbid tubes smaller than $^5/_8$ or $^3/_4$ in diameter in fouling services because they are difficult to clean.

Most steam-jet ejectors are used to remove air and other noncondensable gases from process condenser vents. Although motive steam is added to the mixture, the noncondensable vapor fraction is still large enough that the ejector system designer has to consider each ejector condenser to be a partial condenser. The designer has the choice of using a compromise design in one surface condenser body or (rarely) breaking the operation into two steps. The two-step design has a main condenser with only modest gas cooling, followed by a vent condenser designed for effective gas cooling. Some power plant condensers and ejector intercondensers are carefully designed to accomplish both tasks effectively in one special body.

Figure 5.10 is one example of a design which accomplishes both tasks in one body. The shell-side vapor inlet area has an open construction to enable the incoming vapor to spread out and penetrate the tube bundle at moderate velocity. The area approaching the vent has air baffles closely spaced to raise the velocity of the small vent stream for most effective gas cooling. In the alternate design of this



1. Gas vapor inlet
2. Gas vapor outlet
3. Condensate drain
4. Coolant inlet
5. Coolant outlet
6. Vent
7. Drain
8. Bonnet type front and rear head
9. Channel type front and rear head
10. Channel cover
11. Pass partition
12. Floating head bonnet
13. Floating head backing flange
14. Tube
15. Shell
16. Shell cover
17. Impingement plate
18. Tube sheet
19. Floating tube sheet
20. Tie rods and spacers
21. Transverse baffles or support plates
22. Support feet

Figure 5-9 Surface heat exchanger nomenclature. *(Courtesy Heat Exchange Institute)*

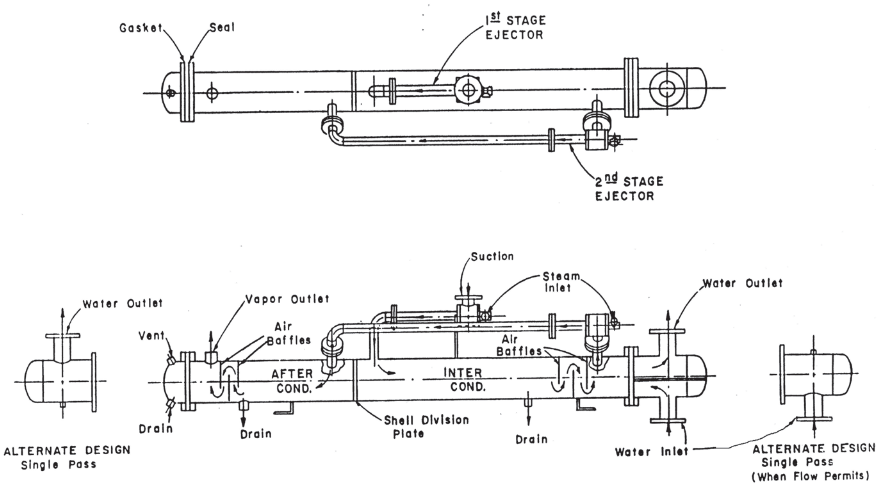


Figure 5.10 Surface condenser design for vent gas cooling, intercondenser and aftercondenser in one shell for low cost. *(Courtesy Graham Manufacturing Co.)*

example, the cooling water makes a single pass and the intercondenser vent is cooled by the coldest incoming water to reduce the load to the second stage. Other designs may have longitudinal shell-side baffles to keep warm condensate away from the exit gas stream. Note that this example happens to be an integrated design which has both intercondenser and aftercondenser in one shell as a cost reduction feature.

Configurations

Example surface condenser configurations given so far in this chapter have been the most common arrangement: horizontal, with vapor on the shell side and water on the tube side. Other configurations are common, and each has advantages and disadvantages. The two basic choices are the placement of the process vapor and the orientation of the condenser body. The tube bundle type is another important geometry factor.

Placement of fluids. The simplest rules for avoiding problems are: make water flow upward to be self-venting, make condensing vapor flow downward so the condensate is self-draining, and locate the vent outlet at the place most distant from the vapor inlet and cooled by the coolest entering water.

The interaction of many design factors makes it difficult to address each one separately. Here I will begin by describing an "ideal" horizontal configuration and an "ideal" vertical configuration. Then I will examine some variations on each configuration and mention some advantages and disadvantages of each.

I have my preferences, which will be evident here, and I have worked with people who have stronger preferences. Many designs exist which violate some of the simple design guidelines and would be rejected by some users and manufacturers. However, so many of the "violator" designs are apparently working OK that one cannot objectively make a flat statement that they will not work. The specific experience of each manufacturer permits it to design with confidence in some of these gray areas. Unfortunately, users often lack that experience and must form their own judgments of the manufacturer's competence and the risks they are willing to take.

Horizontal, vapor in shell. In this configuration the vapor usually enters the shell through a top nozzle at one end. The vapor flow and condensate flow are both downward, the condensate flow driven mostly by gravity. The shear between the high-velocity vapor and the liquid condensate helps move the condensate rapidly downward, thinning the liquid film on the tubes and increasing the heat-transfer action. As the uncondensed vapors turn and flow horizontally parallel to the tubes, they encounter flow baffles which direct the flow back and forth in a cross-flow pattern. These baffles become more closely spaced as the volume of uncondensed vapor decreases. The heat exchanger is primarily functioning as a condenser at the vapor inlet and as a gas cooler at the vapor outlet, with a gradual transition in function and geometry along the shell. Condensate drains out of a bottom nozzle near the middle of the shell, and the vent gas exits from a side or top nozzle at the end opposite the vapor entrance.

The tube side has cool water entering the bottom of the head at the end opposite the vapor inlet. If the design permits a single water pass,the water exits from a top nozzle in the head at the vapor inlet end. If the water makes multiple passes, the gas cooling zone may be baffled and shielded so that the gas contacts only the cool inlet water on the bottom portion of the shell, with the vent nozzle located in the side of the shell near the bottom. A horizontal partition may shield the gas cooling zone to prevent warm condensate from dropping into the cooled gas. If the tube bundle is removable, the partition may be sealed against the shell with flexible strips to reduce bypassing (this seldom works). In a large fixed-tube sheet design, this partition may be welded.

Vapor-liquid equilibrium calculations are complicated by the fact that the vent gases are in contact with only the last liquid condensate formed. Much of the steam will have condensed first and gone directly to the drain without the additional cooling experienced in the gas cooling zone.

An impingement plate must be located at the shell-side vapor inlet to avoid erosion damage from the ejector stage discharging into it. The tubes must be supported to resist vibrations caused by the highvelocity vapors flowing across them. The entire system will drain completely through the shell-side drain connection and the water inlet nozzle.

Vertical, vapor in tube. In this configuration the body of the condenser is vertical, the process vapors are in the tubes, and the water is on the shell side. The vapors enter the top head and flow downward inside the tubes. The liquid condensate forms a film on the inside of the tubes and flows downward under the influence of gravity and the shear from the downward gas flow. In the lower portion of the tubes, the liquid film is thickest, the gas velocity is lowest, and the heat-transfer performance is lowest.

This results in a larger total area compared to a horizontal condenser with the same terminal condensing temperature difference and vent approach temperature difference. If costly or very special materials such as graphite are required for the process side, the vapor-in-tube may be cheaper than the vapor-in-shell configuration. With costly tube materials, the designer may choose to accept a higher pressure drop and higher temperature differences to reduce the purchase price.

Because the water enters at the bottom of the shell and flows upward, an effective counterflow arrangement exists. This creates maximum subcooling of the liquid condensate. The condensate drops out of the tubes into the lower head, along with the cooled uncondensed gases. The lower head usually has a bottom drain and a side vent provided with a hood to minimize liquid entrainment. The vent gases leave nearly in equilibrium with the condensate.

The water side requires special attention in design and operation. The baffles must be properly spaced and fit to keep water velocities high enough to avoid bypassing and low-flow areas which collect silt. Efficient use of expensive tube area is often attained by using small cross-baffle windows and omitting tubes from the baffle windows. On large condensers the top tube sheet requires special venting of the space above the shell-side water outlet nozzle. An extra drain connection must be provided to drain the space below the water inlet nozzle.

Vertical, vapor in shell. I am probably unfairly prejudiced against this configuration, and I admit my limited knowledge of heat exchanger design. I have seen some variations of this configuration that have bothered me. The water is on the tube side and commonly is in a two-pass arrangement. Some U-bend designs orient the U-bend downward, where it cannot be drained and must be removed to prevent freezing damage during a cold-weather shutdown. If water velocities are sufficiently high, the water-side path will vent itself.

Another variation that bothers me is the arrangement in which the vapors flow upward on the shell side. I will not deny that these condensers seem to work, but the counterflow pattern between rising vapor and falling condensate seems to invite flooding and increased pressure drop. Also, for a mixture of volatile vapors, I would expect to see some rectification effect because of the resemblance to the flow pattern in a distillation column. This effect will make the vent contain more volatiles than would be calculated on the basis of a model in which the exit vapors are in equilibrium with the exit condensate. Such an error would not be detected by normal testing using air and water vapor as loads.

Vertical, vapor in tube, upflow. This configuration violates several guidelines and seems to have no merit for ejector condensers. I include it here only because the configuration is occasionally used in process applications, often for good reasons, and because of this a precedent might be adopted mistakenly for ejector use.

The vapor flows upward as the condensate film falls down, increasing the pressure drop and inviting flooding. The potential for flooding may be checked against flooding criteria [6]. Some rectification is to be expected, complicating the estimates of vapor in the vent gas. For maximum thermal effectiveness the water should flow downward to establish the desired counterflow temperature profile, but that causes a venting problem on the shell side. The top tube sheet must be vented above the water outlet nozzle.

Horizontal, vapor in tube. This is functionally similar to the vertical vapor-in-tube configuration. The primary difference is that gravity is now acting at right angles to the vapor flow direction. The condensate will run down the sides of the tubes and form a moderately deep "stream" on the bottom of each tube. The gas flow will tend to sweep the condensate along with it, but flooding and high pressure drop may result unless these units are installed with a slight tilt to permit gravity to aid the condensate removal. As little as 2° or 3° tilt is usually sufficient, but some users prefer a minimum of 15°. Water-side venting and draining are simple, with water entering the shell-side bottom at the vapor outlet end and leaving the top at the vapor inlet end.

Horizontal versus vertical. In addition to the functional performance considerations mentioned above, there are installation, operation, and maintenance considerations which will be discussed in later chapters.

Size

Although sizing a partial condenser is a complex task, most users are interested only in a rough estimate of size for making cost estimates or for preliminary layout planning. The methods given here will meet those needs and give you an idea of the task faced by the condenser designer. Perhaps your specifications will be more complete as a result of your heightened awareness.

Area is the most important single measure of the size of a heat exchanger. Although the methods to be described here will enable you to estimate the needed area for a condenser application, they do not begin to define the configuration of that area or the details of the vapor-side and water-side geometry that must be present to create the fluid flow conditions implicit in the correlations behind the estimates. You cannot simply prepare an estimate of the required area, then grab an available heat exchanger that has that area and expect it to work as a partial condenser. You might be lucky, but probably you won't be.

Q = UA ΔT. Having registered my warnings, I proceed. The Fourier equation (integrated, steady-state, overall form) describes how the area relates to other overall performance parameters:

$$Q = UA\,\Delta T \tag{5.6}$$

where Q = total heat load, BTU/h for example. It consists of sensible and latent heat loads, plus heats of solution and reaction. The heat is given up by load vapors and motive steam and is transferred to the cooling water.

U = overall heat-transfer coefficient, BTU/h·ft^2·°F. This represents a weighted average of the condenser conditions, which range from simple vapor condensation near the vapor inlet to complex condensation and vent gas cooling at the vent outlet.

A = heat-transfer area, ft^2. The effective area through which the heat must flow to get to the water, typically referred to the outside area of smooth tubes.

ΔT = the effective average temperature difference between the hot and cold fluids, °F. This represents a weighted average of the local differences between the bulk temperature of the hot and cold fluids.

Rearranging to solve for A,

$$A = \frac{Q}{U\Delta T} \tag{5.7}$$

For very simple situations, this equation is adequate. In a liquid-liquid heat exchanger with two streams of identical liquid flowing at identical rates in a counterflow configuration, the overall heat-transfer coefficient U will be everywhere the same. The temperature difference ΔT will also be everywhere the same. However, if we now simply change the mass flow rate of one stream, we will cause ΔT to vary continuously throughout the exchanger. How do we handle that?

Looking at the basic definitions of the terms in Eqs. (5.6) and (5.7), we see that we are looking for an effective average temperature difference. For circumstances such as this, in which U is essentially constant throughout the exchanger, and in which the ratio of one fluid's specific heat to that of the other fluid remains constant, we may define the appropriate average temperature as

$$\Delta T = \text{LMTD} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln(\Delta T_{\max} / \Delta T_{\min})} \tag{5.8}$$

The log mean temperature difference (LMTD) is based on the temperature differences between the two streams at the inlet and outlet connections, as shown in Fig. 5.11 for sensible heat transfer with countercurrent flow and cocurrent flow, and for condensation of a pure material (no noncondensables) rejecting heat to cooling water. For the partial condenser, the end conditions are not sufficient to describe the thermal performance accurately.

A mathematical detail which is occasionally confusing here is that the LMTD equation cannot handle the trivial case of a constant temperature difference. It computes the natural log of 1.0, which is zero, then attempts to divide that into the numerator, which is also zero. Calculators and computers yield an error message. If the ΔT_{min} is more than half the ΔT_{max}, then a numerical average of the two is usually sufficient. For example, if the end temperature differences are 10 and 20°F, the arithmetic average is 15°F and the LMTD is 14.4°F. That is only a 4 percent error.

So, what about partial condensers? In ejector condensers the presence of noncondensable air and other gases complicates the situation enough that the values of U and ΔT must be defined carefully together. In general, they vary continuously throughout the condenser in a complex way. We may, however, think of the condenser as being a unified collection of several smaller heat exchangers. Each performs part of the overall job, and each has nearly uniform U and ΔT. The total area is the sum of all the small areas. Expressed mathematically for the sum of an infinite number of small areas,

$$A = \int \frac{dQ}{U \Delta T}$$

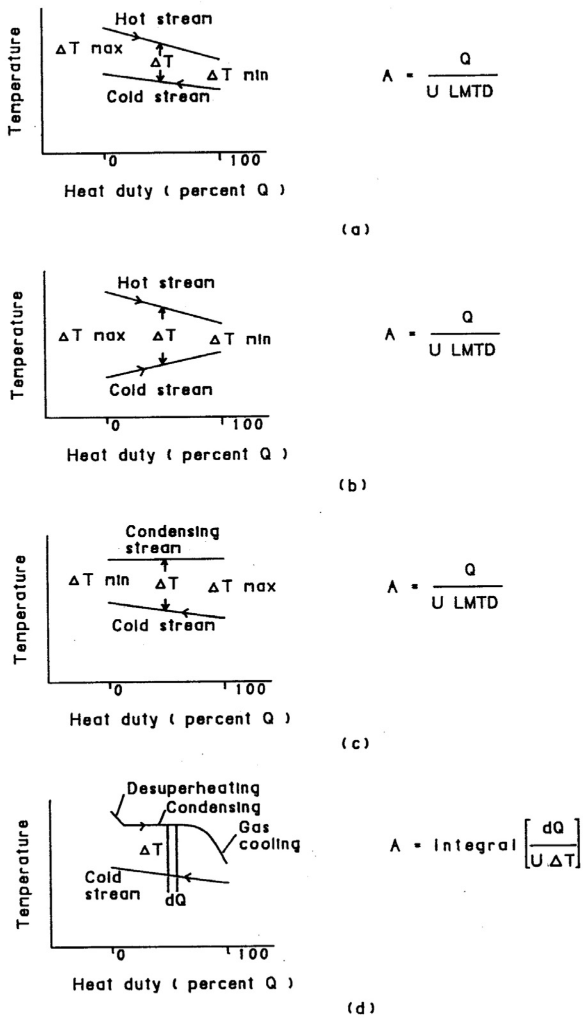


Figure 5.11 Temperature difference versus heat duty for different heat exchanger service: (*a*) sensible transfer, countercurrent; (*b*) sensible transfer, cocurrent; (*c*) condensing, sensible; (*d*) partial condenser.

Ryans and Roper [7] have a good worked example for a mixture of air and water vapor, from which I will soon borrow for an abbreviated example.

First, however, let us examine just what goes into the term U.

Overall heat-transfer coefficient *U*. By its definition, U is seen to be a "conductance" property of a system. It describes how much heat is conducted across a given object (one square foot of heat-transfer surface here) for a given temperature difference. Another useful point of view is to think of the same object as a "resistance" to the flow of heat, generating a temperature difference when heat flows through the object. Designating this property as R, we note that

$$R = \frac{1}{U} \qquad \text{and} \qquad U = \frac{1}{R}$$

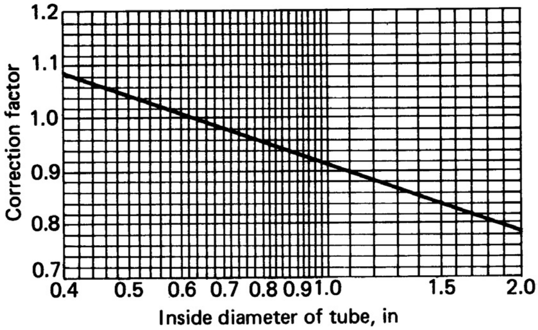
Consider now a small portion of a tube and visualize the path the heat must follow to leave the vapor and arrive at the water. It first has to pass through the slow-moving vapor film and the liquid condensate film to get to the tube wall, then it flows through the tube wall, and finally it has to flow through the slow-moving water film on the other side of the tube until it arrives in the rapid flow of turbulent water. If the tube has been in service for some time, there will probably be some scum or fouling on each surface of the tube to offer more resistance to the heat flow. So we have identified six different resistances along the heat flow path. The overall resistance R is equal to the sum of all these resistances,

$$R = R_{ncv} + R_c + R_{f\text{-}o} + R_w + R_{f\text{-}i} + R_{cw} \tag{5.9}$$

where R_{ncv} = noncondensable vapor film resistance
R_c = condensing film resistance
$R_{f\text{-}o}$ = fouling resistance outside the tube
R_w = resistance of the tube wall
$R_{f\text{-}i}$ = fouling resistance inside the tube
R_{cw} = cooling-water film resistance

Each of these has been studied at length because of its importance. The fouling resistances used in design are a result of field experience and laboratory testing. The sum of the two resistances is usually in the range of 0.001 to 0.003 $\text{h}\cdot\text{ft}^2\cdot°\text{F/BTU}$. The tube-wall resistance is a simple calculation based on the tube geometry and material conductivity. The other three resistances are handled differently. The water-side film resistance and the condensing film resistance are commonly represented as film coefficients h, which are actually conductance terms. Figures 5.12 and 5.13 give the h values for water-side (h_{cw}) and condensing (h_c) coefficients. These values are the reciprocal of the respective resistances. Figure 5.14 gives the noncondensable film resistance.

In combining these into Eq. (5.9), an additional correction must be made. Because the tubes are cylinders and the heat flows radially into them, it is necessary to adjust the inside resistances and coefficients to take into account the outside area of the tube, the measure



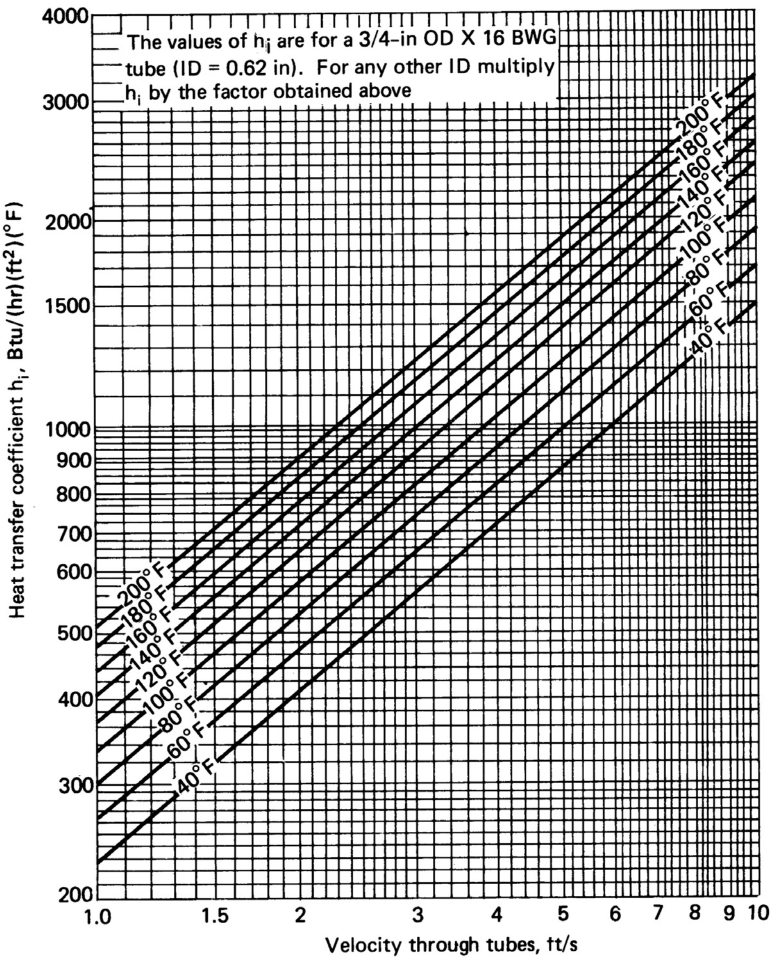


Figure 5.12 Tube-side film coefficients for water. *(From Kern 1950; reproduced in J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation, *McGrawHill, New York, 1986)*

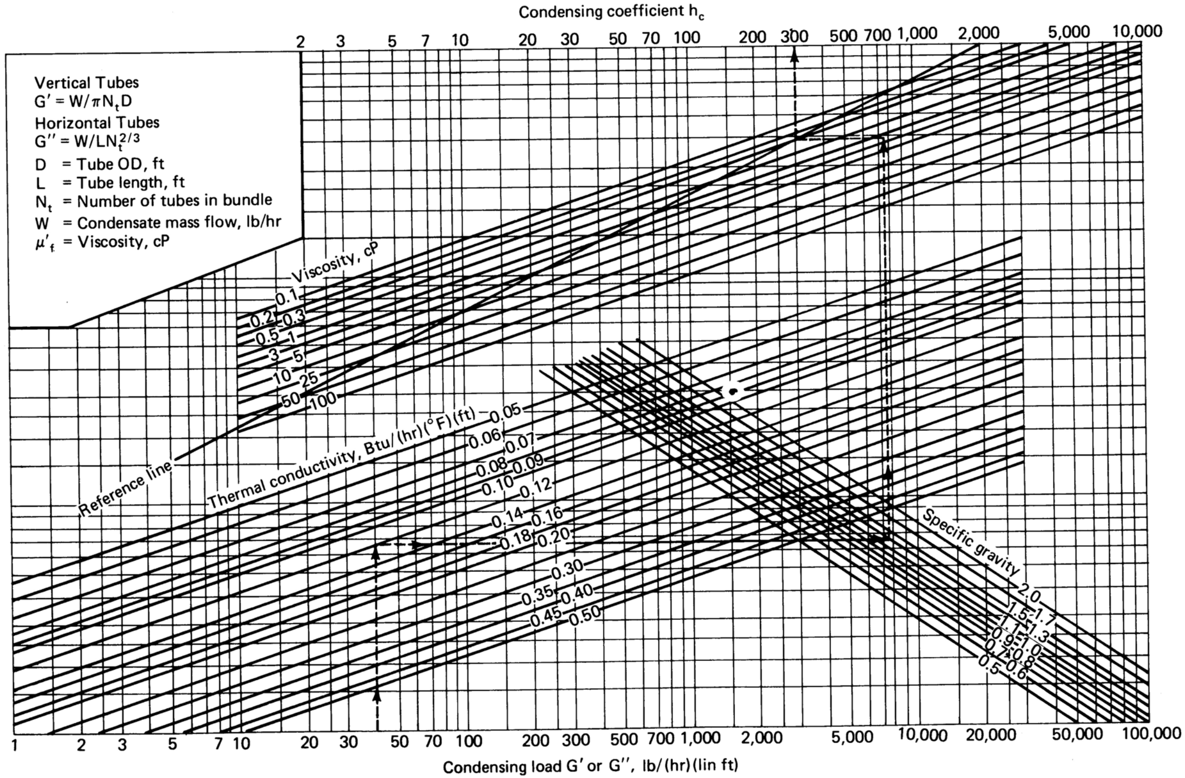


Figure 5.13 Condensing film coefficients. *(From Kern 1950; reproduced in J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986)*

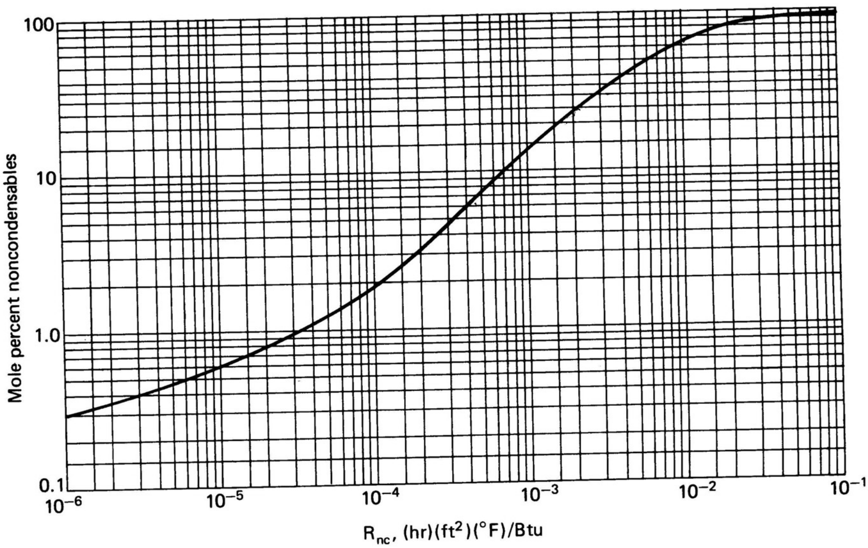


Figure 5.14 Noncondensable film resistances. *(From Kern 1950; reproduced in J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986)*

of heat exchanger size. With these adjustments and reciprocal relationships inserted into Eq. (5.9),

$$\frac{1}{U} = R = R_{ncv} + \frac{1}{h_c} + R_{f-o} + \frac{t_w D_o}{k_w D_{lm}} + \frac{D_o}{D_i}\left(R_{f-i} + \frac{1}{h_{cw}}\right) \tag{5.10}$$

where h_c = condensing coefficient, BTU/h ·ft^2 ·°F
h_{cw} = cooling-water coefficient, BTU/h ·ft^2 ·°F
t_w = tube wall thickness, ft
k_w = tube material conductivity, BTU/h ·ft^2 ·°F
D_i = tube inside diameter, ft
D_o = tube outside diameter, ft
D_{lm} = tube log mean diameter = $\dfrac{D_o - D_i}{\ln(D_o / D_i)}$

We now have the means to estimate the overall U for all or a portion of a horizontal shell-and-tube condenser. Ryans and Roper have done this for a set of conditions representative of common applications (Fig. 5.15). The curve is derived from first principles, and is good enough that one manufacturer thought it had been based on his proprietary curves, derived from a series of special tests.

Next we need to see how the temperature varies within a partial

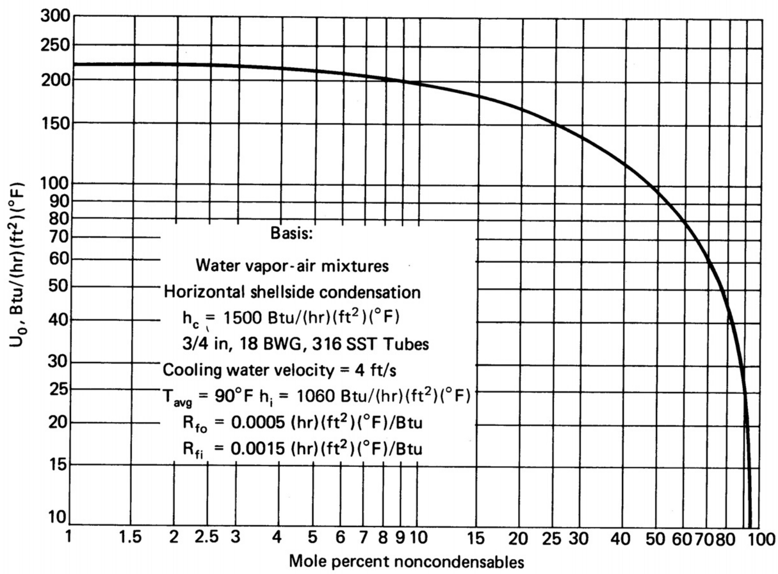


Figure 5.15 Overall heat-transfer coefficients. (*From J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986)*

condenser. Then we will be able to define ΔT at any location and calculate the area required for that portion of the condenser.

Example heat duty and condensing temperature profile. To complete the information required for estimating partial condenser area requirements, we must be able to describe the temperatures versus the heat duty. In other words, we must be able to plot the equivalent of Fig. 5.11*d*. In that diagram, three different zones are present: vapor cooling (desuperheating), condensation, and vent gas cooling. In reality, all three actions may be occurring at any location, but it is a convenient concept.

As an example to illustrate the method, I will use a three-zone version of the Ryans and Roper [7] six-zone example. A surface condenser receives 28 pph air and 248 pph water vapor at 130 torr and 132°F. The stream is to be cooled to 100°F. Find the heat duty and temperature profile, then estimate the total area required. Assume the pressure drop is negligible and latent heats and specific heats are constant.

First, find the dew point, the temperature at which the first drop of condensate appears. In this situation the dew point is the temperature at which water vapor pressure is equal to the partial pressure of water in the gas mixture.

Moles water vapor	= WIMW = 248/18	=	13.78
Moles air	= 28/29	=	0.97
Total moles		=	14.75

$$\text{Water vapor partial pressure} = yP = \frac{13.78}{14.75}(130) = 121 \text{ torr}$$

Dew point, from Fig. 5.3 or steam tables = 132°F

Thus, this mixture is saturated, and no desuperheating is required. Only condensing and gas cooling need be considered. Most of the heat removal will occur in the first few degrees of cooling, then the difficult task of gas cooling begins. The total cooling range is 132 - 100 = 32°F. Arbitrarily I will call the first 25 percent the condensing zone, from 132 to 124°F. From there to 100°F I will call vent gas cooling.

To find the heat duty for this zone, we must first find how much water condensed, then compute the latent heat and. sensible heat effects. The water vapor pressure at 124°F is 98 torr.

$$\text{Water vapor leaving} = \frac{18(98)}{29(130-98)} = 53 \text{ pph}$$

Water vapor condensed = 248 - 53 = 195 pph

Consider now the various heat duty components. The major portion is the latent heat of the condensing vapor, about 1025 BTU/lbm. The air is cooled from 132 to 124°F and has a constant specific heat of 0.25 BTU/lbm·°F. The water vapor is cooled until it condenses, then the liquid is cooled further. As a simplified model, assume the vapor is cooled to the mid-temperature in this zone and the condensate is cooled the rest of the way. Further, assume the condensate follows the uncondensed vapors and is cooled with them. The water vapor and liquid specific heats are 0.42 and 1.0 BTU/lbm·°F, respectively. For this zone the total heat load Q_c is

$$\begin{aligned} Q_c &= Q_{gas\ cooling} + Q_{condensing} + Q_{liquid\ cooling} \\ &= 28(0.25)(8) + 248(0.42)(4) + 195(1025) + 195(1.0)(4) \\ &= 56 + 417 + 199{,}875 + 780 \\ &= 201{,}128 \text{ BTU/h} \end{aligned}$$

Next, consider the gas cooling zone. The vent temperature is 100°F, and the corresponding water vapor pressure is 49 torr.

$$\text{Water vapor leaving} = 28\frac{18(49)}{29(130 - 49)} = 11 \text{ pph}$$

$$\text{Water vapor condensed} = 53 - 11 = 42 \text{ pph}$$

A word here about how I am handling the condensate is appropriate. I chose the model in which the condensate is cooled along with the uncondensed gases. In their worked example Ryans and Roper appear to use this model, but I am not sure. The choice might be more accurate if this were a vertical condenser with the vapor condensing in the tubes. Actually, in a horizontal condenser some condensate forms on the top tubes and is subcooled as it runs down through the tube bundle, then drops out and runs to the drain. A very simplified model would ignore the thermal load associated with sub cooling the condensate. The choice of method for handling condensate cooling will not change the results much. None of the methods is totally descriptive of what is actually happening, and this discussion of diverse approaches to these calculations has prepared you for the sort of diversity you may encounter elsewhere.

To continue, the condensate from the first zone will be cooled to the final temperature. The condensate from the second zone will be cooled from that zone's midtemperature to the final temperature.

$$Q_{(\text{subcooling})} = 195(1.0)(24) + 42(1.0)(12) = 5184 \text{ BTU/h}$$

$$\begin{aligned} Q_{\text{gc}} &= 28(0.25)(24) + 53(0.42)(12) + 42(1025) + 5184 \\ &= 168 + 267 + 43{,}050 + 5184 \\ &= 48{,}669 \text{ BTU/h} \end{aligned}$$

And the total heat duty for this condenser is

$$Q_{\text{total}} = Q_{\text{c}} + Q_{\text{gc}} = 201{,}128 + 48{,}669 = 249{,}797 \text{ BTU/h}$$

which is close to the Ryans and Roper value of 249,600. The difference is due to numerical roundoff and to my choice to subcool all the liquid, which might be different from their method. If they used only modest subcooling, then theirs is probably the better procedure for horizontal condensers with vapor in the shell, but the difference is well within the larger uncertainties in condenser design.

Example area. calculation. We now have the data to estimate the area of the condenser example. For this I follow example 7-4 of Ryans and Roper, which is based on their heat load (example 7-3). Cooling water is supplied at 85°F with a 5°F rise, in a single pass with a 4-ft/s veloc ity in the ¾ in OD, 18 BWG 304 stainless tubes. Condensing-side and water-side fouling factors are given as 0.001 and 0.0015. The condensing coefficient is given as 1500. The water flow rate, by a heat balance, is

$$W_{cw} = \frac{Q}{cp\,(\text{temperature rise})} = \frac{249{,}797}{1.0(5)} 50{,}000 \text{ pph} = 100 \text{ gpm}$$

The cooling-water temperature rise in the gas cooling zone is

$$\Delta T_{gc} = 5.0\left(\frac{48{,}669}{249{,}797}\right) = 1.0\ °\text{F}$$

Now we can plot the temperature versus the heat duty to visualize some of the important factors in this example. Figure 5.16 shows this example, with the Ryans and Roper six-zone example added to show the benefit of using several smaller zones for the estimate. Selecting temperature differences from Fig. 5.16, we can now calculate the LMTD for each zone.

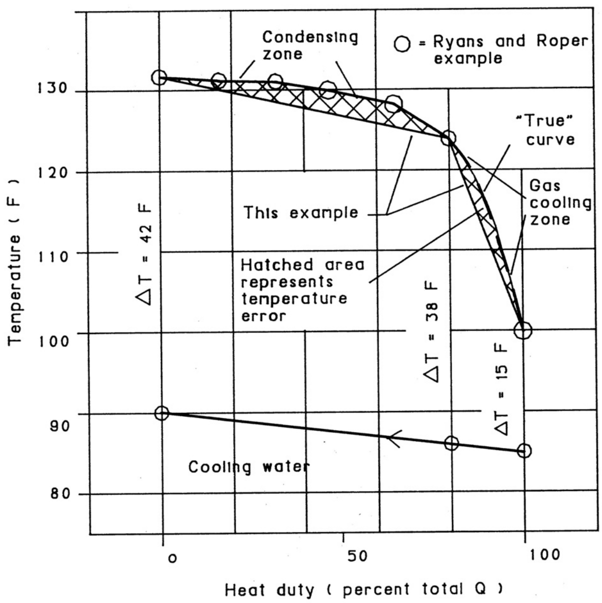


Figure 5.16 Example partial condenser temperature vs. heat duty. *(From J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986)*

$$\text{Condensing LMTD} = \frac{42 - 38}{\ln(42/38)} = 40.0°\text{F}$$

$$\text{Gas cooling LMTD} = \frac{38 - 15}{\ln(38/15)} = 24.7°\text{F}$$

All that is left now is to compute the overall U values for each section, first getting some tube- and water-side data and using Eq. (5.10).

Tube data:

$D_o = 0.75/12 = 0.0625$ ft

$D_i = 0.0543$ ft

$D_{lm} = 0.0583$ ft

$t_w = 0.049/12 = 0.0041$ ft

$k_w = 9.42$ BTU/h · ft ·°F

Water-side coefficient: From Fig. 5.12 with a water velocity of 4 ft/s, $T_{avg} = 87.5$°F, and $D_i = 0.65$ in,

$h_{cw} = 960(0.99) = 950$ BTU/h · ft^2 ·°F

$$\begin{aligned}\frac{1}{U} &= R_{ncv} + \frac{1}{1500} + 0.001 + \frac{0.0041(0.0625)}{9.42(0.0583)} + \frac{0.0625}{0.0543}\left(0.0015 + \frac{1}{950}\right)\\ &= R_{ncv} + \frac{1}{1500} + 0.001 + 0.00047 + 0.00294\\ &= R_{ncv} + 0.0051 \text{ for all zones of the condenser}\end{aligned}$$

Condensing zone:

Moles noncondensables (nc) = 28/29 = 0.97

Moles water vapor in = 248/18 = 13.78

Moles water vapor out = 53/18 = 2.94

Mole percent nc in = [0.97/(0.97 + 13.78)](100) = 6.6 percent

Mole percent nc out = [0.97/(0.97 + 2.94)](100) = 25 percent

From Fig. 5.14, the corresponding values of R_{ncv} are 0.00044 and 0.0022. The log mean value is

$$\text{Log mean } R_{ncv} = \frac{0.0022 - 0.00044}{\ln(0.0022/0.00044)} = 0.00035 \text{ h} \cdot \text{ft}^2\,°\text{F/BTU}$$

so,

$$\frac{1}{U_c} = 0.00035 + 0.0051 = -.00545 \quad \text{and} \quad U_c = 183\ \text{BTU/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

and finally,

$$A_c = \frac{Q_c}{U_c \Delta T_c} = \frac{201{,}128}{183(40)} = 27\text{ft}^2$$

Repeating the procedure for the gas cooling zone,

$$\text{Moles water vapor out} = 11/18 = 0.61$$

$$\text{Mole percent nc out} = \frac{0.97}{0.97 + 0.61}(200) = 61\ \text{percent}$$

From Fig. 5.14, the corresponding value of R_{ncv} is 0.010. The log mean value for the gas cooling zone is

$$\text{Log mean } R_{ncv} = \frac{0.010 - 0.0022}{\ln(0.020/0.0022)} = 0.00515\ \text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F/BTU}$$

so,

$$\frac{1}{U} = 0.00515 + 0.0051 = 0.010 \quad \text{and} \quad U_c = 100\ \text{BTU/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

$$A_{gc} = \frac{Q_{gc}}{U_{gc} \Delta T_{gc}} = \frac{48{,}669}{100(24.7)} = 20\ \text{ft}^2$$

And the total area is the sum of the two zone areas: 27 + 20 = 47 ft^2. This compares closely with the Ryans and Roper calculation of 48 ft^2.

The similarity of the results should not be regarded as a license to shortcut the calculations – "If using two zones is OK, then why not use only one zone?" It does indicate that simplified estimating methods can give useful answers. In this case my choice of 124°F as the temperature which marked the transition between condensing and gas cooling was fortunate, as may be seen by examining Fig. 5.16. I simply picked a convenient temperature in the Ryans and Roper example. Had I picked a much higher or lower temperature, my error of approximation would have been greater, as indicated by the shaded areas in Fig. 5.16.

Detailed, computerized methods. As you have seen from the worked examples, the details of some of the calculations can quickly become tedious. The low cost of computerized calculations for the routine design of heat exchangers permits the designers to include many refinements in the analysis. These refinements may include a large number of heat-transfer zones, the effects of pressure drop and diminishing vapor velocities, multiple-component vapor-liquid equilibrium, the presence of two immiscible liquid phases, and the opportunity to "tweak" the calculations by inserting adjustments based on experience, manufacturer and industry standards, or customer specifications.

Optimization and design quality. Using computers it becomes a trivial task to optimize a design by adjusting many design parameters in a search for the "best" design, whether the most important factor is low purchase price or low operating cost or low combined cost. This search for the "best" must be directed and constrained by an accurate and complete description of the overall design task and by the application of competent engineering judgment as to the reasonableness of the results. For example, the designer will typically define the lowest acceptable values for the terminal condensing temperature difference and the vent approach temperature difference for each surface condenser.

Then the designer allows the computer to search for the best combination of interstage pressures, steam and water usage, and equipment cost. Occasionally an "optimum" design may have an unanticipated flaw, in which case it is necessary to revise the specifications to redirect the design toward a more reasonable result. As an example, an unrestrained optimization might arrive at a design in which a surface condenser vent temperature was only 2°F above the water inlet temperature. Few people would feel comfortable with that design.

Also, fascination with the optimization process should not be allowed to obscure the importance of starting with good data regarding mass flows, physical properties, realistic fouling coefficients, etc., and obtaining a workable, reliable design. As seen previously, a single error such as neglecting to recognize that a condensable vapor's liquid phase is insoluble in water may destroy the value of otherwise careful design work.

Quick and dirty area calculation. For very rough estimates, use Fondrk's method [8], shown in Fig. 5.17. It is offered for precondensers, but there is no intrinsic difference between precondensers and other partial condensers. His curves are for sizing the whole condenser, using only the terminal temperature difference, the concentration of noncondensables, and the quantity of steam condensed. They are based on cooling the noncondensable gases to a temperature 10°F above the cooling-water inlet temperature, an overall fouling factor of 0.003, a water-temperature increase of 40°F or less, water outlet temperature not more than 120°F, fixed-tube-sheet condenser, and modest quantities of oil vapor. If the condensable oil-vapor load

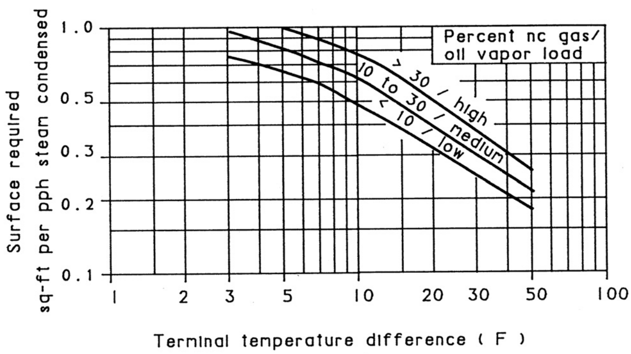


Figure 5.17 Quick area estimate for partial condenser. *(Reproduced with permission from* Petroleum Refiner*, December 1958)*

contributes a considerable portion of the total heat load, some upward adjustment in surface area requirements may be necessary.

From Fig. 5.16 we obtain a terminal temperature difference of 42°F, in previous calculations we found the inlet concentration of noncondensables to be 6.6 percent, and 248 - 11 = 237 pph of steam condenses. From Fig. 5.17 we obtain a value of 0.20 ft^2/pph, which is multiplied by 237 pph to obtain 47 ft^2. How tempting it would be to use this shortcut and bypass all the tedious work. Please do not use this in the wrong places. It is good for quick estimates, and only for that. It is only coincidental that the three methods happen to agree so closely here. Reexamination of the bases for Fig. 5.17 will reveal some significant differences between them and the actual conditions in this example. I would be quite satisfied with an uncertainty of plus or minus 50 percent for this quick method.

External size of shell-side condensers. As a rough rule for shell-side condensers, the shell diameter will be twice the diameter of the discharge of the preceding ejector stage. That makes the superficial velocity of the vapor longitudinal flow about 50 ft/s. In other words, the inlet vapor flowing axially through an empty shell will have a velocity of about 50 ft/s. The vent will typically be half the diameter of the inlet connection, or one-fourth the shell diameter.

Sizing other condenser configurations. The general method for sizing other partial condenser configurations is the same as for horizontal vapor-in-shell condensers. The primary difference is that the condensing coefficients are based on the geometry differences associated with each specific configuration. For example, with water on the shell side, the water-side coefficient is based on cross-flow outside the tubes instead of flow inside the tubes. In general, the area will be larger and/or the pressure drop will be greater for the other configurations. For the vertical vapor-in-tube downflow configuration, the prediction of vapor cooling will be more accurate.

Fouling, cleaning, and operating considerations

The general concern expressed under this topic is that the original geometry may be changed and the heat-transfer capability may become degraded by an accumulation of materials which impede the transfer of heat. This is commonly termed "fouling," and TEMA [9] recognizes five types of fouling:

Precipitation	—	Calcium carbonate crystals
Particulate	—	Clay, sand, silt
Chemical reaction	—	Coking
Corrosion	—	Rust
Biological	—	Algae, fungi, slime, etc.

The net result of the addition of fouling coefficients to heat-transfer calculations is an increase in the heat-transfer surface area. TEMA strongly recommends that fouling coefficients be carefully identified for each side of the heat-transfer surface and not given an artificially high value to provide a safety factor for other uncertainties that may exist in the heat-transfer calculations. It also warns against arbitrarily increasing the heat-transfer surface to allow for fouling. In summary, it advocates the sound practice of identifying the separate elements of a design task in order that the best knowledge of the day can be applied. The practice also keeps all involved people alert to the consequence of any potential change or improvement.

In practice, the experienced condenser designer may consider fouling along with the uncertainties in physical properties, vapor-liquid equilibrium calculations, etc., and use a combination of fouling coefficient and excess area which seems right.

Typical fouling coefficients. Ideally, a process system is designed with fouling coefficients selected for each heat exchanger so that all will operate properly during the intervals between system shutdowns for cleaning and other maintenance.

TEMA offers several pages of recommended fouling coefficients, sometimes abstracted in heat-transfer books. In general, the value for condensing vapors is 0.001 $h \cdot ft^2 \cdot {}^\circ F/BTU$. Some of the exceptions are for clean steam, 0.0005, and oil-bearing exhaust steam, 0.0015-0.002.

A value of 0.002 is suggested for oil-bearing refrigerant vapors and for oil refinery crude vacuum, cracking, and coking overhead vapors.

For water at ejector condenser conditions and a velocity over 3 ft/s the typical value is 0.001. Some exceptions are 0.0005 for seawater, condensate, and boiler feedwater; 0.002 for average river water, muddy or silty water, and boiler blowdown; and 0.003 for untreated spray pond water and hard water.

Antifouling practices. Keeping the water velocity at least 2 ft/s on the shell side and 4 ft/s on the tube side is the simplest way to minimize fouling on the water side. Upper limits, set by considerations of forced vibration wear and erosion, are given in Chap. 10. If the water is on the shell side, the baffle spacing, baffle cuts, and design details which restrict bypassing will avoid the dead spots that permit silting and fouling. Operators should be instructed to keep the water velocities at the design value, even when cold water would otherwise permit some water economy in cool weather. If flow is throttled in cold weather, the water side may become silted and need cleaning to work properly in the demanding warm weather. Operators should backflush the water side regularly to flush out debris.

It is also important to select materials for baffles, tie rods, and spacers to resist fouling and corrosion on the water side.

Vapor-side fouling is best prevented by keeping the fouling vapors away from the ejector or creating temperature and moisture conditions which minimize fouling (if there is an opportunity to do so). As an example, installing a water spray nozzle in the top head of a vertical vapor-in-tube condenser provides an opportunity to periodically flush the top tube sheet to prevent solids buildup there.

Erosion of the tubes at the vapor inlet by the two-phase flow from an ejector-stage discharge is prevented by placing a properly designed impingement plate at the vapor inlet and providing space for the flow to enter the tube bundle at lowered velocity.

If fouling is severe enough to require regular cleaning, try to place the fouling side inside the tubes where it is accessible for cleaning. The tube side is easily cleaned if the tubes are straight, much harder to clean if tubes are U -bend, and impossible to clean mechanically if they are coiled. If both sides will foul, the bundle should be removable for cleaning. Figure 5.18 shows the relative accessibility of different bundle designs. A square-pitch tube array is more accessible for shell-side cleaning than a triangular-pitch array.

Contact condensers using recycled condensate. If significant process-side fouling is expected with surface condensers, consider substituting contact condensers using recycled condensate from the hotwell,

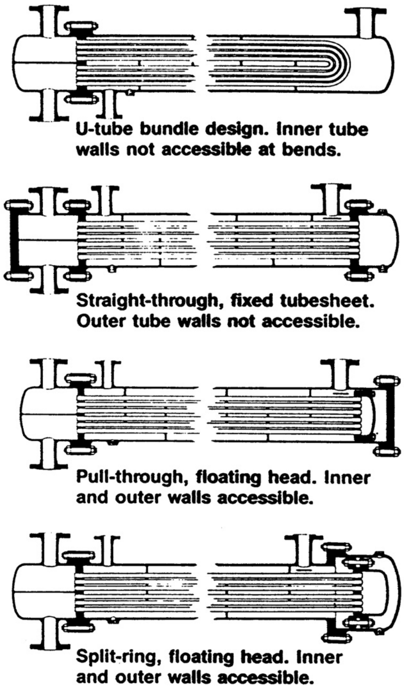


Figure 5.18 Tube bundle accessibility for cleaning and repair. *(Courtesy Nash-Kinema)*

cooled against cooling water in a cleanable plate and frame or some other type of surface exchanger. Closer approach temperatures in the contact condensers and efficient counterflow in the liquid-liquid surface exchanger partially offset the penalty of having two heat exchangers in series in the overall steam-to-water heat-flow path. Surface exchanger cleaning may be done at the hotwell level, contact condensers are easier to inspect and clean, cooling water is not contaminated, steam and water usage are not increased much, and waste treatment cost is not affected.

The presence of volatile process materials in the recycled condenser water must be considered in the vapor-liquid calculations for each condenser. An immiscible phase may be removed in a decanter to minimize recirculation of that material. The system design process is slightly more complex, but requires no special skills.

Steaming out process-side fouling. If process-side fouling is largely removable by steaming with pressurized steam, it may be possible to clean the system in place, *provided* that (1) all the equipment and piping to be pressurized is designed for the pressure and temperature, and careful attention is given to the system design and operating procedures to avoid exposing people or low-pressure equipment to unexpected pressurized steam; and (2) the exchanger is designed to avoid excessive thermal stresses, created, for example, by putting steam in the tubes of a fixed-tube-sheet exchanger with a cold shell.

5.5 Nomenclature

A	heat-transfer area, ft^2
c_p	specific heat at constant pressure, BTU/lbm · °F
D	diameter, in
γ	activity coefficient, dimensionless
g_c	units conversion factor, 32.2 ft·lbm/lbf·s^2
gpm	gallons (U.S.) per minute
h	film conduction coefficient, BTU/h·ft^2·°F
H	energy head, or enthalpy: ft·lbf/lbm (or BTU/lbm)
k_w	conductivity of tube wall material, BTU/h·ft·°F
n	mass molar quantity, lbm/molecular wt.
P^o	vapor pressure of a pure material, torr
P	total pressure, torr
Q	volumetric flow, gpm, or heat load, BTU/h
pph	lbm/h
R_a	lbm motive steam per lbm load air
R_g	universal gas constant = 1546 ft·lbf/lbm·mole·°R
SG	specific gravity; water = 1.0
t_w	tube wall thickness, ft
T	temperature, °R (Rankine)
ΔT	temperature difference, °F
U	overall heat-transfer coefficient, BTU/h·ft^2·°F
v	specific volume = 1/w, ft^3/lbm
V	velocity, ft/s
W	mass flow, lbm/s
x	mole fraction in liquid phase
y	mole fraction in vapor phase

Subscripts

i	generic reference to one component in a mixture
i	tube inside location
o	tube outside location

5.6 References

1. Joseph H. Keenan, Frederick G. Keyes, Philip G. Hill, and Joan G. Moore, *STEAM TABLES: Thermodynamic Properties of Water, Including Vapor, Liquid, and Solid Phases,* John Wiley & Sons, New York, metric units 1969, SI units 1978; Lester Har, John S. Gallagher, and George S. Kell, *NBS / NRC Steam Tables,* Hemisphere Publishing Corp., New York, SI units 1984; and *ASME Steam Tables*, 6th ed., American Society of Mechanical Engineers, New York, 1993.
2. "Vacuum/Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd,England, January 1989.
3. "Ejectors," Bulletin No JVA-7, Jet-Vac Corp.
4. C. H. Chilton and R. H. Perry, *Chemical Engineer's Handbook,* 5th ed., McGraw-Hill, New York, 1973, p. 5-2.
5. *Standards for Steam Surface Condensers*, 8th ed., Heat Exchange Institute, Cleveland, Ohio, January 1984.
6. K. W. McQuillan and P. B. Whalley, "A Comparison between Flooding Correlations and Experimental Flooding Data for Gas-Liquid Flow in Vertical Circular Tubes," *Chemical Engineering Science*, 40 (8): 1425-1440 (1985).
7. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
8. V. V. Fondrk, "Figure What an Ejector Will Cost," *Petroleum Refiner*, December 1958, pp. 101-105.
9. *Standards of the Tubular Exchanger Manufacturers Association*, 7th ed., Tubular Exchanger Manufacturers Association, Tarrytown, N.Y., 1988.
10. C. G. Blatchley, "Selection of Air Ejectors, Part 2," *Chemical Engineering Progress*, 57 (11): 144-150 (1961).

Chapter

6

Pressure Control

6.1 Overview of Pressure Control

In many ejector installations it is desirable to control the pressure maintained in the process system. A prescribed pressure-versus-time schedule may be desired for a batch operation. Pressure control may be needed to avoid product degradation or undesirable surging or "burping" within the system. However, if you want the lowest attainable pressure all the time, then no pressure control is needed. In that case you may wish to at least finish reading this overview to make sure your operation will not be harmed by some side effects of having no control.

One undesirable result of not controlling the system pressure is that the ejector system may "backfire" water vapor into the process system more often than if the pressure were controlled. As we saw in earlier chapters, some ejector stages may not be designed to be stable below 30 to 50 percent of design load. Even if the ejector is designed to be stable to no-load, some water vapor may diffuse back into the process system at low loads. Bleeding a small quantity of dry air or inert gas to establish a modest flow toward the ejector will eliminate the problem [7].

In the typical installation where pressure control is desirable, the design strategy is to design some portion of the process system or the ejector system for a lower pressure or larger capacity than is needed. Then one of several methods is used to dissipate the extra capability to maintain a steady controlled pressure in the process system.

This chapter begins with an explanation of the conditions which acontrol system must satisfy in the process system and the ejector system, then describes several ways of satisfying those conditions while controlling pressure. Many people have strong preferences on this subject, conditioned by their specific experiences. I have found little on this subject in the published literature [1-6, two of which are mine]. In discussing this subject with ejector manufacturers, I found they had little economic incentive to research the subject. I did find qualitative confirmation that the concepts I present here are generally correct. Thus, I offer them with the caution that they are not supported by the body of evidence I would prefer.

If you have a pressure control application that is unusual, I suggest you discuss it with one of the ejector manufacturers to develop a reliable, cost-effective design and installation.

Four general methods of pressure control are described here:

+ Throttling the flow to the ejector
+ Bleeding noncondensable gas, condensable vapor, or both
+ Throttling motive steam externally or by a spindle nozzle
+ Adjusting condenser cooling-water flow (not recommended!)

Before discussing the details of each of these control methods, it is necessary to examine what conditions must be satisfied in the process system and the ejector system.

6.2 System Curves

In most process systems the mass flow of gases going to the ejector increases as the pressure is lowered. By contrast, the ejector capacity decreases as the pressure is lowered. If these two conditions are described by graphs and then combined, we may visualize what is happening in several pressure control systems.

I will first examine the steady-state condition, in which all parameters in the systems (pressures, temperatures, and flows) do not change with time. That idealized situation is a useful concept, even though it is never actually achieved. If we first consider the steady state condition, we are better equipped to examine how the system behaves when it is in transition from one steady-state condition to another. This transient condition includes the special condition in which an ejector is cycling between stable operation and "broken" (unstable) operation with possible backfiring.

Process-system curves

Figure 6.1 shows two types of process-system curves. Figure 6.1*a* covers the uncommon situation in which the total load is noncondensable gas, including air leakage. It is assumed that the process system is below 50 percent of atmospheric pressure and that the total flow is essentially independent of pressure changes below that value.

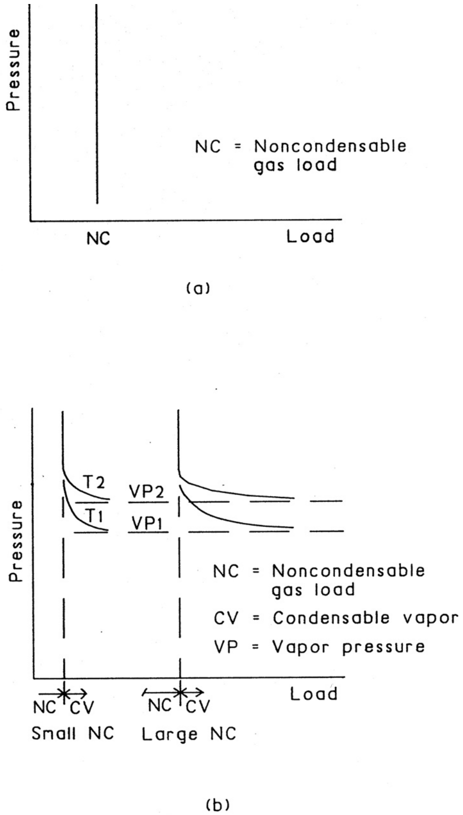


Figure 6.1 Process-system curves: *(a)* non condensable gas only; *(b)* noncondensable gas saturated with process vapor.

Figure 6.1*b* covers the more common situation in which the noncondensable gases are saturated with condensable vapors. At a given temperature, the total load will increase rapidly as the pressure lowers and approaches the vapor pressure of the liquid. At a higher temperature, the limiting vapor pressure will be higher. Note that the higher noncondensable load example curves are simply multiples of the low-load curves.

To illustrate: pressure = 60 torr, temperature = 100°F, and air loads are 10 pph and 100 pph. The water vapor pressure is 49 torr; thus the water vapor load per pound of air (as described in Chaps. 4 and 5) will be

$$\text{Water vapor} = 1.0\,\frac{18(49)}{29(60-49)} = 2.8\ \text{lbm/lbm air}$$

$$= \frac{2.8}{(0.8)(0.99)} = 3.5\ \text{lbm DAE/lbm air}$$

So, at 60 torr the total process-system load based on 10 pph air load is 10(1.0 + 3.5) = 45 pph DAE. The 100-pph air load will result in a total load of 450 pph DAE.

Ejector-system curves

Three different types of ejector curves exist. Single-stage and multistage noncondensing critical-flow ejector systems have a single curve which relates suction pressure to ejector load, provided the motive steam remains constant and the ejector is stable down to zero load, as shown in Fig. 6.2*a*. For multistage condensing ejectors

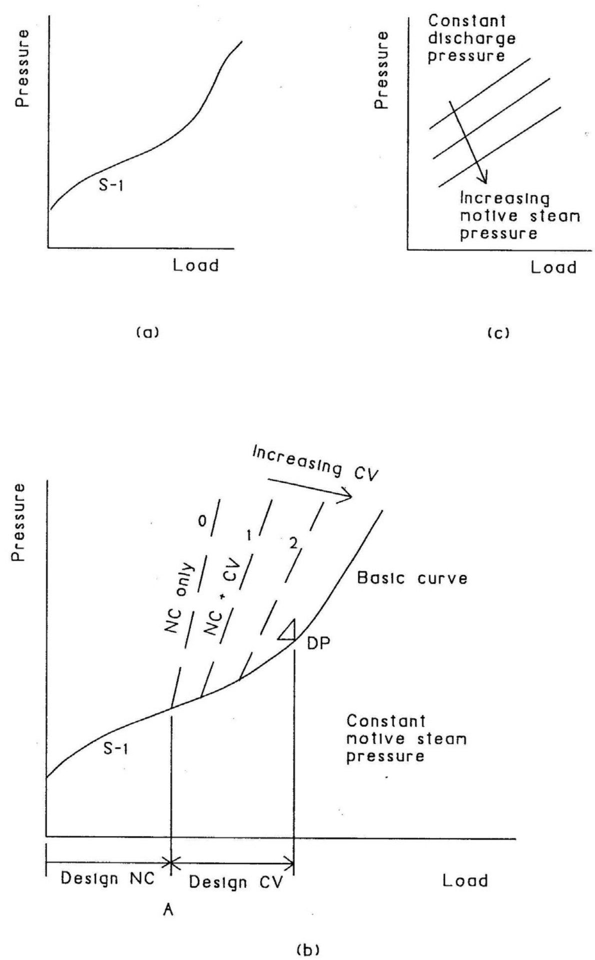


Figure 6.2 Ejector-system curves: (a) single-stage, critical, or multistage noncondensing; (b) multistage condensing; (c) single-stage, noncritical.

and for noncritical (compression ratio less than 2:1) stages, families of curves exist.

For multistage condensing ejectors, a family of "broken" curves is added to the basic curve, as shown by the dashed lines in Fig. 6.2*b*.

The broken (unstable) mode of operation is a natural consequence of exceeding the discharge pressure capability of a stage and does not necessarily result in backfiring or other undesirable behavior. It simply means that the suction pressure has become dependent not only on the load, but also on the motive steam pressure and the discharge pressure.

For each curve shown here, the motive steam pressure is held constant, and the precise location of each dashed-line curve depends upon the sizes of the subsequent stages, the condenser design, and cooling-water flow and temperature. The family of broken curves may be understood this way: If the ejector is loaded with dry air only, it will follow the basic curve until the load equals the design noncondensable load capacity at *A*. As the dry air load increases beyond *A,* a "broken" condition will exist, and the curve will turn sharply upward along the curve labeled 0. If the test is repeated, but this time with a small condensable vapor load established before the air load, the broken curve becomes curve 1. This broken curve is offset from curve 0 by the amount of condensable vapor in the load, and tends to lean a little more to the right. Repeating the test with a larger vapor load leads to curve 2, and so forth.

A little discussion of these curves is in order. They are reproducible for a given set of conditions, such as steam pressure, water temperature and flow, size of subsequent stages, condenser and stage cleanliness, etc. At the point where each broken curve departs from the basic curve, there is a small region of uncertainty in which some cycling between broken and stable operation may occur, possibly causing some backfiring (more about this topic soon). As the broken curves are extended upward, they are found to bend over to the right. In this extreme condition the stage is performing as a noncritical stage, with lower ,than sonic velocity in the diffuser.

This brings us to the subject of noncritical single-stage ejectors, whose performance curves are typified by Fig. 6.2*c*. The noncritical ejector constantly operates in a mode similar to the broken (unstable) mode of a critical ejector. For the noncritical ejector, the suction pressure is a function of three variables: the load, the motive pressure, and the discharge pressure. Depending on the user's needs, the performance curves may be plotted as shown, with the discharge pressure held constant and each curve labeled with the corresponding motive steam pressure. Other curve families can be drawn by holding the motive pressure constant or by holding the suction pressure constant.

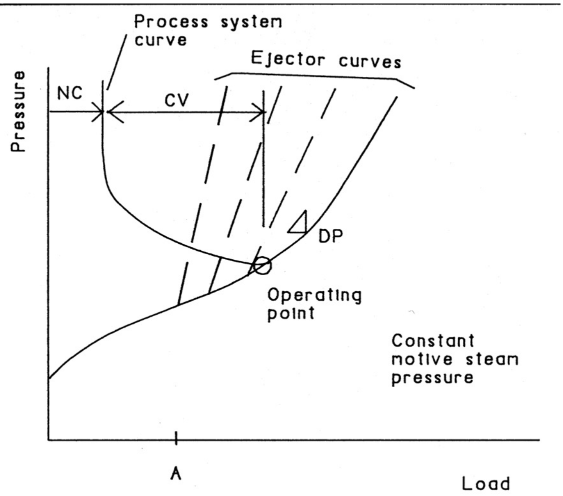
How various control schemes match both system curves

As a first step into this subject, consider the situation in which no control is being introduced. The system and ejector simply interact to find the pressure common to both their curves. For simplicity, the process line pressure drops are assumed to be small. Let us simplify the system curve by considering only one temperature curve and start with the common situation in which the noncondensable gas load is less than the ejector noncondensable design load A. As shown in Fig. 6.3*a,* the operating point at which the curves intersect is on the basic curve. Next, consider the noncondensable load to be greater than A, and observe that the operating point in Fig. 6.3*b* is now on the broken curve which corresponds to the amount of condensable vapor in the load. The operating point is shown here as being slightly above the design pressure. In practice it might be far above the design pressure, depending upon the amount of the noncondensable load and the steepness of the broken curve. I recall my first test of a multistage ejector with an air load, and how startled I was at the steepness and the reproducibility of the broken curve and the abruptness of the transition from the basic curve to the broken curve.

Stable operation without backfiring or cycling. All the examples we have examined so far involve steady_state operation in which there is one operating point which satisfies both the process system and the ejector system. In these examples there is a steady flow of gases from the process system toward the ejector, minimizing the amount of water vapor which flows toward the process system by the process of diffusion. As a practical matter, no water is getting into the process system. If changes in load occur slowly, there will be a smooth transition from one steady-state condition to a nearby one – the system is moving slowly from one operating point to a nearby one.

So, what happens if the load changes occur quickly? And what about the mysterious lower limit of stability identified in Fig. 3.4? If an ejector curve like this is matched with a system operating curve which tries to find an operating point below the lower limit of stability, what happens? Let us find out.

Unstable operation with backfiring and cycling. The ejector performance curves we have encountered so far do not have the complete set of information required to understand just how an ejector backfires and cycles. Soon we will be looking at an expanded set of performance curves which define the operating characteristics of a critical ejector stage. But first consider a field incident which illustrates a simple backfiring and cycling operation.



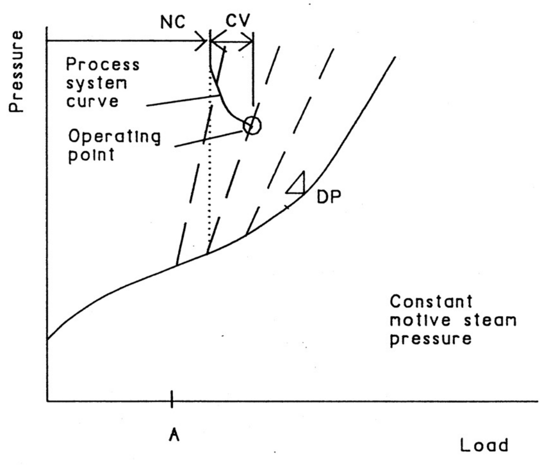


Figure 6.3 Process/ejector operating point, no control: (a) small noncondensable load, A = design noncondensable; (b) large noncondensable load.

A backfiring single-stage hogging ejector. A process plant operator asked an ejector troubleshooter how to make a hogging ejector work right. The single-stage ejector would evacuate a large dry system down to 50 torr, then "break" and slowly rise back to about 200 torr before picking up and resuming the evacuation cycle. It was noisy when it was evacuating properly and quiet when it was broken. The operator wanted it to be quiet all the time.

The explanation and recommendation for improvements was in two parts. First was the explanation that all atmospheric stages are noisy in normal operation and require an aftercondenser or silencer or muffler to suppress the noise. Second came the conclusion that the stage was not quite stable to no-load, and that once it was broken, some operating variable had to change to a less severe condition to permit the ejector to "pick up" and resume stable operation. For this particular stage with the motive steam pressure and atmospheric pressure which existed at the time, the suction break pressure was 50 torr and the pickup pressure was 200 torr. The cycle time was determined primarily by the ratio of the system volume to the ejector capacity. Other factors which could have affected the cyclic performance are the air leakage present in the process system and the temperature and geometry of the internals. The backfiring steam might partially condense in a cool system at the higher pressures, and some warm condensate might flash into vapor as the pressure lowered again.

Figure 6.4*a* shows the relevant performance curves for this ejector. Two features have been added to this type of curve since we defined it in Chap. 3. We now have two discharge pressure curves: the maximum discharge pressure (MDP) curve, which we now also identify as the pickup (PU) curve, and the new break (BK) curve. Recall that we previously identified the MDP curve as the one which is the more conservative because it identifies the ability to resume stable operation from a disrupted condition.

The second new feature that we have added is one broken condition suction curve, labeled with a discharge pressure of 760 torr, which represents normal atmospheric pressure. We could add many other broken curves, but this one is sufficient to explain how the hogging jet was working, and it is a comfortably small step on our learning trip. Especially notice here that this curve has been extended to the left of the zero-load line, representing negative flows-backfiring. At the extreme left the curve turns sharply down, representing the maximum possible reverse flow when the suction pressure is low enough that the reverse flow becomes choked. To the right, the curve rises and flattens, then approaches the basic performance curve, probably asymptotically.

Now, let us "walk through" the operation of the hogging jet and trace the path of the operation along the ejector curves. When the ejector is first turned on, the system is at atmospheric pressure, represented here by the 760-torr line. As the ejector removes air, the system pressure is reduced and we walk down the basic curve toward load *U* (unstable). Keeping an eye on the BK curve, we note that although it is still above 760 torr, the safety margin is getting small

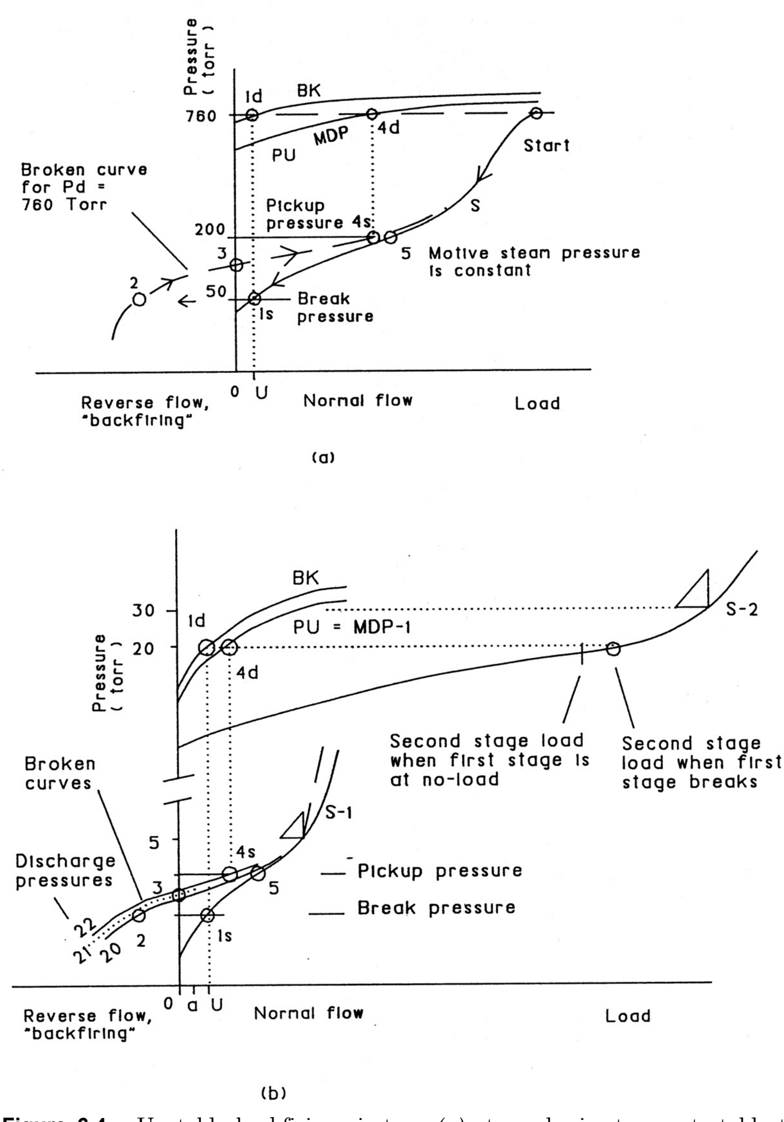


Figure 6.4 Unstable backfiring ejectors: *(a)* atmospheric stage not stable to no-load; *(b)* first stage of multistage condensing ejector with underdesigned booster second stage.

er. When the flow decreases to *U,* the lower limit of stability, we are at point 1, where the pressure is 50 torr. The ejector suddenly breaks, and the performance is now represented by the broken curve.

Because the system pressure is still at 50 torr and cannot change very quickly, we must "leap" from point 1 to point 2. At point 2 the flow has suddenly reversed, and the flow from the ejector back into the process system contains some or all of the motive steam and pos sibly air from the atmosphere. As pressure builds up in the system, we walk from point 2 to point 3, which marks the end of backfiring and the resumption of positive flow of gas through the ejector. As we walk from point 3 toward point 4, we keep an eye on the PU curve and note that it is now approaching 760 torr from the bottom side. As we reach point 4 at a pressure of 200 torr, the pickup pressure reaches 760 torr and we again must leap, this time to point 5, which is back on the basic curve. We have completed one cycle, and the ejector will continue cycling.

On another day or at a different altitude the atmospheric pressure might be 770 torr, in which case we could add a 770-torr broken curve slightly above this one and expect to see the ejector cycle between pressures such as 55 torr and 210 torr. Note also that if the atmospheric pressure were lower, this ejector might be stable down to noload, provided we approached it carefully. If we were to break this ejector at the no-load condition by momentarily turning the steam off or down, or by blocking the vent from the stage discharge, the ejector might backfire for a while, pick up at say 190 torr, and again pull down to a no-load pressure below 50 torr. This example explains why an ejector in marginal condition may work fine some days and get "sick" on other days when the combination of atmospheric pressure, steam pressure, and other factors is outside its operating limits.

Backfiring atmospheric stage of a multistage condensing ejector. A common example of ejector backfiring is the atmospheric stage of a multistage condensing ejector. If the stage is not capable of operating against the actual discharge pressure at no-load, it will backfire and pick up rapidly. The first stages and condensers act as a finite-size "process system" as seen by the last stage. The rapidly cycling pressure will pump water into the pressure measurement apparatus. The cycle time will depend on the air load, specifics of the last-stage curves, and the volume of the ejector system ahead of the last stage.

Multistage ejector which backfires at small loads. Now we are going to look at a multistage ejector which has a booster stage as stage 2. We will see how backfiring occurs in the first stage at small loads. The significant difference between the hogging ejector and the first stage of this ejector is that the hogging ejector always discharges against atmospheric pressure, whereas this stage sees a variable discharge pressure. Here the discharge pressure depends upon the gas load and the design of the second stage. The discharge pressure may cycle rapidly, as often as several times per second. For simplicity here let us consider only the first two stages of a multistage ejector designed for a load of dry air only. Relevant portions of the performance curves of these stages are shown in Fig. 6.4*b*.

This stage combination is not stable to no-load, a steam-economy feature which may cause a problem if water vapor will contaminate the process and if the system pressure is not being controlled.

First let us examine how the system will behave if the air load a is less than the lower limit of stable operation U for this stage combination. After the evacuation has reduced the system pressure below the design pressure of 5 torr, our "walk" down the basic performance curve is the same as the walk down the hogging-ejector curve. When we reach 2 torr, the lower limit of stability, at point 1, the ejector performance is suddenly described by the broken curve corresponding.to the current discharge pressure of 20 torr. Because the system pressure will remain at 2 torr until the new flow conditions create a change, we must "leap" as before from point 1 to point 2.

At point 2 there is a reverse flow of steam from the ejector back into the process system. In this situation the quantity of steam backfiring is a small fraction of the motive flow, thus there is no air flowing backwards through the diffuser. What happens now depends much upon the size of the process system. The suction pressure is still close to 2 torr, but it is rising slowly because air leakage into the process system continues and the backfiring continues.

At point 3 the suction pressure has risen to 2.5 torr and the interstage pressure has risen to 21 torr; point 3 is located at the intersection of a 2.5-torr horizontal line and a 21-torr broken curve, shown as a dotted line. At this condition the backfiring has stopped and the ejector is again removing air from the process system. Soon we reach point 4 at a suction pressure of 3 torr and an interstage pressure of 22 torr, with a total capacity greater than the lower stable limit U. A small "leap" occurs from point 4 to point 5, and the cycle is completed. Notice that the time-averaged net air-handling capacity during the cycle must equal the actual air leakage a. That air in-leakage to the process system is unchanged by the backfiring because the system pressure always remains below 400 torr, establishing critical flow through all air leakage openings.

Qualitatively, this is how backfiring occurs at small loads below the lower stable limit of a first stage which is followed by a marginal design booster stage. Shifting the curves and changing their shapes will alter the amount of backfiring and the range of pressure cycling.

Behavior of an ejector stage when overloaded ("broken"). My need for a better understanding of the behavior of ejector stages in the "broken" state became evident during early drafts and reviews of this chapter. Ejectors "backfire" water vapor into the suction system under some circumstances, and many ejectors are in systems where the pressure is controlled by loading the ejector with noncondensable gas. That method of control often causes a multistage condensing ejector to operate with one or more stages in a broken state.

To help answer some questions that bothered me, Nash-Kinema, Inc., offered to let me conduct a special test of a Y -stage ejector under various overload conditions in a test stand. I gained several useful insights from studying that single page of test data for a few weeks. I offer you the results in the most useful form I can devise, with important warnings.

I am quite pleased with the results. They answer several questions that have nagged at me for years, and they explain some contradictory observations and rules I have encountered. They should be regarded as only preliminary or tentative, however, because they have not been seasoned by time and because they are based on only one test.

Generalizations are based in part upon the laws of similitude and ideal gases. In a more rigorous form, some of the single-line curves might become a family of curves representing different design compression ratios, expansion ratios, and hardware geometry. Some of the simple equations may acquire correction factors for similar reasons. The equations and curves should be used to predict only approximate behavior, with no expectation of high accuracy.

I will describe the test and the use of the results in some detail for two reasons: It is the best way I know of to convey some of the concepts, and it exposes some of the details of the analysis for others to critique or improve upon. If you are not interested in the details of the analysis, you may wish to skip forward to the results.

The test ejector had 3-in suction and discharge connections and a barstock diffuser. It used 252 pph of 400-psig steam, with design-point conditions of 224 pph air at 63 torr and discharge pressures of 242 torr pickup and 247 torr break. The no-load pressure was 7 torr. Suction and discharge piping volumes were small. The performance curves will be shown in detail, representing smoothed interpretations of the data.

Constant-load tests. We applied a test air load using a critical-flow multiple-orifice unit and observed the suction pressure when the stage was in stable operation. Then we began to close the discharge valve and continued closing it until the suction pressure "broke" and jumped, typically 20 torr, to a new steady pressure. We recorded the broken suction pressure and the corresponding discharge pressure. As we continued to close the discharge valve, we obtained pairs of suction and discharge pressures up to 400 torr suction pressure and 500 torr discharge pressure. Then we slowly opened the discharge valve to find the discharge pressure at which the stage resumed stable operation, the discharge pickup pressure. The procedure was repeated for nine loads, from no-load to 256 pph air. The test data are summarized in Table 6.1.

TABLE 6.1 Overload Behavior of a Y-Stage Steam Jet Ejector

Stage size: 3-inch suction and discharge connections
Motive steam: 252 pph at 400 psig, dry and saturated
Test date: October 30, 1992

Load	Suction pressure, torr		Discharge pressure, torr		Discharge pressure: in Hg abs versus suction pressures, in Hg abs[1]						
pph	stable	broken	break	pickup	3	4	6	8	10	12	15
0	6.8	---	---	---	10.6	11.2	12.8	14.4	15.8	17.8	20.2
10	---	22.1[2]	---	---	---	---	---	---	---	---	---
32	18.3	---	232	220	---	---	---	---	---	---	---
64	25.9	44.2	237	227	10.8	11.2	12.2	13.6	15.0	17.0	20.6
100	33.5	---	247	237	---	---	---	---	---	---	---
128	38.6	64.5	247	242	10.6	11.2	12.2	13.6	15.0	16.8	19.2
160	47.5	67.1	247	242	10.4	11.2	12.2	13.8	15.0	16.8	19.2
192	55.1	70.8	247	242	10.2	10.9	12.0	13.4	14.8	16.6	19.0
224	64.7	83.6	252	247	10.2	10.8	11.8	13.2	14.8	16.6	19.2
256	75.4	92.4	247	242	---	10.6	11.6	13.0	14.8	16.4	19.0

[1] Example-at 64 pph load, discharge pressure is 13.6 in. Hg to create suction pressure of 8 in. Hg.
[2] After error correction.
SOURCE: Courtesy Nash-Kinema, Inc.

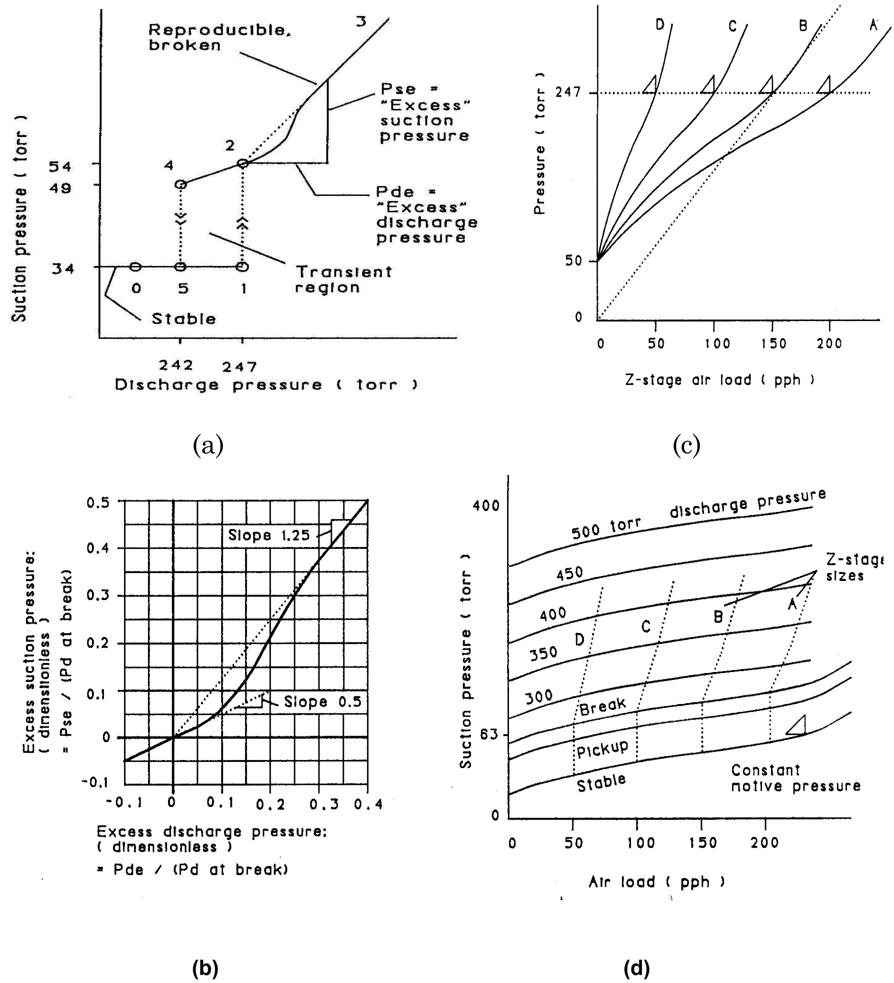


Figure 6.5 Interpretation and presentation of test data for predicting the "broken" state performance of an ejector in a two-stage ejector system: *(a)* constant-load break and pickup test; *(b)* excess suction pressure versus excess discharge pressure, dimensionless, constant load; *(c)* various Z stages to be used with test Y stage to form a two-stage condensing ejector; *(d)* performance curves for two-stage condensing ejector consisting of tested Y stage followed by various sizes of Z stages.

Figure 6.5*a* is a representative test, at a constant load of 100 pph air. Walking through this test, we began at point 0, where the discharge pressure P_d is less than the break pressure for that suction load. The suction pressure P_s remained steady at 34 torr. We closed and opened a valve at the ejector discharge to vary P_d. We slowly increased P_d to the discharge break pressure of 247 torr at point 1, then the suction pressure rose suddenly to the break value of 54 torr at point 2 on the broken curve. As we continued to increase P_d, the operating point moved along the reproducible curve between points 2 and 3.

Now, we reversed the procedure. As we reduced P_d, the operating point moved down the broken curve, passed point 2, and reached the lower end of the curve at point 4. This is the pickup point, where P_d is the pickup pressure of 242 torr and P_s is 49 torr. Then P_s dropped suddenly to the stable value of 34 torr at point 5, completing a break/pickup cycle.

The portion of the chart where P_d is between 242 and 247 torr is of special interest because no simple path describes the transition of the operating point from one curve to the other. Adding a large volume to the suction system will slow the action in a manner to be described later. For now, be advised that a 100-ft^3 suction system will slow the rise from point 1 to point 2 to about 5 s/torr, or about 100 s for the transition. The drop from point 4 to point 5 will take about 75 s.

Let us modify the test procedure by waiting until P_s is somewhere between 34 and 49 torr, then quickly adjusting P_d to 244.5 torr, halfway between the break and pickup pressures. P_s will become nearly constant, drifting slowly up or down as we make slight adjustments in P_d or the motive steam pressure. While in this transient region, we can adjust P_d widely above or below 244.5 torr, and observe the speed of rise or drop to vary in direct proportion to the increase or decrease in P_d.

Generalized constant-load broken curve. To estimate the approximate shape of broken performance curves for ejectors at generalized design conditions, it is useful to create some dimensionless curves and equations.

I will now define some terms to be used in a dimensionless correlation. Refer to Fig. 6.5*a* and observe the definitions of "excess discharge pressure" and "excess suction pressure." As an example, at an operating point where P_s = 70 torr and P_d = 260 torr, the excess suction pressure = 70 - 54 = 16 torr, and the excess discharge pressure = 260 - 247 = 13 torr. Refer to Fig. 6.5*b* for the next step, in which the excess pressures are divided by the break discharge pressure (247 torr for 100-pph load), yielding dimensionless excess pressures. Only the lower portion of the curve is shown, to describe approximately the nonlinear region of the reproducible broken curve in the immediate vicinity of the break and pickup points. It is a well-behaved straight line at higher pressures.

As an example of the use of this curve, let us estimate the suction pressure corresponding to a load of 100 pph air and P_d = 300 torr. In this situation we remember that the break operating point is P_s = 54 torr and P_d = 247 torr. Later, we will find how to estimate these pressures. This excess P_d = 300 - 247 = 53 torr, and in dimensionless form equals 300/247 - 1 = 0.215. From Fig. 6.5*b* we find the dimensionless excess P_s to be 0.23. In dimensional form, the excess P_s = 0.23(247) = 57 torr. Adding this to the broken value of 54 torr yields 111 torr as the estimated broken value of P_s.

Performance curves for a two-stage condensing ejector. The broken behavior of a multistage condensing ejector can be largely understood and predicted by studying how a two-stage condensing ejector works. For simplicity, we will begin by considering ejectors handling only dry air loads. Figure 6.5*c* shows representative performance curves for four different Z stages, which we will combine separately with our tested Y stage. The stages have design air capacities of 50, 100, 150, and 200 pph at a design pressure of 247 torr. We may regard these as the actual air capacities with negligible water vapor, or we may consider the curves to be adjusted for the water vapor accompanying the stated dry air loads.

The observant reader may have noticed that here I refer to the Y-stage discharge break pressure as the design suction pressure for the Z stage, in contrast to the pickup pressure usually preferred. This is because the break pressure operating point is used as a reference in several correlations and examples.

Combining the Y-stage test data with the Z stages produces the family of curves in Fig. 6.5*d*. The dotted lines indicate the estimated broken curves with each of the Z stages. This gives a useful picture of the overall behavior under air overload conditions, but the scale is so compressed that much useful detail is lost in the transient break/pickup zone. For a closer look, we must expand that pressure regime, examine it qualitatively, and do some representative calculations.

Performance curves of the test Y stage in the vicinity of the break and pickup conditions are shown in Fig. 6.6. Because this set of curves is cluttered with details, let us examine them before proceeding. Note the absence of a set of discharge-pressure break and pickup curves. That information is displayed in other ways. The discharge break pressure is represented by tick marks and pressure values below the stable curve. The stable suction curve is a heavy line, as are the important suction break and pickup pressure curves.

To identify those curves and demonstrate their use, let us trace the previous example test with a constant load of 100 pph air. We note

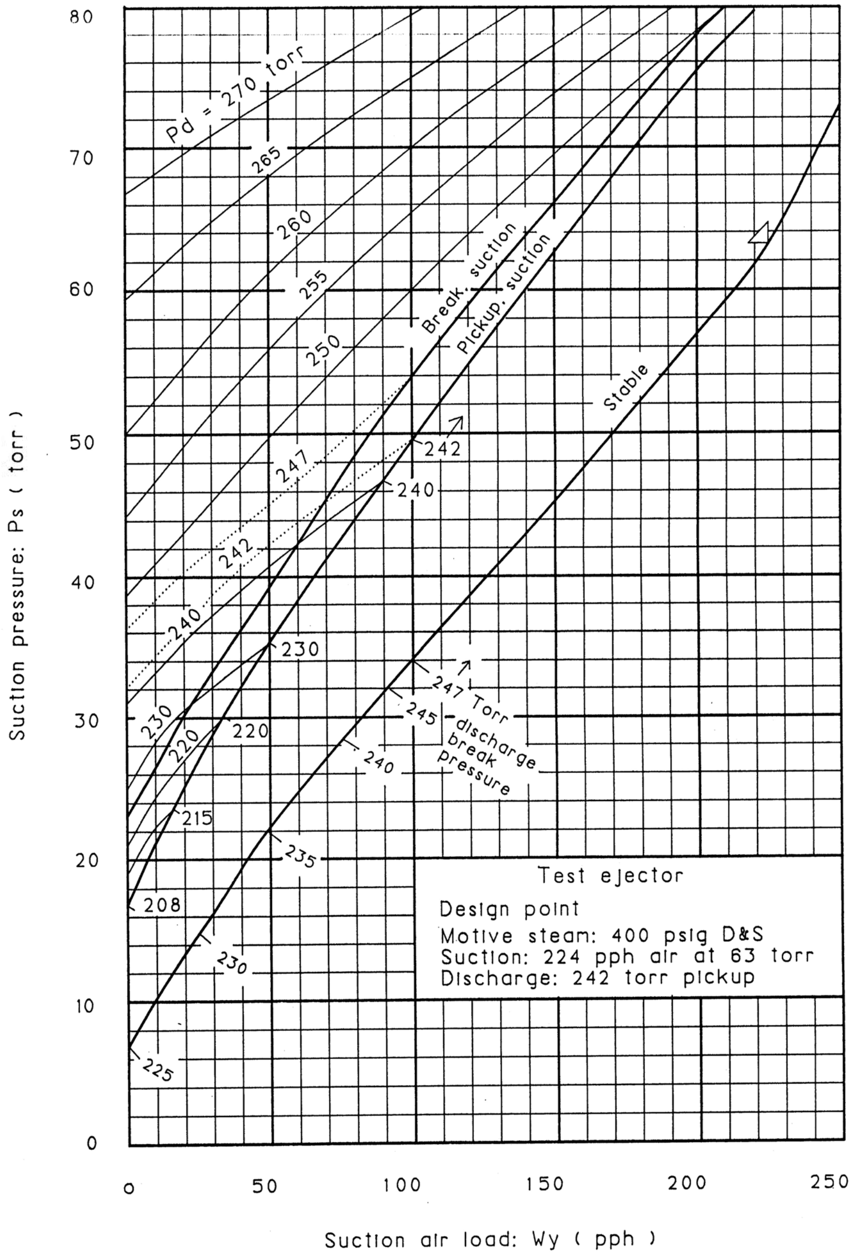


Figure 6.6 Performance curve map for test Y stage, including broken condition operating points.

that the stable suction pressure P_s is 34 torr, and that the discharge break pressure is 247 torr. When we increase P_d to 247 torr, the stage breaks and the operating point moves up to the suction break curve at 54 torr. As we increase the discharge pressure to 260 torr, P_s increases to 70 torr. Then, when we reduce P_d to 242 torr, P_s decreases to 49 torr and the stage picks up. P_s then drops to 34 torr, and the break/pickup test is complete.

We have demonstrated that these curves predict the same behavior as the previous constant load curves did. We have not, however, demonstrated what actually happens in the transition region during a break or pickup event. This is important to know, because we will pass through the transition pressure range during the initial evacuation and during an air overload situation, and because we may be trying to control to a system pressure in this range using a noncondensable gas load. Let us next take a detailed look at the transition dynamics, qualitatively and quantitatively.

Break and pickup cycling in the transient region. The first tool we need is something to slow the "instantaneous" events so they can be observed. Volumetric capacity is that tool, shown in Fig. 6.7 as a suction volume added ahead of the Y stage and an interstage volume added between stages. I used V_z = 10 ft^3 between stages to let that volume be one or two orders of magnitude larger than the ejector, and V_s = 1000 ft^3 for the suction system volume to make it two orders of magnitude greater than the interstage volume. I considered the specified interstage volume to be the effective dry air volume, after correcting for the presence of steam in the condenser. The suction system is easier to idealize, for this purpose being a dry cube, 10 ft on each side.

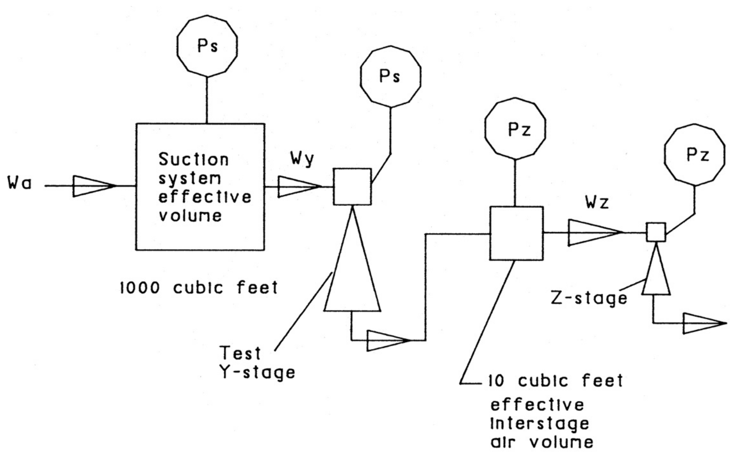


Figure 6.7 Two-stage ejector system, example for dynamic analysis.

The Y-stage ejector is about 2.5 ft overall, with internal velocities at design conditions ranging from 200 to 3000 ft/s, resulting in a time constant of the order of 0.01 s. The time for an air pressure wave to cross the box is of the same order, about 0.01 s.

In summary, this idealized model has a suction volume large enough that its pressure remains nearly constant during a single break/pickup cycle. The interstage volume is much smaller, and the pressure there will vary between the Y-stage break and pickup discharge pressures. The ejector stages are so much smaller that they are assumed to be always in steady-state operation.

P_s is the pressure in the suction volume and the Y-stage suction. P_z is the pressure at the Y-stage discharge, in the interstage volume, and in the Z-stage suction. W_a is the air load into the suction volume, W_y is the air load into the Y stage and the interstage volume, and W_z is the air load into the Z stage.

For the first example analysis, we use a Z stage with a capacity of 150 pph dry air at 247 torr, curve B in Fig. 6.5*c*. At that design point the curve is tangent to the dotted line from the origin, with a slope of 247 torr/150 pph = 1.65 torr/pph. Conversely, the change in capacity with pressure is 0.61 pph/torr. These data will be useful shortly.

Now let us examine the break and pickup behavior in the transient region, referring again to Fig. 6.6. We slowly increase W_a to 150 pph and observe the suction pressure at the Y stage as it increases to 45 torr, then breaks and begins to drift up toward 66 torr. When the first break occurs, we hear a brief change in the sound of the ejector, then several seconds of normal sounds, then a brief change, etc. The brief changes correspond to operation in the broken mode, and the time from the beginning of one break to the beginning of the next break is the cycle time for a break/pickup cycle. Most of the cycle time is spent in the stable mode.

We observe the pressure rising and the cycle time becoming shorter. At a suction pressure of 55 torr, the cycle time is at its minimum, and the ejector spends only half the cycle time in the broken mode. As the suction pressure continues to rise toward 66 torr, the cycle time again increases, and the ejector spends most of its cycle time in the broken mode. Finally, the cycling stops and the ejector remains in the broken mode, with the operating point always on the broken operating curve. If we slowly reduce the load to about 147 pph, the ejector will pick up. The pickup/break cycling will start, and the suction pressure will drop slowly until it again reaches the stable curve at about 44 torr.

What exactly was happening during these events? Can they be predicted quantitatively? How do the different parts of the system interact? This is the fun part! At least, it was very satisfying for me to understand several phenomena that had puzzled me for years.

Let us repeat the test, but this time we will follow the operating point as it moves around on the performance curves, then we will calculate the break/pickup cycle times and the corresponding changes in the suction pressure.

We increase the air load slowly to 150 pph; P_s is 45 torr, and P_z is 247 torr. Suddenly the break occurs and P_s begins rising. Because the first break/pickup cycle is more complicated to analyze than later cycles, let us wait until P_s rises to 46 torr before doing a detailed analysis. When a break occurs at 46 torr, the operating point instantly jumps to the intersection of the lines on which P_d = 247 torr and P_s = 46 torr. Here we find W_y = 55 pph. Think about this a little. The suction volume has 150 pph air load flowing into it and only 55 pph leaving it, so air is accumulating in it at the rate of Acc_s = 95 pph and the pressure is increasing.

Now look at the interstage volume. The pressure there is still 247 torr, and the Z stage is still removing 150 pph air, but the Y stage is supplying only 55 pph The accumulation Acc_z = 55 - 150 = -95 pph. The pressure in this smaller volume changes more rapidly. When this pressure reaches the Y-stage pickup pressure, the Y stage will again resume stable operation. The Y stage pickup pressure will be found at the intersection of P_s = 46 torr and the suction pickup line. By interpolation, we find that the line P_d = 239 torr passes through this point, the stage pickup pressure. The pressure decrease from break to pickup during this cycle is 247 - 239 = 8 torr. At the pickup operating point we find W_y = 88 pph.

From our previous examination of the slope of the Z-stage curve (0.61 torr/pph near the design point), we can estimate the change in Z-stage capacity over this interval to be 0.61(8) = 4.8 pph. This results in a capacity at pickup of 145.2 pph.

The suction accumulation just before pickup is Acc_s = 150 - 88 = 62 pph. The interstage accumulation just before the pickup point is now Acc_z = 88 - 145.2 = -57.2 pph.

Now we have enough information to calculate cycle times and pressure changes. We find the cycle times to be determined by the interstage volume, the values of Acc_z, and the change in P_d between break and pickup. Ideal gas laws will show the pressure rise rate in the suction and the interstage systems to be

$$P_{xr} = \frac{\mathrm{Acc_x}}{0.36\mathrm{V_x}} \tag{6.1}$$

where P_{xr} = system pressure rise, torr/s

Acc_x = accumulation of air in system, pph

V_x = volume of system, ft^3

x = s for suction system, z for interstage system

Because the accumulations vary continuously during the cyclic operation, an average value must be used for these simple estimates. I recommend using the logarithmic mean value. Thus, during the broken time interval, Acc_s = lm(95,62) = 77.3 pph and Acc_z = lm(95,57.2) = 74.5 pph.

The average suction pressure rise rate is P_{sr} = 77.3/360 = 0.215 torr/s Also, P_{zr} = -74.5/3.6 = -20.7 torr/s. Thus, the time for the interstage pressure to change from break to pickup (8 torr) is T_{break} = 8/20.7 = 0.387 s. The suction-pressure rise in this time is 0.215(0.387) = 0.083 torr.

The stable (picked-up) half of the cycle is easier to analyze. Upon pickup, the operating point moves to the stable curve and stays there until break at W_y = 154 pph. During this half of the cycle, Acc_s = 150 - 154 = -4 pph. Acc_z begins at 154 - 145.2 = 8.8 pph and ends at 54 - 150 = 4 pph. Use Acc_z = lm(8.8,4) = 6.1 pph. P_{sr} = -4/360 = -0.0111 torr/s, P_{zr} = +6.1/3.6 = 1.69 torr/s, and T_{pickup} = 8/1.69 = 4.73 s. The suction-pressure change is -0.0111(4.73) = -0.053 torr.

The net pressure change in this break/pickup cycle is +0.083 0.053 = 0.030 torr. The total cycle time is 0.387 + 4.73 = 5.12 s. And the average rate of rise of the suction pressure

$$P_{sr} = 0.030/5.12 = 0.0059 \text{ torr/s}$$

As an overall check on this result, we may step back and look at the entire system, noting that it is being supplied with a constant 150 pph air, and that the Z-stage capacity varies between 145.2 and 150, averaging 147.6 pph. Thus, Acc_s = 150 - 147.6 = 2.4 pph, and P_{sr} = 2.4/360 = 0.0067 torr/s. This is within 15 percent of agreement with the step-by-step calculations.

Consider now the contributions of several parameters to the results we have seen, remembering that we slowly loaded the ejector system to the break load and kept the load constant during the transient condition. The suction volume affects only the rate of rise of the suction pressure. The primary contributions of the Y stage are the stable curve and the discharge break/pickup pressures corresponding to the Z-stage capacity. The interstage volume affects only the speed of cycling between break and pickup. The Z stage contributes only its capacity at the Y-stage discharge break pressure.

The Z-stage curve slope in the break/pickup region is determined solely by its design point, assuming that its design point here is its maximum-efficiency point. An off-design mismatch here would alter the slope somewhat, slightly changing the calculated results. The Y-stage discharge break/pickup pressure interval, 8 torr in this example, thus combines with the Z-stage design point to determine the range between break and pickup loads in the Z stage.

Repeating this analysis at P_s = 55 torr shows the total cycle time to be about 0.8 s, the minimum. This corresponds to a frequency of 1.25 cycles per second. Because the test stand volume between the ejector and the throttling valve was about 0.1 ft^3, the approximate cycle frequency was about 100 cycles per second. This is well within the audible frequency range and corresponds with observations of "humming" sounds reported during some tests.

The broken and picked-up time intervals are equal, and the suction-pressure rise rate is unchanged. Observe that the horizontal distance between the stable curve and the average of the broken and pickup operating points is about 80 pph for this Y stage, over a wide range. At P_s = 46 torr, most of this interval is to the left of the airhandling capacity; thus the broken interval is brief and the stable interval is long. Near 66 torr, the reverse situation may be seen: The pickup interval is brief and the broken interval is long.

Now a brief word about the first few break/pickup cycles. The first break cycle has a very short break time and a very long picked-up time. If we use the previous simplifying assumption that P_s remains constant during the cycle, we calculate the interstage accumulation to be zero just before again reaching the break condition, so that the cycle is never completed. Therefore we must calculate the small rise in pressure during the cycle by a trial-and-error process. Then the accumulation rate will be nonzero as the next break condition is approached.

An alternative analysis method, which avoids this problem, is to determine the initial suction and interstage pressures and rates of rise, then project them for a time interval much shorter than the cycle time. Using the new pressures and rates, another step is taken. This tedious method is well suited to computer analysis, provided the curves are properly represented. The results are comparable.

Rate of suction-pressure rise. If you plan to control system pressure within the transient pressure range by loading the ejector with non-condensable gas, it is useful to be able to predict the rate of pressure rise as you design the pressure control system. Here is an estimating method.

$$P_{sr} = \frac{W_a - 0.5(W_{z,bk} + W_{z,pu})}{0.36V_s} \tag{6.2}$$

where P_{sr} = suction pressure rise, torr/s
W_x = total air load to system, pph
$W_{z,bk}$ = Z-stage air capacity at break, pph
$W_{z,pu}$ = Z stage air capacity at pickup, pph
V_s = suction system volume, ft^3

As an example, consider the test system, which had a suction volume of the order of 0.5 ft^3. For the previous 150 pph constant-load example, $W_a = W_{z,bk} = 150$ pph and $W_{z,pu} = 145.2$ pph.

$$P_{sr} = \frac{150 - 0.5(150 + 145.2)}{0.36(0.5)} = 13 \text{ torr/s}$$

At that rate, the 20-torr increase in pressure at the break would occur in about 1.5 s, "instantaneous" to the casual observer. A 50-ft^3 system (very small for this size Z stage) would experience a rise of 0.13 torr/s, slow enough for the control system to make continual corrections and maintain the desired pressure in this transient region.

Predicting backfiring. Because water vapor is sometimes a nuisance or a costly contaminant in process systems, it is useful to predict the circumstances under which motive steam will backfire from the ejector into the process system. By examining the performance curves for the test Y stage, we can make some generalizations about this stage and tentatively extend them to other ejector stages and multistage combinations. We observed that when the test stage broke at $P_s = 45$ torr, the broken operating point was at 50 pph, one-third of the capacity before the break occurred. If the break occurs at a lower P_s, the broken operating point will move to the left.

At $P_s = 36+$ torr, the break and pickup discharge pressures are 247 and 242 torr, and the broken operating point is on the zero-capacity line. Breaking at $P_s < 36$ torr (air load less than 110 pph) will result in the broken operating point being to the left of the origin, indicating negative flow, backfiring water vapor into the suction system. The design compression ratio at this borderline condition is 242/36 = 6.7. Because the load at the break point is removed only about 50 percent from the design load of this stage, we may anticipate that a stage designed for a ratio of 6.7 would behave approximately the same way. Thus, we may generalize that stages breaking at a compression ratio greater than 6:1 will probably backfire. I found this confirmed by a rule of thumb I encountered recently.

It is the break compression ratio, not the stage design compression ratio, that determines whether backfiring will occur. As an example, assume the test stage is the first stage of a two-stage ejector designed for 80 pph air and roughly 144 pph condensable vapor at 63 torr. The design-point compression ratio is 242/63 = 3.8, but the "worst-case" break-point compression ratio when loaded with noncondensable gas is 220/30 = 7.3. Backfiring is expected with this stage when it is loaded with only noncondensable gas at pressures between 30 and 36 torr. To prevent backfiring in that pressure range, at least 30 pph condensable vapor must be present in the load to raise the break suction pressure above 36 torr.

Detecting backfiring in an existing ejector. Qualitative symptoms of backfiring are repetitive break and pickup sounds, a pulsating manometer or pressure gauge, water in the manometer, and heating of the suction line at pressures above 100 torr. Pressure pulsations may be too rapid for the pressure gauge to respond. Suction-line heating may be hard to detect at lower pressures because the saturation temperature is low.

To test whether a two-stage ejector will backfire, slowly increase the dry air load and observe the suction pressure at which the break occurs. Maintaining that air load, measure the discharge break pressure (Z-stage suction). Next, remove the air load from the Y stage and insert it at the suction to the Z stage, adjusting it until the Z stage is again at the break pressure. Measure the Y-stage suction pressure, which is its no-load pressure at the break discharge pressure. If this pressure is higher than the suction pressure at which the stage first broke, then some backfiring is occurring in the transient-pressure range.

Avoiding backfiring. If you prefer to control pressure with a noncondensable gas bleed, yet want a design that intrinsically avoids backfiring, then you must specify that design requirement. The manufacturer has two options. A direct approach is to size the secondary stages (following the first condenser) with sufficient air-handling capacity to support the first stage beyond the suction pressure below which backfiring is possible. This requires more steam and water, larger secondary stages and condensers, and higher first cost. Another way to avoid backfiring is to use more stages, resulting in a smaller compression ratio in the stage before the first condenser, leading to equal or lower steam and water usage and higher first cost.

If you specify this requirement, I recommend that you ask for confirming performance curves containing the information necessary to demonstrate compliance with the specification. Because the field test for compliance is simple, I recommend that you perform it when the ejector is new, and periodically thereafter. As the ejector wears and fouls, it will backfire more readily.

Condensing ejectors with three or more stages. If one stage breaks and backfires because of air overload, then generally all stages ahead of it will also backfire. Typically the Y stage is the first to break, but if the ejector has a large Z stage for fast evacuation, then the break may occur first in the X stage.

When the break occurs as the result of a slowly increasing air load in the Y stage, as we have examined, it creates a larger effect in the X stage. Assume, for example, that the X stage has a suction pressure at break of 10 torr. We found earlier that the Y-stage transient-pressure rise was about 20 torr, 30 percent of the design suction pressure. The corresponding rise in the X stage will be roughly 3 to 5 torr, *plus* 1.25 times the 20-torr rise in the Y stage, for a total rise of about 30 torr. Thus, the X-stage suction pressure at break will move from 10 torr to about 40 torr in a constant-load test.

The rate at which the suction pressure rises is predicted by Eq. (6.2), which contains only the suction volume, the break air load, and the Y/Z-stage interaction. Even large booster stages will not increase the basic air-handling capability.

Broken curves above the transition zone. To predict the approximate shape of the air overload curve for a two-stage ejector, use the Z-stage and Y-stage curves as shown in Fig. 6.5*c* and *d* to create the broken curves shown in Fig. 6.5*d* as dotted lines. For example, use the 200pph size Z stage, curve *A*. Find the load at which the Z-stage suction pressure equals the Y-stage discharge break pressure. Mark that as the break load and draw a dotted vertical line to the suction-pressure break line. This is the break transition region. Now increase the load by a small increment and find the corresponding pressure at the Z stage. On the Y-stage curves, locate the corresponding operating point at the intersection of the airload line with the discharge-pressure line. Extend the dotted line to this point, and continue the procedure.

Generalizing results to other ejectors. The simplest way to generalize the results of this test to other Y stages and X stages is to use the approximations suggested by the ideal gas laws and laws of similitude and relabel the curves in Fig. 6.6 appropriately.

Capacity is the simplest change, accomplished by simply relabeling the load scale. If the design load were 10 times as large at the same pressure conditions, then we would simply multiply the load scale values by 10 and multiply the motive steam usage by 10. The larger unit would be a little more efficient. If we reduce the capacity, we follow the same general procedure, correcting the steam usage for size as described in Chap. 4.

Changing the suction and discharge pressures is simple if the compression ratio does not change and if the motive steam pressure changes in proportion. For example, if all pressures were reduced by a factor of 1:4, the motive pressure would be about 104 psia, the design P_s = 16 torr, and the design P_d = 61 torr.

Changing compression ratios becomes more difficult. Within perhaps 50 percent of the design point on this curve, we may assume this curve resembles the curve of a stage designed for the new compression ratio. Below 50 percent on this curve (compression ratios greater than 6), I regard the curves as qualitatively representative, but not useful for accurate quantitative predictions.

Changing steam pressure within the range of 100 to 400 psig involves a steam-pressure multiplier adjustment, and the expectation is that the curves will not change much in the region where compression ratios are less than 10.

Relabeling the test curves to represent another ejector stage. To illustrate how Fig. 6.6 may be relabeled to predict the behavior of another stage, assume that we wish to estimate the performance curves for an ejector using 150 psig steam to compress 50 pph air from 10 torr to 70 torr pickup pressure, a compression ratio of 7.0. Because the design compression ratio is greater than 6, we predict that the ejector will backfire unless the secondaries are designed for a larger air load. This pressure combination is in the X-stage range.

Inspecting the curve, we find the compression ratio of 7.0 to correspond to a point on the stable curve at which P_s = 35 torr and W_a = 105 pph. The adjustment scale factor for pressure is 10/35 = 0.29, and the adjustment scale factor for load and steam usage is 50/105 = 0.48. The steam pressure to maintain similitude is 0.29(400 + 14.7) = 120 psia, or 105 psig. At this pressure the steam consumption would be 0.48(252) = 121 pph. Correcting for the actual pressure of 150 psig, using Fig. 4.9*a*, the estimated steam usage is 121/1.07 = 113 pph. No size correction factor is required.

Now examine Fig. 6.8 to see how to mark up Fig. 6.6 to represent the new X stage, using the pressure correction factor of 0.29 and a load correction factor of 0.48. First, move the design-point sYmbol to the new design point and label the capacity and pressure at that point. Then, line out the old numbers on the flow scale and replace them with the new scale numbers. The new load value of 40 pph corresponds to 40/0.48 = 83 pph on the old scale. Next, revise the pressure scale; the new 20 torr label corresponds to 20/0.29 = 69 torr on the old scale. Finally, revise all the P_d pressure values in the chart; old 260 torr becomes 260(0.29) = 75 torr, old 242 becomes 70, etc.

Generalized parameters for stages operating in the broken mode. Figure 6.9 contains several useful parameters obtained from analysis of the test curves. Three parameters. are plotted versus compression ratio, with dotted lines as reminders that the conditions most remote from the design point for the tested stage are to be used with caution.

Observe that here the compression ratio is defined as the design-point discharge pickup pressure divided by the design-point suction pressure. Two of the parameters may be used to calculate broken con

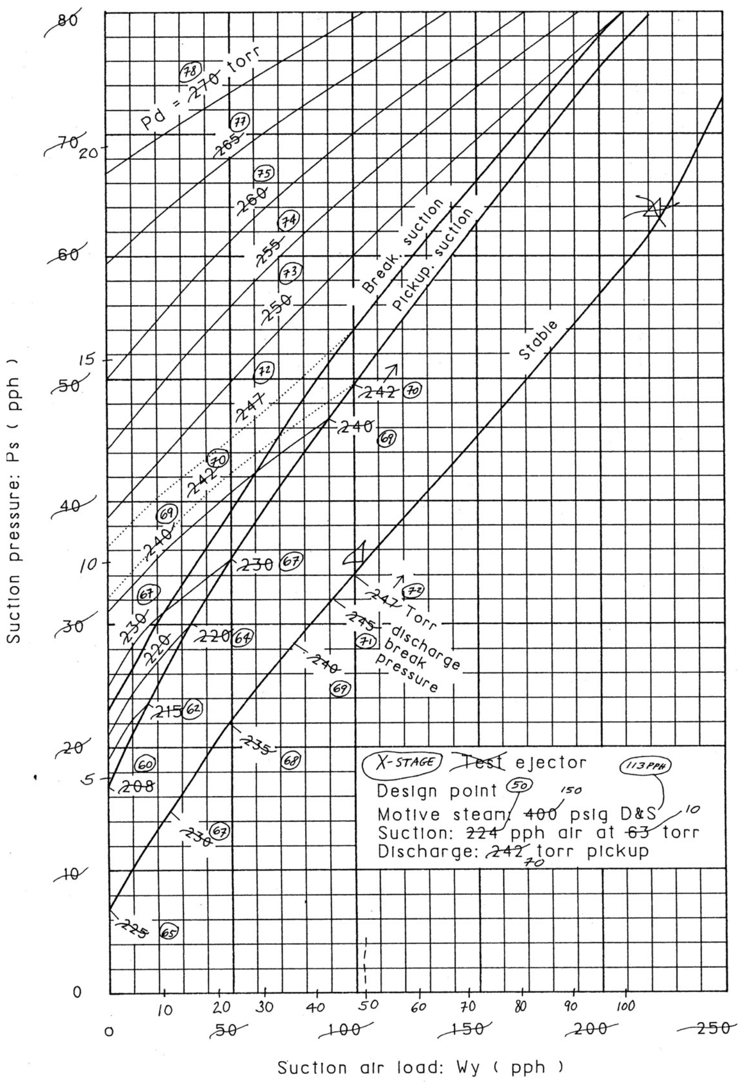


Figure 6.8 Test stage curves rescaled to estimate X-stage curves.

dition curves to supplement a set of conventional curves, which contains only the stable suction curve and the discharge pickup pressure (MDP) curve. The third parameter predicts the minimum break/pickup cycle time in the transient region.

As an example, assume that we have a standard set of ejector curves and wish to create some constant-load data points at a condition where the suction pressure is 15 torr, the discharge pickup pressure is 80 torr, and the air load is 50 pph.

The compression ratio is 80/15 = 5.3. From Fig. 6.9a, we find the

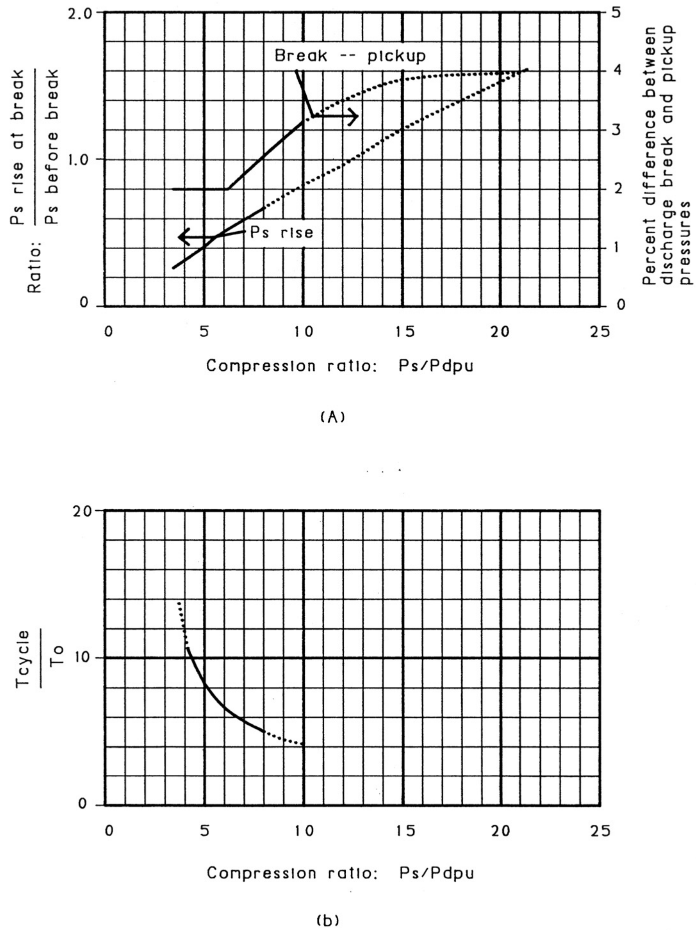


Figure 6.9 Break/pickup parameters for ejector stages operating in the transient region: *(a)* suction-pressure rise and discharge-pressure break/pickup difference; *(b)* minimum time to complete a break/pickup cycle in the transient region, dimensionless.

suction-pressure rise ratio to be 0.45, so the suction pressure will rise 0.45(15) = 7 torr, to 22 torr, after the break. Also from Fig. 6.9*a*, we find that the interval between the break and pickup pressures is 2.0 percent of the .discharge pickup pressure, or 1.6 torr, resulting in a break pressure of 81.6 torr. Use Fig. 6.5*b* to construct the rest of the data for this constant-load condition.

Repeat the construction at several loads and draw smooth curves through the break and pickup points. Then draw smooth curves through lines of constant P_d. The result is an approximation of Fig. 6.6.

To estimate the break/pickup cycle time midway through a transition region, first obtain the value of T_{cycle}/T_o from Fig. 6.9*b*, 7.5 for CR = 5.3. Next calculate T_o,

$$T_o = \frac{0.36 V_z P_{d\mathrm{bpu}}}{W_{az}} \tag{6.3}$$

where T_o = reference time, s
V_z = effective air volume between stages, ft^3
$P_{d\mathrm{bpu}}$ = Y-stage break pressure minus pickup pressure, torr
W_{az} = air capacity of Z stage at break, pph

In this example, let V_z be 1.0 ft^3.

$$T_o = \frac{0.36(1.0)(1.6)}{50} = 0.0115 \text{ s}$$

and T_{cycle} = 7.5(0.0115) = 0.086 s, corresponding to a cyclic frequency of 12 cycles per second.

More information is needed. Here I encourage other people to publish descriptions of these phenomena. I believe there is plenty of room for improvement of my current treatment of this subject. It is important in selecting ejectors, specifying them, designing the instrument system for controlling pressure in the vacuum system, and operating the ejectors.

6.3 Process Condenser Noncondensables "Mask" Excess Area

A distillation column is a typical system in which pressure control is desired. Visualize a system that is initially at steady state with vapor flowing up, liquid flowing down, no noncondensable gases in the system, and the condenser exactly matched to the vapor production rate. Now introduce a steady flow of air leakage, instrument blowback gases, etc. The upward flow of vapors in the system sweeps all noncondensable gases toward the process condenser. In the condenser, the noncondensable gases accumulate in the space most remote from the vapor inlet. If the noncondensable gas is not vented, the gas volume increases until it begins to cover some of the heat-transfer area, "masking" it from the condensing operation. Because part of its area is masked by the blanket of noncondensables, the condenser is unable to condense vapor as fast as it is created, and the pressure begins to rise.

As the pressure rises, the condensing temperature rises, increasing the temperature difference and restoring the heat-transfer capacity of the condenser. By adjusting a valve at the condenser vent, we may adjust the volume of noncondensable gases, thus adjusting the effective heat-transfer area and the condensing pressure. If the process system pressure is less than atmospheric pressure, a pumping device such as an ejector is needed to remove the vent gases. This leads us to the first pressure control system, simply throttling the flow from the process system to the ejector.

If we want to lower the system pressure, we simply open the valve a little to remove more gases and expose more heat-transfer area. Conversely, to raise the pressure, we close the valve a little. How fast will the pressure rise? It depends on how fast the noncondensable gases accumulate, which is in turn dependent on the total of air leakage, instrument blowback gases, dissolved gases liberated from feed streams, and noncondensable gases created by the process. If the system is very leaktight and clean, the rate of pressure rise may be too slow, even with the vent valve completely closed. A small quantity of noncondensable gas may be introduced between the condenser and the control valve to speed the response. Nitrogen is often used for this, but atmospheric air or other gases may be acceptable and more convenient.

6.4 Pressure Control Methods

Now that we have looked in some detail at performance curves for a process system and some ejectors, we are ready to look at how they interact as we use some different methods to control the pressure. Remember, we control pressure by using some method to raise it to a higher level than the process and the ejector would arrive at without any control.

An important aspect of any control system is how it responds to a change in the system or ejector curves. We usually assume the ejector to be well behaved and adequately described by curves that do not change rapidly in response to steam pressure, water temperature, wear, and fouling. The process system will vary its curve as the flow of noncondensables varies, or the temperatures and composition change during a batch operation, or the decision is made to operate the process at a different temperature. It is useful to attempt to answer the question, how much will the system pressure change if the load changes by one pound per hour? The question which immediately follows is, how fast will it move to the new condition? I will give my best answers to the "how much" question, but can give only general responses to "how fast." First I will respond to "how fast."

In any complex dynamic situation, it is often true that one mechanism is slow enough relative to the other mechanisms that it may be considered as controlling the entire response rate-"how fast." I believe this will usually be the process system, more specifically, the noncondensable-gas "mask" that controls the effective area in the process condenser. Or, it may be the vapor space in the distillation column, which requires time to gain or lose the total vapor mass associated with changes in system pressure. The ejector, with its small volumes and high velocities, will usually have a time constant much less than 1 s. The time constant is defined variously, but is typically considered to be the time taken to move 50 or 90 percent of the distance toward a new steady-state condition after a sudden ("step") change in the load. The time constant of large process systems will be of the order of tens of seconds or longer.

Often a large system can be isolated from its ejector for several minutes while the ejector is tested, and the pressure will be found to have increased only a little. The process lines connecting these components are designed for high gas velocities, and the typical operating velocity in them is often only a fraction of design.

A careful analysis of the system will begin with pressure sampling location and frequency, and will include consideration of the interconnecting piping, the volumes of all important process vessels, and the flow characteristics and stroking speed of the control valves.

Returning to the "how much" question, the general procedure is a little different from that illustrated in Fig. 6.3. In these graphs the noncondensable gas flow is less than the design value. Again, for simplicity the line-pressure drops are assumed to be small. Refer to Fig. 6.10 and note that the process system and the ejector now have separate operating points, which are effectively "joined" by the control system. A second feature to note is the slope of the ejector curve in the situations where the pressure is controlled by adding a gas load to the ejector first stage.

After finding how a system responds to a small change in air leakage, the next question to answer is, how will this system behave if the pressure control point is moved suddenly up or down a large amount, or if there is a brief loss of steam pressure? The possibility of backfiring is important if water vapor in the product is more than a minor nuisance.

Suction throttling

The simplest way to get rid of the extra capacity of the ejector is to throttle the flow from the process system to satisfy that system's needs, and permit the ejector to find the pressure which corresponds to that flow. Figure 6.10*a* shows the operating points for this method. The air load is less than half the design value, and the process-system

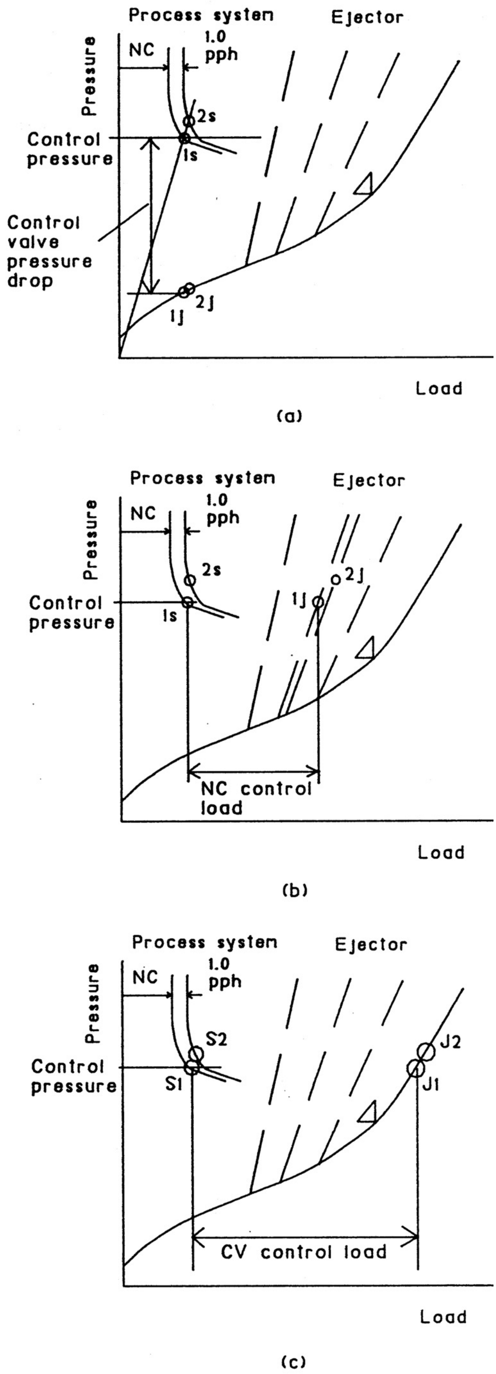


Figure 6.10 Process-and ejector-system curves with pressure control: *(a)* suction throttling; *(b)* noncondensable gas control load; *(c)* condensable vapor control load.

pressure is well above the ejector design pressure, a situation which is representative of my experience. Note that the mass flow is identical for both systems and that the pressure drop across the valve is a large fraction of the absolute pressure, establishing critical flow across the valve (sonic velocity through the opening).

Thus, the mass flow depends only on the valve geometry and the upstream process conditions and is independent of the ejector pressure. Point 1s is the system operating point, and point 1j is the corresponding ejector operating point. The valve now controls the removal of gas, and its "curve" slope is determined by a straight line through the origin. If the air load is now increased by 1 pph and the valve position is not changed, the new process-system operating curve will be offset to the right, and the new operating point will be at the intersection of that curve with the valve curve, point 2s. The corresponding ejector operating point will be 2j. When the rise in system pressure is detected by the control system, it will slowly open the valve until the pressure is again at the desired value. The valve curve will have been rotated to the right.

Now, what happens during big changes, such as a sudden increase in the air leakage or a reset to a much higher pressure? An air-leakage increase will move the system operating point upward along the valve "curve" as in the 1 pph example. The control system will detect the pressure rise and open the control valve. This will rotate the valve curve to the right to permit more flow to the ejector until the pressure is restored. The ejector will see a larger flow and respond with an increase in the suction pressure, but that will have no influence on the flow through the valve if critical flow still exists.

If the pressure controller is suddenly reset to a much higher value, the throttle valve will close, a little nitrogen will be "puffed" into the process system (see Figure 6.11), and the throttle valve will then open to its new setting. If the pressure is suddenly reset to a much lower value, the throttle valve will open to load up the ejector. If this valve opens suddenly, the ejector may be momentarily overloaded by the noncondensable gas flushed out of the process system, and this may possibly create a small amount of backfiring which might flow past the control valve and into the process system. A slow opening rate is thus preferred.

Recommended suction throttling system. Figure 6.11 shows my personal preference for pressure control, throttling the flow of gases to the ejector. Typically, the gas flow is less than half the design value and the ejector suction pressure is less than half the process-system pressure, establishing critical flow across the throttling valve. Usually the

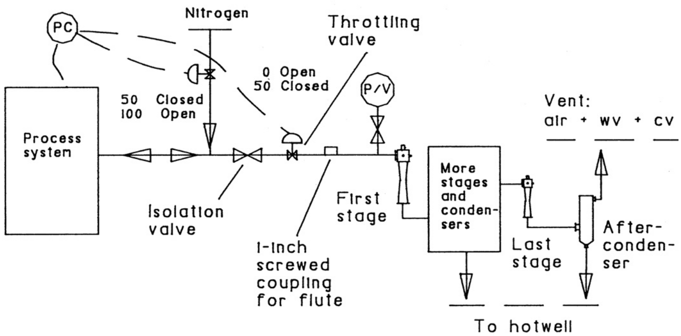


Figure 6.11 Pressure control configuration, suction throttling.

valve will be a butterfly valve, which need not be a tight-shutoff design because the isolating valve is closed for testing and working on the ejector. In normal operation, the throttling valve is in the nearly closed position and the nitrogen valve is completely closed. Because this produces a sonic velocity jet of gas along the edges of the valve, the valve should be located at least 5 pipe diameters from the ejector suction to permit flow redistribution.

If a minor upset occurs or if the pressure control point is reset to a higher pressure, the valve may close completely. If the rate of pressure rise is found to be too slow (a very tight system with minimum air leaks and instrument blowback gas), then the nitrogen valve will open briefly to blow nitrogen back into the process system until the necessary volume of noncondensable gases has accumulated there. Then the nitrogen valve will close and the throttle valve will reopen.

The steady-state load to the ejector system will be only the gases from the process system, typically air leakage saturated with process vapors. If the ejector is multistage with intercondensers, the last stage will usually handle only the air leakage plus a small amount of water vapor and process vapor from the last intercondenser. If an aftercondenser is present and is vented to the air, the air-leakage rate may be measured by slipping a collapsed plastic garbage bag over the vent opening and observing the time required to fill it (see Chap. 8 for more details). Or the pressure may be read at the last-stage suction and used with its performance curve to estimate the air leakage. A 1-in screwed coupling is shown in the line to the first stage, as a convenient location for connecting a "flute/piccolo" multiple-orifice air meter to test the air-handling capacity and calibrate the last stage.

Some people concerned about monitoring air leakage will connect a pressure transducer to the last-stage suction and continuously record the pressure at the last stage. They monitor this record, noting any gradual increase in leakage with time and marking the timing of "spikes" to track down operating activities or process changes that create undesired air leaks.

This control method has several advantages. The shape of the first-stage curve is unimportant, no overload capability is required, and stability to no-load is of value primarily for troubleshooting. Nitrogen usage is essentially zero, as it is used only briefly during occasional transients if the current air leakage is too small for an adequate response time. Air pollution by any process gases which accompany air from the aftercondenser is minimized. The control system is inherently suitable for activation by an emergency shutdown system. The throttle valve closes, and nitrogen is admitted to pressurize the system and dilute the oxygen which continues to enter the system as leakage. Although the control valve may not shut perfectly tight, it will reduce any air in-leakage at that location to a very small value, which is diluted at its source by the large flow of nitrogen gas.

One disadvantage is that the cost of the throttling valve may be significant. This cost may be reduced by selecting a valve size smaller than the line size, with long reducers to minimize the head loss. Also, the pressure drop across a fully open butterfly valve is about one velocity head, which translates to 2 percent of the absolute pressure for 200-fps velocity at the ejector inlet, and 5 percent for 300 fps. A 5 percent pressure drop would increase steam usage of the ejector about 2 percent if it were included in a rigorous design procedure, but in my experience this would be neglected.

The rangability of the valve may be exceeded if the actual leakage is very small and the process-system pressure is much higher than the ejector design suction pressure. The rangability is typically stated as being in the region of 50:1 to 200:1. If a valve is selected to offer minimal pressure drop at an ejector design condition of 10 pph air at 5 torr, and is then required to control by throttling an actual air leakage of 2 pph at a system pressure of 100 torr, it will be handling an upstream volumetric flow that is only 1 percent of the design-point volumetric flow.

If this causes a problem, one solution is to adjust the ranges of the two control valves to cause the nitrogen valve to open before the throttling valve closes completely. That will give the control valve more flow to work with, at the cost of using more nitrogen and losing the ability to monitor air leakage. A better long-term solution is to place a second, smaller control valve in parallel with the primary one and switch manually or automatically from the large to the small valve when the process pressure is raised.

Another consideration is that the first stage will typically be operating well below its design-point pressure, which may allow ice to form if the pressure drops below 5 torr and the stage is not equipped with a diffuser steam jacket or steam superheater. The ice formation may alter the ejector performance enough to upset the control system or subsequent operation or testing. Adding a steam jacket to the diffuser is a common fix.

Loading ejector with noncondensable gas

A very common method of controlling ejectors is to bleed atmospheric air or another more acceptable noncondensable gas to the ejector suction, loading up the ejector as shown in Fig. 6.10*b*. Observe that the process system and the ejector are now at the same pressure (assuming pressure drops are negligible), and that the two operating points (1s and 1j) are separated by a distance which represents the load gas flow. The ejector operating point is on a broken curve defined by the condensable vapor present in the load. If the ejector has been designed for a large air overload and not much vapor load, and if the system pressure is not much above the design suction pressure, then the ejector operating point may be on its basic curve instead of a broken curve.

To find the effect of a change of 1 pph in the air leakage, first consider that the extra air will be accompanied by a saturation quantity of process vapor, causing the operating point 2j to appear on a broken curve slightly to the right and offset by the amount of the condensable vapor with the incremental 1 pph air. This new broken curve will be nearly parallel to the previous one, so we can find point 2j by moving horizontally from 1j to the new broken line, then 1 pph to the right, then up to the new broken line. Point 2s will be offset to the left from 2j by the amount of the load gas. Although this method shows where each of the operating points will move, taking into consideration the vapors flowing with the air, a simpler rule exists. The increase in the system pressure is equal to the change in air load multiplied by the slope of the broken curve at the ejector operating point.

If the ejector operating point lies on the basic curve, then the slope of that curve defines the pressure excursion created by a 1 pph increase in the air load. Note that the slope must be multiplied by the air-load increase plus the corresponding vapor-load increase. The slope of the basic curve is less than that of the broken curves. This benefit is obtained at the cost of increased size of subsequent stages and condensers, with a higher first cost and greater steam and cooling-water usage.

If the air load suddenly increases or if the pressure control point is raised, then the air flow to the ejector will suddenly increase. If the control air load is increased quickly, some of it may flow back into the process system. Also, the back pressure seen by the first stage will increase so much that some backfiring may occur, as described previously, sending pulses of water vapor back toward the process system. Slowing the control-valve action can prevent that cause of backfiring, but sudden "spikes" of brief air-load increases may cause backfiring if the design is not intrinsically "backfire-proof."

Noncondensable gas bleed control configuration. Figure 6.12*a* shows some key features of a process/ejector system in which pressure is controlled by loading the ejector with noncondensable gas. The biggest advantage of this control system is its simplicity. The control valve is small, simple, and easy to size because the maximum required flow is usually no more than three to five times the design

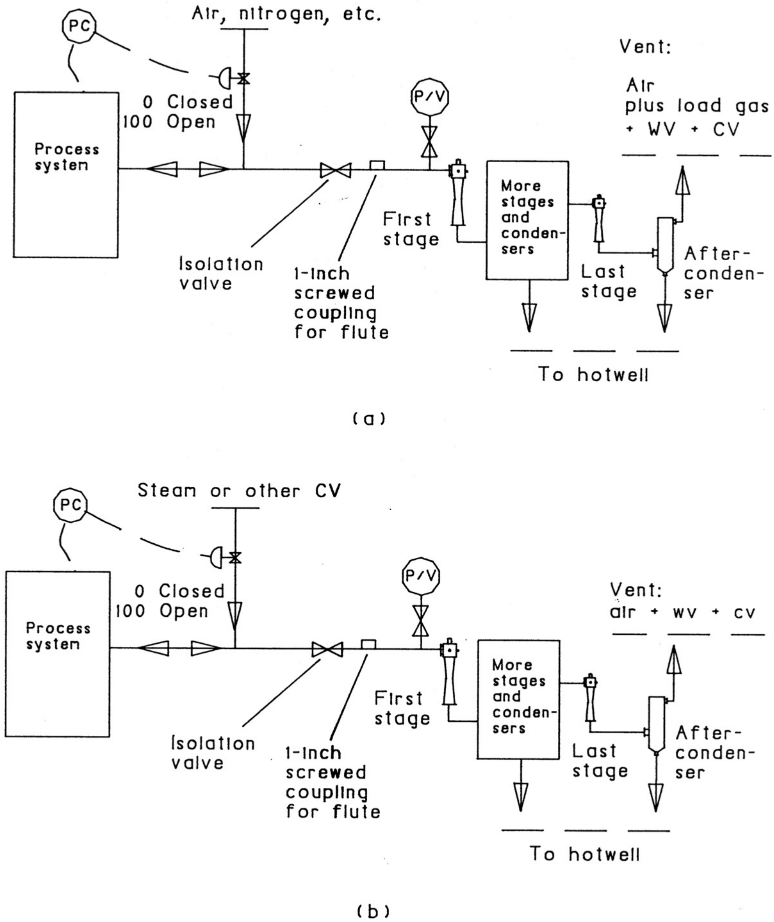


Figure 6.12 Pressure control configuration, adding external load: *(a)* noncondensable gas bleed; *(b)* condensable vapor bleed.

air load. If atmospheric air or a low-cost gas is available, the cost is negligible. When the ejector is isolated from the process system for testing or maintenance or during an operating upset, the control system may be used to pressurize the process system.

Typically, the air leakage is less than half the design air load and the process-system pressure is at or above the ejector design suction pressure. The result is that critical flow exists in the control valve, making that flow independent of downstream pressure changes in the vacuum producer system.

The control gas constitutes most of the ejector load, and the ejector operating point is usually on a broken curve. The flow of noncondensable gas required to "break" the ejector is usually somewhat larger than the design value for several reasons: Some design margin may have been intentionally added by the manufacturer, the steam pressure will be above the design value, cooling water will be cooler than design and usually flowing above the design rate, condensers will be cleaner than the design condition, and the atmospheric stage discharge pressure is usually less than design. If an extended stable range has been specified, typically to a 50 percent air overload, then the total flow will be about 50 percent more than described above.

A disadvantage of this control method which happens to bother me personally is that it obscures the actual air leakage, so that an operator or troubleshooter cannot tell how much of the load to the last stage is air leakage and how much is control load gas. I am biased because most of my field experience has been with "sick" ejector vacuum systems, and this feature makes it harder to diagnose problems.

If air contamination or product loss through the last-stage vent is a consideration, then this control method is the worst. The noncondensable gas flow will typically be above the design value, including any air-overload specification, resulting in maximum product loss to the atmosphere.

If the process vapors can form an explosive mixture with air, then some other noncondensable gas must be used. Nitrogen is almost always acceptable, but may have a significant cost. Safety consideration must be given to the presence of the control gas in the atmospheric vent, which in some systems will be the hotwell area.

Loading ejector with condensable vapor

This control method is similar to the common method of loading the ejector with noncondensable gas, the difference being that the ejector is always operating on its basic curve, as shown in Fig. 6.10*c*. Here it is assumed as before that the air leakage is less than the design value. The pressure rise corresponding to an increase of 1 pph in air leakage will be as before, the slope at the ejector operating point multiplied by 1 pph air plus the corresponding increase in condensable vapor load.

A large sudden increase in air leakage will simply raise the system pressure along the basic curve, provided the design air capacity is not exceeded. If the pressure control point is raised very much, the control valve may open quickly and send condensable gas back into the process system as well as forward to load the ejector. As before, slowing the valve action will correct this.

Condensable vapor bleed control configuration. Figure 6.12*b* shows a typical configuration in which condensable vapor (often low-pressure steam) is used to load the ejector. It is nearly identical to the noncondensable gas bleed control except for the source of the control fluid and the conditions at the vent. As with the throttling control method, the aftercondenser vent now has only the actual air leakage plus a saturation load of water vapor and process vapor. The presence of the extra water vapor in the first intercondenser will tend to remove more condensable process vapor if the liquid phase is soluble in water.

Advantages of this control method are that the pressure excursion resulting from a small increase in load is minimized and the atmospheric vent flow is only the actual air leakage and the accompanying water vapor and process vapor. I find this control method preferred by some ejector engineers, possibly because it resembles the test situations in their shops. I would speculate on another reason why this might appeal to an ejector manufacturer on an emotional level. It loads up the first stage to utilize its full design pumping capacity, "keeping it busy." Somehow the idea of a lot of ejectors loaded to only a small fraction of their design capacity most of the time may seem like underutilized investment, although it may make perfect economic sense from the viewpoint of a user.

Disadvantages of this control method are that the cost of the condensable control fluid may be significant and that the control valve must be carefully specified to resist wire drawing by erosive wet steam in the critical-flow situation, which is typical. During transient conditions when moving to a higher operating pressure, some of the increased vapor load may flow back into the process system.

Recirculated mixture load

Recirculating a gas mixture from somewhere in the ejector system back to the suction is another method of controlling pressure. It has a mixture of the advantages and disadvantages of the systems described pre

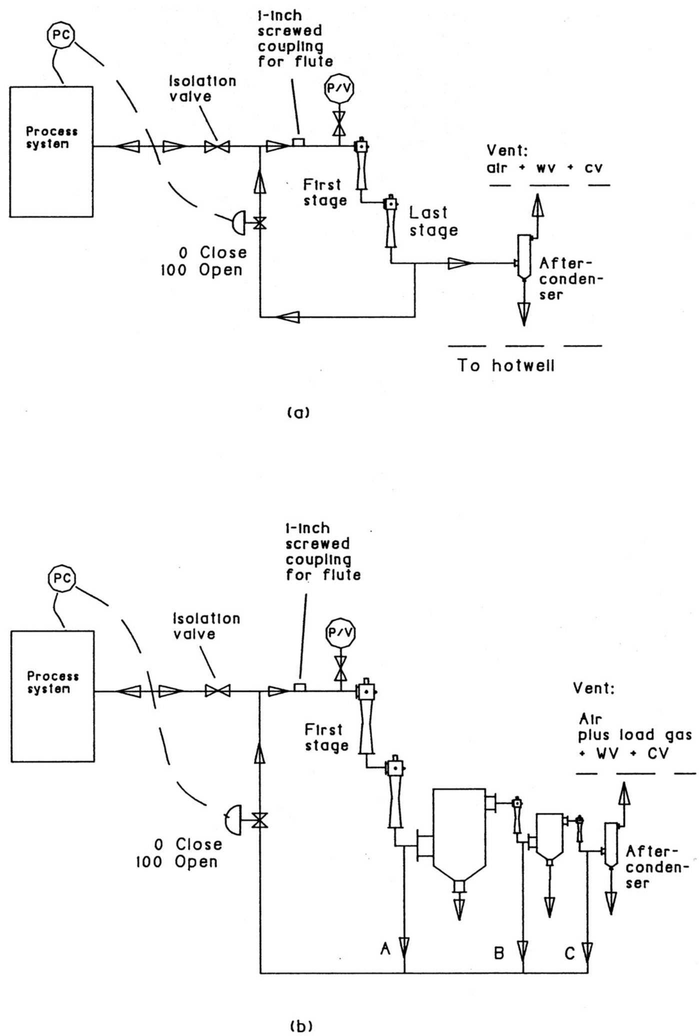


Figure 6.13 Pressure control configuration, recirculated gas and vapor: *(a)* single-stage, or multistage noncondensing; *(b)* multistage condensing.

viously. The primary advantage is that the load gas is free and adds no materials to the system that are not already present. As shown in Fig. 6.13*a*, the gas is obtained from the discharge of a single-stage or noncondensing ejector. For a condensing ejector, it is obtained either from the discharge of the last stage or from an appropriate interstage location, as shown in Fig. 6.13*b*. Because no external gases are added, the aftercondenser vent contains only the air leakage and saturation quantities of water vapor and process vapors.

The operating curves for a single-stage or multistage noncondensing ejector will resemble those in Fig. 6.2*a*. The curves for condensing ejectors will resemble those in Fig. 6.2*b* and *c*. Noncondensing ejectors will always operate on their basic curves provided the suction pressure is above any lower stable limit.

The operation of the condensing ejector system can be much more complex. Design and operation of the control system depends upon the number of stages, the ratio of vapor to noncondensable gas in both the ejector design conditions and the actual operating conditions, the motive steam flow in each stage, and the interstage pressures and temperatures. In some ways it is easier to analyze each specific example than it is to generalize in a useful manner. Here I will offer some observations and suggestions for general strategies and leave you to work out the details for your specific application, contacting your ejector manufacturer to resolve special problems.

Perhaps the first question to answer is, will the ejector operate on its basic curve or on one of its broken curves? Let us start by examining the situation in which the current air leakage and condensable vapor load are far below the design values and the desired pressure is at or above the design point. Referring to Fig. 6.10*b* and *c*, we observe the air flow and total flow corresponding to the desired operating pressure. Referring to Fig. 6.13*b*, we see that we have a choice of extracting our recirculated mixture from *A*, just before the first intercondenser; *B*, just before the second intercondenser; or *C*, just before the aftercondenser.

Let us start by selecting *C* and look to our data sheets to find the motive steam flow at that point. If the motive steam flow exceeds the capacity of the basic curve, then we may be able to operate the first stage on its basic curve by recirculating a portion of the discharge from the last stage. Visualize what happens as we begin to recirculate a small quantity of the discharge. Most of it will be water vapor, but some will be air. These combine with the air and vapor from the process and go to the first stage. We now compare the total air load and the total combined load with the values we previously noted were needed at the possible operating points. If both values are below the required values, we must recirculate more of the discharge.

Note that as we increase the fraction recirculated, we begin to build up the air recirculation within the ejector system. Eventually we will build up enough total load or air load that we satisfy the ejector needs. As a simple example, if we recirculate half the discharge flow, then the total air load seen by the first stage will be twice the air flow coming from the process system. Recirculating 75 percent will cause the total air load seen by the first stage to be four times the air leakage, etc.

Let us repeat the analysis, but this time consider the situation in which the motive steam to the last stage is much less than the total capacity of the first stage at the desired pressure. If we were to suddenly recirculate the total discharge flow, we would begin by operating far down on the ejector suction curve, but there would be a buildup of recycled air until the air load equaled the air capacity; the ejector operating point would then move up along a broken curve until the pressure reached the desired value. Then the control valve would partially close and steady state would be reached at a condition where most of the discharge would be recycled, and the final vent would equal the process-system air leakage.

In designing the control-valve system, care must be taken to anticipate whether a high suction pressure will cause the valve to experience subcritical flow. The recirculated mixture may have moisture present, and the materials and design should be selected to resist wire drawing.

What about recirculating from location A, just before the first intercondenser? We go through the same analysis steps that we used for the final-stage discharge, with the difference that here there is always enough steam flow present to load the first stage so that it can operate on its basic curve.

However, design or operating problems can arise because the absolute pressure here is much lower than atmospheric pressure, the interstage pressure may be lower than its design value with low water temperature and low air load, and the desired system pressure may be much higher than the design suction pressure-possibly higher than the actual or design interstage pressure! The recirculating valve will be larger than the one from the last stage, and typically a little smaller than the vapor line entering the first condenser.

This brings us to the next logical question, what about recirculating from location *B*, the second intercondenser? As might be expected, this is better in some ways and worse in others. The pressure is higher, so the process-system pressure can be raised higher without affecting the recycle flow, the control valve can be smaller because of the higher pressure, and the last stage sees only the actual air load, but the total steam available here may not be adequate to load the first stage to operate on its basic curve.

In summary, the advantages of recirculation are that the load fluid is free and the aftercondenser vent contains only the air leakage plus some water vapor and process vapor. Another potential benefit can be realized by recirculating from the last stage if there is no aftercondenser and the last stage discharges to the air through a silencer or mufl1er. The size of the steam plume will be related to the current air leakage, providing a visible air-leakage analog "meter."

Disadvantages are that some of the wet control gas may drift back into the process system or some brief backfiring may occur in response to air leakage spikes or sudden increases in the pressure control point. Selection of the proper extraction point for the recirculated gases requires more design effort to anticipate the maximum required process-system pressure, analyze the ejector design and "worst-case" extraction point pressure, and size the control valve. If the extraction point follows the last stage, then the last stage will handle recirculated air along with the basic air leakage, preventing you from using the pressure there as an accurate measure of air leakage.

Throttling motive steam with low compression ratio

This control method is practical only for single-stage ejectors with a compression ratio less than 2:1. As shown in Fig. 6.2c, the motive steam may be throttled to shift the ejector performance curve. The range of motive steam pressures is limited. As a more energy-efficient version of this control method, some larger ejectors may have a spindle nozzle, in which the throat area of the nozzle is adjusted by advancing or withdrawing a slender spindle along the nozzle center-line. The benefit of this method is that the full steam pressure drop can be converted to velocity in the nozzle. The disadvantages are the cost of the special nozzle and its operating mechanism, and the slightly compromised nozzle design.

Throttling cooling water to a condenser

Only limited pressure control is possible by throttling water to a process condenser. Condensers have limited rangability on water flow: contact condensers will allow vapor bypassing due to water flow mal distribution below 30 to 50 percent of design water flow [1], and surface condensers may silt up at low water flows. The primary control mechanism is that the water outlet temperature rises, increasing the condensing temperature and pressure. The net result is to shift the system operating curve somewhat.

Throttling water flow to an ejector intercondenser [2] is less desirable because it shortens the ejector stable operating range.

6.5 Manual versus Automatic Control

Manual control can involve any of the methods described above. The choice of methods is strongly influenced by the high cost of operator labor and the amount of pressure variation which can be tolerated. Each of the control methods produces a predictable maximum pres sure excursion from an air-load increase of 1 pph. An ejector operating on its basic curve will usually experience a smaller pressure rise than if it were operating on a broken curve. For this reason, a condensable vapor bleed, recycling of steam-rich vapor, or use of an extended stable operating curve is often preferred for manual operation. A weakness of the common practice of specifying "50 percent air overload" to obtain an extended stable range is that the real objective is to achieve a minimum slope of the curve on which the ejector operating point moves. It would be more precise to specify the maximum slope as "2 torr per pph increase in DAE load," for example. Warning! That was a completely arbitrary example; it should not be regarded as a general-purpose rule or recommendation!

For automatic control, it is not cost-effective to specify an extended stable operating range to improve controllability unless there is a rational quantitative basis for it. It is "in the direction of goodness," but at an operating-cost and first-cost price.

Even in pilot plant operations I encourage the use of automatic controls to minimize upsets and operator distraction.

6.6 Capacity Control

Critical-flow ejectors are basically fixed-capacity devices, as described in Chaps. 3 and 4. If you want a large range of capacity at minimum operating cost, consider using multiple units in parallel. One or more are operated to attain the desired capacity. They may be identical for design simplicity and common spare parts (33, 33, and 33 percent), or they may be sized for maximum flexibility (14, 29, and 57 percent) so that combinations may be operated to closely approximate the capacity desired at any time. Isolating valves prevent reverse flow through inactive units.

As seen before, single-stage noncritical flow ejectors (compression ratio less than 2:1) have a modest range of capacity adjustment through throttling the motive steam.

6.7 Control Steam and Water Flow for Economy

When cooling water is colder than design or when the condensing load to a condenser is less than design, some water may be conserved by throttling the water flow within limits. Or, the steam pressure to a preceding stage may be reduced in recognition of the reduced discharge pressure. The practice is totally reasonable and is strictly a matter of economics, careful design, and proper operation. With the advent of inexpensive sophisticated computer control, a strategy can be developed that gets maximum savings with negligible risk.

The application requires ejector performance curves and condenser design data, and should be coordinated with the ejector manufacturer. It may require a small amount of testing to determine the as-built stage characteristics precisely.

6.8 Nomenclature

Acc_s	rate of air accumulation in suction system, pph
Acc_z	rate of air accumulation in intercondenser, pph
a	air (noncondensables) load, pph
BK	discharge pressure at which a stage begins broken operation
CV	condensable vapor, pph
DAE	dry air equivalent, pph
fps	feet per second, velocity
MDP	see PU
NC	noncondensable vapor, pph
PC	pressure controller
pph	lbm/h
P_s	pressure in suction system, torr
P_{sr}	pressure rise in suction system, torr/s
P_{zr}	pressure rise in intercondenser, torr/s
P_d	pressure at stage discharge, torr
$P_{d\mathrm{bpu}}$	difference between discharge break and pickup pressures, torr
PU	discharge pressure at which a stage resumes stable operation
P_z	pressure at Z-stage suction, torr
P/V	pressure/vacuum gauge (for approximate readings)
S	suction curve for stable operation
T	temperature
T_{cycle}	time to complete one break/pickup cycle in transient region, s
T_o	time basis for estimating break/pickup cycle time, s
torr	absolute pressure unit, 1 mmHg
U	unstable, lower limit of stable curve, pph
VP	vapor pressure, torr
V_s	volume of suction system, ft^3
V_z	volume of intercondenser, ft^3
W_a	air load to suction system, pph

WV water vapor, pph

W_y current air load handled by Y stage, pph

W_z current air load handled by Z stage, pph

W_{zbk} air capacity of Z stage at break, pph

W_{zpu} air capacity of Z stage at pickup, pph

6.9 References

1. C. G. Blatchley, "Controlling Ejector Performance," bulletin 10-56, Schutte and Koerting Div, Ketema.
2. G. B. Knight, "Five Ways to Automatically Control Pressure for Ejector Vacuum Systems," *Chemical Engineering*, Mar. 23, 1959, pp. 171-174.
3. R. B. Power, "Steam-Jet Air Ejectors: Specification, Evaluation, and Operation," ASME Paper 63-WA-143. Reprinted by *Hydrocarbon Processing and Petroleum Refiner*, 43 (2): 121ft' and 43 (3): 138ft' (1964).
4. R. B. Power, "Steam-Jet Air Ejectors," 10-part series, *Oil and Gas Equipment*, October 1965-July 1966.
5. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation,* McGraw-Hill, New York 1986.
6. W. H. Mathews, "How to Control Vacuum on Your Steam Ejector Systems," *Southern Power & Industry*, 6 pages, date unknown.
7. V. V. Fondrk, "Steam Jet Ejector Performance and Costs," *Powerfax*, Spring 1958, pp. 22-25.

Chapter

7

Installation

7.1 Overview

If you have read the previous chapters, you are familiar with how each part of an ejector system works and how the parts work together to perform the pumping job. Now we have the task of designing the physical home of the ejector system and installing the hardware in such a way that the operating people will find it easy to keep the ejector system running in a reliable manner.

After you have studied this chapter, you will be able to walk through an ejector installation and identify inconveniences and major trouble spots. If the ejector has been operating for a time, probably the major trouble spots will have been corrected. But who knows? A new look often reveals "obvious" defects that the permanent residents have come to overlook. You might help them out some!

What makes an ejector installation reliable and easy to operate?

Ejector stages and condensers are fluid-flow devices that require steam and cooling water at the proper conditions to operate properly. Gravity removes condensate and affects several of the fluid-flow phenomena. The hardware must be arranged to enable the proper fluid-flow conditions to exist all the time.

Production and maintenance people have many tasks to perform to operate the ejectors and keep them in working condition. They must start them up and shut them down; detect problems, find the source of each problem, and fix it; and periodically test, inspect, and maintain the ejectors. In doing all this, they read gauges, operate valves, fill and drain equipment, open equipment or remove it to inspect and maintain it, and conduct leak tests and performance tests. You want them to be able to do all this safely.

To help them do their jobs well, I offer you the guidelines in this chapter. This chapter is a collection of "good practice" suggestions gathered mostly from ejector literature and confirmed and augmented by my experience. These suggestions represent sincere efforts on the part of many people to provide constructive feedback on the design and operation of reliable ejector systems. I realize it may be difficult to decide just how much of the "good stuff' you can afford in your design, especially if your job is overrunning its budget and you are not sure whether all the features are essential. In addition, some guidelines may be difficult to follow, and some will be contradictory a design feature that gains one benefit may incur a cost and perhaps contribute to a problem in another area. Welcome to the world of design!

The good news is that an ejector system can often give years of reliable service without many of these features. The bad news is that each deficiency in your installation will contribute to the total probable cost of operation through a combination of downtime or off-design operation, operator labor and management distraction, and equipment damage.

It is amazing at times to compare the equipment performance expectations of different people who have developed their expectations in different environments. We all take pride in doing good work, and some people work happily in circumstances that would frustrate other people who have been "spoiled" by expecting more. If you are afflicted with doubts and uncertainties about how to install ejectors, this chapter should comfort you. If you are casually installing them with a minimum of design attention, I hope to afflict you enough to raise your expectations.

Operating department review

If the installation is being designed by someone outside the organization that will operate it, I recommend that an operating department representative with ejector experience review the design before it is released. Although it is often difficult to express design criteria clearly before design work is done, most people are quite able to identify problems in the designs of systems with which they have operating experience. Usually the reviewer will be involved regularly during the entire design process.

Layout

Figure 7.1 is the flow diagram for a four-stage ejector with contact condensers. It contains many of the hardware features that can be present in an ejector installation. The individual hardware components will be discussed in this chapter.

The first time attention is given to an ejector installation may be in layout planning, "Where will we put it?" Usually that means high

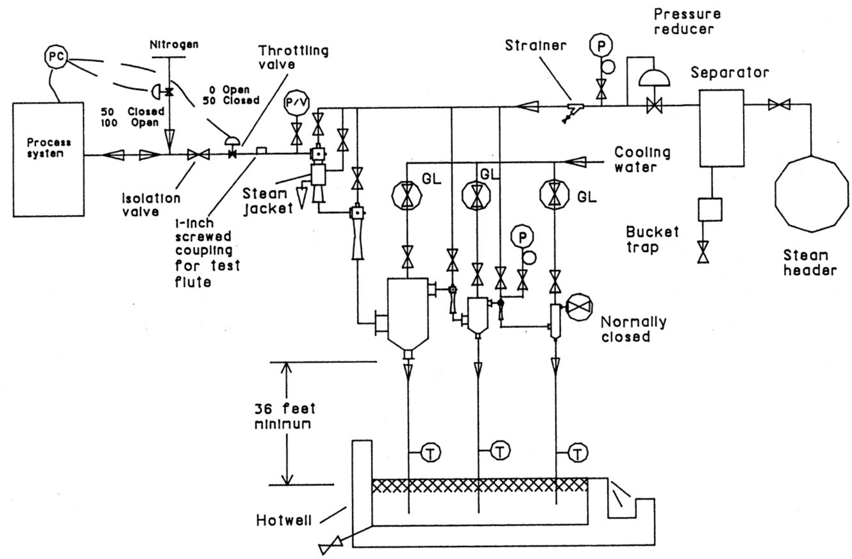


Figure 7.1 Multistage condensing ejector flow diagram.

enough up in a structure that gravity can remove condensate from the condensers. Occasionally that is a platform just for ejectors, and there may be a temptation to keep it very small to save money. Please consider the tasks that people must perform there and give them convenient access and enough space to do a good job safely. Making several trips up a ladder to bring test or maintenance equipment to an ejector can result in fatigue. Then the work is not done as well, and fatigue may lead to inattention and an injury.

Perhaps you can tell that this is being written by a person more than 60 years old! Even before I reached chronological maturity, I was impressed by how difficult it can be to find the problem in an ejector system when much of my time and energy is consumed in coping with deficiencies in the installation design. Of course, no one expects perfection in the working environment, but the ejectors in well-designed installations do seem to work better.

Test both your preliminary layout and your "final" layout or model to see whether some of the common tasks around an ejector are made difficult or dangerous. Some common tasks are:

- Turning steam or water on and off to individual stages and condensers
- Draining condensate from low spots during startup
- Connecting (vacuum) pressure gauges to stages for pressure measurements (not on the bottom of the air chamber!)

- Isolating the ejector briefly by closing a leaktight valve between the ejector and the process system
- Doing a flashlight inspection of steam nozzles in place, looking for pipe scale or other flow obstructions
- Removing steam nozzle assemblies and water spray nozzles
- Inserting slip blanks in the flanged joint at a stage inlet connection to conduct a no-load test while troubleshooting, or at the hotwell to conduct a hydrostatic leak test
- Loosening and swinging aside the line at the last-stage discharge to see whether it works better discharging directly to the atmosphere
- Slipping a plastic garbage bag over the aftercondenser vent to measure the air leakage
- Removing heat exchanger tube bundles and ejector stages without major movement of piping or other equipment

The above is a positive statement of some guidelines. In learning a topic, it is useful to hear both positive and negative views, because the emotional word pictures may be easier to remember. Here I will list some of the frustrating conditions I have encountered while helping process plants diagnose and fix sick vacuum systems:

- No pressure taps on the suction chamber of a small stainless steel stage, or a pressure tap accessible only on the bottom of the suction chamber where water drained into the absolute pressure manometer.
- Steam piping to the ejector stage was along the axis of the steam nozzle, preventing easy inspection of the nozzle for obstructions.
- There was only one steam valve for steam supply to all stages, so that individual stages could not be operated by themselves for a stage-by-stage test.
- Screwed connections made performing a no-load test very tedious because there were no flanges for slip blank insertion.
- There was no isolating valve between the ejector and the process system. This saved the cost of one 10-in stainless steel valve, but delayed solving a major startup problem for several weeks.
- A large single-stage ejector was located outside a railing on the structure. I got burned a little attaching a vacuum gauge for test, and loosening the discharge line for discharge-to-air test was difficult and hazardous.
- A low spot in the suction line collected condensate during shutdown and flashed off under vacuum to interfere with testing. The valve had to be opened and the condensate dumped back into the

process (Yuk!) to proceed with test. In a similar situation, a drain valve permitted us to proceed with a test after we dumped several gallons of condensate on the work site of a maintenance crew below us. We all frowned at the people who designed the installation, and we added rerouting the drain line to the "To do" list.

- The aftercondenser vent was too high for the "garbage-bag" flow test without bringing a ladder to the ejector structure, or the vent line discharge was several feet beyond the edge of the structure, or the aftercondenser drain leg was not sealed in the hotwell. All of these situations prevented our conducting a simple, valuable air-leakage volumetric flow test.
- The ejector structure was accessible only by ladder, requiring multiple trips to carry tools and instruments to the test level. This is a modest inconvenience, but I expect that stairs result in fewer falls and better operation and maintenance.
- Ejector stages were located on two levels, with the last stage requiring ladder access for testing and troubleshooting. This error was caught on paper, but I had to argue strenuously to get it corrected on a project experiencing a cost overrun. I convinced them that the plant would be forced to spend money to correct this defect if the engineer did not eliminate it in the first place.

Access clearances

Make sure that space is available to remove steam nozzles and water spray nozzles without having to move equipment or process lines or the distribution headers for steam and water. See Fig. 7.2. Manufacturers' drawings will indicate the space requirements. Usually the steam nozzles can be removed from the outside, but some older designs required removal of the stage to insert a rod through the diffuser to push out the nozzle.

When estimating the equipment weight for the design of the supporting structure, remember to allow for the weight of the entire system filled with water during hydrostatic testing and leak testing.

7.2 Process Piping

The piping from the process system to the ejector and between ejector stages must be large enough to avoid excessive pressure drops, sloped to drain well, and free of large air-leakage sources.

If it is known or suspected that excessive liquid entrainment will be present in the suction line, add a properly designed entrainment separator and drain the liquid back to the process or forward into the hotwell.

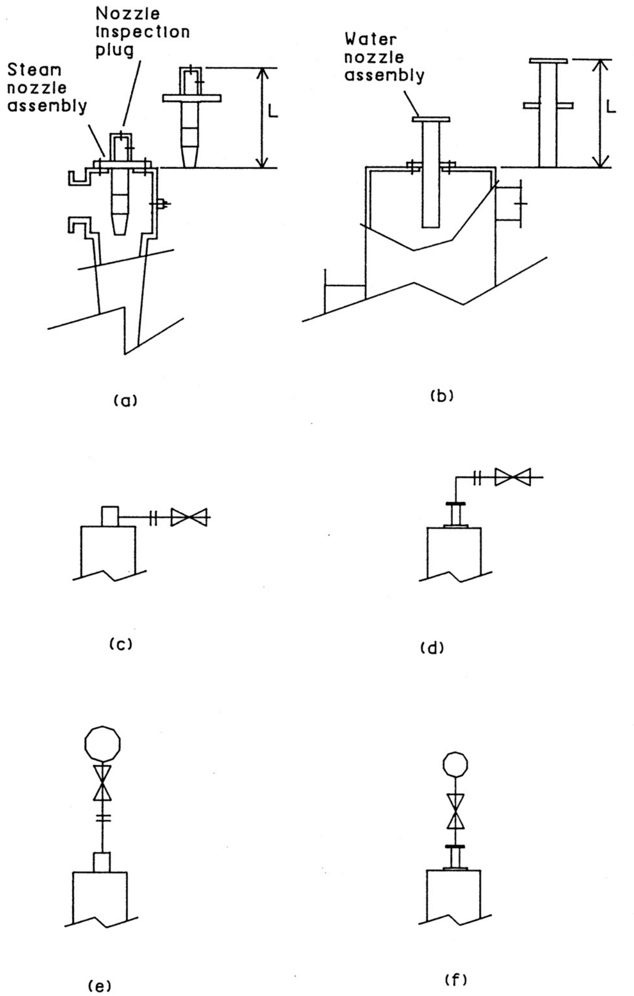


Figure 7.2 Clearances for removal of steam and water nozzles: *(a)* steam nozzle; *(b)* water nozzle; *(c)* accessible steam nozzle; *(d)* accessible water nozzle; *(e)* blocked access steam nozzle; *(f)* blocked access water nozzle.

Line sizing

As a useful guide, the process line to the ejector will usually be the same size as the ejector suction. A large difference between them is cause to suspect an error. I remember seeing a 2-in process line connected to a 6-in ejector suction and having a mental picture of an elephant sucking on a straw! Figure 7.3 may be useful for estimating the

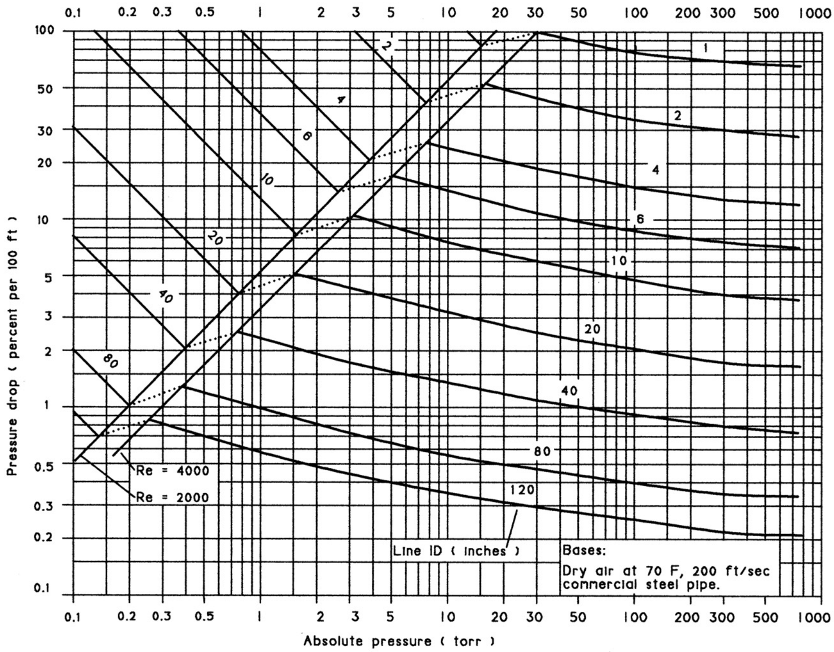


Figure 7.3 Pressure drop in vacuum lines.

pressure drop in different line sizes. It is based on actual inside diameters in inches and a velocity of 200 ft/s of dry air at 70°F in commercial steel (roughness) piping. The flow is considered to be incompressible. If the pressure drop is greater than, say, 10 percent of the absolute pressure, the total length of pipe should be treated as several shorter sections with the pressure dropped appropriately at the beginning of successive sections. The chart is intended primarily for the rough vacuum range above 1 torr, but indicates what happens at lower pressures. See Ryans and Roper [1] for a treatment of the flow regimes at lower pressures.

Approximate corrections for different velocities and different gas densities may be made as follows:

$$P_{drop} = P_{d0}\left(\frac{V}{200}\right)^2\left(\frac{\mathrm{MW}}{29}\right)\left(\frac{530}{T_F+460}\right) \tag{7.1}$$

where P_{drop} = pressure drop, percent absolute pressure per 100 ft
P_{d0} = pressure drop from Fig. 7.3, same units
V = design vapor velocity in line, ft/s
MW = molecular weight of vapor mixture
T_F = temperature of vapor mixture, °F

For example, find the pressure drop ina 20-in line at 10 torr, with a gas velocity of 250 ft/s, molecular weight of 35, and temperature of 150°F.

$$P_{drop} = 3.3\left(\frac{250}{200}\right)^2\left(\frac{35}{29}\right)\left(\frac{530}{150+460}\right) = 4.5 \text{ percent per } 100 \text{ ft}$$

Avoid check valves in ejector suction lines because they usually introduce too much pressure drop, even for low-pressure-drop swing-check designs. They may be appropriate for a single-stage ejector, but the pressure drop should be carefully estimated and accommodated in the design.

Air-leakage considerations

Air leakage is an important consideration for ejector systems, especially for multistage ejectors with intercondensers. Even if the first stages are very large for pumping condensable vapors, the last stages may be much smaller, handling mostly air leakage. When the accumulation of all sources of air leaks overloads the last stages, then the system must be tightened up to reduce the air leakage to below the ejector design limit. Thus, the combination of the ejector air capacity and the various potential air-leakage sources in the vacuum system will establish the cost of maintaining the system to keep the air leakage acceptably small. When you are designing the ejector installation, you have control over some of the air-leakage sources.

Although you do have available the specifications which define the design air leakage, your job is made a little harder because there are no precise data identifying the air leakage associated with each of the hundreds of potential leakage points in a large process system. However, several useful guidelines are available.

The Heat Exchange Institute standard [2] recognizes that removing air from a low-pressure system is more expensive and that the effect on operations of an air leak may be more severe; thus a low-pressure system will have a more stringent standard than would apply to systems operating at higher pressures. Low-pressure systems will thus be designed for smaller air leakage, and more effort is expended in the design and operation to keep the air leakage manageable. The HEI standard contains a chart which quantifies the air leakage associated with different system volumes at different design pressures or systems regarded as "commercially tight."

An older reference by Jackson, reproduced here as Table 7.1, lists nominal air-leakage values assigned to some common fittings. It was prepared by an ejector manufacturer in 1948 [3] and is based on "reasonably good workmanship and periodic attention," plus the observa-

TABLE 7.1 Air Leakage Allowances for Common Fittings

Type of fittings	Allowance, pph
Screwed connections in sizes up to 2 in	0.1
Screwed connections in sizes above 2 in	0.2
Flanged connections in sizes up to 6 in	0.5
Flanged connections 6 to 24 in, including manholes	0.8
Flanged connections 24 in to 6 ft	1.1
Flanged connections above 6 ft	2.0
Packed valves up to ½-in stem diameter	0.5
Packed valves above ½-in stem diameter	1.0
Lubricated plug valves	0.1
Petcocks	0.2
Sight glasses	1.0
Gauge glasses, including gauge cocks	2.0
Liquid sealed stuffing box for shaft of agitators, pumps, etc., per inch of shaft diameter	0.3
Ordinary stuffing box per inch of diameter	1.5
Safety valves and vacuum breakers per inch of nominal size	1.0

SOURCE: D. H. Jackson, "Selection and Use of Ejectors," *Chemical Engineering Progress*, 44(5): (1948). Courtesy Chemical Engineering Progress, AIChE.

tion that "it is general practice to estimate an appreciable amount of air leakage." It was written at a time when energy prices were low and there was a scramble to build and operate processing plants that worked, even if they were a little costly to operate. Most of those plants have long since been replaced by more cost-effective plants.

Although the absolute values of these air leakages may be adjusted downward in recognition of current energy costs and more stringent standards for design and operation of the equipment containing the components, the list does show us which types of components we should use sparingly and maintain carefully.

Ryans and Roper [1], ejector users, have combined the HEI approach with that of Jackson and added an estimate of the leakage from metal porosity and weld-line cracks. In effect they define the air leakage associated with several components in a "commercially tight" system for any size system and components and any system operating pressure. For a brief sample of some leakage values using their method, see Table 7.2. The air-leakage values are based on component sizes of 1 in and 10 in diameter and a system pressure of 1 torr, with multipliers for other pressures. My intent here is to give you quick guidelines. I refer you to Ryans and Roper for the details of their method.

The air-leakage tables emphasize the large air leaks associated with packed valve stems and rotary shaft seals. Improvements are liquid-sealed glands, mechanical face seals, and bellows valvestem seals.

TABLE 7.2 Commercially Tight Air Leakage Allowances for Rough Vacuum Systems

Component	Allowance, pph at 1 torr*	
	1 in	10 in
Static seals		
Threaded connections	0.05	0.5
Conventional gasket seals	0.016	0.16
O-rings	0.006	0.06
Thermally cycled gasket seals		
Temperature less than 200°F	0.016	0.16
Temperature between 200 and 400°F	0.06	0.6
Temperature above 400°F	0.10	1.0
Rotary seals†		
Packing glands	0.80	8.0
Mechanical seals	0.3	3.0
Valves used to isolate system		
Plug cock	0.03	0.3
Ball	0.06	0.6
Globe	0.06	0.6
Gate	0.13	1.3
Valves used to throttle control gas into vacuum system†	0.8	---
Access ports	0.06	0.6
Viewing windows	0.05	0.5

*Multiplier for 10 torr is 2; multiplier for 100 torr or higher is 4.

†Single-element seals represented here. Leakage is much less for packing glands sealed with liquid and for double mechanical seals.

SOURCE: J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation,* McGraw-Hill, New York, 1986. Courtesy McGraw-Hill.

Instrument-connection gas-blowback orifice nipples should be carefully specified for vacuum service, recognizing that blowback gas has a larger volume at low absolute pressures and that the ejector air capacity is limited.

A globe or needle valve used at a vacuum test connection should be oriented so that the valve seats against the vacuum – the flow arrow on the body should point away from the vacuum system. Any leaks at the valve-stem packing will be active only during testing and will be detected during the routine leak testing of the test system.

Piping forces

Thermal expansion from hot process gases or from ejector backfiring may apply excessive forces to ejector components, especially if the stages are nonmetallic materials such as ceramic or graphite. Use expansion joints or flex connections where needed to protect stages from excessive loads. Narrow diffuser throats on fabricated stages may bend or break if overstressed.

7.3 Stages

Stages may be installed in any position, but it is best if they discharge straight down to drain completely. In any other position, condensate and entrainment may collect and interfere with smooth startup, testing, and troubleshooting.

If discharge is up, insulate the diffuser to minimize condensation and possible recycling of condensate, especially on low-absolute-pressure stages. Check with the manufacturer for vertical upward installation of the first stage of a four-, five-, or six-stage ejector.

If the ejector has multiple elements, locate isolating valves at the suction of stages which may be inactivated. If isolating valves are located at the discharge of inactive elements, protect the stages against overpressure by adding properly sized relief valves to the suction chambers unless the stages are designed and hydrostatically tested for the maximum available steam pressure.

Be careful when installing stages which have identical external dimensions. They may be different inside. The last two stages on small ejectors may have identical external dimensions. In a congested construction site, the stages of two different ejector systems may get mixed up.

An elbow on the stage discharge is a cheap consumable "target" to protect the next stage or condenser from erosion by high-velocity water droplets or ice crystals present in the discharge hot gas mixture.

Freezing effects below 5 torr

The triple-point pressure of water is 4.6 torr (the temperature is 32°F, O°C). In stages that operate below 4.6 torr, the steam emerges from the steam nozzle cold, with ice crystals! Ice can build up on the diffuser and/or nozzle, altering the flow geometry and degrading the performance. Two alternative methods are used by ejector manufacturers to prevent icing: superheat the steam or warm the metal surface above O°C.

An electric superheater must be compatible with the specifications for other similar electrical equipment in the area, and must be protected against burnout when the steam flow stops. Other superheater types are offered by manufacturers or provided by users. If you have superheated steam that is truly superheated, you are unusually fortunate.

Steam jackets are commonly used on the diffuser inlet cone or the entire "venturi" portion of the diffuser. Sometimes the steam nozzle is also heated. The design objectives are to warm the metal surfaces to at least O°C and to avoid mechanical damage from high pressure, vacuum collapse, and freezing. Overpressuring a jacket may even col lapse the diffuser from external pressure. Even if the high pressure does not do any mechanical damage, the high-temperature steam may overheat the stage enough to reduce its performance if the manufacturer did not design for this. See the manufacturer's specifications for the steam jacket.

If the steam jacket is not designed for full steam pressure, add a pressure regulator and a pressure relief valve, possibly set as low as 5 psig. If it is not designed for vacuum, use a vacuum relief device or omit traps and valves from the drain line. A manual adjustment method is to introduce steam through a small valve until a small, steady flow occurs at the open drain line.

Less common heating methods use heat-transfer liquids or electric heaters. One method applicable to a stage without a jacket is to wrap the diffuser inlet cone with closely spaced copper steam coils embedded in heat-conductive mastic, as seen in Fig. 7.4. The jacket may be added after startup. First run the ejector at no-load to identify the ice-formation zone and mark the area of frost formation outside the diffuser. Then turn off the ejector, and apply the mastic,

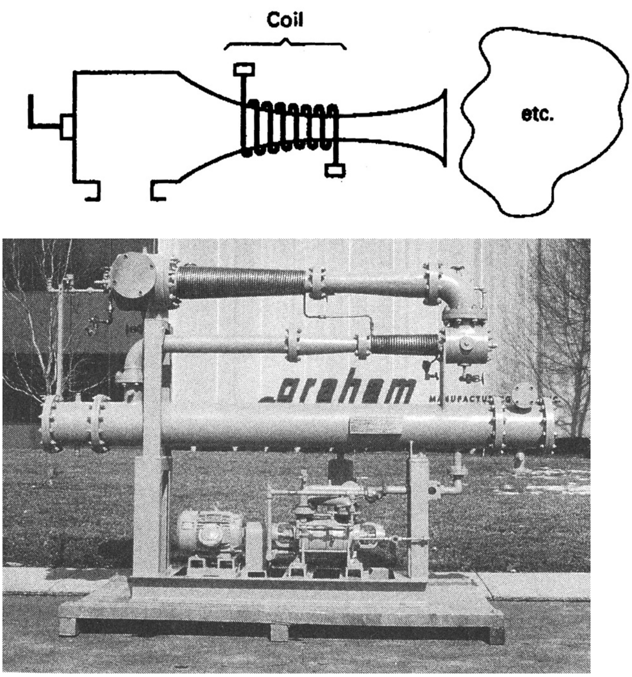


Figure 7.4 Steam coils on diffusers to prevent icing. *(Courtesy Graham Manufacturing Co., Inc.)*

and wrap the coils of flexible copper tubing to cover a zone 50 percent longer than the observed frost zone. A disadvantage of this method of diffuser heating is that unless the stage is vertical, the coil will not drain easily and must be blown out to minimize the chance of freezing.

Last-stage discharge

In several ways the details of the last-stage installation are more critical to ejector operation than those of any other stage. The last stage discharges to the atmosphere, with the attendant noise, steam plume, and air pollution. If the ejector operation is interrupted, air may be sucked back into the process system and create a hazardous situation or contaminate the product. Anything done to solve these problems adds to the discharge pressure, which may "break" the stage's operation and cause the rest of the ejector system to backfire. The solutions have to be within the last-stage limitations, or the last stage must be modified or replaced with one having a higher discharge-pressure capability.

Some of the approaches taken to meet one or more of the design objectives are shown in Figs. 7.5 and 7.6. In Fig. 7.5 the stage is shown oriented up and down in several configurations not involving a hotwell. It may also be oriented horizontally. A check valve is sometimes added to the discharge line to prevent air from being sucked into the process system during upsets, but check valves are not regarded as highly reliable. They may be low-pressure-drop swing-check designs for horizontal installation, or they may be ball- or piston-check designs, which provide a more positive flow check at the expense of a higher pressure drop, for which the last stage must be designed.

It is poor practice to add a short discharge line to the atmosphere with no silencing device because the noise is distracting and may exceed legal limits [4] and cause hearing damage to nearby workers. A long vent line may act as a muffler, and should be one or two sizes larger than the ejector discharge connection to avoid excessive pressure drop. Almost any device which is designed to reduce the noise will do the job.

As an example, the unsilenced sound power level for 100 pph of 150-psig steam discharging directly to air is about 120 dB (re 1.0E-12 W). The OSHA [4] limit for 8-h exposure is 92 dB, and a commercial silencer will reduce the noise by more than 30 dB.

Figure 7.6 shows various treatments of the last stage connected to a hotwell. The simplest arrangement is to discharge the mixture of steam and air where it will contact the hotwell overflow, which will

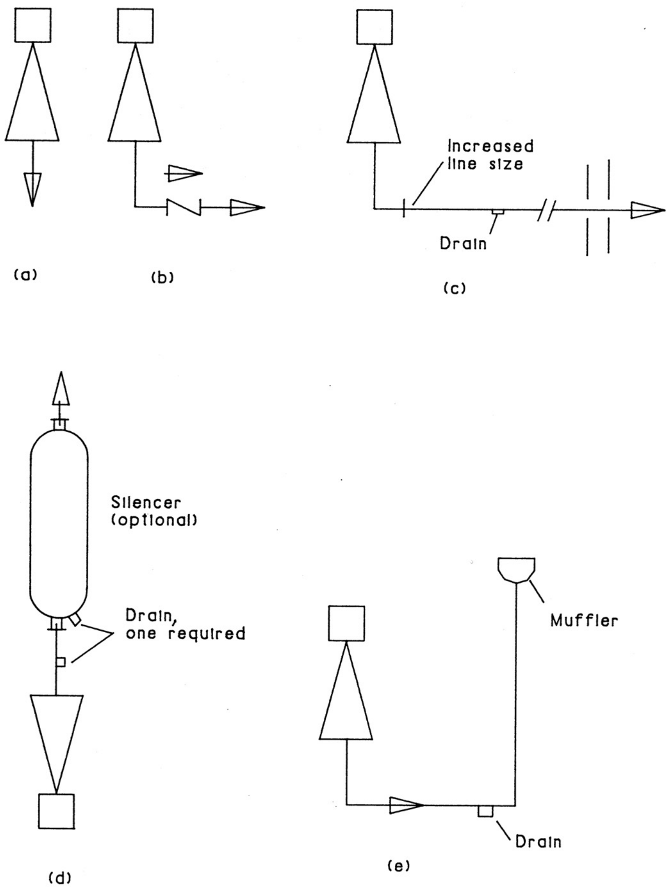


Figure 7.5 Last-stage discharge arrangements, no hotwell: *(a)* down; *(b)* check valve; *(c)* long vent line; *(d)* up, silencer optional for any position; *(e)* muller.

condense some of the steam without creating much more back pressure. The size of the line from the last stage to the hotwell should equal or exceed the size of the discharge connection on the last stage. Adding a water spray to the downpipe reduces the steam released to the air at the hotwell. Sealing the leg in the hotwell gets rid of the air backflow problem but increases the discharge pressure by the depth of submergence. If a water spray is not used in the downpipe, some banging may occur as the steam meets the cool water. For small steam flows the noise may be acceptable, but for larger units some noise control may be needed.

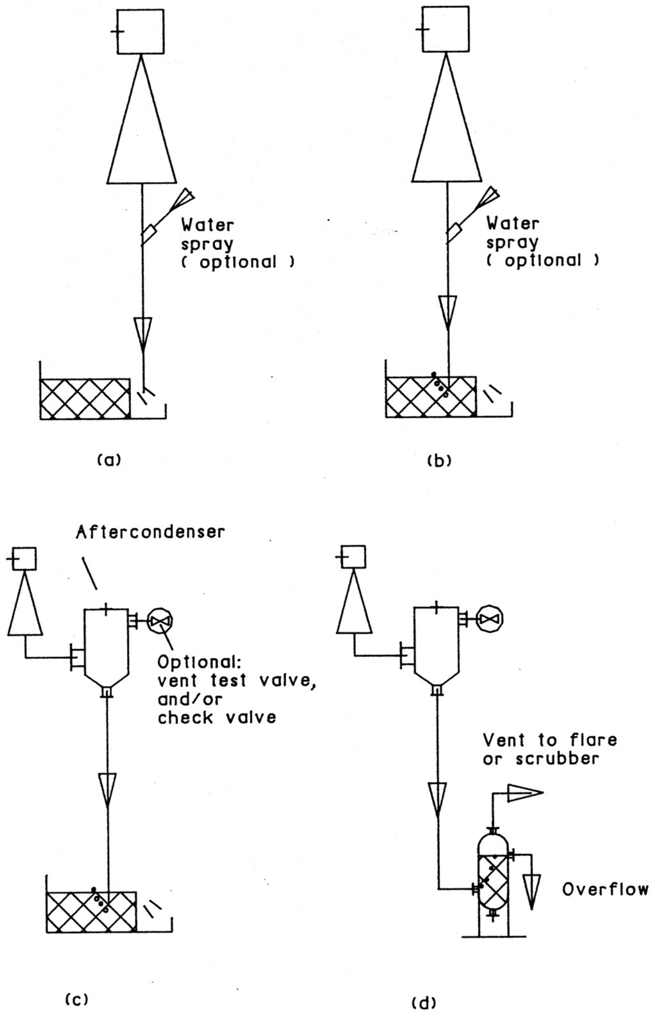


Figure 7.6 Last-stage discharge arrangements, hotwell, sealpot, and aftercondenser: *(a)* to hotwell overflow; *(b)* sealed in hotwell; *(c)* aftercondenser, vent test valve optional; *(d)* sealpot, with pressurized vent.

Adding the fabricated diffuser end to the downpipe as shown in Fig. 7.7 will reduce that noise. The sparger design shown is an inverted tee with perforations to inject the steam into the water in the form of multiple jets. A suggested design basis is to size the holes for 100 ft/s velocity, representing a compromise between low pressure drop and effectiveness. Using that basis, the combined hole area will be about twice the ejector discharge connection area.

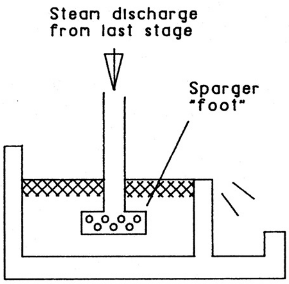


Figure 7.7 Fabricated sparger "foot" for last-stage hotwell drainleg.

An aftercondenser may be added after the ejector, with the drain leg submerged in the hotwell. The aftercondenser may be vented directly to the air or through a check valve to prevent reverse flow. Or it may have a normally closed test valve which forces the vent gas down the drain leg for a positive seal against reverse flow, yet retains the test option of venting momentarily to the air to measure air leakage or see whether reducing the discharge pressure makes the last stage work better. I like this option.

If the last stage is sealed in the hotwell (sealpot) and if the sealpot is enclosed so that it can be vented to a flare stack or waste gas processor, then the design discharge pressure of the last stage must be adequate for the expected pressure plus an allowance for ejector aging.

It is not uncommon to have the design evolve as the job progresses and to encounter a situation in which the discharge pressure exceeds the design value. Contact the manufacturer to work out a fix. It may be as simple as a new steam nozzle for the last stage or a new last stage that is dimensionally interchangeable with the existing one. Or it may make sense to add an additional small stage to boost the discharge gas to the required pressure.

7.4 Condensers

Condensers must be installed so that several fluid-flow phenomena occur in the proper manner: The vapors enter without a large pressure drop, condensed liquid drains away by gravity, the vent gas leaves easily, and the cooling water follows the proper flow path with the proper velocity.

Water supply and condensate removal are discussed later as separate topics.

It is convenient and acceptable to have test pressure taps located at the inlet and vent nozzles. Gauge glasses are a breakage and airleak source, and this offsets their occasional use in troubleshooting. With convenient test connections, a gauge glass can be added or improvised quickly.

Contact (barometric) condensers

Contact condensers must be installed vertically, and levelness is important to maintain uniform water distribution, especially if water is distributed by a weir.

Surface condensers

Surface condensers must be installed in the position for which they were designed. Any other position may cause problems with venting or draining. If it is desirable to install a condenser contrary to the manufacturer's design, contact the manufacturer. Sometimes something can be done. Provide convenient access for maintenance activities such as removal of the exchanger, or the bundle, or the shell, or individual tubes, depending upon the maintenance practices at the location.

7.5 Steam

The greatest contributors to ejector failure are below-design steam pressure and wet steam. Low pressure, high pressure, high superheat, solid particles, or fouling deposits will cause reduced performance or failure. Moisture erodes nozzles, diffusers, and the equipment into which a stage discharges: another stage, a piping elbow, or a condenser.

The steam pressure delivered to each stage should slightly exceed the design pressure at all times. Extract the ejector steam from the top of the steam header, because the steam in the header will generally not be in equilibrium, and water may accumulate in the bottom of a header. Remove entrained water with a properly sized and oriented separator equipped with a bucket steam trap.

Size the steam line to each stage to keep a high velocity without excessive pressure drop, usually matching the connection size provided by the manufacturer. Insulate the steam lines for steam quality as well as safety. In a multiple-booster ejector, try to arrange for the first-stage steam to come from an active region of the supply header where the steam has the least moisture, not at the end of the header.

One manufacturer told me about an ejector for which he carefully specified a small-diameter steam line to the tiny nozzle on the first stage. He wanted to keep the steam velocity high enough that any liquid present would remain finely dispersed by the turbulence. The user replaced the small line with a larger line, probably using the common "bigger is better" justification, and was troubled by the occasional slugs of water that interrupted performance briefly. If you wish to do something contrary to a manufacturer's recommendation, discuss it with the manufacturer. If there is a mistake, the feedback to the manufacturer may help the next user. If the mistake is yours, you will do this job better and be smarter for the next one.

If the first stage of a multistage ejector uses less steam than would be admitted by the minimum allowable orifice size at the steam design pressure, the manufacturer may require that the steam to that stage be throttled to a lower pressure. Provide a high-quality steampressure gauge with pigtail, and a throttling valve, and identify the requirement clearly for the people who will operate the ejector. After it has been adjusted, the throttling valve at the first stage need not be operated again if the ejector is turned on and off by a master steam valve for the ejector.

Although excessive superheat is seldom a problem, it can keep an ejector from working properly unless the pressure and/or nozzles are altered accordingly. If this is a possible problem, discuss it with the manufacturer.

Remove excessive steam pressure (more than 20 percent high) with a regulator. If full supply pressure would create a hazard in the ejector, place a relief valve after the pressure regulator, size it for the full flow which the regulator can deliver and set it at 20 percent above the controlled delivered pressure.

Protect steam traps and steam nozzles with in-line strainers. Nozzle throats of 3/8 in diameter are usually regarded as requiring no strainer protection.

I do not like the series steam valve arrangement shown in Fig. 7.8. It is intended to force the operator to turn the steam on, stage by stage, starting at the last stage and moving forward. Shutdown is from first

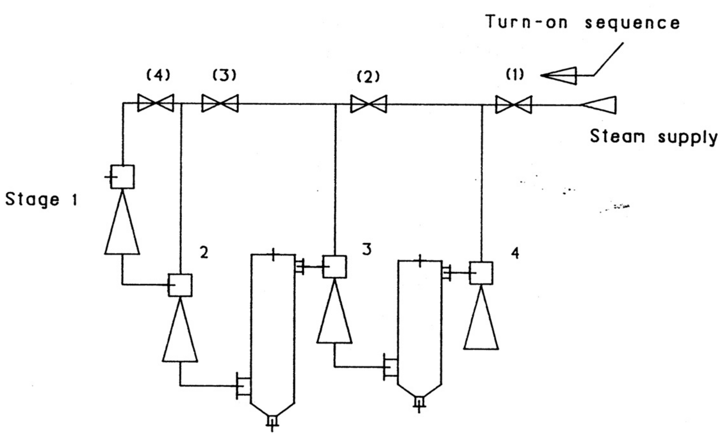


Figure 7.8 Series steam valve arrangement. Pro: Enforces quick-drawdown sequence for manual operation. Con: Valves are larger and all in header. It is less flexible and can't warm up first stages as easily.

to last. The apparent intent is to turn the stages on in the sequence which gives the quickest manually controlled drawdown. Although this arrangement does accomplish that, I think that the quicker drawdown is seldom very beneficial, and I see some other disadvantages.

Because the last-stage valve handles all the steam flow, it will be the largest valve in the set and will tend to be located in the steam supply header for the ejector; this complicates the header design. The same applies to a lesser degree to the other valves. Finally, the first stage (the one most sensitive to moisture in the steam) is located at the end of the steam header, the wettest location in the header.

When I start a multistage ejector, I like to drain the booster stages and warm them up by turning on their steam and leaving them in a "broken" mode until they are warm and drained. Then I turn on the final stages to flash off any remaining moisture and proceed with the testing. This steam piping arrangement requires me to add an air overload after the first condenser or otherwise defeat it.

Place a calibrated, high-quality steam pressure gauge with protective pigtail at or near the steam chest of the last stage. The gauge should be a type not damaged by vacuum. It should be located where it will be visible to the operator who makes adjustments to the pressure regulator. The gauge should have a check screw and socket with slotted link to take care of sudden pressure release.

If steam is to be used as an external load for controlling system pressure, that steam should also be free of excessive moisture.

7.6 Cooling Water

Select a clean water source cool enough that it will not exceed the design water temperature. Contaminated water (containing chemicals, algae, or decaying organics) may foam and release gases. Usually the plant water supply filter keeps out large debris, but small water nozzles may become plugged by debris. If the water is known to contain debris, add a strainer with blowdown valve in the supply line.

Water must be piped in the sequence intended by the manufacturer for a surface condenser. Often it is economical to pipe water through the condensers in series, starting with the first condenser (for example, the XY intercondenser in a four-stage system) and continuing in series through the others, or splitting to flow separately to them.

Locate water-flow adjustment valves and water-outlet thermometers for easy adjustment of flow. For contact condensers, the thermometers must be close to the hotwell to ensure that they will be below the water level (no more than 10 ft above the hotwell water level for intercondensers). An inlet water thermometer should be near the ejector system.

If the water adjustment valves are not near the outlet thermometers, then the adjustment task becomes time-consuming for one person or requires two people to coordinate the valve adjustments and temperature readings. Cooling-water valves tend to be opened completely. Accordingly, use two valves in series per contact condenser: one to adjust the flow, and one to turn it off and on without disturbing the flow adjustment. If the flow adjustment valves can be locked in position, they are less likely to be accidentally readjusted. A master on/off valve for an ejector may be more convenient for normal operation, but it complicates troubleshooting. If water pressure varies widely, add a pressure regulator and make flow adjustments based on the lower, controlled pressure. Or, use a differential temperature control system to maintain the design water temperature rise.

In sizing and routing the supply lines to the ejector condensers, try to prevent even brief starvation of the ejector water supply caused by nearby demand surges.

Water pressure must be adequate for a 15-psi drop in the spray nozzle for most spray-type contact condensers. That requires a supply pressure of 15 psig for aftercondensers, 10 psig for an intercondenser just ahead of a Z stage, and 5 psig for all intercondensers ahead of a Y stage. If a booster pump is present, it should be designed to avoid air in-leakage, which will load up the condensers and ejector stages. Large pressure variations may cause flooding or poor distribution, with breaks in the spray patterns or water curtains.

If a contact condenser has a water distribution weir, the inlet line size should match the inlet connection size.

Some people prefer to use a four-way valve in the water supply to surface condensers so that the water flow may be reversed periodically to help flush away silt and light fouling. Do not rely on a siphoning effect in the discharge line to carry low-pressure water to a surface condenser; it may cause flashing and water hammer.

Vents and drains are identified by the manufacturers. Usually the water will enter through a bottom connection on surface condensers and exit at the top. If the flow direction is downward, make sure the flow velocity is high enough to prevent vapor locking. If in doubt, ask the manufacturer.

7.7 Condensate Removal Drains

They look so simple! What can go wrong with a simple piece of pipe? Well, there is room for some surprises, and some shortcuts can cause problems. Most vacuum condensers are drained by a pipe (barometric leg) leading down to a hotwell, with a vertical height sufficient that gravity can "pump" the liquid from the vacuum to atmospheric (barometric) pressure.

Drainlegs, one per condenser recommended

The most trouble-free arrangement is to run each condenser barometric drain separately to the hotwell without any flanges above the hotwell water level. It is usually false economy to drain one condenser back to a lower-pressure condenser; it may save some piping cost, but it creates problems during testing and troubleshooting, and it may recycle some vapor if condensate flashes as it enters the lower-pressure condenser.

However, a loop seal or a float trap can be used, as shown in Fig. 7.9. A gate valve at the discharge end of the seal loop allows the operator to close the valve briefly to reestablish the seal if an upset causes it to be blown out. I recommend against draining the aftercondenser to an intercondenser through a trap, but some people prefer it to eliminate a very long seal loop or a drainleg to a hotwell.

It is recommended that dimension H in Fig. 7.9*a* be 7 ft for all but special cases [5] and as much as 8 to 11 ft ("up to 15 ft if space is available"), probably including the special cases [7]. Seven feet translates to about 150 torr differential pressure between the condensers, less some allowance for surging. The design intercondenser pressures do not describe the full range of possible conditions that may exist in condensers joined by such a loop seal. If control is by suction throttling, the intercondenser pressures may be very low, especially with cold water in the winter. With control by noncondensable gas load, the intercondenser pressures will generally be slightly above the design values. If you wish to use this arrangement in spite of its dis

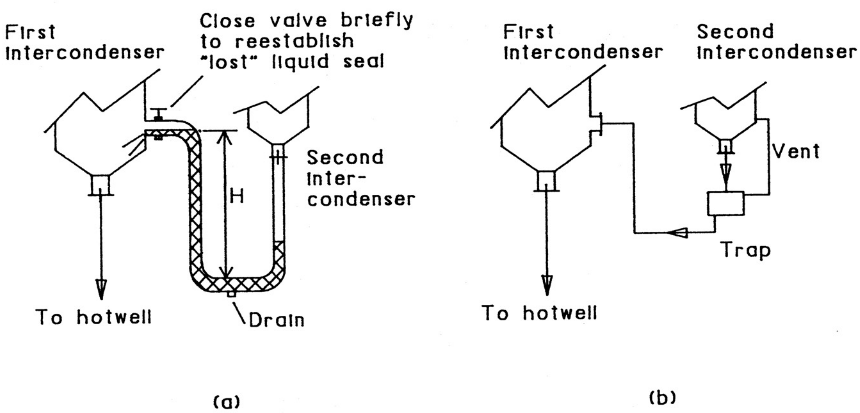


Figure 7.9 Interconnecting drainlegs. Discuss with manufacturer to avoid problems. *(a)* Loop seal; (*b*) trap.

advantages, I recommend you discuss the specific application with the ejector manufacturer.

Horizontal offsets. Avoid horizontal runs because vapor binding may flood the condensers. If a vertical line from the condenser to its hotwell is not possible, use a 45° slope in horizontal offsets to assure positive venting. If horizontal lines must be used, keep them short and vent the downstream end back to the condenser vent to assure positive venting and avoid creating a "trap," as shown in Fig. 7.10. With careful attention it is possible to use a short section of horizontal or "nearly" horizontal drainleg.

If you wish to put a horizontal or nearly horizontal run in a drainleg and do not wish to add a vent line, then you must make sure that the line is self-venting. For the line to be self-venting, its slope should be greater than the hydraulic gradient.

A conservative design approach is to treat the flow as open-channel flow, with the line assumed to run only half filled with liquid. The flow velocity will thus be twice that in the vertical leg. The Reynolds number will be twice as large, and the corresponding friction factor will be a little smaller. Conservatively neglecting the change in friction factor, we find the head loss (H_f) from fluid friction to be four times the head loss in a full line. Converting that to feet of liquid per 100 ft of line (including the effect of any fittings) in the horizontal run gives the hydraulic gradient. Then the line must be installed with a slope which exceeds the hydraulic gradient to permit a large bubble to flow upstream along the top of the pipe until it escapes into the preceding vertical leg.

As an example, consider a small flow of water, 500 pph (1 gpm) of condensate from a surface condenser. Usually the drainleg is sized to match the manufacturer's drain connection, typically resulting in a flow velocity of 1 to 3 ft/s, or lower. Let us assume that we sized it using the conservative equation below for self-venting lines sloped at least 10° above the horizontal, based on water and atmospheric air and standard gravity.

$$d\text{ (in)} = 0.92Q^{0.4}\text{(gpm)} \qquad (7.2)$$

The required line size is $0.92(1.0)^{0.4} = 0.92$; say 1 in.

Consulting a Crane fluid-flow handbook [6], we find that 1 gpm water flow in a 1-in schedule 40 line results in a velocity of 0.37 ft/s and a pressure drop of 0.048 psi per 100 ft. Converting to feet of head loss, 2.31(0.048) = 0.11 ft per 100 ft. Multiplying by 4 for open-channel flow (half full in horizontal run) results in a hydraulic gradient of 0.44 percent. If fittings are present, the gradient will be larger. Now the pipe must be installed with a slope greater than this; perhaps 1 or 2 percent is practical.

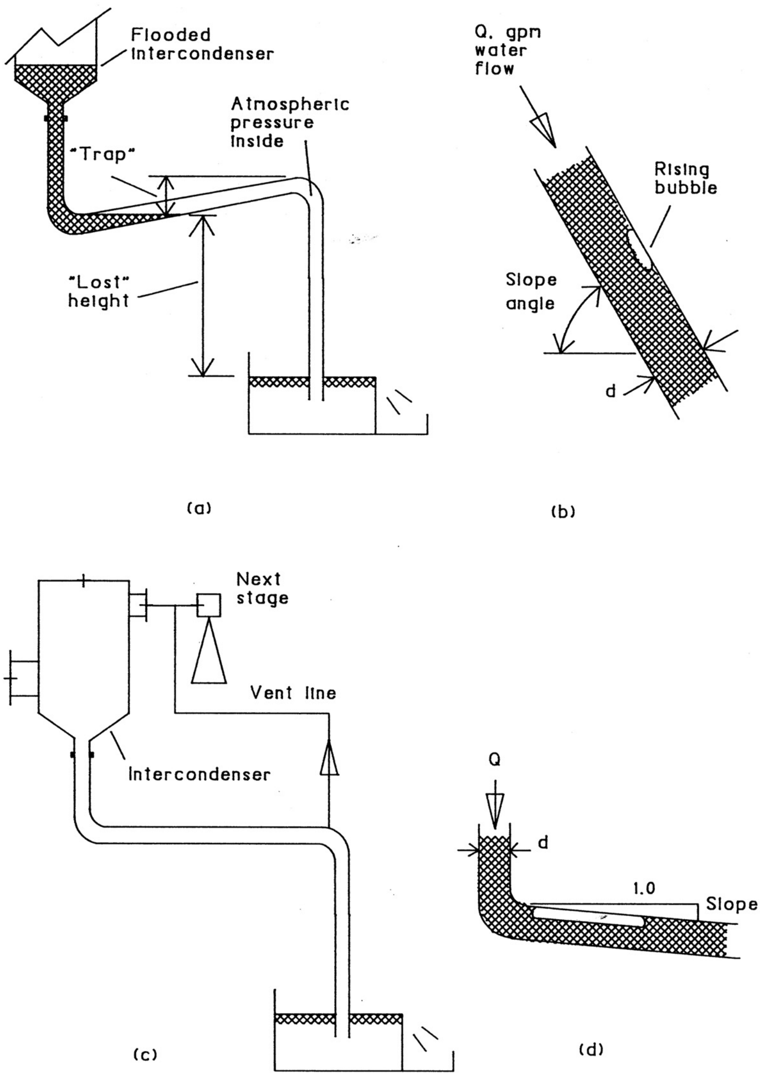


Figure 7.10 Positive venting of drainlegs: *(a)* improper horizontal "trap" run; *(b)* self-venting line, slope angle greater than 10°; *(c)* horizontal run vented at end; *(d)* shallow-slope geometry for self-venting.

Repeating with a flow of 1000 gpm, we find the conservative diameter to be 16 in for the vertical or steep slopes, a velocity of 1.8 ft/s in the full section, and a hydraulic gradient of 0.25 percent in the half-full section.

Reflecting on this picture a little, we can see that a moderately long section of horizontal run will still be self-venting for the larger diameter. A gradient of 0.25 percent results in a slope of 3 in in 100 ft — only one-fifth of the diameter in an unusually long run.

By following the guidelines with care, you can end up with a convenient layout and drainlines that work.

Thermometers. Locate a thermometer in each condenser drainleg to help you adjust the water flow by observing the temperature rise. If the thermometer is within 10 ft of the hotwell, the sensing element will be immersed in the flowing liquid condensate. Above that level, the sensing element may not be positively wetted by a flow of liquid. Desired accuracy is about 1°F for the first intercondenser. A dial thermometer is acceptable for the rest of the condensers. Thermowells permit replacing the thermometers without interrupting operations. Make sure the thermometer does not obstruct the flow.

Drainleg diameter

The drainleg diameter should match the condenser drain size. An oversized drainleg may allow the entrained air bubbles to separate, accumulate into a large bubble below the condenser drain, and cause flooding in the condenser.

Drainleg recommended length 36 ft

The drainleg should have a vertical length sufficient to prevent flooding of the condenser under the worst conditions: high barometric pressure, low condenser pressure, low-density liquid in the drainleg, pressurized hotwell vent, fluid friction in the line, and an allowance for surging. A height of 36 ft above the overflow level in an open hotwell is a conservative guideline if all liquid condensate is soluble in water.

Normal atmospheric pressure will lift cold water about 34 ft in a vacuum, and a 2 percent increase above normal pressure is common.

An ejector system which is shut down and at atmospheric pressure may be suddenly exposed to a low absolute pressure. If the isolating valve between the ejector and a large vacuum system is suddenly opened, the entire ejector system will soon reach the pressure in the vacuum system. Atmospheric pressure in the hotwell will force water up each drainleg, and the water may surge above the final steady-state level. If the water vapor pressure is higher than the system pressure, the top layer of water may boil until it cools.

Condensate may have a density less than that of water. Normally water is the dominant material present in the condensate, especially for contact condensers cooled with water. A problem may arise, however, in the special situation where the process vapor condensate is not soluble in water and where surface condensers are used. If the vapor condensate is lighter than water (toluene, for example) and the drainleg liquid velocity is low, then the insoluble liquid phase will be continuous, and water condensate will descend through it as the discontinuous phase. A conservative design approach is to assume that the drainleg is entirely filled with the insoluble liquid and calculate the height to which the hotwell pressure will raise it in a vacuum.

Because a hotwell (sealpot) handling immiscible liquids is often used as a decanter to separate the two liquid phases, the liquid above the liquid entry level will be almost pure "oil" phase. This condition ensures that when the ejector system is restarted after experiencing atmospheric pressure, all drainlegs will be filled with pure "oil" liquid.

In some applications the hotwell vapors are toxic or a nuisance and must be disposed of by scrubbing or burning in a flare tower. The hotwell must then be enclosed, and the vent will be above atmospheric pressure. This pressure must be added to the design atmospheric pressure when calculating the required drainleg height.

Finally, the head loss caused by fluid friction must be considered. Usually this is so small that it may be neglected, but the calculation is routine for system designers, and the needed data are in the manufacturer's drawings and design specifications.

Collecting all the above considerations into one equation and referring to Fig. 7.11,

$$L_{\mathrm{sl}} = \frac{2.31(P_v + P_a) + L_{\mathrm{hw}}}{\mathrm{sg}} + H_f + \mathrm{SF} \tag{7.3}$$

where, L_{sl} = length of seal leg, ft
P_a = atmospheric pressure, psia
P_v = gauge pressure at hotwell liquid surface, psig
sg = specific gravity of condensate; water = 1.0
L_{hw} = submergence of seal leg outlet in hotwell, ft
H_f = head loss from fluid friction, ft of condensate
SF = safety factor allowance for surging and increases in atmospheric pressure; use 1 ft

As an example of the effects of an adverse combination of circumstances, consider the situation in which toluene condensate flows down a drainleg along with water from a surface condenser. The drain line is generously sized for a low velocity, so the immiscible toluene forms the continuous phase. The lower end of the drainleg is submerged 2 ft in the hotwell. The hotwell vent has a design back

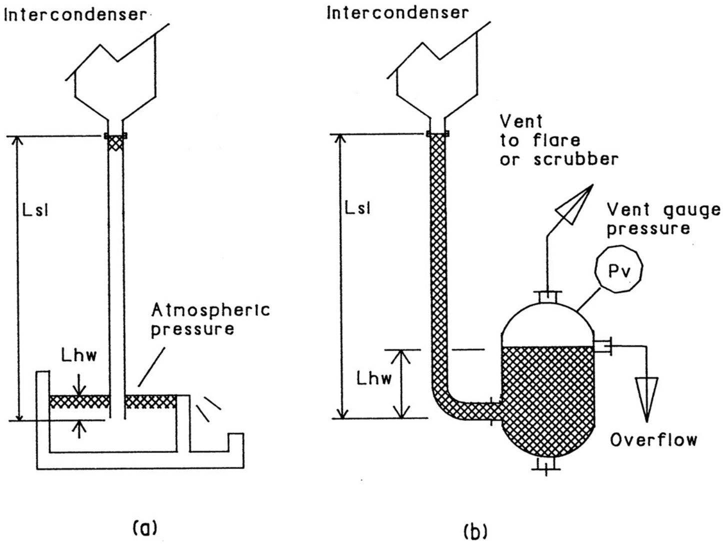


Figure 7.11 Drain leg height requirements: *(a)* open hotwell; *(b)* sealpot with pressurized vent line.

pressure of 3 psig, and the design atmospheric pressure at this altitude is 14.5 psia. The calculated head loss from fluid friction is less than 0.1 ft of condensate.

$$L_{sl} = \frac{2.31(3.0 + 14.5) + 2.0}{0.87} + 0.1 + 1.0 = 50\ \text{ft}$$

Imagine the problem created if these considerations had been overlooked and the common value of 34 ft (plus 2 ft submergence) had been used! One solution would be to raise the ejector structure and/or lower the hotwell. Another would be to use an ejector or mechanical blower to reduce the vent back pressure. Not good!

Low-level condensers

When an elevated ejector installation with barometric drainlegs is not practical or cost-effective, the condensers may be located at a lower level and the condensate removed by special condensate pumps. A liquid-level controller is required to maintain adequate liquid pressure at the pump inlet.

A condensate pump is designed to avoid cavitation. As liquid enters the pump, it accelerates and experiences a reduction in pressure. If the absolute pressure of the liquid is less than the vapor pressure of the liquid, some of the liquid will flash into vapor. This vapor reduces the average fluid density inside the pump and drastically reduces the pump's ability to increase the fluid pressure. As the vapor bubbles move into higher-pressure areas, they collapse, producing a very high shock pressure at the precise point where they collapse. Those bubbles which collapse near the pump metal surfaces will destroy the metal quickly.

The condition in which the pump can perform successfully without damage is described quantitatively by the experimentally measured net positive suction head required (NPSHR). It is defined as the total head at the pump inlet (pressure head plus velocity head) less the vapor pressure of the liquid, both usually measured in feet of the liquid being pumped. The pump installation must meet this requirement, with a small safety factor. The NPSH available (NPSHA) in the system is defined the same way as that for the pump and is physically equivalent to the vertical distance between the liquid level in the condensate receiver and the centerline of the pump inlet. This assumes that the condensate is just saturated at the liquid surface and that friction losses in the short line to the pump are negligible. Subcooling the condensate increases the NPSHA, and line friction losses reduce it.

Because vertical space is at a premium in a low-level installation, it is desirable to use a pump with a low NPSHR. The NPSHR is often plotted versus flow on the pump performance curve. Low-NPSHR pumps are designed to run at lower than normal pump speeds, have large carefully shaped inlets, and cost more than regular pumps. I mention all this to keep you from using a regular pump without considering the NPSH factor. It is worth remembering later on, because replacing a failed low-NPSHR pump with a regular pump will not work.

The condensate piping from the receiver to the pump should be short with no valves and sloped down toward the pump. The pump casing should be vented to the condenser. Locate a globe valve and a liquid-level-controlled throttle valve in the discharge line (not the suction line). See Fig. 7.12 for a low-level three-stage ejector installation. Note that the liquid-level sensor on the receiver and the flow-control throttling valve on the pump discharge are required but not shown. See Chap. 11 for more on this subject.

An alternative treatment of low-level condensers is to have two receivers, with a timer or liquid-level sensor which switches to the empty receiver when the first is full, then drains or pressures the condensate from the first, then switches again when the other receiver is full. The emptied receiver should be slowly vented to the condenser to avoid a bobble in the vacuum caused by suddenly dumping a charge of air into the condenser when the receivers are switched.

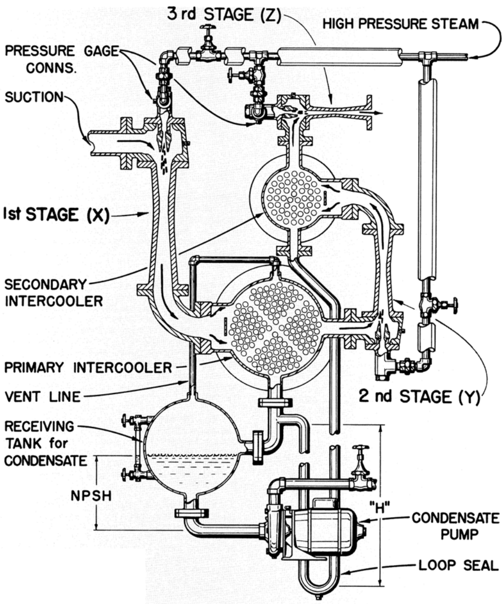


Figure 7.12 Low-level installation with condensate pump. *(Courtesy Jet-Vac Corp.)*

7.8 Hotwells or Sealpots

I have used the term "hotwell" to refer generically to the device to which barometric drainlegs deliver the liquids removed from condensers. Schematically the device is often drawn as an open-topped sump near ground level, with a simple weir overflow. The term "sealpot" is probably more representative of good current practice, which consists of some type of fabricated metal container. This is better suited for containing the liquids and vapors and minimizing the possibility of environmental contamination at this location.

The older practice of using a concrete sump with a simple overflow to the sewer is not appropriate unless the service is extremely clean – a steam-jet refrigeration system, for example. The condensate of most process vapors should not be discharged to the environment without treatment. A concrete sump constitutes an underground storage tank and must be treated accordingly.

A variety of recommendations on this subject are given in the ejector literature. I will lean here toward the conservative, assuming you are familiar with the drainleg guidelines discussed previously. See Fig. 7.13 for several hotwell examples.

Size hotwell sealing volume (free surface area multiplied by the least of the drainleg submergences) to fill all drainlegs to at least 36 ft, plus an additional 50 percent. The least drainleg submergence

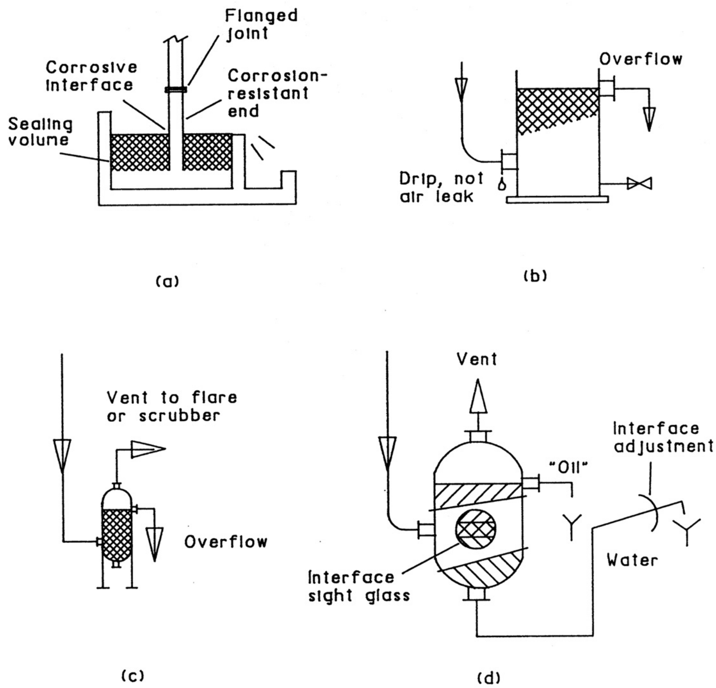


Figure 7.13 Hotwells and sealpots: *(a)* top entering; *(b)* side entering; *(c)* pressurized vent; *(d)* integral decanter.

often is on the aftercondenser. The factor of 50 percent is a modest allowance for some evaporation loss during shutdown and for some surging and splashing during operation. I have seen more conservative recommendations of 100 and 200 percent. Another criterion I have encountered but did not research is to size the hotwell volume for at least 30 s of flow. That seems to be based on contact condensers, with their larger drainleg flow rates.

If the hotwell is enclosed because the gases and vapors present are corrosive, odorous, or toxic, vent it to a low-back-pressure area. Size the vent system for the largest expected flow, usually during the initial evacuation period. Hogging jets will increase this air flow if they also discharge here. Remember that if the hotwell pressure is above atmospheric pressure, this will raise the liquid higher in the drainlegs and raise the discharge pressure seen by the last stage if its vent is sealed in the hotwell.

A hotwell vessel may double as a decanter for recovering immiscible liquids mixed with the steam condensate. Design the drainleg height requirements along with the decanter because the location of the liquid surface may be different from that in an open hotwell.

Top versus side entry

Top entry to an open concrete hotwell is the configuration typically shown in the ejector literature. The bottom of the drainleg should be 2 or 3 diameters above the bottom of the hotwell to allow for unrestricted flow and some accumulation of debris. This is simple and easy to understand and design, but it has some disadvantages. Corrosion is most rapid at the air/water interface where the drainleg enters vertically into the hotwell. The pressure inside the drainleg is a little below atmospheric pressure, so a leak at this location will allow air to bubble up the drainleg and load up the ejector stages. Sometimes the first symptom of this may be that the last stage discharges a stream of water because the rising air bubbles resulted in flooding of the previous condenser.

A correction for this problem is to use a corrosion-resistant material for the bottom part of the drainleg, with a flanged joint above the hotwell for easy replacement. The flanged joint itself becomes another potential air-leakage site, but it is a convenient location for adding a slip blank for hydrostatic leak tests.

Another approach is to bring the drainlegs into the hotwell on the side below the water level. This is easier to do with a metal fabricated sealpot. The entrance is typically flanged. If the drainlegs arrive below the air/water interface, ordinary pipe material may be used, and any leak at the flanged joint results in drips instead of air inleakage. The configuration makes it easy to swing the pipe aside to insert a slip blank or to divert the condensate flow for testing.

7.9 Troubleshooting

An isolating valve between the first stage and the process system helps in locating the source of vacuum problems and doing brief testing and maintenance on the ejector without shutting down the process system. Flanges between stages and condensers permit inserting slip blanks for isolated testing of stages and testing for leaks. Shutoff valves at each stage and condenser permit selective operation of stages and condensers. At least one vacuum-pressure test connection should be accessible in each stage (not at the bottom). If the stage does not have a test-pressure tap in the suction chamber, add one to the top or side of the suction line ($^1/_4$-, $^3/_8$-, or $^1/_2$-in NPT). Steam-pressure taps close to each stage steam nozzle should also be present.

A very convenient feature is an aftercondenser vent valve which is normally closed to seal the atmospheric vent in the hotwell, but can be opened for a garbage-bag airflow test or to test the effect of reducing the last-stage discharge pressure. Other vent configurations should provide easy and safe access for the garbage-bag test.

When continuous monitoring of the air-leakage rate is desired, an exhaust-air meter may be added to the aftercondenser vent. The low-pressure-drop orifices have a useful flow range of 4:1.

Another method is to connect an absolute-pressure transmitter to the suction chamber of the last stage and monitor that pressure continuously. With a performance curve for the last stage, it is easy to output a continuous record of the air leakage. If that stage becomes worn or fouled, the pressure will rise, and maintenance can be performed before failure occurs. Another advantage of this monitoring method is that it may be used regardless of the configuration following the last stage.

A gauge glass on the first condenser shows the water level there to detect flooding, but it is fragile and adds leakage sites. Test plugs at the proper elevations permit a tester to improvise a gauge tube with transparent flexible tubing.

7.10 Drainage

Ideally all equipment and piping should drain dry when liquid flow stops. Low spots in stages or piping should be drained for quick startup, preferably automatically where large volumes of liquid may accumulate. See Fig. 7.14 for some examples. This includes the motive steam chest and steam piping, as shown in Fig. 7.14*d*.

Freezing damage is also a consideration in cool climates. A diffuser steam jacket that is full of condensate and valved off may be damaged by freezing if that stage operates below 5 torr, or the diffuser may collapse from external pressure. Steam trace all lines which may have standing water in them (even steam lines!).

7.11 Safety

Insulate all steam-heated surfaces to avoid burns: steam lines, steam jackets, and the discharge end of ejector stages. It is not necessary to insulate the ejector suction chamber and intercondensers for safety, but your cold weather experience may indicate that insulation is necessary to prevent freezing in normal operation in very cold weather.

Avoid valving arrangements which permit steam pressure to be applied to components which may not have been designed for more than a 15 psig hydrostatic leak test. Do not put valves in barometric legs. Do put relief valves on low-level systems.

Do not permit operator error to overpressure components which are not designed for full steam pressure; add relief valves if in doubt. Locate relief valves in each section of the system which may be valved off and subjected to high-pressure steam. The pressure setting should be based on the weakest component in that section, usually a con

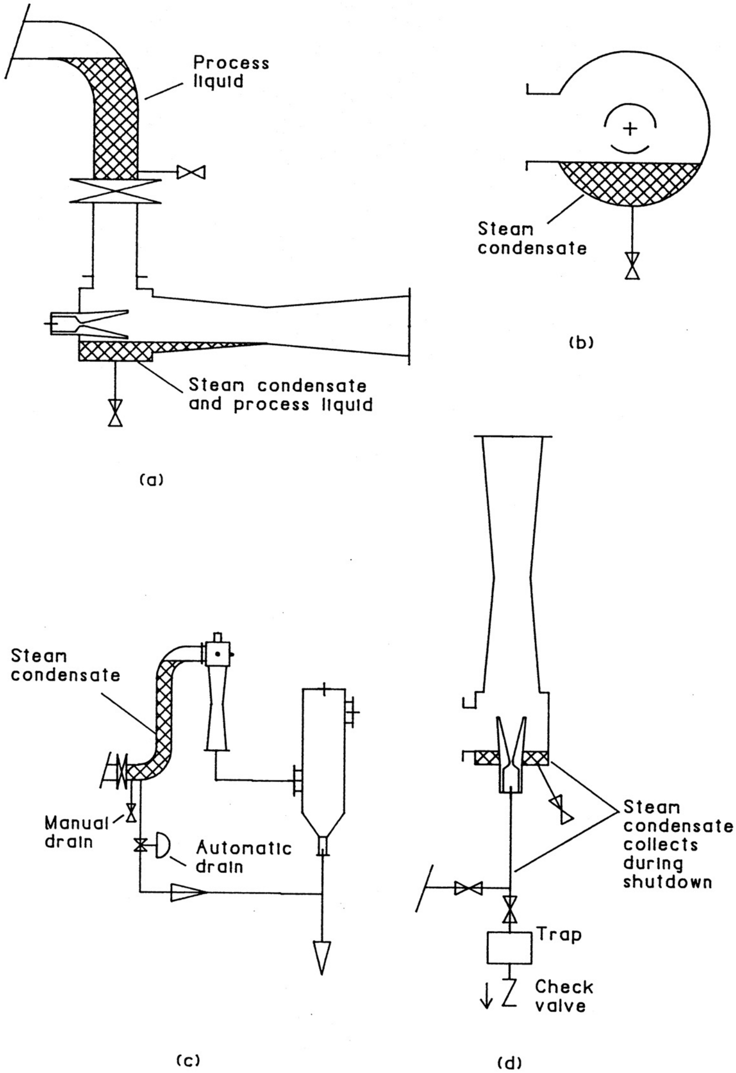


Figure 7.14 Low spots to drain: *(a)* horizontal stage, top suction; *(b)* horizontal stage, side suction; *(c)* nondraining suction line; *(d)* vertical stage, discharge up.

denser. Unless specifically designed for higher pressures, components should not be expected to be suitable for more than a 15 psig hydrostatic leak test. The relief capacity should be based on motive steam pressure at least 25 percent over the design pressure and steam nozzle throat areas 10 percent in excess of design. See the HEI standards [2] for more on this subject.

However, if all components have been designed for high pressure and tested for strength, the remaining task to ensure continued safety is to document this exception properly so that all new and replacement components will be mechanically strong and not brittle (cast iron, carbon, ceramics). People accustomed to using low-pressure components for ejector systems may otherwise make repairs or replacements with unsafe components.

Vacuum itself can collapse large vessels that are not designed for it. Although such accidents are infrequent, vacuum is dangerous if applied to a large area of the body or to fragile organs such as eyes or ears. Entry into a vacuum vessel should be treated with as much care as entry into a pressure vessel.

7.12 Nomenclature

fps	feet per second, velocity
H_f	head loss from fluid friction, ft of condensate
L_{sl}	length of seal leg, ft
L_{hw}	submergence of seal leg outlet in hotwell, ft
P_a	atmospheric pressure, psia
PC	pressure controller
pph	lbm/h
P/V	pressure/vacuum gauge (for approximate readings)
P_v	vent pressure, psig
SF	safety factor allowance for surging and atmospheric pressure variations; use 1 ft
sg	specific gravity of condensate; water = 1.0
T	temperature
torr	absolute pressure unit, 1 mmHg

7.13 References

1. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
2. *Standards for Steam Jet Vacuum Systems*, 4th ed., Heat Exchange Institute, Inc., Cleveland, OH, 1988.
3. D. H. Jackson, "Selection and Use of Ejectors," *Chemical Engineering Progress*, 44 (5): 347-352 (1948).
4. OSHA standards, Paragraph 1910.95, "Occupational Noise Exposure," Table G-16, 29 CFR Chapter XVII (7-1-91 ed.), pp. 204-220.
5. "Instructions for Installing and Operating Multistage Ejectors with Barometric Intercoolers for Vacuum Service," Bulletin JVS-ll, Jet-Vac Corp.
6. "Flow of Fluids through Valves, Fittings, and Pipes," Technical Paper No. 410, Crane Co., 1982.
7. "Instruction Manual, Steam Jet Air Ejectors," Bulletin J-50-7, Foster Wheeler Corp, 1950.

Answer to Problem Number 6

Find test P_s in torr.

$P_s = P_{atm} - P_{vacuum}$:

P_{vacuum} = 24.7 in. Hg (760 torr / 30 in Hg) = -626 torr

$P_{atm} = P_{barom} + P_{corr\text{-}elev}$:

P_{barom} = 14.5 psia (760 torr/14.7 psia) = 750 torr

$P_{corr\text{-}elev}$ =500 ft air (lbf/13.3 ft^3)(ft^2/144in2)(760 torr in^2/14.7 lbf)= +14 torr

P_{atm} = 750 + 14 = 764 torr

P_s = 764 –626 = 138 torr

Problem Number 7

At one test condition an ejector stage receives a load of 25 pph air, saturated with water vapor at 93 torr and 100 °F. What is the water vapor load and the total load as DAE to that stage?

Answer on page 442

Chapter

8

Operation, Testing, Troubleshooting, and Maintenance

8.1 Overview

The purpose of this chapter is to help you get the best performance out of an existing ejector, assuming the ejector and its installation were designed and fabricated properly. If any major defect does exist, it will be discovered by the testing and troubleshooting procedures. Then Chaps. 10 and 7 will guide you through an appropriate redesign of the ejector or its installation.

The best single measure of my success in writing this chapter is whether it persuades you to do more field testing of your ejectors. The understanding and control you can gain by doing some selective testing will yield satisfaction and improved operations.

Good vacuum results from good jets in a good system. To keep the system in good condition for maintaining good vacuum, it is necessary to respect the need for leak-tightness in all parts of the process and ejector systems which see vacuum in normal operation. Batch operations are especially demanding. Many valves are opened and closed. Connections are made and broken, sometimes including large gasketed openings. Under the press of time in routine operations, it is easy to damage or foul the vacuum-sealing metal surfaces or gaskets, especially when loading solids or withdrawing samples. All such air-leakage sources need tender loving care.

This chapter covers four major topics. The arrangement matches the chronological events in the life of an ejector, corresponds to the organizational specialization within many companies, and is convenient to me for writing. In a smoothly running plant, however, the activities blend together. For example, it is a good idea to measure air leakage routinely after any maintenance or major operation likely to introduce or aggravate leaks. If the last-stage suction pressure is monitored continuously, this task is automated.

8.2 Operation

Startup

The startup may be done by the production department, a special startup team, or some combination which may include specialists familiar with portions of the system that are believed to need special attention. A preliminary walk-through inspection by someone with ejector experience is an effective way to spot visually apparent errors in the installation before the actual startup begins. The "new eyes" may spot problems overlooked by busy design and construction people.

Steam lines typically contain debris that takes time to flush out. Blow down steam strainers initially and again after 15 days and 30 days, then inspect them once a year. If a steam-pressure regulator is present, adjust the steam pressure to the specified design pressure plus about 5 percent. Steam jackets, if present on the low-pressure stages, may require low-pressure steam or a manual adjustment to assure an adequate flow of low-pressure steam.

If no regulator is present and the steam pressure exceeds the design pressure by more than 10 percent, throttle it to design plus 10 percent. If a steam separator and pressure regulator are not present, consider adding them before startup. Low steam pressure and wet steam are some of the most common causes of ejector troubles.

Water lines should be flushed to remove debris. Hotwells (sealpots) which do not receive water from contact condensers should be filled with water to provide the initial sealing liquid immediately on startup.

If a two-valve arrangement is used for each condenser to separate the on/off and flow-control functions, now is the time to make the flow adjustments. Turn on the steam to all stages and water to all condensers. Adjust the flow-control valve on the first condenser until that condenser's water temperature rise is equal to the design rise. Repeat with the next condenser, and so on. Then seal the flow-control valves.

If design data are not immediately available, you may use 20°F as the approximate temperature rise in the last intercondenser and 10°F for the first of two intercondensers [1]. The actual design values may be less than these if the design water temperature is abnormally high or if the ejectors have been designed for the lowest possible steam use.

Contact condensers may flood if the water flow is too high. Surface condensers will operate with high water flows, but erosion may result from water velocities higher than 10 ft/s, depending on the tube materials.

Low flow rates in contact condensers will result in poor distribution of the water, permitting uncondensed vapor to bypass to the vent connection, where it will overload the next stage. Velocities lower than 2 ft/s in the shell side and 4 ft/s in the tubes of surface condensers will allow silting and fouling to occur.

If the water flows in series through the first (surface) condenser and then splits to additional condensers, make the flow adjustments for the split flows in an appropriate manner.

Initial operating test

I recommend that this be done routinely on new installations, even if a shop test was conducted and a full set of performance curves is available. The results provide a benchmark for future operating and maintenance work. The testing trains operators and assures that the ejector delivers substantial performance (or contract performance). If a shop test was conducted (as is recommended for critical, custom designs), this field test merely tests the installed configuration for installation problems. If any problem is found, there is time to work on it before the rest of the startup begins. If a deficiency is found in the ejector design, it is best to work immediately with the manufacturer to correct it before the job gets "cold."

Isolate the ejector from the process system, using a blind flange if you are not sure the isolating valve is completely tight. Drain the ejector system and dry it out, and supply the steam and cooling water at their proper conditions. Then measure the pressure at the suction chamber of each stage with no-load and with several air loads to at least 50 percent beyond design air capacity. Note that this test does not confirm the condensable vapor handling capacity. That test is much more difficult to conduct in the field, and is not usually done.

If you plan to turn the ejector on stage by stage for quickest evacuation of the process system, now is the time to extend the test a little so that you can prepare "crossover" curves that show the pressure at which each successive stage should be turned on. Turn off the stages ahead of the first condenser and repeat the air-load test, then turn off the next stage and repeat, and so on. The test points should be spaced closely enough that the crossover pressures can be identified. For example, it might be determined that when the last stage has reduced the system pressure to 300 torr, evacuation will proceed faster if the next-to-last stage is turned on.

The attentive reader may observe that the very fastest evacuation is obtained by arranging that the last stage handle only dry air until the crossover pressure is reached, and that the crossover pressure is slightly different when it is handling wet air. I agree, but I prefer to use the more conservative curves and operate with water in all condensers before turning on any steam. The difference is small.

The air-load test yields much more valuable data than the simpler, common no-load test. The no-load test merely confirms that the ejector system can maintain a low pressure with very small air loads. Its weakness is that it may conceal major defects which are easily revealed by a quick air-load test.

I will briefly describe three examples which I use to justify the air-load tests. My most painful one was a "heartbreaking" extended start-up in which we found that the last-stage configuration could support the previous stages only when the air load was small enough that the aftercondenser could flush it down to the hotwell. In a sequence of tests on another ejector, I discovered that the maximum air-handling capacity had been greatly reduced by an undersized long vent line that had recently been added to the atmospheric vent. The third story is from an associate, who described a situation in which an ejector found to have only 80 percent of design capacity was about to be replaced "before it fails completely," until someone checked and found that it had had exactly that capacity when it was first installed!

Use the test results to confirm the performance curve for the last stage if that curve is available. If the curve is not available, prepare it from the test data, adjusting the data for the cooling-water temperature during the test. Or, run a separate test by admitting a dry air load to the last stage only. The curves for this stage are useful in diagnosing ejector problems, measuring air leaks, and predicting ejector failure.

Performance deficiencies. What should you do if the ejector appears to be defective? If it will not maintain the desired suction pressure with the design air load, then perform your own troubleshooting tests as described later in this chapter. Work with the manufacturer, if needed, to resolve the problem promptly. If the capacity is low at the design suction pressure, or if the steam usage rate calculated from the actual nozzle throat diameters is higher than the value in the purchase contract, then a judgment must be made as to whether the deficiency justifies some action with the manufacturer. Chapter 10 discusses this subject.

I recommend feedback to the manufacturer on your deficiency findings, whether or not you seek a contract adjustment. If you have made a mistake, the manufacturer will help you identify it, and you will avoid being erroneously critical of the manufacturer. If the manufacturer has made a mistake, the feedback is useful and provides the manufacturer with an opportunity to offer an adjustment. If the deficiency seems to fall within the area of "substantial performance," then you have clarified the nature of such contractual agreements, and you will be able to enter into the next contract with a better understanding of what approach you wish to take.

Normal operation

The startup is completed, and you are operating the ejector on a routine basis. The ejector has been demonstrated to work, and now the primary objective is reliable operation with minimum operator attention. It is reasonable to expect rock-solid performance from an ejector in a clean service. Some attention must be given to monitoring the performance and maintaining the system, but the procedures for doing so are routine and readily learnable.

If the ejector has been properly installed, the job of turning it on and off is relatively simple. Begin by closing off all openings through which air would leak, fill the hotwells if necessary, and remove all slip blanks which might have been inserted for troubleshooting or leak testing. If an aftercondenser is present and has a vent test valve which is normally closed, make sure that it has not been left open, because air could get sucked back into the process system if the ejector operation is disrupted.

Steam and cooling-water conditions may need adjustments as described in the initial operating test. If the steam pressure is excessive, throttle it to the proper value. If the steam pressure is controlled by a local pressure regulator, check the pressure and adjust it if necessary. Remember to turn on the steam in the steam jackets, if present. The cooling-water flows may need to be adjusted if the flow-control valve adjustments have been altered or if flow control is to be established by the off/on valve. If a water-flow adjustment must be made each time the ejector is turned on, it may be convenient to record the position of each valve so that the adjustment can be quickly reset by counting turns or otherwise noting the valve position.

The fallacy that ejectors require "tuning." Some people hold the erroneous belief that ejectors are basically temperamental and must be "coaxed" into operating properly. I will try here to uproot that belief, because it may prevent those people from taking the proper actions to make their ejector systems reliable and simple to operate. Because it is hard to give up an idea that has been respected as a guideline, I will speculate on how the erroneous belief may have originated.

I think the belief originated with people who had good intentions and wanted to do a good job, but who were the victims of ignorance about how ejectors work, a lack of proper instruments and ejector performance data, and (sometimes) defective ejectors or installations.

A defective ejector system has one or more defects in the ejector design and installation, has a deficiency in the steam or water supply, or has become worn or fouled. Removing the deficiencies or replacing the ejector is generally appropriate to achieve the high reliability that a user can expect from an ejector.

Shooting in the dark. An operator who is asked to make an ejector work but has no guidelines other than a process vacuum measurement may begin to adjust steam and water valves in a try-it-and-see manner because that is all there is to work with. The time the process system takes to respond to changes may be long enough that any favorable results will strongly encourage the operator to remember the last action and repeat it.

A little extra steam pressure does not usually cause a problem, but low steam pressure will cause the ejector to fail. Too much water will flood a contact condenser, and too little water will cause the ejector to fail. So attention is focused on the water-flow valves. The range of acceptable valve positions will vary with the water pressure and temperature. Different operators may use different procedures for finding the acceptable valve position. The result is that the adjustment valve may have a number of different marks, each indicating a position that worked on a given day. The summer positions will differ from the winter positions if water temperature varies with the seasons.

Mis-sized steam nozzles. I know of a situation in which the steam nozzles were supplied with oversized throats and it was explained that the ejector would perform properly when the steam pressure to each stage was throttled to the (unspecified) pickup pressure. That is an unacceptable complexity for the equipment operators, and even raises questions of ethics. If you should encounter such a situation, then you may consider these options:

1. Do nothing. The extra steam flow will usually result in extra reliability. Capacity may be slightly reduced at the design point, and air capacity may be slightly increased. Extra steam and water will be consumed.
2. Obtain properly sized nozzles to control your operating costs while retaining design specification reliability.
3. Determine the steam pickup pressure for each stage from the manufacturer or by test and adjust the pressure to each stage to equal

it. Add a little extra for the last stage. You may wish to install steam valves designed for throttling service.

Looking for false economies. Some large ejectors use enough steam and cooling water that cutting back on steam or water during off-peak conditions appears attractive.

Avoid the temptation to "tune" an ejector for lowest steam and water consumption during normal operation or even with a known applied air load when isolated from the operating system. One mistake which is sometimes made when the cooling-water supply temperature is somewhat below the design temperature is to adjust the water flow to attain the condenser design water discharge or condensate outlet temperature. Even if the reduced flow should be adequate for condenser operation at the moment, a rise in water temperature in warmer weather will cause the ejector to fail.

If steam is throttled to reduce its usage rate with a given air load, the resulting decrease in suction pressure gives the misleading appearance of improving the operation of the ejector. However, its reliability has been reduced. If one or more stages are on the verge of "breaking" because of the throttling, a momentary upset caused by a brief drop in header pressure or a slug of condensate passing through a nozzle will cause the ejector to fail. The steam pressure must then be raised to resume normal stable operation.

As a result, the belief that tuning is necessary becomes a self-fulfilling prophecy! The tuned ejector is like a trap waiting to be sprung, and must be readjusted after each incident of disruption. It is difficult to justify the high labor cost and operating interruptions that result from this misguided attempt at economy.

The places for economy. Only for large ejectors is it worthwhile to invest the analysis, design, and operating effort to save steam and cooling water during off-peak operating conditions. If you think it may be worth the effort, see Chap. 11 for some suggestions in this area and discuss the possibility with your ejector manufacturer.

For the majority of ejector applications, the most effective time to consider steam and water economy is when the system is being designed and the purchase specifications are being written. The steam and water usage are established primarily by the design load and pressure, the number of stages and intercondensers, the steam and cooling-water conditions, and any financial incentive offered to the manufacturer for reducing the usage of steam and water. The appropriate procedure is to avoid specifying an excessive load at lower than necessary suction pressure, then focus on making the ejector system reliable and easy to operate and troubleshoot. The result will be a balanced design containing the desired compromises between first cost, operating cost, and reliability.

Turning the ejector on. On/off operation is often adequate for small ejector systems: Turn on the cooling water and the steam, wait for the steam supply to clear itself of liquid condensate and create a vacuum, wait for the ejector to clear itself of liquid in low spots and create the desired no-load suction pressure, then open the valve to the process system and observe the system pressure as the ejector evacuates the process system. This operation could be automated.

If the first stage is not stable at no-load, there are several possible reasons. The ejector may not have been designed to be stable to no-load, the first stage may have steam pressure lower than design, the nozzle may be fouled, steam jackets on the diffuser and nozzle may not be working properly, the diffuser may be worn or fouled, the subsequent stage or condenser may not be working properly, or the cooling water may be too warm or its flow may be restricted. Adding a small air load may make the stage stable by moving it away from the no-load condition.

A strong rushing sound will be heard during initial evacuation, quieting down to a low hum when low pressure is reached and all stages are in their stable range.

Where startup speed is important, especially with larger ejector systems which may have low spots where liquid condensate accumulates during shutdowns, much time may be saved by a more detailed procedure in which the ejector is started up stage by stage. First turn on the water to protect surface condensers from overheating. Then turn on the steam to the stages ahead of the first condenser to flush and dry the steam system while you drain liquid from the low spots in the first stages and the piping between the first stage and the isolating valve. After the liquid has been drained and the metal has become warm, turn on the steam to the rest of the stages and close the drain valves. Observe the suction pressure until it pulls down to the normal no-load pressure. Now you are ready to evacuate the process system.

To evacuate the process system in the shortest possible time, starting from atmospheric pressure, turn off all stages except the last stage and open the isolating valve between the ejector and the process system. When the pressure at the ejector first stage reaches the crossover pressure (usually between 200 and 300 torr), turn on the next-to-last stage. When the pressure reaches the next crossover pressure (usually between 75 and 125 torr), turn on all the rest of the stages.

If process considerations permit, you may shorten the evacuation time by using steam to flush air from the system before you start the evacuation.

Improving evacuation speed. If you want to shorten the evacuation time even more, there are several things you may do. First, see whether a large air load is responsible. Measure the air load using one of the methods described in Sec. 8.3, "Testing," and compare it with the design air-handling capacity. A large air load may reduce the excess capacity required for evacuation. Next, consider whether a one- or two-stage hogging ejector in parallel with the ejector or its last stages will deliver the desired performance. It might perform double duty as an emergency spare to the normal service ejector.

Restarting with system under vacuum. If the process system is already under vacuum, it is better to have all the stages operating before the isolating valve is opened. That will minimize the possibility of sucking ejector steam back into the process system. Some backfiring may occur anyway until the system reaches steady state. Opening the isolating valve slowly may reduce the backfiring, but may lengthen the evacuation time.

Poor vacuum. If the pressure in the process system cannot be brought down to the desired design level, begin the troubleshooting procedure to identify the problem.

Monitoring to predict failures. Some vacuum failures can be anticipated and prevented or quickly corrected if the system is monitored properly.

The causes of vacuum failure may be grouped into two broad classes: sudden changes and gradual changes. Sudden changes include such dramatic events as loss of steam pressure or cooling-water pressure, the arrival of a large piece of pipe scale at a steam nozzle, or accidental opening of a drain valve which admits a large air leakage. There is often no advance warning of these sudden changes. However, gradual changes such as increasing air leakage, fouling, and wear or corrosion can be detected by several types of monitoring, and corrective action can be taken to minimize lost production.

If pressure control is by throttling the flow from the process system, then the load to the last stage will indicate the air leakage. It can be estimated by measuring the pressure at the suction of the last stage and referring to the performance curve of the last stage, and by using the garbage-bag test on the aftercondenser vent. If the two estimates agree, then the air leakage is accurately known. If the pressure measurement suggests a larger air leakage, then the stage may have become fouled or worn, or the previous condenser may not be working properly.

In some installations, an absolute-pressure transmitter at the suction of the last stage provides a continuous reading which is recorded on a chart or by a computerized logging system. When a sudden rise in pressure indicates an increase in air leakage, the timing of the increase is often a good indication of which recent action caused the increase in air leakage: switching to a spare pump, operating a leaky valve, withdrawing a sample, or connecting to an adjacent system which has a noticeable air leak.

The position of the pressure control valve will indicate how close the ejector is to being overloaded. If a throttling valve is nearly open or if a supplementary load bleed valve is nearly closed, then overloading may be imminent.

Testing the ejector performance periodically can be made a part of the normal operating procedure. Covered in more detail in Secs. 8.3 and 8.5, "Testing" and "Maintenance," such a test can be conducted during system shutdowns. Or, a carefully planned no-load and design air-load test can be conducted in a few seconds while the ejector is briefly isolated from the process system. In many process systems such a brief test will have a negligible effect on operations.

After these data are compared with operating experience and troubleshooting and maintenance observations, the frequency and type of monitoring can be adjusted to obtain adequate information at minimum cost.

Shutting down. Shutting down the ejector is a simple operation. Close the isolating valve to separate the ejector from the process system, shut off the steam, then shut off the water. If water flow adjustments are required each time the water is turned on, you may prefer to leave the water running during brief shutdowns. In freezing weather, drain everything dry or leave fluids flowing for freeze protection when the process system is shut down.

8.3 Testing

Many users fail to get the best performance out of their ejector systems because they either neglect to test the ejector performance or do some testing with inadequate procedures or instruments. Be assured, however, that if you have an understanding of the basic behavior of single-stage and multistage ejectors, plus the manufacturer's data describing design conditions and performance curves, and some modest field-quality test instruments, you have all you need to get the best performance out of your ejectors by testing to identify performance problems.

The easiest way to learn how to test ejectors is to observe or participate in a test. One good way to start is to observe the test of an ejector on the manufacturer's test stand. If it is an ejector you will be responsible for operating, the benefits increase dramatically. The manufacturer wants to make sure you are satisfied with the performance, and your increased responsibility for its proper operation in your facilities will sharpen your senses. You will enter the field installation/startup/operation activities with the positive attitude that it will work if the proper conditions are met. If you have any problems, you will have the names and telephone numbers of people with whom you have worked for several hours and with whom you can begin any problem solving in a friendly, cooperative manner.

Objectives

The objectives of shop and field tests are usually different. Shop tests are typically done indoors under comfortable ambient conditions. Precision measurements of flows, pressures, and temperatures are possible, and the tests of individual stages or assembled units are intended primarily to confirm that the stages have been properly designed and fabricated. With some care, the test of an assembled unit can approach the design conditions, but a precise duplication of all design conditions, including surface condenser fouling and the condensing of organic vapors, is usually approximate at best. It is more costly to test an assembled unit, especially if it has large contact condensers.

In the field the tests are often done under adverse conditions, such as combinations of darkness, wind, heat or cold, and rain or snow. The ejectors are typically located in elevated structures accessible by stairs, ladders, or elevators. Simplicity, reliability, and convenience are highly desirable characteristics of instruments and procedures. Tests are of the assembled system, and the emphasis is on substantial performance rather than on precision. Field tests are thus used to confirm the general health of the ejector and to guide troubleshooting when the system is not working properly. They are used to answer the questions, is it working? and what must we do to get it working?

Advance warnings of vacuum failures

Some vacuum failures are the result of defects which accumulate steadily over a period of time. There are several methods for predicting some of these failures. Sudden failure caused by loss of steam or cooling water or the appearance of an overwhelmingly large air leak is outside the scope of this book. Slowly developing air leaks or ejector defects can be monitored, and corrective maintenance can be taken in an orderly manner.

Monitoring air leaks is simple if the system pressure control method is not bleeding noncondensable gas to the ejector and if the ejector has a condenser before the last stage. The last-stage suction pressure plus its performance curve will tell you the current noncondensable load. A garbage-bag test will tell you the flow at the aftercondenser or hotwell vent. When the air load exceeds the ejector design air load, the suction pressure will rise rapidly.

Ejector overall condition assessment requires regular testing and inspections for the greatest benefit. Adequate test procedures and records, combined with periodic maintenance inspections, will help you predict when your ejector needs maintenance to prevent failure from fouling, corrosion, and wear. Understanding the importance of the last stage is the starting point. As explained in Chap. 3, if the last stage is not working properly, nothing can make the other stages work properly.

The simplest measure of the safety margin in the last stage is to measure the steam pickup pressure for that stage with the design or near-design air load. If a safety margin exists, the pickup pressure for that stage will be below the design steam pressure. The difference between the steam pickup pressure and the normal steam supply pressure is a measure of the operational safety margin. This measurement should be done when the ejector is first placed in operation, to establish an "as new" reference.

Another measure of the safety margin is the stable range of the last stage. If the last stage deteriorates so that its maximum discharge pressure (MDP or pickup pressure) is less than the actual pressure it sees, the suction pressure will be somewhat above the stable performance curve and may exceed the MDP of the preceding stage. Because the MDP curve generally rises with increasing air load, one symptom of impending failure may be that the last stage becomes unstable below increasingly larger air loads. Its MDP curve is lower than when it was new. No-load and design air-load tests of the total ejector system may not reveal this developing condition.

One appropriate test, therefore, is to apply an air load to find the maximum air load at which the ejector is stable (multistage condensing ejectors), then find the lowest load at which the last stage is stable. The difference between the two air loads is a rough measure of the safety margin. As the lower limit approaches the upper limit, the ejector is approaching total failure.

A third measure of the safety margin is to measure the MDP as described in Chap. 3. That may be awkward to arrange in the field, depending on how the discharge from the last stage is handled. The difference between the MDP and the actual discharge pressure is a measure of the safety margin in the last stage.

Only hints of impending failure may be found by making measurements during normal operation. If the process system pressure is controlled by throttling or bleeding condensable vapor to a condensing ejector, the last stage may be in a stable or unstable condition, depending on the air leak and discharge pressure at the time. Measuring the suction pressure at the last stage is then informative only if you also measure the noncondensable flow in the vent.

Note that if the process-system pressure is controlled by bleeding noncondensable gas, the last stage may always be fully loaded, concealing the deterioration of its MDP curve until the ejector suddenly stops working. Measuring the suction pressure at the last stage will always show it near its design value.

Chapter 6 describes a test to determine whether an ejector will backfire at no-load or air overload.

Routine tests are the best source of advance warnings.

Test procedures, general

Stage testing procedures are described in Chap. 3. In general, the ejector is turned on in the proper manner and allowed to warm up, then various air loads including no-load are applied. For each load condition, the pressures are measured at the suction of one or more stages. Occasionally, it is appropriate to measure the discharge pressure of the last stage.

Troubleshooting symptoms may include noises; steam pressure gauge pulsations resulting from steam moisture slugs passing through the nozzles; the pressure drop across a condenser; the temperature patterns in the ejector stages, suction piping, and condenser shells; the temperature of steam throttled to the atmosphere; liquid levels in condensers or the hotwell; vent gas flow; and condensate flow and composition.

Troubleshooting procedures include isolating portions of the system by valves and/or slip blanks or blind flanges; testing for air leaks using hydrostatic tests, low-pressure air tests, or the use of a sensitive leak detection gas system; and disassembly of stages, condensers, and hotwells looking for visually apparent defects.

Testing standards

Two national testing standards for steam-jet ejectors are applicable in the United States [2, 3]. The Heat Exchange Institute (HEI) standards contain about 60 pages of performance test standards and related vacuum engineering data. The American Society of Mechanical Engineers (ASME/ANSI) standard is 27 pages and applies entirely to performance testing. In both standards the emphasis is on accuracy, calibration, standardization, and contractual performance for test stand conditions. The focus is on individual stage testing in the immediate vicinity of the design point, and "unnecessary testing" is discouraged.

European standards [14, 15, 16] will lead to identical results, for engineering purposes.

Because the simplest field tests usually include no-load pressure measurements, it is desirable to have stage performance curves that include the no-load condition. Most manufacturers' curves do extend from no-load to somewhat beyond the design load.

These standards are a valuable point of departure for planning shop tests and field tests. Shop testing is discussed some in Chap. 10 of this book in connection with procuring ejectors.

Field tests

Field testing typically involves a relaxation of requirements for precision, so that the following uncertainties in common measurements are often acceptable: 10 percent in load, 2 to 5 percent in suction pressure, 2 percent in motive steam pressure, and 1°F for cooling water.

Measuring temperatures. Temperature measurements are used to set the water flow rates in condensers, to estimate the vapor saturation load in condenser vents, and occasionally to check the quality of steam in a throttling calorimeter or equivalent test.

The most demanding of these applications is setting the water flow rate in the first intercondenser when the water temperature rise is 10°F or less. A 1°error at both the inlet and the outlet would create a 20 percent uncertainty in flow measurement! Fortunately, temperature measurement accuracy usually exceeds this requirement if quality thermowells are installed properly. Subsequent intercondenser and aftercondenser flow rates are less critical.

Measuring pressures. Accurate pressure measurement is the most valuable single tool for testing ejectors.

Multistage ejectors will not work if the steam pressure is too low. Test results are incompletely defined if the steam pressure is not known. Ejector loads are measured by using known orifice geometries and measuring the upstream pressure for critical-flow orifices, or upstream and downstream pressures for subcritical flow. The overall compression task for which the ejector is used is measured by observing the absolute pressure at the ejector suction and discharge. The process pressure control system is driven by a measurement of the pressure in the process system.

The HEI standards' treatment of pressure in vacuum systems was summarized in a 1966 article [6]. It predates the development of the capacitance diaphragm gauge and describes primarily shop test methods, but is a useful brief summary.

Steam pressure. At least one accurately calibrated quality steam-pressure gauge should be used for a test. Desirable protective features include a check screw and socket, with a slotted link to avoid damage from a sudden pressure release, and a pigtail to protect the steam gauge from the heat. A good gauge at each stage is best.

Note that the pressure at one stage is not necessarily the same as the pressure at others, because there may be obstructions in the steam piping between them. Because the last stage is the one most adversely affected by poor steam pressure, locate the best gauge near that stage, or near the steam pressure regulator if one is present.

Vacuum pressure. The instruments required for adequate pressure measurement in ejector vacuum systems range from simple to complex, rugged to fragile, and inexpensive to costly. A single-stage ejector maintaining a system pressure of 500 torr may be adequately operated and maintained using a simple pressure/vacuum compound Bourdon tube gauge or a simple mercury manometer reading absolute or differential (gauge) pressure. For troubleshooting the last one or two stages of an ejector, a manometer may be improvised using some mercury, transparent plastic or glass tubing, and a cup or bottle.

At the other extreme, an ejector with four or more stages may be designed to maintain a system pressure of less than 1 torr and require the use of two or more complex gauges which are fragile and/or expensive.

Roper and Ryans [4, 5] give excellent reviews of this subject. Here I am borrowing heavily from their work. Reading their review leads to the conclusion that the one gauge which covers the full range of pressures encountered with vacuum ejectors and yields high accuracy is the rugged capacitance diaphragm gauge (also called the capacitance manometer).

Manufacturers' shop test conditions are such that it is appropriate for them to use an assortment of gauges, including complex electronic devices which are sensitive to the physical properties of the gases surrounding the pressure measurement element. In the field, however, the emphasis is on reliability, simplicity, and low price. I will review here the simple devices and associated procedures that are generally adequate for pressure measurements from atmospheric pressure to about 1 torr.

If you wish to make accurate measurements of pressures below 1 torr, I recommend that you do some reading of Roper and Ryans [4] and the manufacturers' literature. Be prepared to pay appropriately for quality instruments, a quality calibration stand, training and

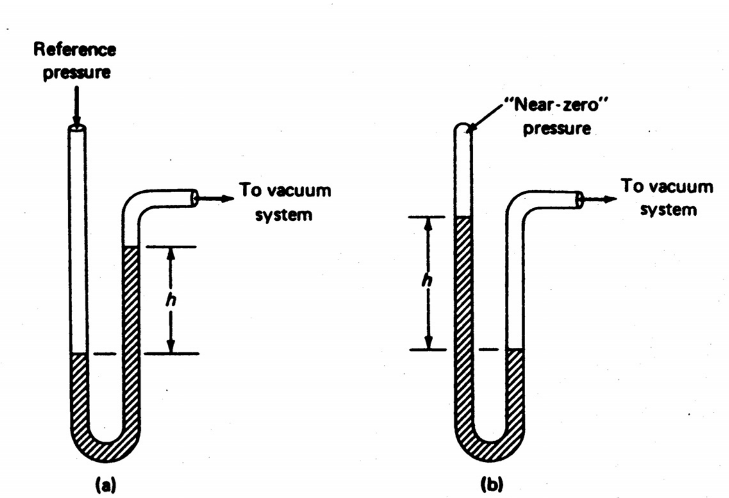


Figure 8.1 Liquid manometer: *(a)* open leg; *(b)* closed leg. (*From J. L. Ryans and D. L. Roper*, Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986*)

learning time for the people who will do the work, and regular recalibration of the transfer standards upon which the accuracy of your measurements will depend.

Liquid manometer. The two common forms of the liquid manometer are shown schematically in Fig. 8.1. The manometer is basically a simple U tube formed from large-bore transparent tubing and partially filled with mercury or low-vapor-pressure oil for ejector work. Water is used for the low differential pressures in ventilating systems. One liquid column ("leg") is exposed to the pressure being measured, and the other is exposed to a reference pressure. The many variations on this basic design include a compact concentric-tube arrangement, a simple closed-leg manometer cushioned inside a test tube for protection, and short and long metal-jacketed commercial models.

The pressure measurement is given as the vertical difference between column heights (h in Fig. 8.1), plus an identification of the liquid, plus a description of the reference pressure. Some examples are:

- Millimeters of mercury absolute (mmHg abs, torr), referenced to "near-zero" pressure in an open end or a presumably clean closed-end manometer
- Inches of water differential
- Inches of mercury gauge, referenced to unspecified barometric pressure

- Inches of mercury vacuum, referenced to unspecified barometric pressure
- Millimeters butyl phthalate absolute

When maximum precision is desired, appropriate corrections must be made for the effects of temperature on the liquid density and for the possible deviation of the local gravity from the standard value. Large-bore tubing and careful cleaning procedures minimize errors resulting from dirt or contamination of the liquid and tubing. With extreme care and micrometer-type instruments, accurate measurements can be made with mercury down to fractions of a torr. These refinements are not justified or practical in most field-test situations.

The practical lower limit for direct visual observation of the difference in liquid levels is about 1 mm, which is 1 torr with liquid mercury or about 0.1 torr with an oil such as butyl phthalate. Cleanliness is the other limitation. The manometer is prepared by carefully cleaning the tube, evacuating it to a pressure well below the range of useful pressure measurements (near-zero), then carefully filling it with clean liquid. Oil manometers should be connected to a near-zero vacuum pump after a test, to determine whether they have become contaminated. The effect of contaminants in the closed leg is to lower the apparent value of the pressure being measured. Thus, a contaminated gauge may indicate a below-zero pressure when connected to the vacuum pump. If the near-zero in the closed leg is no longer acceptable, then the previous test measurements are suspect. Oil manometers may be left connected to a vacuum pump between tests to decontaminate them and keep them clean.

Manometers are fragile gauges and must be handled with care. The glass tubing is fragile, and the liquid may be lost if the device is tipped. A general-purpose manometer with a metal case and adjustable scales to measure liquid levels is a 4-ft-long object which can be awkward to carry to the test site and place in a protected location that is close to the measurement points and easy to read. If gas or liquid is allowed to contaminate the closed leg of an absolute gauge, the reference pressure will not be near-zero, and the gauge must be cleaned and refilled to restore accuracy.

The manometer is an easy gauge to work with because you can see the pressure you are measuring and understand its significance. A pressure measurement of 15 torr will be seen as a 15-mm difference between liquid levels in a mercury-filled closed-leg manometer. A closed-leg unit with a 15-in (400-torr) scale is convenient for most vacuum measurements. I have found it convenient to carry a 300-mm metal scale for reading the manometer. The scale is always clean and easy to read. It is quicker than reading the two manometer scales, adding the two readings, and converting the total to torr if the scales read in inches. With care you can minimize the error that arises from the distance between the scale and the levels you are comparing. It is accurate enough for most troubleshooting at pressures to about 5 torr with mercury and 0.5 torr with oil. If the liquid is one of the low-vapor-pressure oils with a density close to that of water, you have the minor burden of dividing the measurement by about 13.6 to obtain the pressure in mercury units.

An open-leg manometer is typically used to measure the discharge pressure at the last stage. For high-back-pressure installations, an accurate Bourdon gauge may be more appropriate. The gauge pressure must be added to the current (adjusted) barometric pressure to arrive at the absolute pressure. Note that sometimes these pressures must be converted to the desired units: torr, psia, or inches of mercury absolute.

Increasing concern over environmental pollution, workplace safety, and process contamination (especially for food products, pharmaceuticals, and photographic intermediates) has led many companies or plants to forbid the use of mercury manometers. A broken manometer usually results in loss of the entire mercury charge, and mercury could get sucked into the process system. It has the toxic properties typical of heavy metals.

Barometer. When vacuum measurements are made with a manometer open to the atmosphere and the readings are to be converted to absolute pressure, then the current atmospheric pressure should be measured. A barometer is used for this measurement. It is typically a closed-leg manometer with provisions for measuring atmospheric pressure to within a fraction of a millimeter. Because the barometer may be located at a different elevation from the field tests and at a much different temperature, corrections may be appropriate if accuracy is to be retained when measuring pressures below 20 torr. This nuisance and delay is enough to encourage most people to use absolute-pressure gauges for most ejector work.

Bourdon gauge. This common gauge, shown in Fig. 8.2, contains a C-shaped (Bourdon) tube that is surrounded by atmospheric pressure and filled with gas or liquid at the pressure to be measured. If the internal pressure is greater than atmospheric pressure, the Bourdon tube will slightly straighten out, causing the pointer linkage to move the pointer to the scale position which indicates the (gauge) pressure. If the internal pressure is less than atmospheric pressure, the reverse motion occurs. Compound gauges which have scales for both vacuum and pressure are commonly used on ejector systems as simple, low-accuracy indicators of vacuum and pressure. Do not assume that the

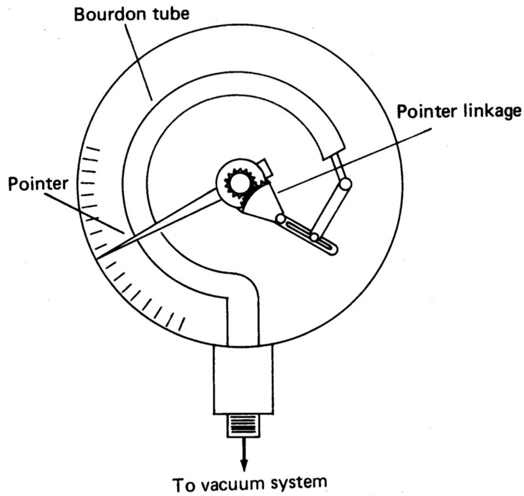


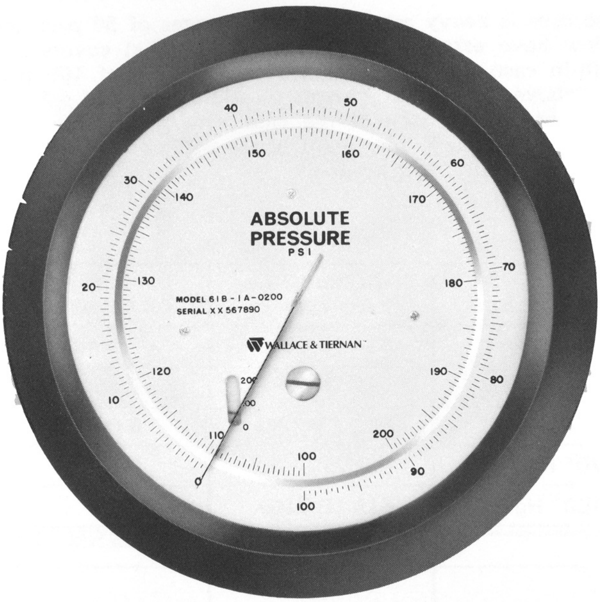
Figure 8.2 Bourdon gauge. (*From J. L. Ryans and D. L. Roper*, Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986*)

vacuum readings are accurate. It is not unusual to see readings that look like 32-inHg vacuum, a physical impossibility.

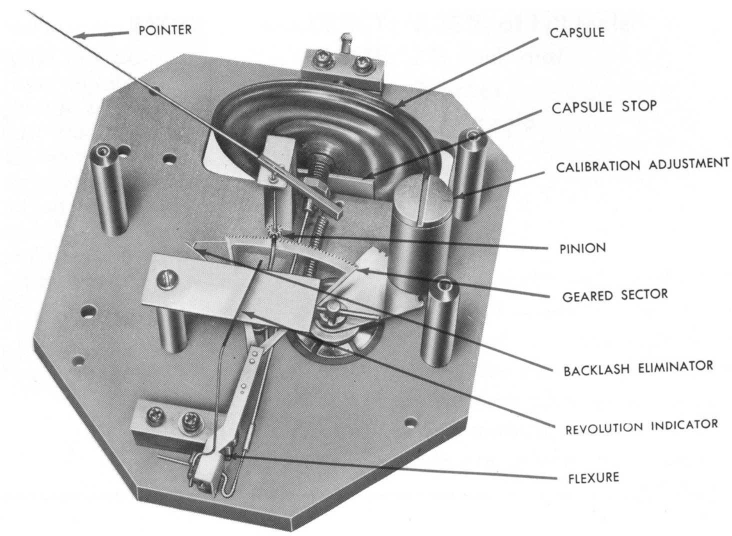
Mechanical diaphragm gauge. The mechanical diaphragm gauge, shown in Fig. 8.3, is a differential pressure gauge in which the pressure sensing element is a thin metal diaphragm. One side of the diaphragm sees process system gas pressure, and the other side sees the near-zero pressure in a sealed evacuated capsule. A high-precision mechanical linkage translates the small motion of the diaphragm into rotation of a long, thin pointer which traverses a finely graduated scale. The gauge is suitable for absolute-pressure readings as low as 0.5 torr, depending on the pressure range for which the specific gauge model was designed. Like the Bourdon tube gauge, this type of gauge is quite portable. The precision scale makes it easy to read the absolute pressure directly.

The benefits are achieved by constructing the gauge of expensive materials, using delicate precision parts, and exposing the moving parts directly to the system whose pressure is being measured. The pressure connection to the case permits the process gases, liquids, and solids to fill the case. A knockout pot, cold trap, or other device may be required to protect the gauge internals from contamination.

With proper care and protection, this gauge can give years of useful service in the field. It must be calibrated against a reference, and it can be used to check the accuracy of locally mounted gauges and transmitters.



(a)



(b)

Figure 8.3 Mechanical diaphragm gauge: *(a)* front view; *(b)* gauge internals. *(Courtesy Wallace & Tiernan, Inc.)*

Many variations of this absolute-pressure gauge exist, some using a Bourdon-type sensing element. Several manufacturers should be contacted to find the type best matching your needs.

McLeod gauge. When the gas pressure is less than 1 torr, measuring the force it exerts on a measurement device becomes very difficult. The McLeod gauge, shown in Fig. 8.4, is a manometer which uses a different approach. It is based on the ideal gas law relationship between the pressure and volume of a gas at constant temperature. Briefly, a known volume of gas at the unknown pressure is captured and compressed to a much smaller measured volume at the same temperature. The pressure of the gas increases in proportion to the ratio of the volumes, becoming high enough that it can be measured accurately. The measured pressure is then divided by the volume ratio to yield the pressure at which the gas was captured.

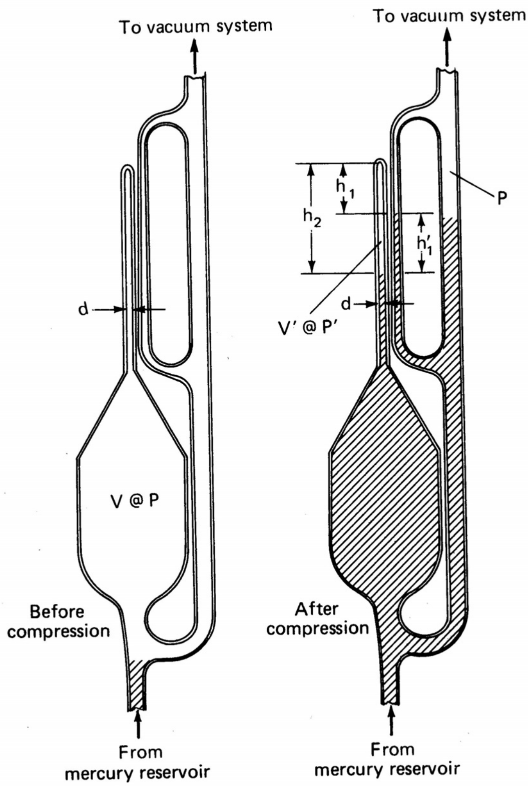


Figure 8.4 McLeod gauge, basic operation. (*From J. L. Ryans and D. L. Roper*, Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986*)

For example, if a sample is taken and compressed to 1/1000 of the sample volume at the same temperature, its pressure will increase by a factor of 1000. If that pressure is measured as 15 torr, then the original pressure must have been 0.015 torr (or 1.5E-3 torr in scientific notation).

An important characteristic of the gauge which limits its usefulness and leads to occasional errors in measurement is that any condensable vapors in the gas sample may partially condense as the pressure is raised. Then the final volume will be smaller, and the reading will yield an erroneously low pressure. Accordingly, the gauge must be protected by a cold trap if condensable vapors are present. One strategy to be used with some variations of the basic design is to perform the test twice, using two different compression ratios. If the results are the same, then no condensation occurred and the results are accurate.

The gauge is typically made of glass, and mercury is the working liquid. The action of trapping a gas sample and compressing it is accomplished by causing mercury to flow upward in the gauge, either by rotating the whole gauge or by the action of a piston which forces the mercury upward from a reservoir. The entire operation yields one reading, giving no indication of the pressure changes which might have occurred since the sample was taken. It is a common practice, accordingly, to observe the pressure with a less accurate, continuously indicating electronic gauge, and operate the McLeod gauge only when the pressure is steady, or at least in a range of interest.

One example of a rugged, portable McLeod gauge is shown in Fig. 8.5. It rests on its self-contained pedestal and is connected to the vacuum system by a flexible tube. To obtain a pressure reading, you pick it up by the handle, rotate it 90° until the tubes and scales are vertical, wait a few seconds, then read the pressure directly from the position of the mercury column against its scale. Several pressure-range options are available. The 0-50-torr range scale has good resolution to about 0.1 torr, and a 0-5000-μm (5-torr) model has a dual scale with useful resolution from 1 μm to 5 torr. Options include a hose clamp or ball valve for protective isolation and a renewable chemical desiccant trap to exclude condensable vapors. The device weighs from 7 to 17 lb, depending on the model.

Capacitance diaphragm (manometer) gauge. The instrument of choice among engineers who want accurate vacuum-pressure measurements in the field is the capacitance diaphragm gauge, also called a capacitance manometer. Its operating principle is shown in Fig. 8.6. It is a variation on the mechanical diaphragm gauge. Here the mechanical linkage and pointer are replaced by extremely sensitive

Figure 8.5 McLeod gauge, rugged portable unit.
(Courtesy Stokes Vacuum, Inc.)

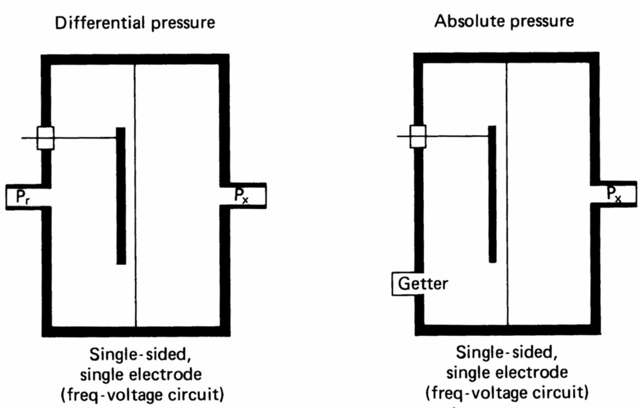


Figure 8.6 Capacitance diaphragm (manometer) pressure sensor.
(Courtesy MKS Instruments, Inc.)

electronic components. The pressure to be measured is applied to one side of a very thin mechanical diaphragm, and the other side of the diaphragm sees near-zero in a sealed evacuated capsule. A metal plate is located near the diaphragm, and an electronic circuit measures the changes in capacitance between the diaphragm and the plate caused by microscopic motion of the diaphragm in response to changes in the measured pressure. The changes in capacitance are interpreted by calibrated circuits and translated into electronic signals, which may be displayed as pressure readings or delivered to an electronic system for automatic process control and data logging.

High accuracy is maintained over a wide range of pressures. For example, a gauge rated at 0-1000 torr has a resolution of 0.1 torr (0.01 percent of full scale) and an accuracy of 0.5 percent of reading! Other range options are available for lower pressures. Because the sensitive device is affected by temperature changes, the readings must be adjusted for ambient temperature changes. If your pressure measurement system is subjected to large changes in ambient temperature, you may choose the option of having this measurement and adjustment done automatically. A constant-temperature heated sensor is an option available to reduce the effect of varying ambient temperatures and to prevent corrosion and fouling from condensation of process vapors on the delicate sensing elements. If fouling does occur, the device is easily cleaned.

Figure 8.7 Portable capacitance diaphragm gauge. (*Courtesy MKS Instruments, Inc.*)

A fully portable battery-operated capacitance diaphragm gauge system is shown in Fig. 8.7. The heated sensor option is not available on portable systems. Usually a calculated correction based upon the ambient temperature and the specified temperature coefficient will yield acceptable accuracy.

Force-balance pressure transmitter. This transmitter, shown in Fig. 8.8, is another variation of the mechanical gauges which use a diaphragm, Bourdon tube, or other device as the pressure-sensing element. When the process pressure displaces the diaphragm (or other type of element), that motion is transmitted through a force bar to an external motion detector. The motion detector controls a restoring force which restores the diaphragm to its undisplaced position. It also generates a pneumatic or electrical signal which represents the measured pressure. The signal then may be used to display the pressure, control it, or record it.

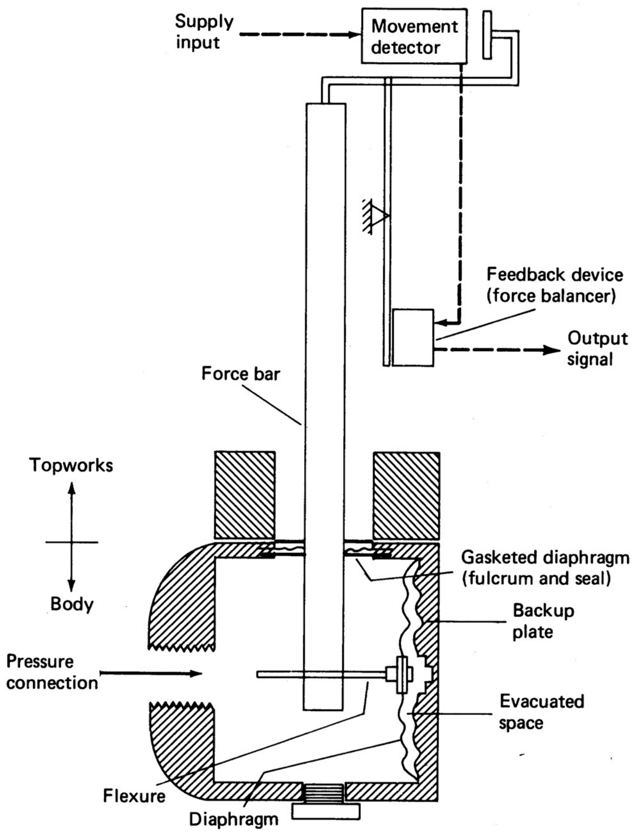


Figure 8.8 Force-balance transmitter. (*From J. L. Ryans and D. L. Roper*, Process Vacuum System Design and Operation, *McGraw-Hill, New York, 1986*)

TABLE 8.1 Comparative Advantages of Vacuum Gauges

Advantage	Liquid manometer	McLeod	Bourdon tube	Mechanical diaphragm	Capacitance manometer	Force-balance	Thermocouple, Pirani
Readings possible with condensables present	Yes	No	Yes	Yes	Yes	Yes	Yes
Measurement is composition-independent	Yes	Yes	Yes	Yes	Yes	Yes	No
Internals isolated from process gases	No	No	Yes	No	Yes	Yes	Yes
Pressure easily read from a remote location	No	No	No	No	Yes	Yes	Yes
Portable	Yes	Yes	Yes	Yes	No	No	No
Readings independent of ambient conditions	Yes*	Yes	No	Yes	Yes†	Yes	Yes†
Rugged construction	No	No	Yes	No	Yes	Yes	Yes

*Open-ended designs are pressure-sensitive.
†Temperature-sensitive. Many models are heated or readings are temperature-compensated.
SOURCE: J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.

Still used in older plants, this device gives reliable performance and acceptable measurement accuracy down to about 1 torr. Linearity is good, and hysteresis and dead-band effects are minimal.

Other pressure gauges and transmitters. Here I will briefly review several other pressure measurement devices.

Another method of measuring pressure is to measure a gas physical property which is related to pressure in a known manner. Thermal conductivity gauges employ a heated wire element whose heat loss is a measure of gas conductivity. If the composition of the gas mixture is known, the pressure may be determined. The thermocouple gauge and the Pirani gauge are two examples of this application. The output is an electrical signal, for display or control.

The oil-filled capacitive transmitter is a modification of the capacitance diaphragm gauge. An isolating diaphragm is placed between the sensing diaphragm and the process, and the space between the two diaphragms is filled with silicon oil. Protection from corrosion and physical damage is gained at some loss in sensitivity. The useful range is down to 1 torr.

Selecting vacuum gauges and transmitters. For additional guidelines in selecting the proper gauges and transmitters for measuring pressures in vacuum, see Tables 8.1, 8.2, and 8.3 and Fig. 8.9. Again, this useful information is borrowed from Roper and Ryans [4, 5].

Avoiding vacuum-pressure measurement errors. Although measuring pressure in a vacuum system seems simple enough ("Just connect the gauge and read it!"), several things can go wrong. Even a simple mercury manometer reading can be in error if there is a leak in the measurement system or if the closed leg is contaminated. Condensable vapors may be present in the gases sampled by a McLeod gauge.

TABLE 8.2 Cost and Accuracy of Vacuum Gauges

	Manometers	Mechanical gauges	Variable-capacitance electronic gauges
Reading error, %*	1-10	2-20	0.2-2.0
Operating temperature limits, °F	40-120	-50-400	-20-300
Retention of calibration	Excellent†	Fair to good	Very good
Resistance to shock and vibration	Poor	Fair to good	Good to very good
Cost, U.S. dollars ‡	50-1000	50-2000	700-2500

*Expressed as percent of reading.

†The liquid manometer and the McLeod gauge are primary calibration standards. Retention of calibration is excellent because the calibration of the instrument can be calculated from first principles.

‡Costs are as of June 1988. Costs for variable-capacitance electronic gauges are based on absolute-pressure gauges. Prices quoted do not include a readout or recorder.

SOURCE: D.L. Roper and J.L. Ryans, "Select the Right Vacuum Gauge," *Chemical Engineering*, March 1989, pp. 125-144. Courtesy McGraw-Hill.

TABLE 8.3 Characteristics of Transmitter Types Used to Control Vacuum Systems

	Force-balance	Oil-filled capacitive	Capacitance manometer	Resonant wire	Piezo resistive
Range, torr	0-1520	0-1397	0-1000	0-1400	0-780
Minimum span, torr	0-10	0-50	0-10	0-9.5	0-10
Maximum span, torr	0-1520	0-1397	0-1000	0-1400	0-780
Accuracy*	±0.5% to ±5% span	±0.25% span	±0.5% reading	±0.3% span	±0.1 % to ±0.5% span
Operating temperature limits, °F	-20-180	-40-220	-20-120	-40-100	-40-200
Temperature effects†	Total: ±1 % span/ 100°F at maximum span; ±3% span/ 100°F at minimum span	Total: ±1 % span/ 100°F at maximum span; ±3.5% span/ 100°F at minimum span	Zero shift: ±0.56% full scale/100°F; Span shift: ±2.2% reading/100°F	Total: ±1 % span/ 100°F at maximum span; ±7% span/ 100°F, at minimum span	Total: ±0.50%span/ 100°F at maximum span; ±2.2% span/100°F at minimum span
Resistance to shock and vibration	Fair to good	Good	Good	Good	Excellent
Retention of calibration	Fair	Good	Very good	Fair	Very good
Weight, lb	25-35	12	6	11	10
Overpressure limit, psi	85-135	2000	5-20	135	15
Cost, U.S. dollars‡	1700	1100	1000	1100	1700

*All accuracies are zero-based and include the combined effects oflinearity, hysteresis, and repeatability.
†Temperature effects change with changes in span, suppression of zero, and a number of other factors that are unique to the transmitter.
‡Costs listed are list price before discounts. Cost for piezo-resistive transmitter does not include field communicator.
SOURCE: D. L. Roper and J. L. Ryans. "Select the Right Vacuum Gauge," *Chemical Engineering*, March 1989, pp. 125-144. Courtesy McGraw-Hill.

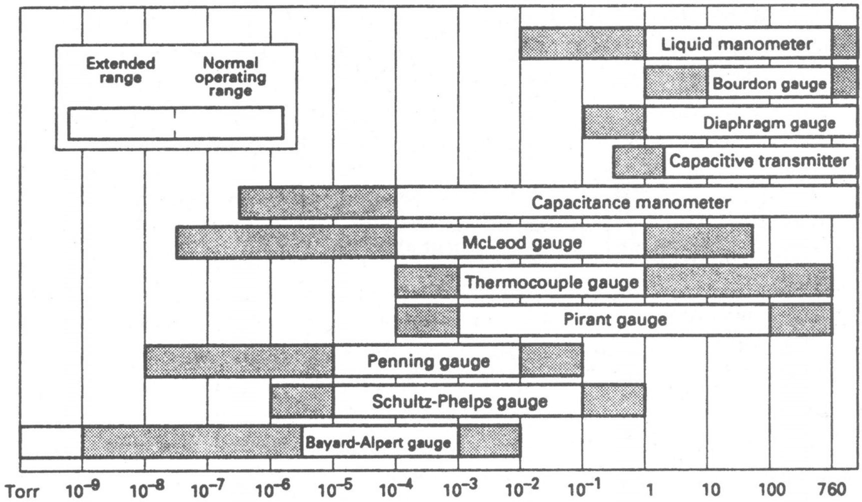


Figure 8.9 Pressure ranges covered by vacuum gauges. (*From D. L. Roper and J. L. Ryans, "Select the Right Vacuum Gauge,"* Chemical Engineering, *March 1989, pp. 125-144, Courtesy McGraw-Hill.*)

Mechanical gauges may be off calibration or damaged by fouling or corrosion. Finally, even a good gauge will not yield the proper pressure reading if it is connected to the wrong location!

Measure pressure at the proper location. Ejector-stage performance curves are based on suction pressure being measured at the ejector suction chamber. Ideally the pressure is measured at a position approximately 90° from the inlet nozzle, and at the top or side if the stage is horizontal, as shown in Fig. 8.10*a*. Sometimes it is not convenient to measure there. Some inexpensive designs do not have pressure taps, sometimes access is blocked by nearby equipment or structures, and sometimes the only available connection is underneath the suction chamber on a horizontal stage. Any liquid at that low spot will interfere with pressure measurements and may damage or foul the gauge.

If you must measure at another location, think carefully about anything in the arrangement which might cause the pressure there to be different (usually higher) from that in the suction chamber. One such example is shown in Fig. 8.10*c*, which shows the pressure being measured directly opposite a small opening through which a gas load is being introduced. That is not the best way to introduce the load; the ejector performs best when the flow enters through the suction connection with a uniform velocity profile. A second source of error was that the gas jet split around the steam nozzle and rejoined to have an impact on the pressure opening, creating an erroneously high-pressure reading.

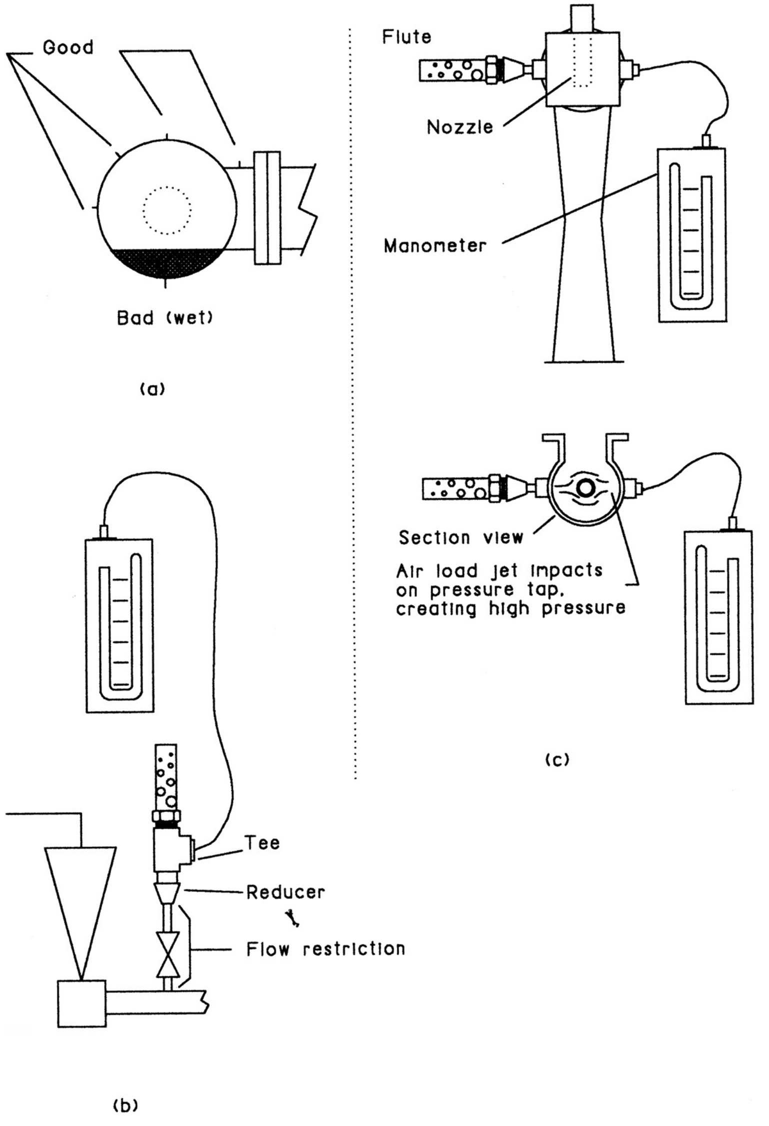


Figure 8.10 Good and bad locations for pressure measurements: (*a*) horizontal stage, good and bad locations; (*b*) in a small-diameter line with flow restriction, bad; (*c*) facing a jet of gas, bad.

If the piping between the gauge and the ejector stage is narrow and if any flow exists in it, then the pressure drop may introduce a large error. Figure 8.10*b* shows one such error which was diagnosed during a telephone consultation. The ejector no-load pressure was proper, but the introduction of an air load soon caused the measured pressure to rise in direct proportion to the air load. A plot of the data showed the linear shape of the curve, and an inquiry confirmed that the load gas was flowing through the narrow connection to the process system. In effect, the setup created a crude critical-flow meter. The solution was to use another location for the pressure measurement.

Check for gauge leaks. Leaks are always present in a pressure-measurement system. This results in the gauge pressure always being a little higher than the actual pressure at the location being measured. Sometimes the leaks produce unacceptably high errors. Always arrange for a leak test in a field test situation. It requires only the addition of a small test valve and a few seconds of time for the test.

Early in my ejector training, my mentor, Bob Hagen, told me about a field service trip which revealed to an embarrassed client that a leak in the hose to his vacuum gauge explained the apparent problem. I will tell you a story from my own experience to emphasize the need to check for leaks.

One day I received a call to help start up two ejectors. I left my office and carried my jet test briefcase into the nearby plant. My client was a Ph.D. chemist wearing jeans and enjoying his "hands-on" experience in the startup of his unit. He gently kidded me about being a "dude," wearing a coat and tie in the plant environment. I told him I used to get dirty testing ejectors, but had developed tools and procedures that did not require my getting dirty very often. We proceeded up the ladder, and he pointed to the gauge on the first three-stage ejector. I inserted a leak test valve where the gauge hose connected to the ejector and explained that routinely my first step is to check for leaks. I opened the test valve for a few seconds to establish a vacuum in the gauge, then closed the valve to watch for a leak. The mercury column moved up almost as fast as the red flush in his face. He saw instantly what the problem was and got some vacuum grease to help fix it. In a few minutes the leak was fixed and both ejectors tested OK. I repacked my briefcase and showed him my hands: "See, clean hands!" We both laughed.

Figure 8.11*a* shows a vacuum testing arrangement and identifies several common leakage locations. Although test standards and guides to vacuum practice recommend using quality large-bore vacuum hose, it is very common to use ¼-in plastic tubing in the field because of its availability. Because all test configurations should be tested routinely for leaks, there is not much penalty for using the smaller tubing. If you do not detect a significant pressure rise in several seconds, and if the pressure drops quickly when you open the test valve, then the pressure measurement system is adequately tight.

A typical leak test valve is shown in Fig. 8.11*b*. I recommend installing it with the flow arrow pointing away from the vacuum.

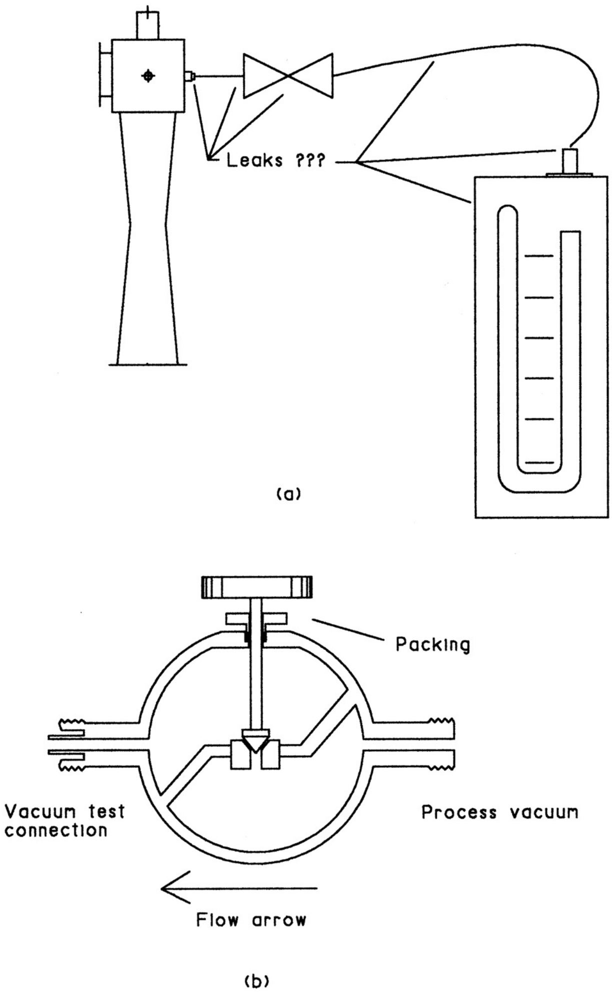


Figure 8.11 Vacuum gauge system leak detection: (*a*) leakage locations; (*b*) leak detection valve.

Then the valve will be tightly sealed against leaks in normal operation if the valves are left installed for convenience in future tests. Of course, that places the valve packing inside the gauge system being tested for leaks, but you are already alerted to detect and fix leaks, so it takes only a few seconds to detect and minimize any leak there.

Cold traps and liquid traps. Liquids and condensable vapors interfere with gauge operation and may damage the gauge. The cold trap shown in Fig. 8.12 is a two-stage trap which will remove liquids and condense some of the condensable vapor near the gauge. Note

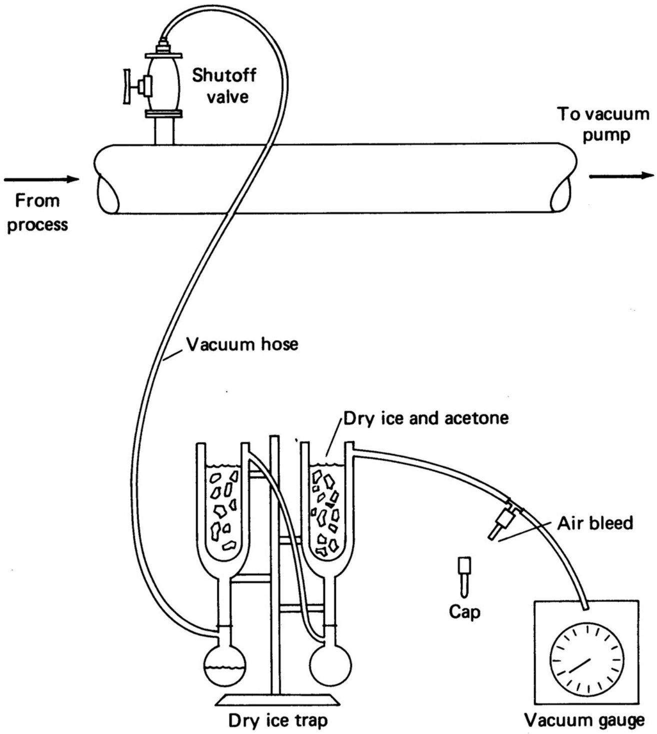


Figure 8.12 Cold trap. *(From J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986)*

especially the use of a tiny air bleed which sweeps condensable vapors toward the vacuum system between measurements, ensuring that the gauge will see only air during the measurement. Of course, the process vapors will move toward the gauge by diffusion and pressure changes when the bleed orifice is capped.

Sometimes an uncooled trap is placed ahead of a gauge to intercept slugs of water pumped toward the gauge by a backfiring stage. Because the pulsating flow contains a spray with the slugs, some moisture will appear in the gauge unless the trap is carefully designed and not allowed to fill up.

Calibrating vacuum gauges. An uncalibrated vacuum gauge can be very costly. An unwary person may assume that the measured results are accurate, then operate the process system at an undesirable condition or spend money and time trying to solve a nonexistent problem. Some minimal calibration capability is essential to create confidence in vacuum pressure measurements.

Figure 8.13 shows the elements of a basic calibration system. Key

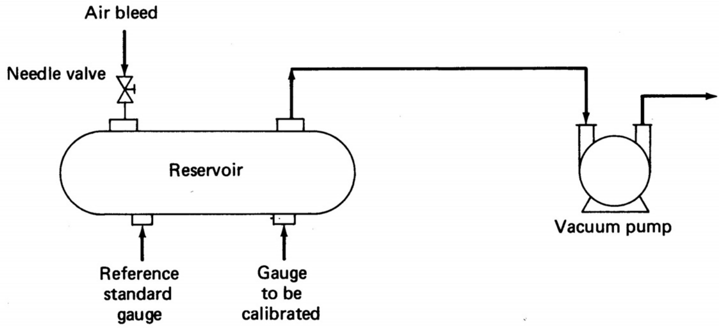


Figure 8.13 Vacuum gauge calibration arrangement. *(From J. L. Ryans and D. L. Roper,* Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986)*

elements are the vacuum pump and the reference standard gauge. For rough vacuum gauges the pump may be a small one-stage or two-stage oil-filled rotary piston pump which will maintain a pressure well below 1 torr. Typically, it will have a gas ballast feature to permit it to cleanse itself of modest contamination by condensable vapors. The reference gauge may be a McLeod gauge or a reliable transfer standard such as a capacitance diaphragm gauge. In practice, the gauges are connected to the reservoir and the air bleed is adjusted until the reference gauge indicates a steady reading at the desired pressure. Then the gauge to be calibrated is adjusted to obtain that reading. If a McLeod gauge is the reference, it may speed the test work to use a continuous-reading electronic gauge as an indication of the approximate pressure and to establish that the pressure is stable before using the McLeod gauge to verify the pressure.

One warning about Fig. 8.13: It should be regarded as a top view, not a side view! Never connect a precision pressure measurement device to the bottom of a reservoir or pipe; there may be liquid or dirt present. The reservoir may be simply a large-diameter header with appropriate valves and fittings to connect the pump, gauges, and air bleed. A very large reservoir will lengthen the time to stabilize the pressure after each pressure change or gauge change.

For detailed information about vacuum gauge calibration and vacuum practice in the rough vacuum and lower-pressure ranges, refer to documents such as References 4, 7, 8, and 9. They give enough additional references for you to build up a library on the subject. Manufacturers of precision gauges offer standard and custom-designed calibration systems. Some are portable enough for field calibrations.

Useful test kit hardware. I have found it desirable to keep a small satchel stocked with a collection of items useful for testing ejectors. My first field test of an ejector was in the winter, and I recall the feel-ing when a small pipe plug dropped out of my cold fingers and rattled down through the structure, where it was lost in the snow. I resolved then to keep extra small objects on hand for future tests. The Appendix has a list of recommended contents of a test kit. That may be useful to you in planning tests. I will mention the usefulness of some items here and elsewhere.

The small items are expendable and often can be left at the test site. On the other hand, it is a nuisance to locate replacements, so I attempt to replace them when we are at the ground-level shops after the tests. When the job is successful, the plant is in a generous mood and will often contribute some more useful items for the test kit. Thus, the kit becomes a "pool" into which each job takes and puts as needed.

I have already mentioned the metal scale graduated in millimeters and the air leak test valve. I carry several ¼-in inlet needle valves, pipe bushings to adapt them to larger-size pressure taps, and adapters to the test tubing. Teflon tape is superior to pipe dope because you get unnecessarily dirty using pipe dope and dirty hands smudge the test data sheets. A 10-in adjustable end wrench and 10-in slip-joint pliers are handy for making test connections and removing inspection pipe plugs. A multiple-orifice critical-flow air meter (flute or piccolo) with 1-in NPT male connection and a reducer to ½-in NPT is essential for easy and accurate introduction of test air loads.

For recording the results, a notebook is very handy. It forces the data into chronological sequence, which is useful when you forget to enter the date and time with your observations. For this reason, data forms are extremely helpful. They prompt you to enter data you might otherwise forget. Blank forms for test data, performance curves, and inspection results are all useful; sample forms are provided in the Appendix.

Thin metal plates sized for 1-in through 4-in flanges are useful in doing no-load tests on small stages and performing low-pressure leak tests. A flashlight, an inspection mirror, and a foot of utility wire are useful for inspecting steam nozzles and removing solid chunks which may be blocking the steam flow. A plastic garbage bag or several feet of 6-in-diameter thin-film polyethylene tubing knotted at one end is useful for vent gas flow measurements.

Test clothing. Wear natural fiber clothing when testing or working with steam-jet ejectors. Synthetic materials may melt and stick to your skin if you accidentally touch hot metal surfaces. Vigorous dancing has resulted from neglect of this rule.

Loading the ejector with air and steam. Adding a known air load to the field test of an ejector is a simple way to bring the usefulness of a field test close to that of a shop test. Previously in this chapter I described the several benefits. Now we look at how it is done.

The simplest, and in some ways the neatest, technique is available if the manufacturer has added (as at least one manufacturer has) a critical-flow air orifice to the first stage of an ejector, locating it in the side of the suction chamber or in a tap on the side of the suction connection. The orifice is sized to meter an air load equal to or slightly below the design load, and is equipped with a protective sealing cap. To perform a quick test while the ejector is operating, attach a vacuum gauge, close the isolation valve to obtain a no-load pressure, then remove the orifice cap to obtain an air-load pressure. If the two pressures substantially match the manufacturer's performance curves, the ejector is healthy. If not, more testing is indicated at the next convenient opportunity.

This is neat, but some precautions are required. The jet of air entering the system will produce an erroneous pressure reading if it impinges on the pressure measurement location. The orifice must be inspected at each use to make sure it has not been obstructed by fouling or enlarged by corrosion. The orifice cap should have a chain to prevent its being lost while it is removed. And finally, the test would yield a better prediction of failure if pressure measurements were also taken at the last stage. That is where most ejector troubles start.

Metering air with the multiple-orifice unit (flute, piccolo). This is one of my favorite tools. I invented it, although someone else preceded me. I devised it as a short section of ½-in stainless steel pipe with a series of graduated-size, rounded-entrance orifices. It had an NPT male thread on one end to fit into the opening that was typically made available by removing the pressure/vacuum gauge near the first stage of most of the ejectors I worked with. The other end of the pipe was capped, and the unused orifices were covered with tape strips. Hole combinations were untaped to create the desired air load. The device could be carried in my pocket and was equal in accuracy to bulky test manifolds I had seen in shop tests. It was far more useful than the limited-range rotameters I had previously used, and it worked in any position. When I learned that these devices had been marketed for years, I decided to buy them instead of making them.

Figure 8.14 shows a typical configuration. It is made of a copper alloy with a 1-in NPT male threaded outlet and a series of stainless steel precision orifices sized 0.5, 1, 2,..., to 64 pph dry air from standard atmospheric conditions into a vacuum below one-half atmospheric pressure. Instructions with the unit include corrections for nonstandard atmospheric pressure and temperature. Such corrections are seldom needed in the field, especially when doing troubleshooting.

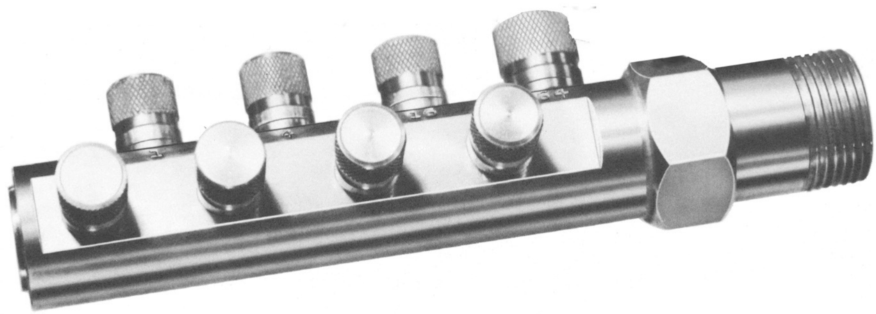


Figure 8.14 Multiple orifice critical-flow meter (flute, piccolo). *(Courtesy J. D. Pedersen)*

Each orifice has a tight-sealing threaded cap. By opening the appropriate combination of orifices, any load between 0.5 and 127.5 pph can be created, in increments of 0.5 pph. Other sizes are available, and variations may be made to order. The Appendix tells how to size and shape critical-flow orifices, but it is probably not cost-effective to make your own multiple-orifice units.

I ordered a variation of the basic design, omitting the screwed caps as items which get lost. The orifice size sequence I prefer is 1, 2, 3, 5, 10, 20, 30, 50, and 50. The binary/decimal sequence has numbers that are easier to think with. I had a □-in NPT pressure tap added at the closed end of the unit. That permits you to measure pressure there to make corrections for subcritical flow, if necessary. This is for test circumstances in which there may be flow restrictions in the piping from the device to the ejector which might result in subcritical flow through the orifices. Even if the path to the ejector suction connection is short and unobstructed, this is still the best place to measure pressure when correcting for subcritical flow, when the pressure inside the device is greater than 400 torr.

I am aware of a larger opening at that end being used to introduce condensable vapor (steam) for mixed-load (air plus steam) tests. Another way to meter steam to an ejector as a test load is to throttle steam into one end of an open-ended can which is placed loosely over the flute. The flute then becomes a critical-flow orifice metering superheated atmospheric-pressure steam into the vacuum. Two flutes, one covered with a steam can, could be used for a mixed-load test in the field. Seldom is there a need for a condensable-load test in the field.

Introducing steam as a load. Steam may be introduced as a metered or unmetered load when testing ejectors. Unmetered steam may be introduced at an intermediate ejector stage to load that stage without appearing as a load to any subsequent stages which follow a condenser.

For example, you may wish to perform a field test to determine the maximum discharge pressure of the first stage at different air loads and steam pressures. One field test setup that may not require rearranging the ejector components is to inject steam through a throttling valve to the suction of the second stage. By adjusting the throttling valve, you can load the second stage to adjust the first-stage discharge pressure while you observe the suction pressure at both the first and second stages.

For loading an ejector with a metered steam flow, the use of critical-flow orifices is routine. The Appendix tells you the orifice size and shape and offers some guidelines for the test configuration and procedure. See the HEI [2] or ASME [3] standards for more comprehensive guidelines. Here I will emphasize that the steam must be dry, the upstream pressure must be accurately known, the temperature must be known if the steam is superheated, the downstream pressure must be less than half the upstream absolute pressure, and the upstream piping diameter should be at least two or three times the orifice diameter. Because this test is costly, I strongly recommend reading one of the standards mentioned above and reviewing the test plan with an experienced person to assure success.

Steam condenser performance tests. Although this topic is outside the scope of this book, I found a reference that may be useful to those readers who are responsible for maintaining and testing large power steam condensers. Freneau [10] describes procedures for determining the true pressure drop in main condensers. You refer to the ejector first- and last-stage performance curves while measuring pressures at several locations in the system when the system is operating. I have not used the procedure, but it looks useful.

Measuring process vapor and steam flows. Occasionally, it is desirable to know the mass flow rate of motive steam and condensable vapor in an ejector system. Motive steam flows are easy to estimate accurately if you know the nozzle throat diameters and the steam pressure. The condensable-vapor flow is most easily measured by intercepting liquid condensate at the hotwell and measuring it by weight or volume. If the materials present are not regarded as toxic or hostile to the environment, it may be a simple task to monitor the overflow from a hotwell or to intercept the flow from one or more drainlegs on the ejector being tested.

If the motive steam flow is known with confidence and a reliable estimate can be made of the water vapor leaving a condenser vent, then the steam condensate flow rate will be known with confidence. Of course, it is necessary to know the approximate flow of noncondensable gases in the vent before you can make the vapor-liquid equilibrium calculations. If the steam condensate flow is known, it is only necessary to analyze a sample of that condensate stream to determine the mass flow of condensed process vapor which accompanies it. The test can be repeated with subsequent condensers until sufficient data are collected. If the condensable liquid is not soluble in water, the sample can be observed visually for an on-the-spot estimate of the relative volumes and mass flow rates of water and condensable liquid.

If the ejector load contains significant amounts of water vapor, then it is necessary to intercept and measure the flow volume or mass. Diverting the flow into a calibrated container for a measured time interval may be a practical way to get a sample, as shown in Fig. 8.15.

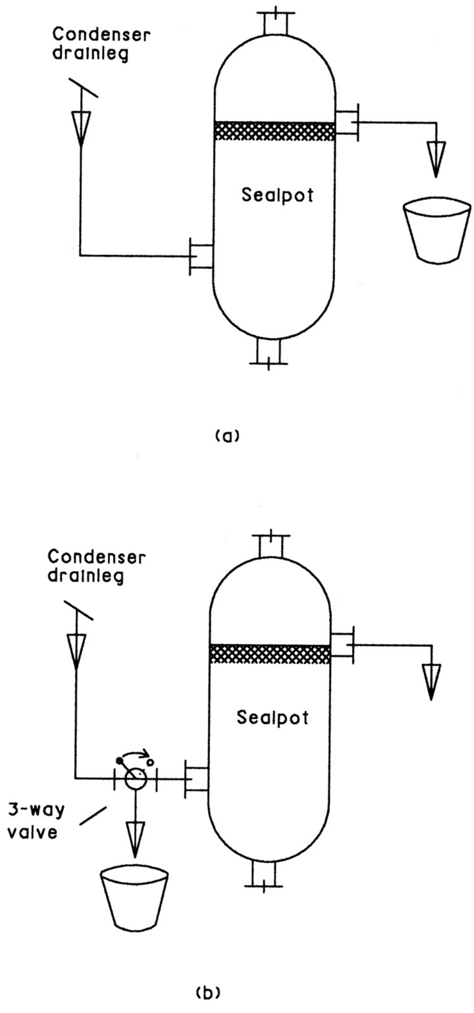


Figure 8.15 Measuring condensate flow: (*a*) sealpot overflow; (*b*) diverting drainleg flow.

Recording test data

Record all the test data you obtain. Enter the date and time frequently. Use the blank forms in the Appendix to record your test data and inspection findings. Plot the test data on blank performance curve forms or on copies of the manufacturer's curves as you go along. The forms will prompt you for data you might otherwise forget to observe or record, and the curves will help you interpret the results and determine their significance as the test proceeds. A good test rule is never to erase original data! If you make a mistake, neatly line through the error, leaving it readable, then enter the correct data alongside. If you are not quite sure, mark it with a question mark. Truth is what you are looking for, and often the lined-out data prove valuable. After the test is over, you may copy the data onto clean forms for neatness, but do not throw the dirty form away until the job is over. You may have made some errors in copying.

Do not be intimidated by the form. It is only an aid to help you do the best job. Add observations in the margins or in the spaces for data, and record your thinking and speculations and questions. If you are very organized, you may keep notes in a notebook about what you are doing and what data types you are collecting. Because time and space are limited at many outdoor field test sites, and sometimes the weather is bad, I often fill out several test data sheets in sequence with data and lots of notes. I try to plot at least the most important load/pressure data as I go along.

At the close of the job, look again at the forms while the job is fresh in your mind and consider whether some changes might make the form more useful to you in future tests. If so, a few minutes now to revise the form will save you time in the rush of future tests.

8.4 Troubleshooting

"The vacuum system/ejector is not working right; fix it!" This is sometimes the first inkling some people are given that such a thing as an ejector even exists. Although most people have some idea of what a pump looks like and how it works – their home may have a sump pump and pumps in the dishwasher and clothes washer, and their car has a few pumps – very few people are prepared by domestic experiences for working with ejectors. Nothing moves in them! The working principles do not reveal themselves readily from a visual inspection of a disassembled ejector. There is help, however. If you have read all or the recommended portions of this book, you are prepared to begin troubleshooting ejector problems with the best of them. Please try to retain a sense of humor and be tolerant, because you will make some mistakes as you put your new ejector knowledge into practice. You have a lot of company, and help is available both inside and outside this book. You can do it.

Troubleshooting is the fun-and-frustration topic that people discuss most when they get together and talk about ejectors. The many stories that are told have an interesting mix of lore and science, simple and complex causes of problems, and the "aha!" moments when the cause of a perplexing ejector problem is finally understood. It is an opportunity to show off their skills and innovative ability as well as to confess their mistakes.

In reading what I have just written, I realize my memory is weak in recalling my own mistakes.

Most field problems have a simple explanation, usually failure to observe one or two of the many guidelines offered in the instructions for installing and using ejectors. Accordingly, it makes sense to reread your instructions if your ejector is not working right. This book is a consolidation of such guidelines from many sources, and is therefore more likely to describe the defect in your system.

Even if you have the necessary guidelines at hand to help you avoid the problem, there is a practical limit to how much you should invest in preventing problems. At some point it is more cost-effective to "get on with it" and try to operate the ejector, then identify and correct any remaining problems.

Poor steam is the single most common cause of ejector problems. Low pressure, high moisture content, scale and sediment, and fouling impurities are major offenders. Looking first at the steam supply adequacy is not a bad policy, but the scope of potential problems is large enough that a more powerful strategy is needed.

The role of a troubleshooter

Everyone who troubleshoots ejectors has a different role-even the same person on different occasions. You may have complete responsibility for operating, testing, troubleshooting, and maintaining ejectors in your area and may do it all yourself. Or, you may be an ejector specialist within a unit, plant, or company and work with many people in many locations. I will offer suggestions to the specialist who is working with a brand new problem in an organization and location new to his or her experience, because it is the most demanding role. Most readers will have a simpler role, and if that is true for you, I ask you to forgive me for the extra detail and read quickly over it.

Selecting a troubleshooting strategy

Two diverse approaches are offered to troubleshooters: the checklist and the very generalized strategy. I recommend the generalized strategy; it will take a little longer to learn and to get started, but it is more flexible and positive, and uses data more effectively.

A checklist is attractive because it is a handy way to start "right now" to solve the problem. Often it works fine. Using it, you follow a detailed test and inspection procedure, perhaps written for your specific equipment. The sequence of steps reflects the writer's (usually the manufacturer's) judgment as to what the most common problem areas are. Often a checklist is not available for the specific equipment. If a user has many identical installations, it may be useful to develop such a checklist tailored to the specific equipment. The manufacturer may prepare one or help you prepare your own.

A deficiency of the checklist is its brevity: "Do this; if..., do that." The brevity minimizes the user's learning of why the actions are taken. Also, the checklist must be rewritten for each different installation. The user is likely to reach a dead end if an instruction is not understood or if an unforeseen circumstance arises.

I recommend, instead, adopting a generalized, broad-front strategy which uses a mixture of personal experience and knowledge of ejectors in general and the system in particular, considers all of the symptoms available, and offers a "last ditch" stage-by-stage meticulous approach. Although it takes a little more time to get started with such a strategy, it does develop generalized skills for all jets and other equipment. It encourages a judgment-directed strategy, playing the odds to pursue the course of action that seems to offer the highest probability of benefit for the cost and time involved. No one procedure is always the most cost-effective. It encourages managed risk taking and can involve managers in the strategy sessions for big problems.

Generalized, broad-front strategy

The method involves judgments arising from knowledge of three related subjects: problem symptoms, circumstantial evidence, and failure modes. These are defined below.

1. Problem symptoms are the data which have alerted someone to the apparent existence of a problem, and which have thereby prompted the troubleshooting process. Typical data are high vacuum-system pressure, water contamination of product, noise, visible leaks, out-of-range pressures or temperatures. You must also notice whether faulty operation is steady, cycling, or intermittent.
2. Circumstantial evidence is that portion of the mass of available information which has potential value in solving the problem at hand. It includes descriptions of recent changes, the history of this equipment, user logs, and manufacturers' field service reports.

3. Failure modes are the patterns of failure peculiar to ejectors and vacuum systems. When one portion of a system fails, it often causes other portions to fail. For example, a piece of scale obstructing the steam nozzle of the last stage will cause the suction pressure there to rise, which causes the preceding stage suction pressure to rise,..., which causes the process system pressure to rise.

   Knowledge of the different ways ejectors can fail and the resultant failure modes leads to familiarity with the corresponding patterns of symptoms. With that knowledge you will be prepared to look for and recognize informative patterns in the data at hand, and plan tests and inspections to obtain more data. The Appendix has a description of several common failure modes and corresponding symptoms. This information should be learned in advance of troubleshooting to minimize guessing and false moves during a crisis.

When you start troubleshooting as an individual or on a team, be sure to start with a definition of the problem assignment from the manager involved to understand the objectives and how the manager ranks the priorities of time, cost, quality, and reliability. There is nothing more empty than solving the wrong problem.

Next, interview the people familiar with the situation and otherwise gather data about problem symptoms and relevant circumstances. Then hypothesize a list of probable failure causes and test them against the symptoms. Rank the probable failure causes according to your judgment, based on the circumstances and your estimate of the cost and time to check out each theory. Start with the most promising and get the information to check it out. Depending on the needs of the job, more than one action may be started: order new parts rush, test, schedule a shutdown, rig up a temporary fix, etc.

As you search for the problem(s), remember that there may be more than one. One of my horror stories involves a trouble-prone ejector application that had about ten distinct problems over a period from the bid review process through the first year of operation. Three of the problems involved the manufacturer, and the rest were caused by the user. The troubleshooting job is not finished until a test convinces you that the ejector is OK, and the manager to whom you report your findings is satisfied.

You will have an even greater responsibility if your role is that of a specialist or consultant. If you see an opportunity to prevent or minimize future problems in the area, you will want to give advice or recommendations to the people who can best use them. Following up by communicating this in writing has the greatest impact. It is harder to overlook and easier to implement.

Walk-through inspection. A quick walk-through orientation tour of the ejector installation is appropriate after the manager has defined the problem-solving task. It will prepare you for the sit-down interview with the people who will give you the many details. After that meeting, you will probably have a preliminary plan for a detailed walk-through inspection of the installation, followed by a series of tests and inspections of the ejector system.

In your detailed walk-through inspection, look for obvious installation errors or highly informative symptoms: pressures, temperatures, sounds, etc. Plugged steam nozzles will allow stages to remain cool, and warm condenser walls will indicate low water flows or poor water distribution. The orientation from this inspection will be useful if a more detailed study is necessary. Put a thermometer in the discharge from the last stage at no-load (makeshift throttling calorimeter): 240°F or higher is OK, 220°F is too low. Create or mark up a schematic flow diagram to show the important details in the process and ejector system. Include all valves, gauges, and pressure tap openings.

Detailed study. Begin a more detailed study by collecting all operating data that may be gotten quickly without disturbing the operation of the system. Interview people; measure pressures, temperatures, and flows. Especially important are the last-stage suction pressure and the volume of the aftercondenser vent flow. Interview the operators and examine the records to see whether relevant changes may have occurred in pressures, flows, temperatures, quality, and cleanliness. Has any equipment maintenance been done recently?

Inventory and examine the design data on hand: instruction book, purchase specifications, performance curves, and manufacturers' specifications for minimum steam pressure, maximum superheat, maximum water supply temperature, and design conditions for each stage and condenser.

Keep a notebook and record observations as you make them, especially of visual inspection of disassembled parts. Color photographs may be useful to supplement the written descriptions of fouling deposits. Be alert to any sign of a flow obstruction: poorly centered gaskets, fouling, and scale.

Divide and conquer. Assuming that your only problem symptom is "poor vacuum," your first task is to determine whether the problem(s) are with the ejector or in the process system. Often, operating people will point to the ejector as the culprit simply because they have not identified the problem as being in the process system. It is convenient to attribute the problem to the remaining unknown area, and ejectors often represent a big unknown. You and I can help change that!

You have already made your detailed survey and have measured all the pressures, temperatures, and flows without changing the system operation – right? If not, do so now and come back to this spot. Now we are going to do things that will affect the process operation. You have obtained permission and advised them – right? Here we go.

Our first step is to determine whether the problem is in the ejector or the process system. Close the isolation valve at the ejector inlet or insert a blind flange there and measure the suction pressure at the first and last stages. Check the temperature of the inlet line, listen for sounds of backfiring, and identify any other unusual process conditions. If the ejector looks OK (no-load pressures agree with curves, or are about 20 percent of design suction pressures), add an air load of perhaps 80 percent of design and see whether the substantial-load pressures are OK.

If the ejector looks OK, the problem appears to be in the process system. The problem in a system which has operated properly for some time is usually an air leak or excessive condensable vapor.

Air-leakage measurement. An air leak may be indicated by the garbage-bag test or another measurement of the aftercondenser vent flow, or by the suction pressure of the last stage before the tests began. The air-load test you conducted after isolating the ejector will have confirmed the adequacy of the last-stage curve as an airflow meter.

If the air leak is well below the design air load of the ejector, then air leakage is not a problem unless product contamination by air is a concern. If the airflow is known to exceed the ejector design capacity, then a search for the air leaks should begin promptly. If an airflow measurement was not made during the detailed inspection, you may wish to take it now. Remember that it will take some time for the airflow at the last stage to reach steady-state. If that is not convenient, you may estimate the air leakage into the process system by the pressure rise method.

Pressure rise (vacuum drop). In a dry vacuum system at ambient temperature and a pressure below 400 torr, critical flow will exist in the air-leakage paths into the system. The pressure will rise in direct proportion to the air-leakage rate and inversely with the system volume. Solving for the air-leakage rate in that ideal system when the volume is known,

$$W_a = 0.006 \text{ (rise) } (V) \tag{8.1}$$

where W_a = air leakage, pph
rise = measured rate of pressure rise, torr/min
V = system volume, ft^3

For example, a leak test of a 200-ft^3 system shows a pressure rise of 50 torr in 15 min. The air leakage is

$$W_a = 0.006\left(\frac{50}{15}\right)(200) = 4 \text{ pph}$$

This method has some limitations. If the system is warmer or colder than ambient, the calculated leakage will be correspondingly high or low. If the system contains condensable vapors which condense as the pressure is raised, that will "buffer" the pressure and reduce the apparent air leakage. If the pressure rises above 400 torr, subcritical flow will exist, and the equation is not applicable. Finally, the size of the system is often not known, requiring a two-step test.

To perform the two-step test, first measure the pressure rise rate with the unknown air leakage. Then repeat the test, but with the addition of a known air leakage. The known air leakage might equal the ejector design air load.

Then,

$$W_a = \frac{W_t}{(rise_{t+a} / rise_a) - 1} \tag{8.2}$$

where W_t = test air load, pph

rise_a = rise with unknown air leakage, torr/min

rise_{t+a} = rise with test + unknown air leakage, torr/min

In the previous example, assume that V is not known. Adding a 10 pph test air load, we observe a rise of 120 torr in 10 min.

$$\text{rise}_{t+a} = \frac{120}{10} = 12 \text{ torr/min}$$

$$\text{rise}_a = \frac{50}{15} = 3.33 \text{ torr/min}$$

$$\frac{\text{rise}_{t+a}}{\text{rise}_a} = \frac{12}{3.33} = 3.6$$

$$\text{W}_a = \frac{10}{3.6 - 1} = 3.8 \text{ pph}$$

which agrees closely with the previous calculation.

We now have enough data to estimate the system volume, to save us time in future tests of this system.

$$V = \frac{168\, W_a}{\text{rise}_a} \tag{8.3}$$

$$V = 168\, \frac{3.8}{3.33} = 192 \text{ ft}^3$$

which is close enough.

Air leak source detection. When tests indicate that an excessive air leak exists, start searching to find it. Leaks sometimes take a lot of searching to locate. Only very large leaks are noticeably audible at the source. A critical-flow orifice sized for 10 pph of atmospheric air is only about 0.1 in diameter. Below 20 pph the leakage path is small and often obscured from direct acoustic observation. Background noise levels may mask the sound from even large leaks.

Some simple, low-technology leak detection methods include placing foam (from a shaving foam dispenser) on suspected leak areas and watching it get sucked into leaks; hydrostatic testing and watching for water leaks; 5-psig air tests in which suspected leak areas are painted with soap solution to generate bubbles at leaks; and even the use of a candle, whose flame will be sucked toward a large leak. These take time, or require a shutdown, or may contaminate the system, or may create a fire or explosion hazard. If the equipment and piping are covered with insulation, this complicates the search for leaks. Access for inspection is limited, and the water leaking during a hydrostatic test may travel some distance underneath the insulation before it becomes visible. Occasionally, a fluorescent dye is added to the water to make small leaks more visible. Isolated leaks that small will not generally be a problem in a rough vacuum system. Note that equipment with a flexible inner liner such as rubber may leak air under a vacuum, but not leak with internal pressure. The flexible material forms an inward-oriented check valve at the leak site.

Make sure the system has been designed for the water weight for a hydrostatic test, and do not exceed the pressure capability of the weakest element in the system.

The operating circumstances immediately preceding the appearance of the leak may indicate which valves, pumps, flanged joints, etc., are the first places to look. If the last-stage suction pressure is continuously monitored because the control method permits it to measure the noncondensable gas flow, that record will show the precise time and approximate magnitude of the increase in air leakage. Consulting the operating log or questioning the operators may reveal events which indicate where to look.

Inert gas leak detectors may be used to detect leaks without shutting down the process system. A noncondensable tracer gas which is inert to the process system and alien to the area, helium for example, is discharged toward likely leak sources while the ejector vent flow is monitored by a device which can detect minute quantities of the trace gas. Taylor [11] briefly describes such a procedure, using a cold trap and mass spectrometer to monitor the gas stream for helium. Other gases may also be used, but they should be selected carefully to avoid interference from gases in the plant environment. I recall the story an engineer told on himself about his witnessing a shop leak test of equipment intended for a critical service. Something was interfering with the readings, creating the false appearance of a leak, until the rest of the group suddenly turned to him and directed him to extinguish his cigar or leave the test area. The cigar smoke was the culprit.

Finding the ejector problems. You have isolated the ejector from the process system and you do not have the manufacturer's curves at hand? Soon I will give you some approximate no-load pressures to use as guides until you get curves or create your own after the ejector works properly. They may tell you where to start looking for problems.

Various troubleshooting checklists and charts have been prepared. They are a starting guide for you until you develop experience in your own plant situation. One such chart [12] ranks the probable sources of the problem, showing a shift in rankings depending on the ejector and condenser types. The general pattern is clear, however: Poor steam is the major culprit, pressure measurement is a contender, and high discharge pressure follows. Air leaks usually come next, followed by the infrequent and obscure causes of failure.

An effective strategy, therefore, is to give your first attention to these potential problem sources before you shift to a deliberate stage-by-stage diagnosis.

No-load suction pressures. Typical no-load suction pressures if the steam and geometry are OK:

Z stage, 50-75 torr

Y stage, 5-10 torr

X stage, 1-3 torr

The normal no-load pressures will tend to be higher if the ejector has two intercondensers, if the last stage is designed for a high discharge pressure, or if motive steam pressure is below 75 psig.

Steam quality. First check the steam pressure with a reliable pressure gauge, protected by a pigtail if possible. It should be equal to or above the design steam pressure specified by the manufacturer, often marked on the nameplate or stamped on the stages (look at the flanges).

Next check the steam quality, the moisture content. A simple test is to crack a steam valve to the air and observe the appearance of the steam plume. If the plume of steam is transparent for some distance before it becomes a white cloud, the moisture content is acceptably low. If the plume is white as it emerges, moisture is present, and that may be a problem. If a sturdy thermometer is available, immerse it in the discharge from the last stage at no-load, or otherwise improvise a throttling calorimeter. An exhaust temperature of above 240°F is best; 220°F is too low [13].

If the steam is wet, check for inoperative steam traps. Make sure the steam piping comes from the top or side of the steam header and that the piping to the ejector is insulated. A separator may be necessary to deliver dry steam.

The last shall be first. Here we are, looking at a sick ejector and wondering where to start. The steam supply looks OK, the vacuum pressure gauge seems OK, and the discharge pressure either is unknown or tests OK. Now it is time for a step-by-step methodical approach, starting with the last stage.

Why start there? I will explain that soon, but first I wish to dispel a misconception. Experience with other equipment such as air compressors and multistage pumps might misdirect us toward wanting to start with the first stage. Someone accustomed to reading process flow diagrams might also be inclined to start with the first stage encountered by the load from the process system.

I recall a telephone call from a plant engineer who was troubleshooting a three-stage ejector. He had been working on it for several days and reported his most recent test: running the first stage all by itself! He noted that it pulled down to perhaps 600+ torr, and wanted to know how to make it work better. I asked him what he would have done if it had worked fine by itself. Would he have needed to turn on the last two stages? We both laughed and got down to business. I tell this not to put him down, but to emphasize that even an experienced engineer needs to understand how ejectors work to get the best performance out of them.

To finish the story, I suggested beginning with the last stage to get it working properly: It did not do much better than the first stage had done, and I suggested that the steam nozzle might be partially obstructed. He doubted that, because they had just added a strainer to the steam line. I speculated that the steam line alterations might have loosened some pipe scale which then lodged in the nozzle, and that a flashlight inspection through the inspection plug might reveal it. He called back 30 minutes later with the ejector working fine. They fished out the scale with some wire.

Now to explain why we start with the last stage. In Chap. 3 we learned that ejector stages designed for a compression greater than 2:1 (most ejectors) are basically designed to pump from a given suction pressure and discharge to a given discharge pressure. If the design discharge pressure (MDP) is exceeded, then the ejector suction pressure will increase greatly. In turn, that effect will cascade forward, causing all the preceding stages to operate poorly. It is sort of like building a house of cards, layer upon layer. Failure at one level (stage) causes a failure in all layers (stages) resting upon it.

Troubleshooting individual stages. Whenever the suction pressure of a stage matches its performance curve at no-load and at the design air load, then it and the subsequent stages are apparently OK, and the next step is to test the preceding stage. If not, continue as described below.

Place a flute in the suction line to the first stage and shut off all stages except the last.

Connect a vacuum/pressure Bourdon gauge to the last stage until stable pressure readings indicate that it is safe to attach a more accurate, sensitive gauge.

First test the no-load suction pressure. Because the last stage must always discharge against atmospheric pressure, even at no-load, it requires more testing than the other stages. If the suction pressure is higher than 50-75 torr or is erratic, the stage may not be stable at small loads. You may find that raising the steam pressure 10 percent will stabilize the ejector. If so, the stage may not have been designed to be stable at small loads, or the stage is worn or fouled, or the discharge pressure is too high.

Return the steam pressure to its design value and open some air orifices to add the design air load to the ejector. For troubleshooting the last stage, neglect the small amount of water vapor that may be added to the load as it flows through the condensers. If the ejector is stable and the proper suction pressure is reached, then you know the discharge pressure is OK and the stage is simply not stable to no-load. Vary the air load to find the value below which the ejector is not stable. The difference between the design load and the lower stable limit represents the margin of safety in the stage.

If the last stage is not stable with any air load, or only when the steam pressure is raised, then you should still suspect steam blockage, wear, or fouling, or high discharge pressure. Consider the installation and its records to judge which one to try first. Sometimes it is simple to remove a plug in the steam chest for a visual inspection of the nozzle, or you may loosen four bolts and allow the stage to discharge directly to the atmosphere for the lowest possible discharge pressure. If the nozzle is blocked, clean it. If the discharge pressure seems excessive, look for the source of unusual back pressure and fix it if you can. If a pressure measurement shows the discharge pressure to be within the ejector specification, then remove the stage and inspect the nozzle and diffuser for fouling, wear, and corrosion as described in Sec. 8.5, "Maintenance."

Replace defective parts as appropriate. If you replace a nozzle, replace the diffuser also, unless a careful inspection shows it to have its original dimensions. Sometimes a nozzle and diffuser will enlarge together as they wear. The extra steam passed by the enlarged nozzle compensates for the diffuser enlargement. The new nozzle will not pass enough steam to "fill" the enlarged diffuser, and the stage will fail to work properly.

If the stage is stable, but is not able to maintain the required pressures at no-load and design air load, then an air leak or steam leak at the steam nozzle should be suspected. A garbage-bag test of the vent on an aftercondenser or hotwell may indicate whether a significant leak is present. Or, you may insert a slip blank in the line to the stage suction and repeat the test. If the pressure is still too high, remove the stage for inspection, looking for evidence of a leaking steam nozzle, as described in Sec. 8.5, "Maintenance." If an air leak is detected, you may use the leak-detection procedures described previously. You may wish to move the test slip blank forward stage by stage until the air leak source is localized somewhat.

If the suction pressure is acceptable at no-load, but is higher than the design value with the design air load, this indicates that the preceding condenser may not be working properly. Check the vent air flow or last-stage load to make sure there is no air leak in the ejector system. If the condenser vent temperature is too high, the air load will carry with it an excessive load of water vapor. If that seems to be the problem, check the condenser as described below.

Test the other stages in a similar manner, with three differences: The discharge pressure decreases with load because of the "lap" between the MDP curve for a stage and the suction curve of the next stage, at lower pressures the water vapor content of the load becomes significant, and at suction pressures below 5 torr ice may form on unheated steam nozzles and diffusers. The effect of a decrease in the MDP of the other stages is to reduce the maximum air-handling capacity, which will be evident in the airload tests. Another important difference is that the other stages have larger diffuser throats, making them less sensitive to small amounts of wear or fouling.

Call for help. If you have reached this point without finding the problem, it is time to call the manufacturer for help. Most problems are solved by one or two calls. Assemble your test data and all of the manufacturer's data that describe the ejector, including the serial number and/or job number. If this is a startup problem, begin with the assumption that there is probably something wrong with the installation or your observations, and possibly something wrong with the ejector. You can expect to talk to a knowledgeable ejector specialist who is sympathetic to your needs.

After taking a few minutes to orient the specialist to your situation, including the extent of your personal experience in working with ejectors, you will jointly "walk through" your design information and test data. Typically, you will develop a list of additional tests and inspections to conduct before calling back. Do call back, even if you get the ejector working fine. The courtesy will brighten the specialist's day and make for a better future relationship.

If you call back with the ejector still not working, the two of you may develop additional tests or inspections for you to perform. Or, the urgency of the situation and the complexity of the problem may indicate that the ejector specialist should come to your site to help get the problem solved quickly. Expect to pay for the specialist's time and expenses unless the ejector is under warranty and the manufacturer has clearly erred in the design or fabrication.

Inspecting condensers. You will probably inspect the condensers during your detailed walk-through inspection, or as a result of suspicions raised by tests which suggest a large flow of condensable vapor from the vent to the next stage.

Make sure the cooling-water supply temperature is adequate and that the proper flow exists. Feel contact condenser bodies for hot spots indicating poor water distribution, look in the top opening (at atmospheric pressure!) to observe the water distribution pattern if water enters at the side. The vent connection should be cool, about 5 to 15°F above the water inlet temperature. A spray condenser will have uniformly cool walls, starting a little below the spray nozzle and warming near the bottom. Disk and doughnut or side-to-side tray distributors will be cool where the water curtains wet the wall and warmer where the curtain is away from the wall. Tap gently with a hammer (not if the condenser is ceramic!) on the side to detect flooding (dull sound) or empty (ringing sound).

If you can measure the temperature of the vent flow, compare it with the design temperatures. For contact condensers and those surface condensers with special vent gas cooling arrangements the vent temperature may be from 5 to 15°F above the cooling-water inlet temperature.

To detect tube leaks in surface condensers, turn off all steam and leave water pressure on the condensers. If a large tube leak is present, water will continue to flow out of the hotwell.

Flooding in a condenser can be detected by a gauge glass improvised from clear flexible tubing between the condenser drain and vent locations. Symptoms of flooding include a pulsating suction pressure in the next stage, water in the discharge from the next stage (the last-stage discharge might be visible), and residue in the suction connection and suction chamber of the next stage.

Loop seals between condensers may be plugged or blown open. Or a trap may have failed in the drainleg from an aftercondenser to an intercondenser (I recommend against this configuration).

If you suspect that the condenser is being overloaded with condensable vapor, the condensable vapor flow may be measured by intercepting the flow from the first and second condensers at the hotwell and analyzing it as described previously.

Odds and ends. Here are a few ideas and suggestions to close out this topic.

First, be careful not to apply full steam pressure to low-pressure equipment. In the process of modifying the piping by adding slip blanks and closing isolating valves, it is easy to isolate a system so that high-pressure motive steam may be applied to stages, condensers, sealpots, or gauges that were not designed to withstand the pressure.

If raising the steam pressure improves the performance of a multistage condensing ejector, check for possible overload by noncondensable vapor, marginal condenser water temperature or flow, or marginal MDP capability of the last stage.

On low-level condensing units, wrong condensate level in the condensate sump may flood the condenser or starve (cavitate) the condensate removal pump. The condensate pump must be the low NPSHR type: slow speed and low head.

8.5 Maintenance

Different maintenance strategies for handling ejector problems include correction, prevention, and avoidance. The default choice is to run the ejector until something fails, then correct the problem with a minimum repair or a complete inspection and repair. In a preventive program the ejector is tested and inspected periodically, with the time intervals between the activities extended as experience indicates it is proper to do so. By problem avoidance, I refer to the expensive practice of replacing the entire system every few years on an arbitrary schedule. I once heard of such a practice, and I have no more details. It simply adds hearsay support to my conviction that many ejector users are suffering economically from a lack of information about how to get the most out of their ejectors.

Data files and record keeping

Depending on the organization within a plant, useful ejector data may be found in files kept by engineering, production, and maintenance. All files should be available for people doing troubleshooting and maintenance, and records of the activities should be kept in a file that is accessible at any time. File control is a problem, because one careless misplacement of a file may lose years of valuable data and result in costly delays of future work. One solution is to permit only copies to leave the file area.

Complete file information is extremely helpful, but unfortunately rare. Useful information to keep in the file, not necessarily in order of importance, includes the following: process system item number; plant property number; purchase order number and purchase specifications; and manufacturer's job number and serial number, performance specs, performance curves and stage-by-stage design conditions, drawings, nozzle and diffuser throat dimensions, nozzle positions, parts list, and utilities usages.

Periodic performance tests

Many ejectors in clean services run for years without trouble. For them, a simple, quick performance test is adequate to spot developing problems. I recommend testing a new ejector every 6 months, then adjusting the time between tests based on your test and inspection findings. A minimum routine test for a multistage condensing ejector involves measurement of the suction pressures at the first and last stages while varying the test air load from no-load, to minimum stable load for the last stage, to maximum air-handling load. This test should require only a few minutes, and many process systems will be relatively unaffected by removing the ejector for a few minutes.

If fouling of condensers or process piping is known or suspected, then pressure drops may be measured occasionally to detect the extent of fouling.

Spring is inspection season

I recommend disassembling and testing each ejector at least once a year or at each major shutdown/turnaround, adjusting the schedule based on your inspection and test findings. Spring is a good time for inspecting ejectors, before surface cooling water becomes warm.

As can be seen in Figs. 8.16 and 8.17, there are many parts to keep track of in a simple two-stage condensing ejector. It is easy to mix up or misplace parts and reassemble the ejector improperly. Note the stage numbers on nozzles, spacers, and diffusers. Replace old gaskets with new ones of the same thickness, being especially alert to avoid altering the distance between the nozzle and the diffuser.

Identify the type, amount, and location of fouling, wear, and corrosion. Disassemble and inspect stages, condensers, and interconnecting piping. Look for gaskets that are improperly centered or that have

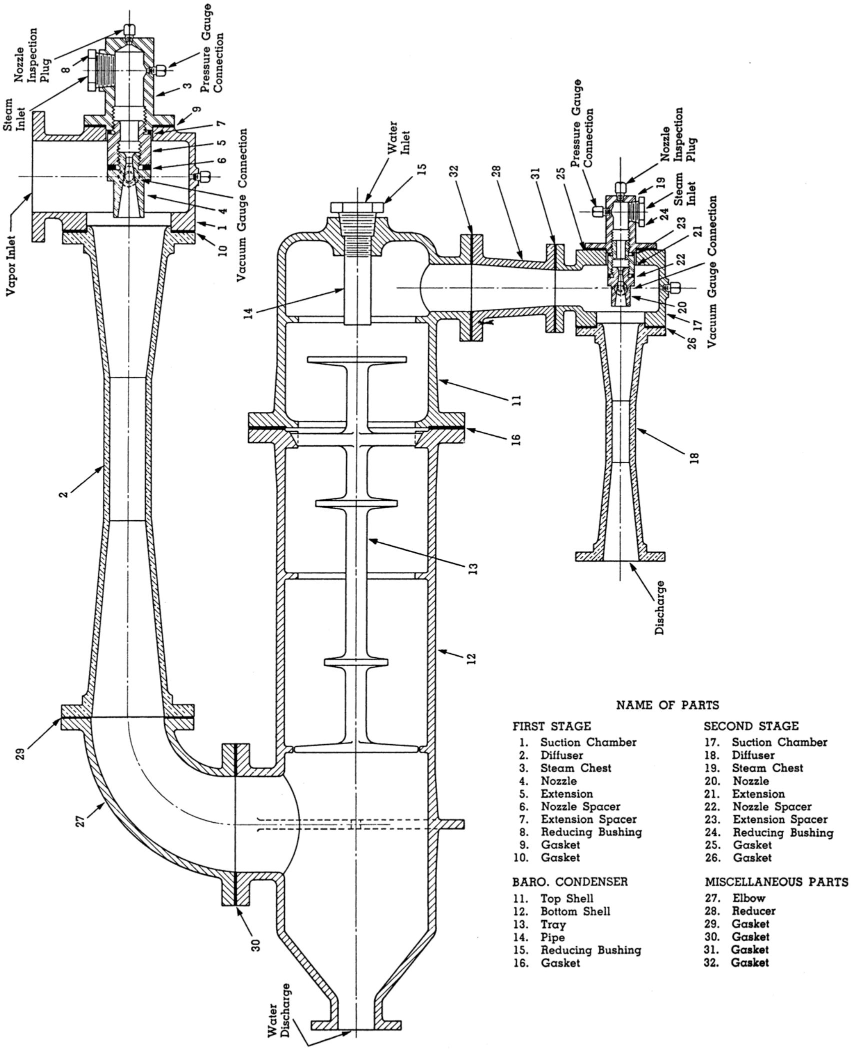
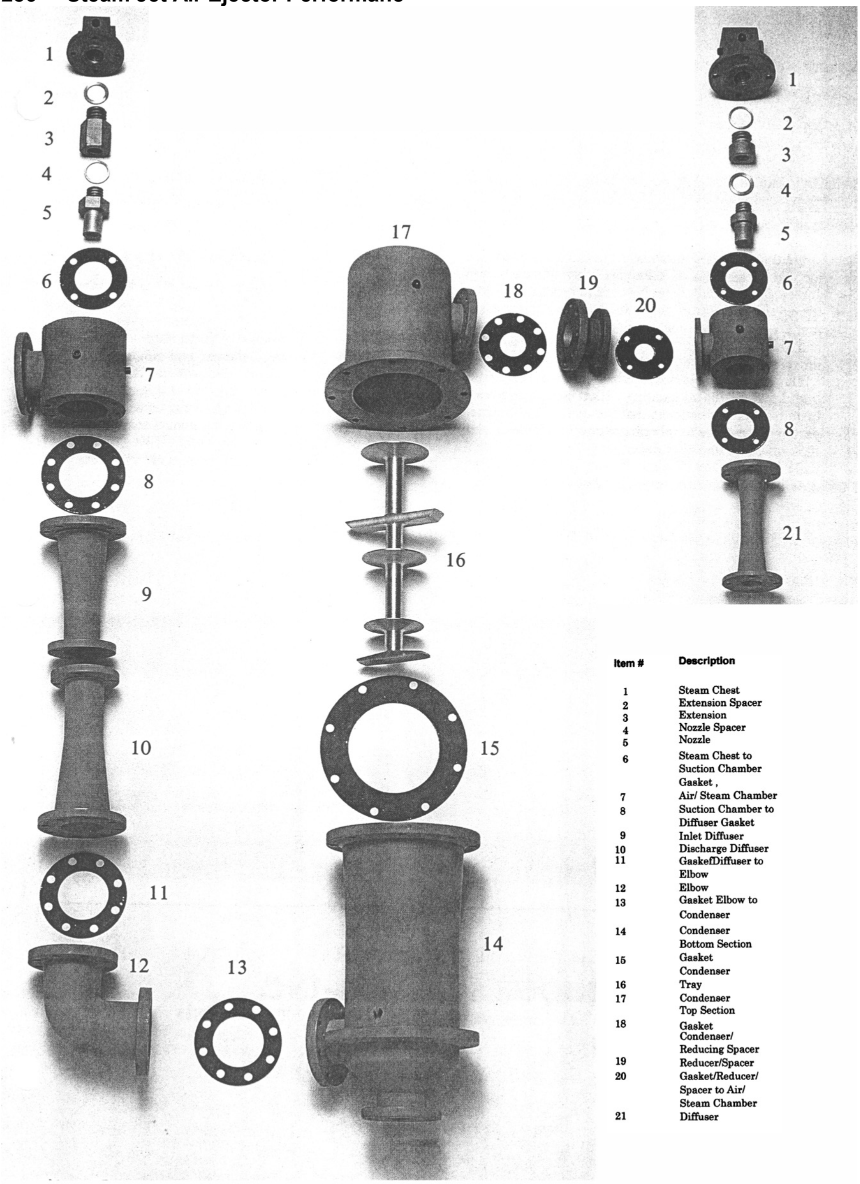


Figure 8.16 Two-stage ejector with contact intercondenser, assembly. *(Courtesy Unique Systems, Inc.)*

undersized openings which create back pressure. Look for corrosion, erosion, and fouling at the piping elbow often present at the condenser vapor inlet, and in tubes at the condenser shell-side vapor inlet.

Inspect nozzles, diffusers, and relief valves if present. Wire drawing and gray or tan streaks may indicate steam leaks at the nozzle



Item #	Description
1	Steam Chest
2	Extension Spacer
3	Extension
4	Nozzle Spacer
5	Nozzle
6	Steam Chest to Suction Chamber Gasket ,
7	Air/ Steam Chamber
8	Suction Chamber to Diffuser Gasket
9	Inlet Diffuser
10	Discharge Diffuser
11	GaskefDiffuser to Elbow
12	Elbow
13	Gasket Elbow to Condenser
14	Condenser Bottom Section
15	Gasket Condenser
16	Tray
17	Condenser Top Section
18	Gasket Condenser/ Reducing Spacer
19	Reducer/Spacer
20	Gasket/Reducer/ Spacer to Air/ Steam Chamber
21	Diffuser

Figure 8.17 Two-stage ejector with contact intercondenser, disassembled. *(Courtesy Unique Systems, Inc.)*

assembly joints. If it is necessary to face off the base of a nozzle or spacer to remove a wire-drawing groove, replace the metal gasket with a correspondingly thicker one so that the nozzle will be replaced at the same distance from the diffuser. Diffuser streaks and grooves indicate erosion by wet steam. Pockmarks are from corrosion.

Clean nozzles carefully to avoid enlarging the throats or altering the geometry, using soft metal scrapers or proper-size drills and reamers. A nozzle with a dent in the edge of the outlet cone should be replaced with a spare; the original can be either repaired and used as a spare or discarded.

Contact condenser internals should be inspected for broken, fouled, or missing water distribution hardware. Check the water inlet distributor or spray nozzle(s) for fouling. Check the water outlet and drainlegs for fouling. Inspect surface condensers for wear, fouling, and corrosion at inlet and outlet nozzles and heads.

Condenser water carryover is indicated by scale deposits at the inlet of the next stage. The cause may be excessive flow or fouled internals of a contact condenser, or an air leak in the drainleg.

Inspect and blow down steam-line strainers.

Leak-test the ejector system with a low-pressure hydrostatic test after reassembly. Use blind flanges, not permanent valves. If valves are left in, they may be accidentally closed, overpressuring equipment or forcing steam back to the process or into the cooling water. In general, avoid overpressuring the low-pressure equipment with steam or water.

Wear allowances. When you examine a worn nozzle or diffuser, you ask the question, is this item OK to use? As you might expect, you will get a different set of criteria from everyone you ask. Those manufacturers who have committed themselves to written guidelines [13, 17-19] are reflecting different perceptions of what a typical user needs. Some are quite conservative, and some expect a user to be willing to take a little more risk. All of them are good references.

Important considerations are the location and size of the stage. A small last stage with a ½-in diffuser throat is much more sensitive to wear or fouling than a Y stage with a 3-in throat. The last stage must always discharge against atmospheric pressure, whereas the Y stage can continue to function at slightly reduced capacity if its MDP curve is lowered.

Any wear or fouling alters the ejector flow geometry and reduces its performance. The safety factor in the last-stage discharge pressure and the motive steam pressure, plus the "lap" between the MDP and suction curves between stages, will allow for continued performance at slightly reduced capacity in spite of minor defects. Eventually, a defect is large enough that performance is no longer acceptable.

This uncertainty reinforces my conviction that a regular test program is a cost-effective way to measure ejector performance in a meaningful way, with a useful measure of the safety factor in the performance. When you are considering replacing worn components, the best combination of data is test results *plus* inspection results.

Measure the throat diameter in each nozzle and diffuser and compare your measurements with the dimensional data provided by the manufacturer. If you have several ejector systems to maintain, a set of pin gauges will improve your measurements of nozzle throats. If you have several identical ejector systems, a set of go/no-go gauge rods for the nozzles and diffusers will speed the inspection work and simplify the decision-making process. Some manufacturers can supply such gauges for each stage of their ejectors.

The conservative advice is to replace parts which show measurable wear or are pitted, rough, or cut. A 1 or 2 percent increase in the diameter of a nozzle or diffuser will probably not cause a problem if it is reasonably uniform.

Clean the nozzles and diffusers carefully to avoid enlarging them. Polish with emery paper. Avoid denting the thin rim at the nozzle discharge; a dent will disrupt the smooth flow pattern.

It is generally uneconomical to repair nozzles or diffusers.

Temporary fixes. Additional wear of the diffuser throat may still be acceptable if the nozzle throat size is increased more. A temporary fix is to rebore the nozzle throat, increasing its area by 1.25 times the percent increase in the diffuser throat area [19]. For example, if the diffuser throat diameter has increased by 3 percent through uniform wear, its area will have increased by 6.1 percent. The nozzle throat area should be increased by 6.1(1.25) = 7.6 percent, corresponding to a diameter increase of 3.7 percent. After you rebore the nozzle, reround the entrance to maintain a smooth entry and restore the nozzle throat length to about ¼ diameter. This example corresponds to one recommended upper limit for nozzle diameter increase [18].

If you wish to enlarge a nozzle more than that to make the stage stable or to compensate for a worn diffuser, discuss it with the manufacturer.

If you wish to try other temporary fixes to keep operating until a replacement part arrives, discuss them with the manufacturer.

In a pinch, you may be able to get an ejector working again by replacing a defective last stage with a good one from another ejector if the design pressures and sizes are reasonably matched. As described in Chap. 3, this may easily create a restricted air-capacity ejector or result in costly unutilized extra capacity in the last stage. Replacing the bad stage with a new one will restore it to the original balanced design.

Interchangeability of parts and stages. The interchangeability of some of the smaller ejector stages and components has some advantages, but creates some potential problems.

When a service is found to be more corrosive than expected, a moderate upgrade is to replace the stages, using materials that have more corrosion resistance. Some metal parts may be interchangeable with parts of other alloys. With some manufacturers, certain stages made of nonmetallic materials are dimensionally interchangeable with metal stages, with the possible exception of the steam connections.

A potential mixup problem exists, however, when externally identical stages have different internals! In smaller sizes, the last two stages may have the same external dimensions.

Accordingly, be careful to avoid mixing components and stages. Observe the part markings on nozzles, nozzle spacers, diffusers, and suction chambers. Part marks may indicate the stage number or letter (1,2,3, or X,Y,Z, or A,B,C, or Th,T,S, etc.). Also make sure nozzle spacers are kept with the nozzles. If parts are not so marked, carefully record the markings and dimensions of all parts. If parts appear to be mismatched or mixed up, be extremely careful to keep accurate records and try to resolve the problem before reassembling. If you do not have the manufacturer's parts list, call the manufacturer to help resolve your problem. It is especially easy to mix parts if more than one ejector system is taken apart at the same site.

Nozzle assembly. Accurate steam nozzle positioning is important for proper operation of the stage, and steam leaks will reduce ejector capacity. Make sure the proper thickness gaskets are in place, clean the nozzle and spacer threads, and coat these threads with molybdenum disulfide lubricant. If a soft copper gasket is used, replace it with a new one of the same thickness or anneal the old one.

Make sure that nozzle spacers are reinstalled in the proper stages if they are present. If a grooved surface is faced off, replace the gasket at that location with a correspondingly thicker gasket and record the change in the equipment record file.

Install soft gaskets on both sides of a graphite or porcelain nozzle flange. Tighten the flange bolts evenly. Use new gaskets each time with nonmetallic gaskets.

Special care for fragile materials and air-leak sources. If nozzles are plastic or graphite, check throat size for shrinkage (swelling) from extended hot service. Graphite, Haveg, and porcelain stages and condensers are fragile, and they often have protective metal casings and flange backings. Flange faces of soft materials are easily scratched, and the brittle materials are easily chipped or broken. Flanges in these systems should be tightened with torque wrenches. Haveg spare nozzles should be stored in airtight plastic bags to prevent cracking.

Give careful attention to potential air-leakage sources in the process system and the ejector system, especially packed joints and large-diameter flanged connections. Use clean tools, and avoid damaging the metal surfaces against which the gaskets seat. Do not fold gaskets.

Retest after maintenance

Retest the ejector after maintenance by measuring suction pressure at each stage at zero load and at design air load using a properly sized test orifice or test flute/piccolo. Determine the margin of safety in the last stage.

8.6 Nomenclature

MDP	maximum discharge pressure for a stage, torr
P_a	atmospheric pressure, psia
PU	pickup pressure, either MDP in torr or motive steam in psig
pph	lbm/h
rise	rate of pressure rise, torr/min
torr	absolute pressure unit, 1 mmHg
V	process system volume, ft^3
W_a	mass flow of air, pph
W_t	mass flow of test air, pph

8.7 References

1. Roger G. Mote, Air Movement and Vacuum Devices, chap. 5, "Jet Ejectors and Condensers," Technomic Publishing Co., Westport, CT, 1981.
2. *Standards for Steam Jet Vacuum Systems*, 4th ed., Heat Exchange Institute, Inc., Cleveland, OH, 1988.
3. "Ejectors, Performance Test Codes," ASME/ANSI PTC 24-1976, reaffirmed 1982.
4. D. L. Roper and J. L. Ryans, "Select the Right Vacuum Gauge," *Chemical Engineering*, March 1989, 125-144.
5. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
6. Heat Exchange Institute, "Steam-Jet Vacuum Measurement," *Instruments & Control Systems*, 39: 127-130 (1966).
7. W. Steckelmacher, "The Calibration of Vacuum Gages," *Vacuum*, 37 (8/9): 651-657 (1987).
8. N. S. Harris, *Modern Vacuum Practice*, McGraw-Hill, New York, 1989.
9. R. W. Hyland and R. L. Shaffer, "Recommended Practices for the Calibration and Use of Capacitance Diaphragm Gauges as Transfer Standards," *J. Vac. Sci. Technol.* A, 9 (6): 2843-2863 (1991).
10. Philip Freneau, "Condenser and Steam-Jet Air Ejector: A Team for Maintaining Exhaust Vacuum," *Power*, June-July 1945, 3-7.
11. Paul Taylor, "To Pull High Vacuum Try Steam Jet Ejectors," *Plant Engineering*, Aug. 8, 1968, 109-11.

12. "Vacuum/Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd, England, January 1989.
13. "Instructions for Installing and Operating Steam Jet Ejectors," Bulletin JVS-10, Jet-Vac Division of Artisan Industries.
14. VDMA, "Dampfstrahl-Vakuumpumpen und Dampfstrahl-Verdichter, Messregeln und Arbeitsmittel" (Steam-jet vacuum pumps and steam-jet compressors, methods and instruments for testing), work sheets No. 24294, sheets 1 and 2, 1971.
15. "Messregeln fUr Dampfstrahl-vakuumpumpen und Dampfstrahlkompressoren" (Methods of measurement for steam-jet vacuum pumps and steam-jet compressors), DIN 28430.
16. Association Française de Normalisation, Paris-La Défense, AFNOR NF X 10-541, Eduction 4.1976.
17. "Using Steam Jet Ejectors in Vacuum Systems," Bulletin CR-V75-1, Croll-Reynolds Co., Inc.
18. "Ejector: Operation, Maintenance, and Installation Manual," OMI-JET-184, Graham Manufacturing Co., Inc.
19. Victor V. Fondrk, "Installation, Operation, Maintenance, and Troubleshooting of Ejector Systems," Unique Systems, Inc.

Answer to Problem 3

Assuming the vent gas is essentially dry air with negligible moisture, and remembering that a pound of air has a volume of 13.3 cubic feet at standard atmospheric conditions:

Flow =(3 ft^3 / 34sec) [lbm air / 13.3 ft^3] [3600 sec / hour]

= 24 pph

Note that this unanalyzed gas flow is the net result of air leakage, plus gas dissolved and entrained in the feed, plus reaction products, minus gas dissolved and entrained in cooling water and condensate.

Problem 4

A 2-stage ejector worked OK for 5 years and now does not. An operator reports that when the last stage discharge was opened to the air, warm water spewed out. The contact intercondenser was opened, but no obstructions or wear or damage were found. The hotwell was drained and a light below the drainleg showed the drainleg was clear. What are other explanations, and what actions should be taken to test them.?

Answer on Page 422

Part

3

Specifying and Buying Steam Jet Ejectors

Answer to problem number 8

To get a quick look at how one searches for the lowest operating cost design, let us look at interstage pressures of 100, 125, and 150 torr. Neglect pressure drop in the condenser for simplicity in this first pass.

100 torr intercondenser

stage 1, dae load= $\frac{100}{1.0(0.99)} + \frac{300}{0.81(0.99)}$ = 475 pph [pp 79, 80, 91]

steam [60>100 torr]: Ra = 0.5 [pp 81 – 84]

steam use = 0.5(475) = 238 pph

dewpoint: mols = $\frac{100}{29} + \frac{(300+238)}{18}$ = 3.45+29.9=33.35 [p 138]

vapor pp = (29.9/33.35)100 = 90 torr; dewpoint = 122 $^{\circ}$F [p 110]

water out = 122-3 = 119 $^{\circ}$F; vent = 90+3 = 93 $^{\circ}$F

vapor in vent = 100 (0.42) = 42 pph [p 112]

vapor condensed = 300+238-42 = 496 pph

water flow = $\frac{496(1100)}{1.0(29)500}$ = 38 gpm [pp 430,431]

stage 2, dae load=101+42/0.8 =154 pph; 100>813 torr; R_a = ~7

steam=154(7)=1078 pph

oper. cost = (238+1078)\$300 + (38)\$400 = \$410,000 equivalent investment

125 torr intercondenser

stage 1, [60>125 torr]; R_a = 0.7; steam=0.7(475)=333 pph

vent=100(0.3)=30 pph;

dp mols = 3.45 + (300+333)/18=3.45+35.2=38.7

vapor pp=(35.2/38.7)125=114 torr; dp=130 $^{\circ}$F; water out=127 $^{\circ}$F

vapor cond = 300+333-30=603 pph; water=(603/37)2.2=36 gpm

stage 2, load=101+ 30/0.8=139 dae; steam=5(139)=695 pph

op. cost=(333+695)\$300 + (36)\$400 = \$323,000 eq. inv.

150 torr; op cost =(408+472)\$300 + (37)\$400 = \$279,000

175 torr; op cost =(475+375)\$300 + (38)\$400 = \$270,000 (estimates)

COMMENTS

The lowest operating cost requires an interstage pressure near 175 torr. The water outlet temperature exceeds that allowed by some locations,

---

Problem number 9

Work problem number 8 again, but use the shortcut methods on page 95.

Answers on page number 478

Chapter

9

Selecting a Vacuum Producer

Ejectors and/or Mechanical Pumps

9.1 Overview

The general objective of this book is to help you get the best performance out of your ejector systems. In this chapter, however, I want to make sure that I do not oversell you on using an ejector when another pumping device would be clearly better for you.

Your choice of the vacuum producer for a vacuum system is critical to successful operation of the system. If the vacuum producer does not work right, the system will fail to operate or will operate in an undesirable manner: Pressure will be too high, product will be off-specification or contaminated, production rate will be low, corrosion will be excessive, or maintenance costs will be high. If too much money is spent on the vacuum producer, then the profitability of the plant will be reduced.

In many applications, several types of vacuum pump may be capable of meeting the process specifications. Often, however, there are major differences in reliability, first cost, operating cost, and complexity of operation and maintenance. Accordingly, an appropriate selection strategy is to start with the process specifications, exclude those pump types that are clearly unsuited for the service, make preliminary estimates of first cost and operating costs, then sit down and consider the data. If the application is quite small, then the emphasis will tend to be on reliability, simplicity, and low first cost. For larger applications, the operating costs could be high enough that you may be willing to accept more complexity and a higher first cost to reduce the operating costs.

Usually the decision comes down to a subjective comparison of differences in initial investment and utilities costs versus perceived differences in reliability, operating features, cleanliness, and estimated maintenance costs. Unless convincing quantitative data simplify the decision, it will generally favor the equipment with which the designer is most familiar.

For example, I am more familiar with ejectors and tend to think first of using an ejector. Almost always, an ejector should be considered as one of the alternative vacuum pumping devices. I feel confident that I can install and operate one properly.

On the other hand, I am aware of that bias and have tried to compensate for it. As a supervisor of a machinery technology group 15 years ago, I sponsored a 30-page memorandum for coworkers containing guides to the selection of vacuum pumps. In it were references used here also.

Another preference which must be considered is that of the operating plant, especially if this process system is similar to others with which experience has been accumulated. If you wish to depart from precedent, be prepared to support your choice. It is possible that your choice may solve some ongoing problems – or create some new ones.

Because the focus of this book is primarily ejectors, I will only briefly review some of the other vacuum producers for the rough vacuum range. I will refer you to several other sources of detailed guidelines for selecting vacuum pumps and working with them.

9.2 Vacuum Pump Features

Ejectors have been described in detail throughout this book, so only a brief description will be given here. The mechanical vacuum pumps require some description. For the rough vacuum range, the mechanical vacuum pump is typically a displacement pump. The fundamental action of a displacement pump is to create a cavity which becomes filled with the suction fluid, isolate the cavity from the suction port, reduce the cavity volume a little (for some machines designed to handle compressible fluids), present the cavity to the discharge area, and then further reduce the cavity volume to expel most of the fluid. The action is repeated continuously, with the fluid delivered in intermittent pulses.

The most useful mechanical vacuum pumps have a rotary pumping element, one or two impellers of some sort, supported and driven by a rotating shaft which has bearings and a shaft seal. The reciprocating compressor, in spite of its reputation as a high-efficiency gas pump above atmospheric pressure, is not well adapted to handling small flows in general process industry applications.

The primary advantages of mechanical pumps are their constant volumetric capacity above their design pressure and a high efficiency at pressures above 10 to 30 torr. Their primary weaknesses are that most will be severely damaged by liquid slugs and that many are not available in corrosion-resistant alloys.

In the remainder of this chapter I will describe the basic features of several classes of vacuum pumps, review some important considerations on the interactions between the process system and the vacuum pump, and give you some selection guidelines.

Steam-jet ejectors

Having no moving parts, ejectors may be made of a wide variety of metals and moderate-strength erosion-resistant nonmetals. This versatility permits ejectors to be used for pumping materials which are highly corrosive, abrasive, or both. The smooth, open flow passages permit them to handle entrained solids and liquid slugs without damage.

Ejectors use high-velocity jets of steam or other motive gas to mix with the pumped gases and deliver them to a region at higher pressure. When the overall compression ratio exceeds 10:1, several stages may be arranged in series to do the pumping. Lowest in cost of the vacuum pump types, the ejector is typically highly reliable, and requires only normal skills for operation and maintenance. Disadvantages of the steam-jet ejector are the high operating costs for pressures above 30 torr, limited ability to handle air overloads, and the potential for process and environmental contamination resulting from the use of steam in contact with the process vapors. Special applications may use a process-compatible motive vapor other than steam.

Mechanical vacuum pumps

These pumps are typically driven by a constant-speed electric motor connected to the pump drive shaft by a coupling or V-belts. Small units may be close-coupled, with the impeller and shaft seal mounted on an extended motor shaft. Most pumps have a single rotor inside a casing, bearings to maintain shaft position, one or more shaft seals, and no valves. Pumping stages may be arranged in series for large compression ratios. These pumps differ from one another in the manner by which variable-volume cavities are formed to perform the pumping.

Other important differences are the presence or absence of sealing liquid and lubricants, sensitivity to liquid slugs, the presence of timing gears, auxiliaries required in a complete installation, and the need for controls to protect the pumps from operation under damaging conditions (including no-load for several types). Major advantages of mechanical pumps are their lower utilities cost in the proper application range, good air-overload characteristics (constant volumetric capacity), and compact low-level installations. Disadvantages are the general sensitivity to damage by liquid slugs, the limitations imposed by requirements for sealing liquid or lubricants, and the limited availability of corrosion-resistant materials for most types.

For more detailed descriptions of mechanical pumps and discussions of selection criteria, see References 1 to 6.

Liquid-ring. The importance of this pump in this book is that it is the closest kin to the steam-jet ejector of all the mechanical pumps.

This pump somewhat resembles a centrifugal pump in its internal appearance and operation (see Fig. 9.1). Its rotor has radial vanes, resembling a paddle wheel. The vanes have a small axial clearance with the casing and a generous clearance radially. Liquid (commonly water) is introduced in a controlled manner which causes a ring of liquid to circulate with the impeller. Centrifugal action causes the liquid to follow the contour of the casing (either circular or oval) and to separate from the pumped gas, which is trapped by the liquid ring in pockets between the vanes. The impeller is centered in an oval casing and is off-center in a circular casing. As the impeller rotates, the liquid moves in and out of the pockets between vanes, alternately closing and enlarging the pockets. A suction port is located to admit gas into the enlarging pockets, and a discharge port releases gas from the closing pockets. Because of the pumping action of the liquid, this is sometimes referred to as a "liquid piston" pump.

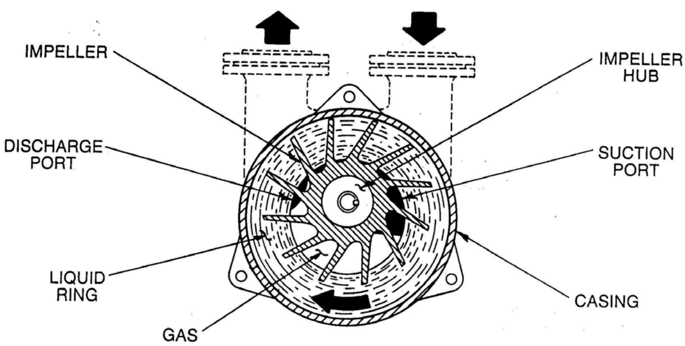


Figure 9.1 Liquid-ring pump operation. *(Courtesy Graham Manufacturing Co., Inc.)*

Although the liquid ring is maintained by the dynamic effect of the rotation, the gas-pumping characteristics more nearly resemble those of other positive-displacement machines. Manufacturers' curves describe the gas-handling ability of each model. Two stages may be combined in one housing.

Because of the intimate mixture of gas and liquid, the compression process is nearly isothermal. Offsetting that efficiency is the high friction loss in the circulating liquid ring, leading to an overall efficiency lower than that of some other gas pumps.

The sealing liquid and process gas must be mutually compatible. If the gas condenses or dissolves in the sealing liquid, that must be considered in the design. Vapor-liquid equilibrium theory is used along with experimental design factors. The vapor pressure of the sealing liquid limits the lower-pressure application range. Evaporation of the sealing liquid into the entering gas reduces the amount of gas flowing into each pocket. It is possible for damaging cavitation to occur at low flows of noncondensable gas, and the control system must protect against this damaging condition.

The reverse effect occurs if the condensable vapors in the pumped gas condense within the pump. Thus, this pump is attractive for applications where the condensable vapor is water at a temperature which permits condensation. One common application is for vacuum service on large steam power condensers. A "hybrid" application involves one or more steam- or air-jet ejector stages followed by a surface condenser and a liquid-ring pump. Because of these factors, most major ejector manufacturers now have a line of liquid-ring pumps as alternatives or companions to ejectors.

See Reference 7 for more details.

Rotary piston, oil-sealed. I chose to describe this pump second not because of its process application, but because of its use on a vacuum-pressure-gauge calibration stand. A small one- or two-stage oil-sealed rotary piston pump creates a very good near-zero pressure for checking and calibrating vacuum gauges, as described briefly in Chap. 8.

In this pump, illustrated in Fig. 9.2, an eccentrically rotating piston moves in a circular path within a circular casing, capturing a pocket of gas with each revolution and sweeping it around to the discharge port. A quantity of recirculating low-vapor-pressure oil is present, sufficient to seal the small mechanical clearances and sweep the last tiny bubble of gas out of the "pocket." The oil helps cool the gas to prevent overheating. Very high compression ratios can be maintained with a single stage, and some models have two stages.

The requirement that oil be present is a limitation on this pump's process application. The oil and process vapors must be mutually

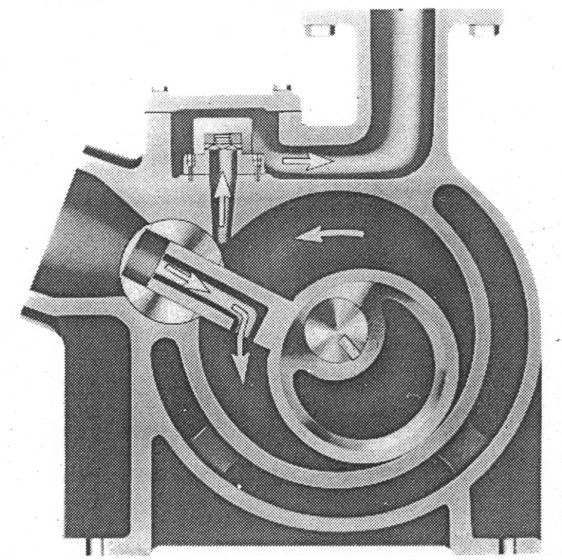


Figure 9.2 Rotary piston pump. *(Courtesy Kinney Vacuum Co.)*

compatible if it is to be used. Volatile vapors will condense in the oil and limit the low-pressure application. A gas ballast feature may clean up the oil contaminated with volatiles, or the oil may have to be replaced.

Materials of construction are limited to standard steels, and precision fits tolerate very little wear or corrosion.

Rotary-lobed "roots" blower. As shown in Fig. 9.3, this pump has two contra-rotating rotors running with close clearances inside an oval casing. The rotors are synchronized with external gears to prevent contact. The pump typically runs dry, but some applications use liquid injection for increased volumetric efficiency and cooling.

The recooling feature shown in Fig. 9.3 recycles cool gas to a carefully selected location at each side of the casing to prevent overheating of the rotor. This permits a larger compression ratio per stage, extending the application of this type of pump over the entire rough vacuum region. Without recooling, the maximum compression ratio at pressures near atmospheric pressure is about 2:1. At pressures below about 10 torr, the gas density is so low that overheating is not a problem, and interstage cooling and recooling are not required.

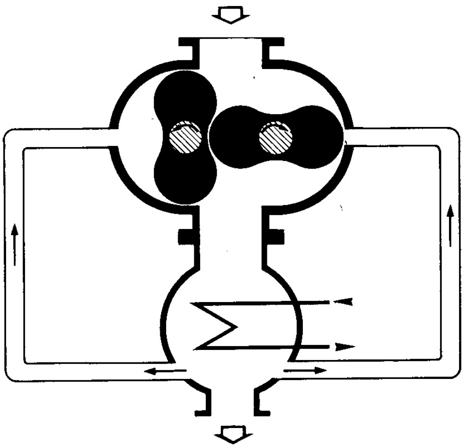


Figure 9.3 Rotary-lobed blower. *(Courtesy Leybold Vacuum Products, Inc.)*

The advantages of this type of pump are that the dry interior minimizes back diffusion of contaminants into the process (only trace quantities of lubricants from the shaft seals are involved); it has high volumetric throughput for a given size machine; the concept is simple, and the maintenance procedures are straightforward.

Its disadvantages are as follows: It is extremely vulnerable to slugs of liquid or overheating, and requires knockout pots and sophisticated protective controls; the choice of corrosion-resistant materials is limited; and timing gears plus four bearings and four shaft seals increase the number of potential mechanical failure sites and require high machinery maintenance skills.

Variations. Several variations of the basic "roots" design are commonly found. A claw-shaped rotor version adds a built-in compression which raises the efficiency and the achievable compression ratio per stage without requiring interstage cooling. Another variation is to increase the number of lobes to three or more to achieve a smoother flow. "Screw" pumps have male and female rotors with different numbers of lobes, plus an axial twist in each rotor that alters the flow pattern into a generally axial flow and produces a built-in compression for higher efficiency.

The so-called dry pumps are applications of these pump types in high-vacuum applications where cleanliness is very important. For these applications, as many as six stages may be arranged in series, mounted on one or more pairs of shafts. Hybrid combinations include "roots" stages followed by claws, rotary piston, rotary vane, or liquid-ring backing pumps.

Sliding-vane. The sliding-vane pump roughly resembles the liquid-ring, except that the impeller vanes extend to touch the casing and are located in radial slots in the rotor. Centrifugal force keeps the vanes pressed against the casing, and they slide in and out of the slots as the rotor turns. The compartments between vanes thus become larger and smaller as the rotor turns. Gas is admitted to the compartments as they become larger, and is pumped out the discharge port as they become smaller. Oil is typically used to seal the vane tips and to control wear at the tips and slots.

Variations include vanes hinged at the rotor, nonlubricated vanes, and vanes supported circumferentially by "rider rings" which absorb the centrifugal forces to reduce tip wear.

Advantages include low cost, availability in small sizes suitable for laboratory applications, and high compression ratios per stage. Vane pumps dominate laboratory applications much as steam jets dominate process applications.

Disadvantages include intolerance for liquid slugs, the need to select a compatible lubricant, the high cost of lubricant if a once-through lubricant system is required, low tolerance for fouling, the need for frequent replacement of worn or broken vanes, limited materials of construction, and the need for high maintenance skills.

Reciprocating. Originally developed as an adaption of water pumps and air compressors, the reciprocating pump/compressor is not well suited for general process vacuum service. The valves must be operated mechanically, because the pressure drop required to operate check valves is excessive. The high compression efficiency is offset by the mechanical friction losses. The volumetric throughput for a given machine size is limited by the low speeds, and the maintenance is high. Significant vibration must be accommodated.

Hybrid systems: ejector/mechanical pump

Hybrid vacuum pump systems have two or more stages consisting of two or more types of pumps, combining the best features of each. The ejector/liquid-ring hybrid is a combination that is well suited for wet vacuum applications [8]. In that hybrid, one or more steam-jet ejector stages are followed by a condenser and a liquid-ring pump. Both types are tolerant of liquid slugs, the ejectors extend the suction pressure below the lower limit of the liquid-ring pump, the liquid-ring pump has excellent air-overload capabilities and low electric power operating cost, and the combination can be supplied in a preengineered skid-mounted low-level package. .

Disadvantages are that the system requires electric power in addition to steam and water, the first cost is higher than that of an ejector, and the operation and maintenance activities require skills in working with two different classes of equipment.

Another hybrid system has an air-jet ejector using atmospheric air as the motive fluid, followed by a liquid-ring or rotary-lobed pump. It is useful where steam is not available, and it makes a compact low-level package, but the operating cost to recompress the motive air to atmospheric pressure is significant.

Some special applications can be handled by a variety of custom-designed hybrid systems involving recirculation of ejector motive steam condensate in contact condensers, chemical reactions in the contact condensers, refrigerated precondensers, and liquid-ring pumps [9]. See Chap. 11 for more on this topic.

Other combinations of mechanical pumps are outside the scope of this book.

Process-vapor-powered ejectors

Motive gases other than steam can lead to major economic improvements [4, 10]. If a process vapor is used as the motive gas, then the system is free of water and the potential for water contamination of the process, and the vapor condensate may be recycled to the process system. Thus, the costs of cleaning up the load vapors for recovery and cleaning up the steam condensate for discharge to the environment are avoided. See Chap. 11 for more on this topic.

Other high-vacuum pumps

Here I will briefly mention some pumping devices used at pressures well below 1 torr. Oil diffusion pumps operate on the ejector entrainment and momentum-transfer principles, but with a very different geometry and with low-pressure oil vapor as the motive fluid. The load gas flow is in the molecular flow region. Turbo-molecular pumps roughly resemble a multistage axial-flow compressor. Adsorption devices and materials act by trapping and holding gas molecules, almost like molecular fly paper.

9.3 Feature Comparisons: Steam Jets versus Mechanical Pumps

Many references contain detailed comparisons of the significant features of various vacuum producers [4, 6, 9, 10].

For a comprehensive comparison of the application pressure ranges of vacuum/pressure pumps, see Fig. 9.4. Note that the pressure scale is in bars and millibars (1 bar = 750.1 torr, 1.013 bar = std atm).

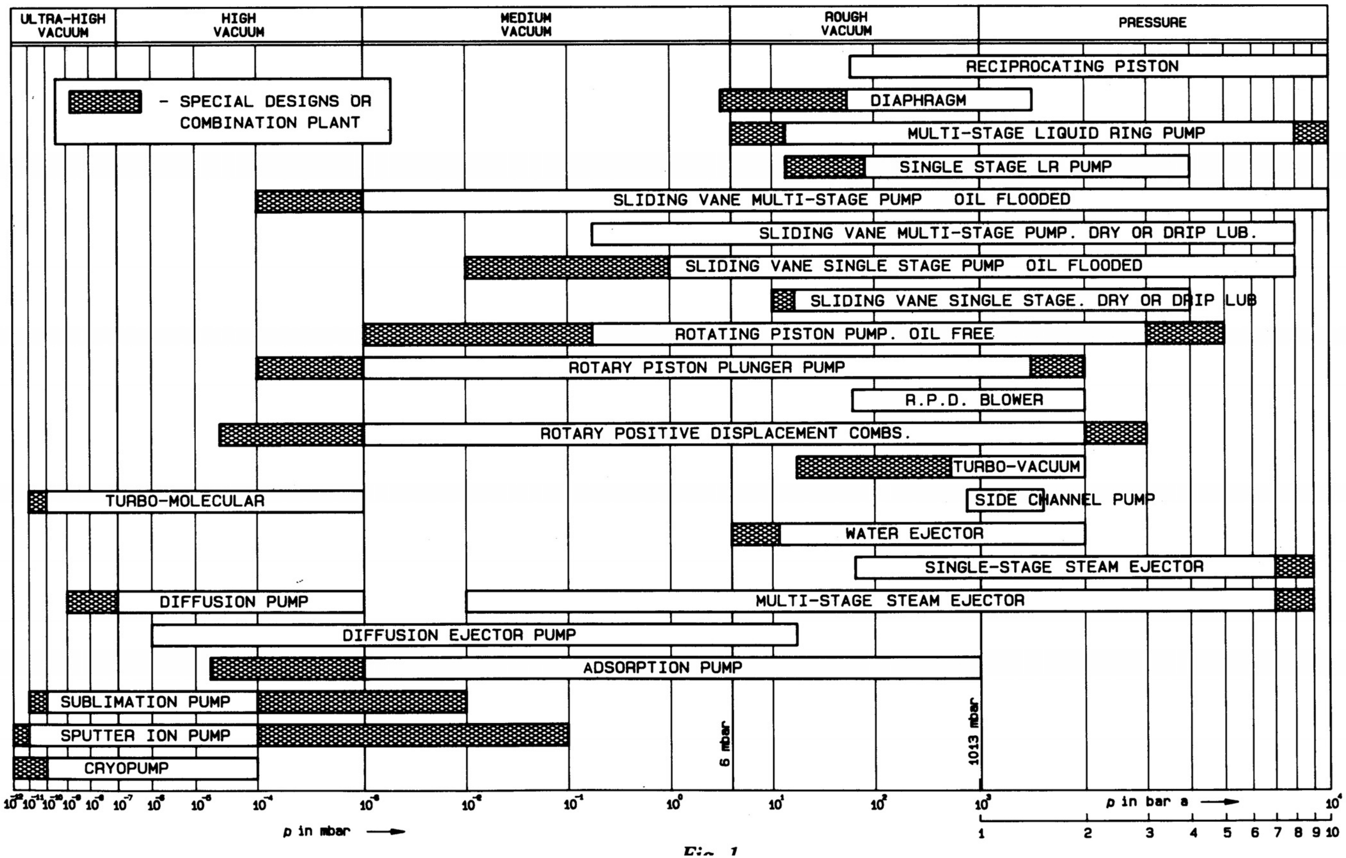


Figure 9.4 Application pressure range of vacuum pumps. *(Courtesy Hick Hargreaves & Co., Ltd.)*

TABLE 9.1 Capacity and Operating Pressure Range for Vacuum Pump Systems Commonly Used in Process Applications

Type	Blind or base pressure	Lower limit for process applications	Single unit capacity range, ft^3/min
Steam ejectors:			
One-stage	50 torr	75 torr	10-1,000,000
Two-stage	4 torr	10 torr	
Three-stage	800 μm*	1.5 torr	
Four-stage	100 μm	250 μm	
Five-stage	10 μm	50 μm	
Six-stage	1 μm	3 μm	
Liquid-ring pumps:			
60°F water-sealed:			
One-stage	50 torr	75 torr	3-18,000
Two-stage	20 torr	40 torr	
Oil-sealed	4 torr	10 torr	
Air ejector first stage	2 torr	10 torr	
Rotary piston pumps:			
One-stage	5 μm	100 μm	3-800
Two-stage	0.001 μm	10 μm	
Rotary vane pumps:			
Operated as a dry compressor	20 torr	50 torr	20-6000
Oil-sealed, rough vacuum pump	0.5 torr	2.0 torr	50-800
Oil-sealed, high-vacuum pump:			
One-stage	5 μm	100 μm	3-150
Two-stage	0.001 μm	10 μm	
Rotary-lobe blowers:			
One-stage	100 torr†	300 torr	30-30,000
Two-stage	10 torr†	60 torr	
Integrated pumping systems:			
Ejector-liquid-ring pump	1 μm	3 μm	100-100,000
Rotary-blower-liquid-ring pump	1.0 torr	5 torr	100-10,000
Rotary blower-rotary-piston pump	0.1 μm	0.10 μm‡	100-30,000
Rotary-blower-rotary-vane pump	20 μm	200 μm§	100-30,000

*1.0 μ= 0.001 torr.

†Based on intercooled design that uses gas admitted to a trapped discharge pocket to cool the blower.

‡Based on using a two-stage rotary-piston pump as the backing pump.

§Based on a two-stage rough vacuum rotary-vane design that exhibits a base pressure of approximately 0.5 torr.

SOURCE: J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation,* McGraw-Hill, New York, 1986. Courtesy McGraw-Hill.

Table 9.1 gives the pressure range and available capacities of variations and hybrids of pump types discussed in this chapter. Table 9.2 compares the reliability of various vacuum pumping systems.

Ryans and Roper [4] devote a chapter to an even-handed treatment of the selection of a vacuum producer.

TABLE 9.2 Comparative Reliabilities of Vacuum Pumping Systems

Considerations affecting reliability	Steam jets	Liquid-ring pumps	Rotary-piston and rotary-vane pumps	Steam-jet – liquid-ring pump systems	Integrated mechanical systems
Tolerance for entrained solids	*	*	Installation of inertial traps and/or bag filters in the suction line is recommended	*	Filters are recommended for applications involving abrasive materials
Tolerance for liquid slugs	*	*	Installation of a knockout pot in the suction line is required	*	A knockout pot is required if rotary-piston or rotary-vane pumps are used
Response to a surge in air leakage	Use an overdesign factor of 2.0 for critical applications. Overloading can result in backstreaming of steam to the process	The pump responds well to surges in air leakage	*	Response depends on staging ratios. Overloading can result in backstreaming of steam to the process	Response depends on staging ratios
Performance in pumping condensables	Intercondensers act as vapor pumps	*	Condensation of process vapors in the pump reduces capacity of oil-sealed pumps	*	Condensation in the blower is usually not a factor. Performance is determined by the backing pump
Response to precondenser failure	Suction pressure for multistage condensing jets shifts with load. Single-stage jets fail immediately	*	Condenser failure invariably requires changing the pump oil	*	Response is determined by the performance of the backing pump

Response to excess discharge pressure	Installation of a condenser between the ejector and the process is recommended to contain backstreaming of steam	*	*	Installation of a condenser between the ejector and the process is recommended to contain backstreaming of steam	Overheating of the blower can be a problem. A temperature limit switch that shuts the blower down is standard
Operation in corrosive environments	*	The simple design of the pump allows fabrication from a wide range of corrosion-resistant materials	Gas ballast and/or operating the pump at high temperatures permits handling corrosives provided they are not corrosive in the vapor phase	The simple design of the system allows fabrication from a wide range of corrosion-resistant materials	Condensation in the blower is not usually a factor. Reliability is determined by the performance of the backing pump
Skills required for field maintenance	*	Maintenance is relatively easy to perform and, with the exception of major overhaul, can be performed on site	Specially trained crews should handle major maintenance	Maintenance is relatively easy to perform and, with the exception of major overhaul, can be performed on site	Blower maintenance is relatively simple, but highly skilled personnel should handle servicing of the gear drives and timing mechanisms

*Indicates standard for comparison.

SOURCE: J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986. Courtesy McGraw-Hill.

Additional feature comparisons

Here are some additional questions and ideas which may help you select your vacuum producer. They are not necessarily discussed in order of importance.

Is this a large system where operating costs are significant, or a small system where reliability and simplicity are emphasized?

Can existing vacuum producers be used with the new system, either to handle overloads or to serve as a backup? Sometimes an old system can be retrofitted. For example, an ejector system's final stages can be replaced with a liquid-ring pump to reduce operating costs and improve air-overload handling.

Do in-house experience and skill levels affect your choice? Are your operators familiar with the equipment operation and troubleshooting? Are in-house or nearby maintenance shops equipped and trained to maintain the equipment, or must it be returned to the manufacturer for rework? Some mechanical pumps contain precision rotors, shaft seals, valves, and timing gears requiring realignment and recalibration.

What is the availability of the required utilities at the installation location: steam, water, electricity, and compressed air?

Is an elevated structure available for installing multistage ejectors with barometric drainlegs?

What is the emphasis on reliability? Consider the availability of replacement parts and the need to stock key parts in-house.

How important is it to avoid contamination of the process system and the environment? Must all wastewater contaminated by process materials be treated before disposal? Will process materials and lubricants contaminate each other? Is it desirable to recover process vapors in the load? Must an after-filter be used to reduce oil contamination in the exhaust air?

Is sensitivity to fouling, wear, or liquid slugging important? Is fouling affected by heat, moisture, or lubricants? If severe fouling occurs periodically, is it desirable to select pump types that are easier to clean?

Is off-design performance important: pressure control methods, air or condensable vapor overload handling, evacuation speed, or the ability to operate at zero load without damage?

Is excess discharge pressure performance important? Ejectors may be designed for almost any discharge pressure, but will fail if the design pressure is exceeded. Many mechanical pumps will continue to operate, at reduced capacity or power overload. However, some mechanical pumps will be damaged quickly.

Are there unusual safety concerns that involve high-pressure steam, electricity, rotating machinery, or violent chemical reactions with pumping vapors or liquids or lubricating oils? Is vacuum pump noise important?

For "house" vacuum systems, a corrosion resistant hybrid ejector/liquid-ring system provides any desired pressure, combined with good air-overload capability and tolerance of liquid slugs. Consider whether several small dedicated ejectors or pumps might be more cost-effective and prevent pressure control interaction and cross-contamination.

9.4 Installed Costs, Operating Costs

Several comparisons of the installed and operating costs of the various vacuum pump types and combinations have been made [1-3, 8-10]. These are useful as guides, but it is difficult to estimate installed costs, and the comparative cost changes over the years. Chapter 10 contains a brief discussion of the economics of utilities operating costs, and the Appendix contains an in-depth discussion of utilities costs, their use in purchase specifications, and a detailed example of tradeoffs in first cost versus total costs over the life of the plant.

One source [6] gives depreciation rates of 10 percent for mechanical pumps and 5 percent for ejectors, and maintenance rates of 6 percent for both. It includes a detailed analysis checklist, and reminds the user to consider the power factor of electric drives at part load.

9.5 Nomenclature

MDP	maximum discharge pressure for a stage, torr
P_a	atmospheric pressure, psia
PU	pickup pressure, either MDP in torr or motive steam in psig
pph	lbm/h
rise	rate of pressure rise, torr/min
torr	absolute pressure unit, 1 mmHg
V	process system volume, ft^3
W_a	mass flow of air, pph
W_t	mass flow of test air, pph

9.6 References

1. E. S. Monroe, "Energy Conservation and Vacuum Pumps," *Chemical Engineering Progress*, 1 (10): 69-73 (1975).
2. Paul W. Patton and Charles F. Joyce, "How to Find the Lowest Cost Vacuum System," *Chemical Engineering*, Feb. 2, 1976, pp. 84-88.
3. J. L. Ryans and Stephen Croll, "Selecting Vacuum Systems," *Chemical Engineering*, Dec. 14, 1981, pp. 72-90.
4. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
5. N. S. Harris, *Modern Vacuum Practice*, McGraw-Hill, New York, 1989.
6. "Vacuum/Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd., England, January 1989.
7. W. Hakin Faragallah, "Liquid Ring Vacuum Pumps and Compressors: Applications and Principles of Operation," Gulf, 1985.
8. J. L. Ryans, "Advantages of Integrated Vacuum Pumping Systems," *Chemical Engineering Progress*, 59-62 (1984).
9. George A. Huff, Jr., "Selecting a Vacuum Producer," *Chemical Engineering*, Mar. 15, 1976, pp. 83-86.
10. Armin Steuber, "Consider These Factors for Optimum Vacuum System Selection," *Hydrocarbon Processing*, September 1982, pp. 267-269.

Chapter 10

Specifying and Buying Ejectors

10.1 Overview

If you have read this book in chapter sequence, you are now prepared to specify and buy an ejector. If this is the first chapter you have read, and if your experience with ejectors is limited, then I urge you to read Chap. 1 to see what I recommend as preparation prior to this chapter.

If this ejector is simply one more piece of equipment in a large project, you may be anxious to finish the project. You may even be irritated at the need to learn about an obscure type of equipment you may never expect to specify again. If so, I sympathize with you. You are one of the people for whom I am writing this book. Using the information at hand, you can be satisfied that you did a good job and that you avoided the mistakes that are often made by uninformed people in a hurry.

Define your objectives

A useful starting point is to define your objectives. If this ejector is part of a larger project, your objectives are probably similar to the general project objectives: low lifetime project cost, or high reliability, or low equipment first cost, or simplicity. Avoid the temptation to think that you can achieve all these objectives. Some are mutually exclusive: low lifetime project cost and low first cost, for example.

I recommend a coordinated approach:

First, specify the smallest load and highest suction pressure with which you can feel comfortable. That can save a lot on first cost and long-term operating costs.

Next, reinvest some of the money you have "saved" by paying attention to the true costs of operating an ejector. Spend a little more for a system that uses less steam and water because it has been carefully engineered and may have more stages and more or larger condensers. Finish by assuring yourself that the installation will be reliable and simple to operate. Spend some more money on the installation details, and supply the plant people with the information they need to do their job well.

You cannot have it all-but you will have a balanced design that will look like it was designed by an expert. You will avoid the extremes of getting an oversized operating-cost hog or an undersized or unreliable unit that requires too much attention.

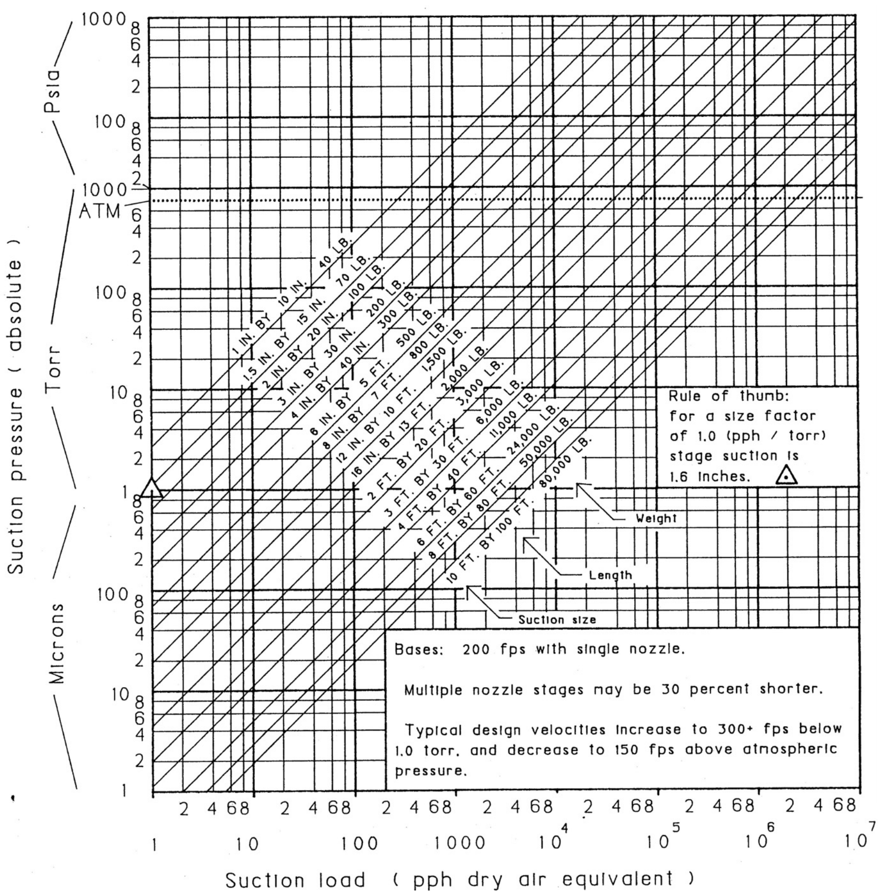


Figure 10.1 Ejector-stage size and weight.

Size and cost estimates. An important aid in establishing your design perspective is an early estimate of the size, first cost, and operating costs of the ejector system. Will it fit in your pocket, in your kitchen, or in a barn? Are the purchase price and annual operating costs less than $5,000 or more than $100,000? Figure 10.1 shows the size of the first stage, and Chap. 5 gives some estimates of condenser sizes. Figure 10.2 gives rough estimates of price, and Chap. 4 shows you how to estimate steam and water usage.

Cost estimates are budget-quality only, and much scatter is evident in any correlation of cost data. Careful analysis of such data will confirm, for example, that a four-stage ejector will generally cost more than a three-stage ejector for the same duty. The difference, however, is less than the wide scatter. Years ago I spent many hours looking at an assortment of ejector cost curves, but quit struggling after I laid my pencil on the log-log graph and observed that it covered all the data. Price movements are caused by seasonal effects, metal price fluctuations, and industry construction activity levels.

Simplicity. Simplicity is sometimes difficult to define. Often an apparent simplification is really a tradeoff in which a simplification

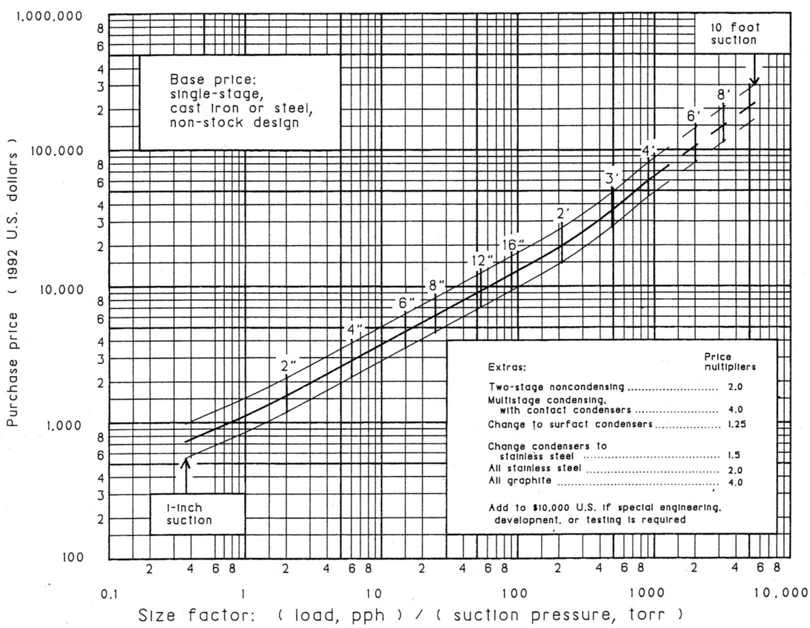


Figure 10.2 Ejector system price, components only, average utility costs. (*Note*: variability is plus or minus 30 percent.)

in one area is matched by a corresponding complexity in another. An example in ejector design is the practice of mounting a stage integrally inside the following contact condenser. Although this saves space and piping, it greatly complicates the tasks of troubleshooting and maintaining the system. Internal leaks, for example, are hard to detect. Another example is the practice of installing a throttle valve and an on-off valve in the water supply to each contact condenser, adding complexity to the design so that the routine field operating procedures can be simplified.

Reliability. An ejector is very reliable if it is properly specified, installed, and operated. The reliability is the result of careful attention to detail throughout the specification, bid evaluation, and installation design activities. In many organizations the person who specifies the ejector also reviews bids and participates in the installation design. Then the responsibility may shift to different people for the construction, startup, and continuing operation. A problem can be created if the careful planning during the specification stage is not communicated well to the operating people.

Detailed specification and data forms are a useful checklist to lead one's thinking through each of the many details which must be considered for a complete job, and to communicate the results of that thinking to the people who need the information. The result is a set of documents that convey the key data describing an ejector installation, providing each person with the basic information needed to prepare bids, evaluate them, witness tests, design the installation, operate the ejectors, and troubleshoot and maintain them.

Forms. The ejector specification form has two pages, Fig. 10.3*a* and *b*. On the first page, you describe the desired performance, objectives, and constraints to the manufacturer. This chapter walks you through the task of filling out your specifications. The Appendix contains a set of brief notes which prompt you to fill out the specifications without this book. Figure 10.4*a* and *b* is a general specification which adds essential details to your specifications.

On the second page of the ejector specification form, Fig. 10.3*b*, the manufacturer is to detail the quotation of an ejector system which meets your specification. This chapter will advise you on how to evaluate the quotations and decide whether to witness the shop performance tests.

The plant will require as-built information about the ejector in order to operate, troubleshoot, and maintain it. After the order is placed and the shop tests are completed, the manufacturer fills out

(My Conpany My Department)	BUYER'S SPECIFICATIONS STEAM JET EJECTOR	CASE: ____ PAGE ____ OF ____ ITEM NO. ____

REV	DATE	DESCRIPTION	BY	APP	
					ITEM NAME
					PROJECT
					LOCATION
					JOB NO. ____ E.O.N. ____
					PROPERTY NO.
					P.O. NO.

DESIGN DATA AND PERFORMANCE REQUIREMENTS

	SYSTEM CONFIGURATION	OPERATING CONDITIONS
1	TYPE SYSTEM ACCEPTABLE:	CASE – / PERCENT OF OPERATING TIME –
2	☐ EJECTOR	SUCTION: PRESSURE, TORR- / TEMP., F/C-
3	☐ HYBRID, EJECTOR/LIQ. RING VAC. PUMP	DISCH. PRESS. TORR (BLIND/DESN)-812/812 /
4	NUMBER OF ELEMENTS– ONE	TEMPERATURE, (MAX.) F/C –
5	NUMBER OF STAGES – BY SELLER	MOTIVE FLUID.– /STEAM
6	PRE-CONDENSER – NOT WANTED	PRESS. PSIG (DESN/MAX-FOR STRENGTH) - /
7	INTER-CONDENSER(S) – ☐ SURFACE ☐ CONTACT	TEMP. F (DESN/MAX-FOR STRENGTH) - /
8	AFTER-CONDENSER – ☐ SURFACE ☐ CONTACT	MINIMUM NOZZLE THROAT, INCH - 0.0938 /
9	CONDENSER DRAINS– ☐ BAROMETRIC LEG ☐ LOW-LEVEL	COND. WATER PRESS., PSIG (MIN/MAX) - /
10	SURFACE CONDENSER– ☐ HORIZONTAL ☐ VERTICAL	TEMPERATURE MAX. F (IN/OUT) - /
11	TUBE: ☐ STRAIGHT ☐ U-TUBE ☐ FTS ☐ PULL-THRU ☐ SPLIT-RING	VELOCITY, FPS: SHELL (MIN/MAX) 2/4 /
12	MIN (O.D./GA)– / ; MAX LENGTH, INCHES:	TUBES (MIN/MAX) 4/8 /
13	TEMA TYPE– "B" /	LIQ. RING PUMP SEALANT - / WATER
14	ASME: CONSTRUCTION ☐ YES ☐ NO STAMP ☐ YES ☐ NO	TEMP/PRESS (F-C /PSIG) - /
15	VAPOR IN– ☐ SHELL ☐ TUBES, DOWNDRAFT ONLY	SITE BAROMETRIC PRESSURE, TORR -
16	OVERALL FOUL. FACTOR (HR-FT**2-F/BTU) -0.002 /	STABLE RANGE, DESN LOAD - 0-100 % /
17	MOTORS, ENCLOSURE– TEFC /	CLOSED SYSTEM DESN PRESS. (PSIG) - NA /
18	VOLTS 460 / ; PHASE 3 / ; CYCLE 60 /	ALLOWABLE NOISE LEVEL (DB) - 90
19	SPACE LIMITS, FT (LENGTH, WIDTH, HEIGHT): / /	HOURS OPERTN THIS CASE (DAY/YEAR) - /
20	GENERAL SPECIFICATIONS ATTACHED	INITIAL EVACUATION REQUIREMENT YES / NO
21	EJECTORS RBP-JGS	TIME, MIN.- / INITIAL TEMP.(F/C 0-
22	CONDENSERS	SYSTEM VOLUME, CUBIC FEET -
23	MOTORS	SUCTION PRESS. TORR (INITIAL/FINAL) - /
24	NOISE LEVEL	ASSUMED AIR LEAKAGE, LB/HR -

	MATERIALS		
25	EJECTORS:	SURFACE CONDENSERS:	LIQUID RING VACUUM PUMP:
26	STEAM NOZZLES 316 SS /	TUBES	CASING MFR STD /
27	STEAM CHESTS MFR STD /	TUBE SHEETS	ROTOR MFR STD /
28	NOZZLE PLATES MFR STD /	CHANNELS	SHAFT MFR STD /
29	SUCTION CHAMBERS MFR STD /	SHELL	SLEEVE MFR STD /
30	DIFFUSERS MFR STD /	BAFFLES 304 SS /	SHAFT SEAL MFR STD /
31	CONTACT CONDENSERS MFR STD /	TIE RODS 304 SS /	BASE PLATE MFR STD /
32		SPACERS 304 SS /	
33			

34	INVESTMENT EQUIVALENTS - ADDED TO EQUIPMENT COST FOR EVALUATED COST
35	STEAM: $ /LB/HR ; COOLING WATER: $ /GPM ; LIQ. RING SEALANT: $ /GPM ; POWER: $ /BHP

	FLUIDS HANDLED				CONDENSABLES			IMMISCIBLE CONDENSABLES ONLY			
36–37	LOAD COMPT	NAME	FLOW PPH	MOL WT.	LAT. HT. AT 50 C BTU/LB	VAPOR PRESS. TORR AT 20 C / 50 C	SOLUBILITY IN WATER PERCENT	SPECIFIC HEAT BTU/LB-F	CONDUCTIVITY BTU/HR-F-FT	VISCOSITY CP	SPECIFIC GRAVITY
38	A	AIR		29							
39	B	WATER VAPOR		18							
40	C										
41	D										
42	E										
43	F										
44	G										

COMMENTS:

FORM RBP-JS-1 PAGE 1 OF 2 (10/93) P1-A

Figure 10.3 Steam jet ejector specification forms: (*a*) buyer's specifications.

(My Company My Department)	MANUFACTURER'S SPECIFICATION AND QUOTATION STEAM JET EJECTOR	CASE: ____ PAGE ____ OF ____ BUYER ITEM NO. ________ DATE ________ MFG REF NO. ________ MFG QUOTATION NO.

		STAGE NO.	MODEL OR SUCTION SIZE, INCH.	STEAM USAGE PPH Δ	SUCTION PRESSURE TORR Δ	DISCHARGE PRESSURE TORR Δ	STEAM JACKET YES/NO	Δ STAGE-TO-STAGE CONDITIONS MAY BE ADJUSTED SLIGHTLY IN PREFABRICATION DESIGN, NOT TO INCREASE STEAM OR COOLING WATER, OR DECREASE CONDENSER AREA NOTES
45	STAGES	1						QUOTATION IS TO INCLUDE "STICK" DWG OR BETTER, TO SHOW STAGE LENGTH, CONDENSER DIAMETER AND LENGTH.
46		2						
47		3						
48		4						
49		5						
50		6						
51		TOTAL STEAM USAGE		SILENCER REQUIRED ☐	MAKE		MODEL	MATERIALS

		CONDENSER LOCATION	COOLING WATER ☐SERIES ☐PARALLEL FLOW GPM	RISE F	VAPOR TEMP F / C INLET	OUTLET	PRESSURE TORR INLET	OUTLET	SHELL DIAM INCHES	SURFACE CONDENSERS HOR/VERT H/V	PASSES SHELL/TUBE	AREA SQ. FEET	TUBES LENGTH INCH	DIAM INCH	GAGE
52–55	CONDENSERS														
56–57		PRE-CONDENSER									/				
58–59		1ST INTER-CONDENSER									/				
60–61		2D INTER-CONDENSER									/				
62–63		3D INTER-CONDENSER									/				
64–65		AFTER-CONDENSER									/				
66		TOTAL, GPM			CODE REQUIREMENTS										
67		TOTAL, RISE, F			AND COMMENTS:										

		VACUUM PUMP	MOTOR	SEAL LIQUID	AUXILIARIES
68	LIQUID RING PUMP				
69		MODEL AND SIZE	BRAKE HP	TYPE–	SEPARATOR
70		NO. STAGES	MOTOR HP	RECOVERY–☐ FULL ☐ PARTIAL	FLOW INDICATOR
71		OPERATING RPM	RPM	☐ ONCE-THROUGH	REGULATING VALVE
72		COUPLING	ENCLOSURE	TEMPERATURE F/C –	PRESSURE GAGE
73		MATERIALS CASE	FRAME SIZE	MOL WT – GPM –	
74		ROTOR	VOLTAGE	VAP PRESS AT F/C TORR	
75		SHAFT	PHASE	VAP PRESS AT F/C TORR	
76		SHAFT SLEEVE	CYCLE	SPECIFIC GRAVITY	
77		SHAFT SEAL	SERVC FACTOR	VISCOSITY	
78		BASEPLATE		MIN SUPPLY PRESS, PSIG –	
79					

		SYSTEM		SPARE PARTS, WITH ORDER: STAGE	NOZZLE	DIFFUSER	LIQUID-RING PUMP	
80	PRICE & DELIVERY			EJECTOR STAGES				
81		BASIC SYSTEM, FOB SHIP PT	$					
82		ACCESSORIES, FOB SHIP PT	$	STAGE	NOZZLE	DIFFUSER		
83		WITNESS PERFORMANCE TEST	$	1	$	$	PUMP	$
84		WITNESS HYDROSTATIC TEST	$	2	$	$	MOTOR	$
85		FOB POINT		3	$	$	SEAL	$
86		FREIGHT EST, RAIL/TRUCK	$	4	$	$	OTHER	$
87		DWGS WKS (AFTER-ORD / APP-CERT)	/	5	$	$		$
88		EQUIP WKS (AFTER-ORD / APP-DWGS)	/	6	$	$		$
89				DELIVERY OF SPARE PARTS LATER, WKS –				

90	FORMS AFTER ORDER	PERFORMANCE CURVES - EJECTOR STAGES	RBP-MPC
91		PERFORMANCE DATA - STAGES AND CONDENSERS	RBP-MPD
92		CONDENSER DATA, SEE GENERAL SPECIFICATIONS.	

REMARKS: ** MANUFACTURER MUST ENTER ALL DATA TO BE CONSIDERED FOR PURCHASE

FORM RBP-JS-2 PAGE 2 OF 2 (10/93) p1-B

Figure 10.3 (*Continued*) Steam jet ejector specification forms: (*b*) manufacturer's specifications and quotation.

(My company and department)

GENERAL SPECIFICATIONS FOR STEAM JET EJECTORS

1. Where MULTIPLE-CASE performance specifications are listed, this specification is intended to cover one ejector capable of meeting the conditions of all cases. Multiple-element operation shall be described.
2. STABLE OPERATION is the operation of all stages of the ejector on their basic performance curves without fluctuation or bobble of the suction pressure when discharging at the specified discharge pressure. The ejector shall meet the requirements of stability when supplied with steam at design conditions, and at a steam pressure 10 percent higher than design. Basic curves and overload curves shall have only positive slopes.
3. DISCHARGE PRESSURE shall be measured at the discharge of the last item supplied by Seller: stages, aftercondenser, or silencer vent, as examples.
4. PICKUP PRESSURE of motive steam is the pressure at which a stage recovers to stable operation at the design discharge pressure. Stage pickup discharge pressure is the pressure at which the ejector recovers to stable operation at the design motive steam pressure.
5. STEAM AND WATER USAGE are to be guaranteed with all nozzles and condensers supplied with steam and cooling water at design conditions. If small booster stages are to use low-pressure steam as an exception, they shall be so identified in the bid and marked for field identification.
6. PERFORMANCE TESTS are to be performed by Seller generally in accordance with the latest edition of Standards for Steam Jet Vacuum Systems, published by the Heat Exchange Institute. The ejector shall meet or exceed the specified capacity, suction pressure, and stability. The steam and water consumption shall not exceed the guaranteed quantities. The test shall be performed and reported in the following manner:
   A. Test of an ejector as an assembled unit is preferred, with each stage loaded with its design air equivalent vapor load at or below its design suction pressure. For each stage the suction and maximum discharge pressure curves shall be determined.
   B. (5) copies of performance curves and data shall be completed (on Buyer's forms if so specified) and forwarded immediately after the shop tests are completed. The basic performance curves shall be plotted from no-load to well beyond the design load (and overload, if specified) to provide data for field troubleshooting. The performance of each stage operating in a "broken" condition with a dry air (only) overload shall be indicated approximately by a dashed line curve. It is necessary that performance data be on hand in the field before equipment startup.
7. CONDENSER MAINTENANCE DATA - (5) copies of a construction materials list shall be supplied to Buyer for use in maintenance and repair of surface condensers. The list shall identify the number, size, type, and materials of all major components of each condenser. Information supplied for code vessels is acceptable.
8. HYDROSTATIC TEST of all vacuum parts of the ejector system shall be performed at an internal pressure of at least 20 psig. All pressure parts other than steam piping shall be hydrostatically tested at an internal pressure at least 50 percent greater than the maximum working pressure.

RBP-JGS Page 1 of 2 (10/93)

Figure 10.4 General specifications for steam jet ejectors.

9. STANDARDS OF CONSTRUCTION
   - A. Unless otherwise specified, materials and details of construction shall be in accordance with the latest edition *of Standards for Steam Jet Vacuum Systems*, published by HEI, and the latest edition of TEMA Standards.
   - B. IMPINGEMENT PLATES shall be installed at the inlet vapor nozzle of surface condensers when vapors are condensed on the shell side. The minimum flow area around the impingement plate shall be twice the area of the inlet nozzle.
   - C. A VENT CONNECTION shall be added to the highest point of the water side of a surface condenser if the condenser water side is not self-venting. If water is on the shell side of a vertical condenser, then (4) drilled and tapped vents as large as possible and equally spaced shall be located in the top tube sheet if necessary to keep the top tube sheet wetted, and the inlet and outlet water connections shall be 180^{o} apart.
   - D. CONDENSATE DRAIN connection on a surface aftercondenser shall be sized to handle condensate plus full vent flow, so that Buyer has the option of closing the normal vent to seal the vent in the hotwell.

10. AUXILIARY EQUIPMENT-Unless otherwise specified, the Seller shall furnish only the following:
    - A. Ejector stages and condensers, and liquid-ring pump, if specified.
    - B. Equipment required by the ejector system to meet performance requirements-steam jacketing, steam superheater, sealing liquid controls, separator, etc.
    - C. Adapter pieces such as Dutchmen, to enable Buyer to assemble the system with commonly available piping and fittings.

    Buyer will supply valves and piping for steam, water, vapor, and condensate.

11. OUTLINE DIMENSION DRAWINGS shall include the following information for each component for detailed engineering completion before receipt of the equipment:
    - A. Location, size, and type of all external openings, including vent, drain, and test connections. Indicate flange ratings and whether bolts straddle normal centerlines.
    - B. Increments of rotation in degrees for all flanges which can be rotated for convenience in equipment layout.
    - C. Operating or maintenance features affecting equipment layout, including clearance required to remove tubes, nozzles, and other components.
    - D. Wet and dry weights of each major component.
    - E. Buyer's numbers: purchase order, item, and property number(s).

12. TEST AND DRAIN CONNECTIONS-Holes suitable for pressure taps and drains shall be drilled, tapped, then plugged in the suction chamber of each ejector stage. Fabricated stages in sizes above 5-in suction connection shall have drain connections suitable for draining the suction chamber if the stage is installed discharging vertically upward. Acceptable sizes are $^{1}/_{4}$, $^{3}/_{8}$, or $^{1}/_{2}$-in pipe taps. Three holes are normally preferred for flexibility in installing and testing.

RBP-JGS Page 2 of 2 (10/93)

Figure 10.4 (*Continued*) General specifications for steam jet ejectors.

the performance curve sheet, Fig. 10.5, and the performance data sheet for condensers and stages, Fig. 10.6. In addition, the general specifications instruct the manufacturer to supply basic information useful to the plant for repairing the surface condensers.

The plant needs a copy of each of the forms. If the operating plant is supported technically by the engineering department that specified the ejector, then it is useful to have a copy of the manufacturer's asbuilt data in the engineering project file also.

10.2 Status of Process Design

In a smooth-running project, the process design will have been completed before the ejector system is specified. Often, however, some of the fuzzy details need to be clarified.

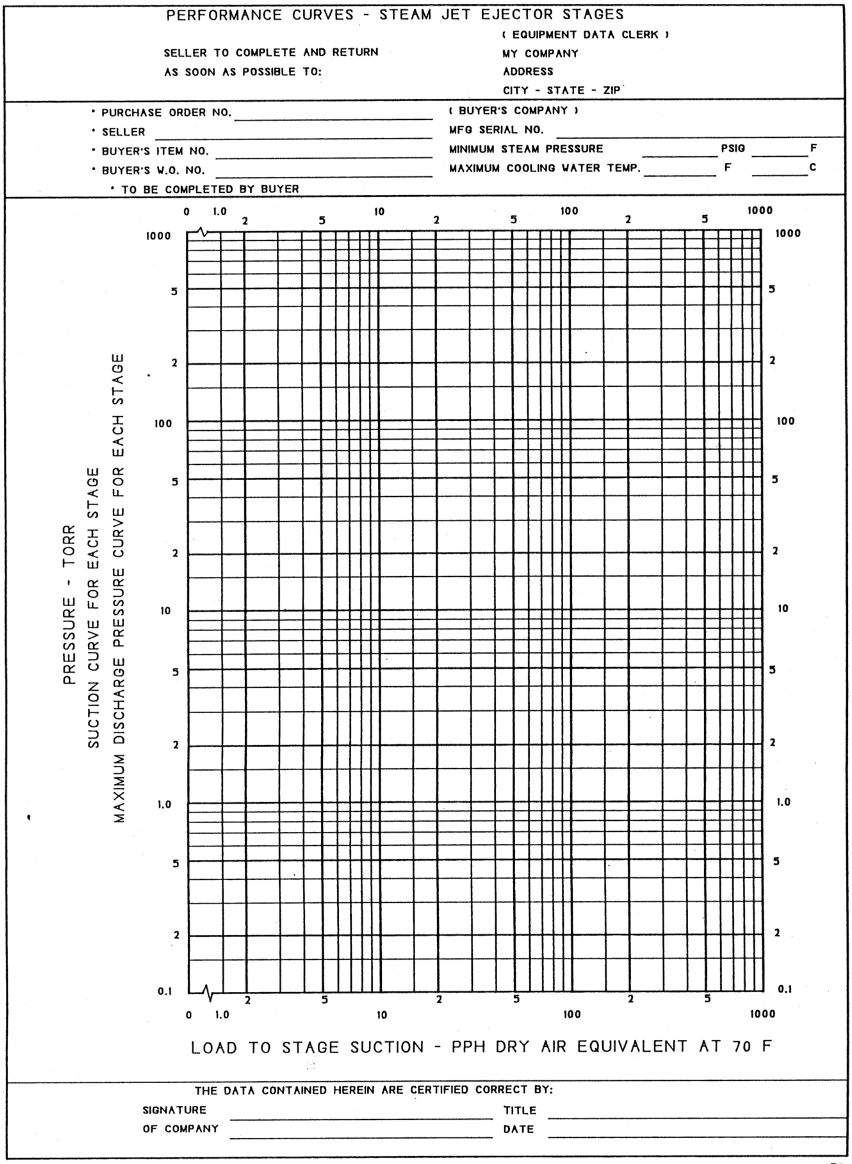
Precondensers

The first of these details is the possible need for a vent condenser (precondenser) between the process system and the ejector to condense some of the condensable vapor as a product conservation measure, an ejector cost reduction measure, or both. If the product condensate from ejector condensers requires treatment as contaminated waste, then that cost also enters into consideration. In general, a precondenser will be justified if it significantly reduces the condensable vapor load to the ejector [1, 2]. Chapter 5 describes the vapor-liquid equilibrium calculations to estimate the load reduction.

A well-designed process condenser using cooling water will cool the vent gases to within a few degrees of the cooling-water supply temperature, and usually an additional water-cooled vent condenser will be difficult to justify economically. A refrigerated vent condenser is not usually justified solely on an energy cost reduction basis. A stripping application may justify a water-cooled precondenser. Chapter 5 shows you how to estimate condenser sizes, and Fig. 10.7 gives you a price estimate for steel contact condensers.

Vapors which condense as solids

A special type of pre condenser is the one in which condensable vapors condense as solids. The solids may be washed down without fouling in a contact condenser if a suitable liquid coolant exists. Or, the solids may be condensed in a surface condenser in which they are allowed to accumulate until they diminish the condenser performance. Then the flow is switched to an alternative surface condenser and the fouled condenser is washed or heated, or both, to remove the solids.



PERFORMANCE CURVES - STEAM JET EJECTOR STAGES

SELLER TO COMPLETE AND RETURN AS SOON AS POSSIBLE TO:

(EQUIPMENT DATA CLERK)
MY COMPANY
ADDRESS
CITY - STATE - ZIP

* PURCHASE ORDER NO. ____________
* SELLER ____________
* BUYER'S ITEM NO. ____________
* BUYER'S W.O. NO. ____________
* TO BE COMPLETED BY BUYER

(BUYER'S COMPANY)
MFG SERIAL NO. ____________
MINIMUM STEAM PRESSURE ______ PSIG ______ F
MAXIMUM COOLING WATER TEMP. ______ F ______ C

THE DATA CONTAINED HEREIN ARE CERTIFIED CORRECT BY:

SIGNATURE OF COMPANY ____________ TITLE ____________

DATE ____________

FORM RBP-MPC (10/93) P3

Figure 10.5 Form for manufacturer's performance curves---stearn-jet ejector stages.

PERFORMANCE DATA - STEAM JET EJECTOR CONDENSERS AND STAGES

SELLER TO COMPLETE AND RETURN AS SOON AS POSSIBLE TO:

(EQUIPMENT DATA CLERK)
MY COMPANY
ADDRESS
CITY - STATE - ZIP

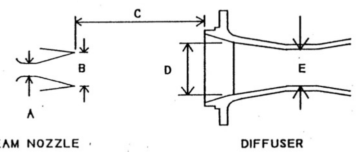
• PURCHASE ORDER NO. __________	(BUYER'S COMPANY)
• SELLER __________	MFG SERIAL NO. __________
• BUYER'S ITEM NO. __________	MINIMUM STEAM PRESSURE _____ PSIG _____ F
• BUYER'S W.O. NO. __________	MAXIMUM COOLING WATER TEMP. _____ F _____ C
• TO BE COMPLETED BY BUYER	MINIMUM COOLING WATER PRESS. _____ PSIG

CONDENSERS

TEMPERATURE SCALE UNLESS OTHERWISE SPECIFIED ☐F ☐C		PRE-CONDENSER	1ST INTER-CONDENSER	2D INTER-CONDENSER	3D INTER-CONDENSER	AFTER-CONDENSER
PRESSURE, VAPOR INLET	TORR					
PRESSURE DROP	TORR					
VAPOR INLET TEMPERATURE						
OUTLET TEMPERATURE						
WATER INLET TEMPERATURE						
OUTLET TEMPERATURE						
REQUIREMENT	GPM					
PRESSURE DROP (SURFACE CONDENSER)	PSI					
SURFACE CONDENSER AREA	SQ FT					
CONTACT CONDENSER WATER FLOW.	GPM (MIN/MAX)	/	/	/	/	/
UNIT SERIAL NUMBER						
CONDENSATE OUTLET TEMPERATURE						

•STAGES

STAGE NUMBER		1	2	3	4	5	6
SIZE CONNECTIONS. (INLET / OUTLET)	INCHES	/	/	/	/	/	/
STEAM INLET	INCHES						
NOZZLE NUMBER							
THROAT DIAM.	INCHES A						
DRILL NUMBER							
STEAM FLOW AT DESIGN CONDITIONS.	PPH						
OUTLET DIAM.	INCHES B						
TIP-TO-DIFFUSER.	INCHES C						
SPACER NUMBER							
LENGTH.	INCHES						
GASKET THICKNESS.	INCH						
MATERIAL AND TREATMENT		/	/	/	/	/	/
DIFFUSER NUMBER							
MOUTH DIAMETER	INCHES D						
THROAT DIAMETER	INCHES E						
DESIGN LOAD. 70 F DRY AIR EQUIVALENT	PPH						
PRESSURE. SUCTION.	TORR						
DISCHARGE.	TORR						
PICK-UP DISCH (DESIGN STEAM & LOAD)	TORR						
PICK-UP STEAM (DESIGN DISCH & LOAD)	PSIG						
PERFORMANCE CURVE NUMBER							



THE DATA CONTAINED HEREIN ARE CERTIFIED CORRECT BY:

SIGNATURE __________
OF COMPANY __________
TITLE __________
DATE __________

FORM RBP-MPD (10/93) P4

Figure 10.6 Form for manufacturer's performance data---steam-jet ejector condensers and stages.

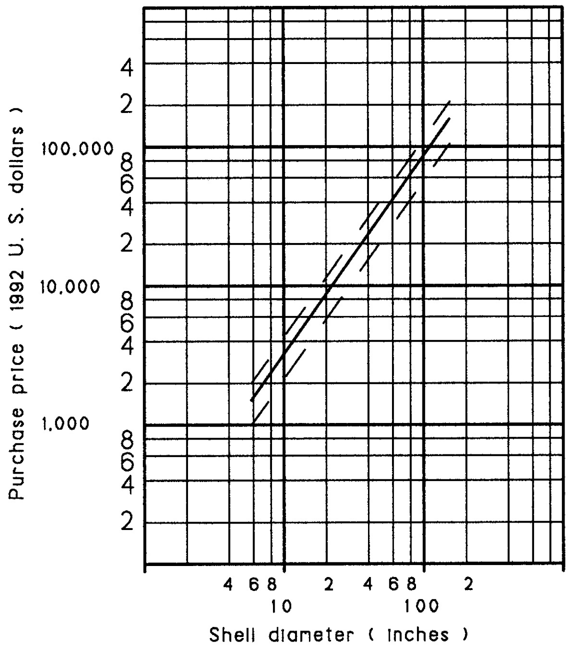


Figure 10.7 Contact condenser price, cast iron or steel.

Condensers in this service have been called sublimators. When vapors condense as solids, it is important to consider the temperature, pressure, and vapor composition at every point throughout the ejector system to prevent fouling. Heated stages may be helpful. Duplicate ejector systems may be another solution: One is in service while the other is being cleaned. Having process vapor replace steam as a motive fluid (Chap. 11) may be another solution if water causes the precipitation.

Condenser retrofit

Environmental considerations are forcing many plants to replace their contact condensers with surface condensers. See Chap. 11 for special considerations to avoid problems.

Sealpots (hotwells) and vents

The details of how the sealpot condensate will be handled and how the aftercondenser and sealpot vents will be handled may affect the ejector layout and performance specifications. If condensate pumps are required, seal pots are typically located above grade to provide a positive flow to the condensate pumps. If the sealpot must be vented to a pressurized vent header, then the sealpot pressure will be above atmospheric pressure also. Both considerations affect the drainleg height and may require elevating the ejector structure. If the ejector vent pressure is higher than normal, then the last-stage discharge pressure must be raised accordingly, or a vent ejector or blower must be added to develop the needed extra pressure.

10.3 Preparing the Ejector Specifications

Now we will walk through the specification forms and discuss each entry. Later, you may find that the specification notes in the Appendix are an adequate prompt. Changes or additions to the forms' content may be made for individual jobs by noting them in the comments section on the forms, making reference to an attachment, or both. You may wish to make extensive changes or permanent changes to the specifications forms, the general specifications, or both.

Your choice of engineering units for temperature, pressure, mass flow, and physical properties should be made to simplify work for you and other people who will work with your ejectors. The manufacturers are accustomed to working with many different combinations of units and can adapt to your needs more readily than you and your associates can adapt to another system. I have prepared the forms using a system of units convenient for many people in the United States. If those units are inconvenient for you, simply change the forms to match your needs (and please change the form numbers and dates!).

Buyer's specifications

This form is shown in Fig. 10.3*a*, and is given form number RBP-JS-1. If you enlarge it so that the rectangular border is 10 in by 7.25 in, the horizontal lines will have a vertical spacing of 6 per inch – convenient for typing the entries if you wish to do so. Modify the form by inserting your company name and address where indicated. If you modify the content of the forms or general specifications, please change the form numbers and dates to avoid confusion.

The heading contains the job data required to identify the ejector. The "case" data at the upper right corner is for those situations in which one ejector is being purchased to operate at more than one design condition. Each of the operating condition cases may be assigned a letter or number, and a separate sheet should be filled out for each case. The requirements may be explained briefly in the comments area at the bottom of the form, or in an attachment referred to in the comments area.

If the specifications contain unusual requirements, make sure that they are understood clearly by the people who need to know: the manufacturers, the installation designers, and the operating plant. The benefits of a good innovation may be lost if the installation is not properly designed and the plant is not properly instructed in its use. You may have to overcome old habits in several people.

Missing data might be requested by one or more manufacturers, but incorrect data may cause problems that are not detected until startup.

System configuration. Here you define the general physical configuration of the vacuum-producing system. If space is limited, describe the space available-length, width, and height---or provide a sketch. If you want a skid-mounted package system, describe your requirements in the Comments section.

Type of system acceptable. Select either an ejector system or a hybrid, ejector/liquid-ring system. If either is acceptable and you are not sure which will look the best to you, mark both to obtain quotations for both.

The number of elements is usually one. Twin-element designs with isolating valves at each stage are standard for high-reliability steam turbine condenser service. Usually they are supplied as a package, complete with interconnecting piping, isolating valves, and pressure relief valves. Occasionally the load is known to vary, or there is a strong incentive for steam economy that leads to a multiple-element design which permits the user to operate with the smallest number of elements that maintains acceptable vacuum. An ejector system designed to handle more than one operating condition might appropriately have multiple elements for one or more stages, especially if your specification stresses the desire for steam economy.

A multiple-booster steam-jet refrigeration system is an example of a special multiple-element ejector. Here, only the first stage has multiple elements. The stages after the first condenser in a refrigeration system are typically single-element.

Multiple-element ejectors are not difficult to operate, but they do add complexity, which leads to occasional operator error. If the steam to an element is turned off before the isolating valve at that stage suction is closed, then reversed flow through the idled stage diffuser will load up its companion stage(s).

Usually the seller can best specify the proper number of stages, based upon your design information and constraints and upon the economic objectives you specify. If, however, you wish to purchase, for example, a three-stage ejector with two intercondensers, then state that requirement precisely.

Condensers. A precondenser is usually not wanted. If one is needed, you may prefer to supply it yourself, the same way that you design and obtain other process condensers. If you want the seller to supply one, alter the preprinted entry.

Intercondensers and aftercondensers should be specified as either surface or contact type. Surface condensers keep process vapors and condensate separated from the cooling water, and they cost more than contact condensers. Contact condensers may do better cooling, and the large flow of water may condense more of any soluble condensable vapors, resulting in slightly lower steam usage. They tend to foul less than surface condensers.

Condenser drain type is identified to make sure that low-level condensers after the first one have suitable vent connections for any trap you may wish to install. Also, the entry is a reminder to you to consider whether you want other options which are often associated with low-level installations: liquid-ring pump to act as combination condensate pump and final stages, a package unit preassembled on a skid, or both.

If you selected surface condensers, indicate whether you have a preference for horizontal or vertical. If you have no preference, do not mark either option. Horizontal orientation is usually cheapest for standard materials, with vapor in the shell. If you prefer or accept vertical, make sure your cooling-water pressure is high enough to provide positive venting to the highest point in the condenser.

Tube and tube bundle types. Indicate the tube type as straight or U-tube. U-tubes may be cheaper than a removable straight-tube bundle, but U-tube exchangers cannot be mechanically cleaned inside the tubes. For straight tubes, indicate whether you accept fixed tube sheets (FTS) or prefer the more expensive pull-through or split-ring (floating-head) designs in which the shell side is also accessible for direct cleaning.

Removable bundles may be especially useful in a dirty service where you maintain spare bundles that may be quickly exchanged with dirty ones. Fixed tube sheet designs permit the use of effective shell-side baffling to perform efficient cooling of the vent gas. Removable-bundle designs attempt to duplicate that baffling effect by using flexible elements to seal the gap between baffle and shell, but with only limited success. With all removable bundles condensing in the shell side, any longitudinal baffle used to improve vent gas cooling is subject to bypassing through the baffle/shell seal. Internally packed joints are subject to troublesome hidden leaks. A condenser with an externally packed head usually has water in the shell and vapor condensing in the tubes, usually vertical. If vapor is placed in horizontal tubes, the exchanger is tilted a few degrees to drain the tubes by gravity.

Give the minimum acceptable tube diameter and gauge. A common preference is $^3/_4$-in minimum diameter for ease of cleaning. Some plants accept $^5/_8$ or $^1/_2$ in diameter, especially if their water is quite clean. Tube walls thinner than 16 gauge are difficult to roll tightly into the tube sheets, so some people accept thinner tube walls only if the tubes are seal-welded to the tube sheets. Maximum tube length is specified to avoid getting "spaghetti"---long, thin condensers that are awkward to handle and take up too much space in the layout. An estimate of the length of the first ejector stage will help you in preliminary layout planning and specifying maximum tube length. Twelve feet is a common limit. A condenser shorter than the first stage may not save space in the layout. Your maintenance shops may express a preference for tube sizes and lengths.

Indicate whether you have a choice for placement of the vapor. Typically, clean vapor is on the shell side and water is located on the tube side, where cleaning is more convenient. If on the tube side of a vertical condenser, downdraft is preferred. If on the tube side of a horizontal condenser, the body will often be tilted a few degrees toward the discharge end to ensure a positive gravity flow to minimize the film depth on the bottom of each tube.

An approach to condenser maintenance used by some plants for small aftercondensers is to accept inexpensive nonrepairable units which are simply replaced when they fail. This may include helicalcoil tube designs or small vapor-in-tube units that are well suited for maximum cooling of aftercondenser vent gases to minimize atmospheric pollution. Smaller tube diameters may be acceptable for these units, in which case special water strainers may be added to protect the small clearances of the water side from fouling by debris. If this option is acceptable or desirable, add it to the specifications.

Fouling factor. An overall fouling factor of 0.002 is preprinted as a conservative default for most situations. This corresponds to a heat-transfer coefficient of 500 BTU/h $\cdot$°F $\cdot$ft^2. A review of the topic in Chap. 5 and with your heat-transfer and maintenance specialists may lead you to use a different value for your plant or for this job. A factor of 0.001 is appropriate for normal cooling-water service and a clean process side. The contribution of the cooling-water condition to this factor is very dependent upon site-specific factors such as water intake location, straining, treatment, design velocity, and operating adjustments of water flow.

Motor specifications may be omitted if you have not selected the hybrid option. You may prefer an enclosure other than TEFC.

General specifications. General specifications are a convenient device for communicating those specifications common to almost every job. I am including only general specifications for ejectors as a preprinted entry, assuming you will use them on each job. You may wish to add references to your own general specifications for condensers, motors, liquid-ring pumps, noise levels, package systems, preparation for export shipping, painting, instruments, or other considerations.

The general specifications state that unless you specify otherwise, the manufacturer is to supply only the ejector system components. You must design the installation, and you must supply the valves, interconnecting piping, and instruments. As an alternative, ejector manufacturers are prepared to supply their equipment, ejectors and liquid-ring pumps, as preassembled, skid-mounted packages complete with interconnecting piping and appropriate valves and instruments, pretested and ready to lift into place. Each strategy has advantages and disadvantages.

Purchasing components only gives you maximum flexibility in designing the installation to fit the location for which the ejector is intended. You retain control over the details which affect many important aspects: ease of operation, testing, and maintenance; safety; and use of plant standards for piping, insulation, and instrumentation. It does, however, require more of your engineering and more on-site time and labor to install.

Purchasing a package gives you access to the expertise of the manufacturer; effectively you are subcontracting much of the installation engineering and thus reducing construction time. If you wish to incorporate your engineering and equipment standards, however, you must supply extensive specifications to guide the manufacturer. Describe the approximate space available and any layout preferences you have. The manufacturer can also supply insulation, applying it only to items that normally run hot and retaining access for testing.

Items such as valves, instruments, separators, strainers, traps, and superheaters must be specified with care to get the quality and reliability desired or to conform to your standards. "Manufacturer's standard" does not establish a minimum quality, and "or equal" is so subjective that it also does not control quality well. If you are obtaining competitive bids, remember that the pressure is on each manufacturer to minimize costs by eliminating all nonessential features. A frequent complaint about commercially designed packages is that they are not always convenient to maintain, sometimes requiring major disassembly to gain access to some parts for routine maintenance.

Operating conditions. If more than one case is specified, fill out this section of the form for each case and attach a written description of the situation. If an economic evaluation is to be made, describe the pattern of usage and the rules for the evaluation. This permits the ejector manufacturer to offer the configuration which represents your lowest evaluated cost.

The suction pressure should be specified in torr (mmHg abs). Other absolute-pressure units are acceptable, but should be specified precisely and consistently throughout the specifications, curves, and data sheets. The suction pressure should be calculated by starting with the process-system design operating pressure and subtracting the calculated pressure drops at the design flow conditions across condensers and interconnecting piping leading to the ejector. Do not specify an artificially low pressure as a means of introducing a large safety factor. It is costly, and it is not the best way to achieve the reliability you desire. Let the suction conditions safety factor be contained only in the load quantity.

The suction temperature affects the sizing of the first stage, and may affect condensation and ice formation in ejectors operating below 5 torr. If the ejector load comes from the vent of a process condenser, the condensable vapors are approximately in equilibrium with the noncondensable gases at this temperature and at the vent pressure. This often directly establishes the size of the condensable vapor load.

Discharge conditions. The discharge pressure is another critical specification. If the discharge pressure seen by the last stage exceeds the design maximum discharge pressure, the ejector will fail. All the stages!

As described in Chap. 3, the typical maximum discharge pressure (MDP) is a minimum at no-load (blind), increases as the load increases, and tends to level out near the design load. The default values for the blind and design MDP are given as 812 torr, which is 1 psig referenced to a standard atmosphere. As described in the general specifications, this is to be measured at the discharge of the last equipment item supplied by the manufacturer: last stage, aftercondenser, or silencer.

This is more conservative than the common values of $1/2$ or 1 psig at the design load. Many such last stages are unstable at small loads. The instability is not enough to "break" the preceding stage, but it does make it difficult to measure the suction pressure accurately. When an unstable ejector pumps water into a mercury manometer, it discourages the person from testing ejectors.

Several benefits result from using the conservative default values. The last stage is easier to test, which is important because that is where most problems are found and where troubleshooting should begin. It provides a comfortable, standard margin of safety for general use. For example, it permits the last-stage discharge line to be sealed in the hotwell without redesigning. the last stage (provided the submergence is not greater than 6 in to 1 ft). Or the last stage or aftercondenser may be vented to a remote location. Even if the extra discharge-pressure capability is not required by this installation immediately, it does permit some piping modifications without changing the ejector. Finally, the extra steam used by the last stage in this manner creates a safety factor at the most useful location in the ejector system. It produces tangible benefits.

Now that you are aware of the implications of changing the discharge-pressure specification, consider some reasons why you may wish to raise or lower it. An abnormally high discharge pressure may result from the need to process the vent gases in a remote treatment system or a flare stack. A modest increase to 2 psig or even 5 psig may be handled by modifying the last one or two stages. Or, it may be more appropriate to simply add an additional ejector stage or mechanical blower (or liquid-ring pump).

If the last-stage discharge or aftercondenser vent is sealed in a sealpot, and if the sealpot vent must go to a pressurized vent header, then the calculation of the specified discharge pressure is a little more complicated. Working backwards from the pressurized vent header to the last-stage discharge, it equals the vent header pressure, plus the pressure drop in the vent line from the sealpot, plus the submergence of the aftercondenser drainleg in the seal pot, plus the pressure drop in the drainleg, plus the pressure drop in the aftercondenser if present, plus a safety margin.

There is no hard limit on the allowable discharge pressure, only the expected increases in operating costs and first costs. A practical requirement is that any such special design should be clearly identified as such to the plant people responsible for using and repairing it. Otherwise, some resourceful person familiar with conventional ejectors may replace a defective last stage with a last stage from another, conventional ejector and find that it will not work.

The estimating data in Chap. 4 will help you estimate the extra steam required by the last stage. Or, you may treat an additional stage as a thermocompressor and size it as described in Chap. 11.

Finally, you may have good reasons for reducing the design discharge pressure – carefully! If your plant is at a high elevation. and you are not worried about possible reuse of the ejectors at a lower elevation, then you may wish to reduce the design discharge pressure.

The site barometric pressure is included later in the specification form, as a reminder for this specification and as a reference for gauge-pressure specifications. If your ejector last stage is large, and therefore less sensitive to wear and fouling, you may wish to reduce the design discharge pressure to save steam if you are sure that the operating discharge pressure will not be excessive. If your vent gases are removed by a header maintained below atmospheric pressure for safety reasons, then you might reduce the design discharge pressure, but not below atmospheric pressure!

The design discharge temperature has significance only if there is an aftercondenser. It gives a quantitative basis for designing the aftercondenser and should be selected to minimize the condensable vapors lost to the atmosphere. It may be from 3 to 20°F above the cooling-water supply temperature. A vapor-liquid equilibrium calculation will guide you in selecting a temperature which yields a compromise between an oversized gas cooler and excessive vapor loss to the atmosphere.

If the vapor loss to the atmosphere is large even when the vent temperature is close to the cooling-water temperature, then you may wish to add a refrigerated vent condenser if a refrigerant is available. Note that if the method of controlling pressure is by throttling the suction flow or bleeding a condensable vapor, then the vent losses calculated for economic analysis may be much less than the maximum amount based on the design air capacity. The typical air leakage may be taken as one-fourth to one-half the design air leakage, as an example, and the vent losses will usually be in direct proportion.

Motive steam. In those rare situations where a motive vapor other than steam is specified, identify it carefully and describe its thermodynamic and physical properties. See Chap. 11 for more on this subject.

The motive steam pressure is one of the most critical specifications on the form. If the operating steam pressure at the ejector steam nozzles is lower than the pressure for which the ejector is designed, the ejector will not work. Do not expect the manufacturer to insert a safety factor here to protect you. The HEI ejector standards [3] emphasize that the safety factors are your responsibility! The manufacturer may choose to add a little margin in the design to match any uncertainty in predicting performance, but incurs a corresponding penalty if operating costs are considered in bid evaluations.

Your problem is that you know that the steam pressure will vary on a seasonal basis and in response to operating changes in other plant areas. If you specify a very low design pressure to improve the reliability, then your ejector may use 10 or 20 percent more steam at typical operating conditions.

I recommend the strategy of specifying a design steam pressure 20 percent below the lowest expected header pressure, then installing a pressure-reducing valve in the local steam line to the ejector. In normal operation the steam pressure may be adjusted to 5 percent above the ejector design pressure. The steam savings will usually pay for the reducing valve quickly. An additional reliability benefit is that if the ejector fails because the last stage has become slightly worn or fouled, you may restore operation temporarily by increasing the steam pressure. In a similar manner, the steam pressure may be increased when performing field tests to make the last stage stable to no-load if it is normally unstable at no-load. It does require discipline, however, to remember to reset the steam pressure for normal operation.

As an example of the economics, consider a steam header that normally runs at 180 psig, but occasionally drops to 150 psig. The conventional approach to designing the ejector system would be to create an operating safety margin by specifying the design steam pressure to be 142 psig, which is 95 percent of the lowest plant steam pressure. Let us assume that this ejector uses 1000 pph steam at the design condition of 142 psig. In normal operation, this ejector will see 180-psig steam and will use 1240 pph, 24 percent more than the design value.

As an alternative, consider placing a steam-pressure regulator ahead of the ejector and designing it for 120-psig steam. Chapter 4 tells us that it will use about 3 percent more steam at the design conditions, 1030 pph. Setting the normal pressure at 126 psig to create a 5 percent operating safety margin will cause it to use 1080 pph steam in normal operation.

In summary, adding the pressure regulator and reducing the design steam pressure reduced the normal steam usage from 1240 pph to 1080 pph. The savings of 160 pph is 16 percent of the "bid" rate for the 142-psig conventional design. Note that a superficial comparison of the two designs would not include consideration of the normal operating steam usage, and would erroneously conclude that the alternative uses 3 percent more steam.

A second benefit of the alternative is that if the ejector fails because of wear or modest fouling, the operators may be able to restore operation temporarily by adjusting the regulator to raise the operating steam pressure.

If several ejectors are installed in the same area, a local steam header may be dedicated to ejector use, with a single pressure regulator. That gains simplicity, but eliminates the option of increasing the pressure to an individual ejector for testing or temporary relief of a problem.

The maximum pressure is specified to identify any unusual strength requirements. If ejector steam is being supplied by reducing high-pressure steam, then failure of the reducing device might apply the high pressure to the ejector steam components.

If the motive steam is dry and saturated, specify "D&S." It is not a good practice to operate ejectors using wet steam: Moisture erodes nozzles and diffusers, and it degrades the performance of low-pressure stages. If the steam is superheated, give that temperature. Superheat has little effect on the design of ejectors, except that the nozzles must have a slightly larger throat diameter to compensate for the reduced steam density. If the operating superheat exceeds your specification, the steam flows may be too low for the ejector to operate properly. If it is less than your specification, the ejector will use more steam and may have icing problems if the design depends on superheated steam for the stages below 5 torr.

In some applications below 5 torr or where solids may precipitate at normal temperatures in stages, the manufacturer may wish to supply an electric superheater to heat the steam to one or more stages. A little superheat assures the absence of moisture, which may momentarily interrupt the flow of steam in tiny nozzles and which erodes nozzles and diffusers.

The specifications for an electric superheater should include the characteristics of the electric power available, the type of hazardous atmosphere (when applicable), and required protection. The heater must be protected against overheating when the steam flow is stopped. A thermometer located in the steam outlet line will help the operator set the thermostat.

Give the maximum temperature possible, as a basis for the mechanical-strength design.

If your motive steam contains noncondensable gases, describe the situation in your specifications. They will enter the system at each steam nozzle and will add to the vent load from each condenser.

Minimum throat diameters. Specifying a minimum acceptable nozzle throat diameter improves the reliability of your ejector. Tiny nozzles are easily plugged or fouled, and have friction losses that reduce the stage efficiency. The reduction in efficiency offsets much of the economy sought by using small nozzles. The diffusers on stages with small nozzles have correspondingly small throats, especially the last stage. I once bought a two-stage ejector that had nozzles with $^1/_{16}$-in throats – and felt guilty about it for some time.

I have selected $^3/_{32}$ in (0.094 in) as a common minimum and used it as a default value on the form. Ryans and Roper [2] recommend that, and $^1/_8$ in for the last stage. Variations on this idea are to specify a minimum steam consumption or minimum diffuser throat diameter for the last stage. You might find that a larger value is appropriate in your plant, especially if the steam or the process tends to foul the nozzles and diffusers. You may wish to put these criteria permanently on the form or in the general specifications.

Manufacturers appreciate your specification of generous minimum nozzle sizes. It reduces the number of calls they will receive from plant people who complain that their ejector is not working right. It also relieves them of the painful choice between quoting you a tiny nozzle they would not want in their own plant and quoting a robust size that will make their bid unattractive. When you look at the investment equivalents from their viewpoint, you will appreciate their situation.

An alternative which a manufacturer may propose for a small first stage on a multistage ejector is to reduce the steam pressure to that stage so that the required motive steam will be delivered by the minimum nozzle throat you have specified.

Condenser water. Give the minimum and maximum pressures at which the water will be supplied. The minimum pressure determines the size of spray nozzles in a contact condenser, or whether a low-pressure-drop weir configuration is required for inlet water distribution. A vacuum contact condenser can actually lift water a few feet, but surface condensers should be supplied with water under positive pressure to assure positive venting and avoid possible cavitation. The maximum pressure might affect the strength design of the condensers.

As you think about the variability of the water pressure, consider whether this variability will make the proper water flow rate in each condenser difficult to establish. When you are doing the installation design, you may wish to install flow control valves to maintain the desired temperature rise for the cooling water in the condensers in spite of water-pressure fluctuations.

The maximum water inlet temperature is almost as critical as the steam pressure, especially for ejectors having two or more intercondensers. Temperatures above the design value will reduce the ejector capacity or cause the ejector to fail. The maximum outlet temperature is commonly given as 122°F (50°C) to avoid the precipitation of solids which foul the condenser. Some people use 140°F (60°C) as a maximum metal temperature in contact with cooling water, a more precise method of preventing fouling. Use a value based on your experience or the advice of your water specialist.

Give the range of allowable water velocities. The preprinted default minimum and maximum velocities are 2 and 4 ft/s for the shell side, and 4 and 6 ft/s for the tube side, common values to avoid the extremes of silting from low velocities and erosion or vibration from high velocities. The maximum velocities are based on copper tubes. The tube-side velocities may be increased to 12 ft/s for copper-nickel alloys and to 20 ft/s for stainless steel, nickel, or titanium.

If water has unusual fouling properties, describe them. If you specified contact condensers and the water contains unusual amounts of dissolved gas, describe the amounts. See Fig. 5.7 for the solubility of air in fresh water, and Ryans and Roper [2] for gas dissolved in sea water.

Liquid-ring pump sealant. Identify the liquid, and supply its physical properties if it is not water or a commonly known material. Important physical properties are specific gravity, viscosity, vapor pressure at two temperatures near cooling-water temperature, solubility with water and any condensable vapors in the load, reactivity, flammability, and toxicity.

Give the supply temperature and pressure.

Site barometric pressure. This provides a reference for gauge pressures and a check on the specified discharge pressure.

Stable range. For most ejectors, specify the default value, 0 to 100 percent of design load. All stages will be stable over that range. The design will be suitable for pressure control by suction throttling or by using a condensable vapor to load the ejector as described in Chap. 6.

If you plan to control the ejector by loading it with noncondensable gas such as air or nitrogen, see Chap. 6 for a description of how the stable operating range affects the behavior of ejectors using different control systems.

This recommendation, together with the recommended high discharge pressures, is more conservative than the practice of some manufacturers. By requiring that the last stage be stable to no-load, you are simplifying field testing of the most important stage in the ejector. Do not relax this requirement on the last stage unless you are comfortable testing Z stages that are not stable to no-load, and unless you are confident of the remaining safety margin.

In special circumstances it may be desirable to relax the stability requirement – carefully. In a large first stage of a large stripping ejector, you may have no objection to minor backfiring of water vapor into the process system during brief upsets, and there may be no need for the booster stage to operate much below its design capacity and suction pressure. You might permit the first stage to be unstable below 30 or 50 percent of design load to reduce the steam consumption. The stage will be stable to lower loads, even to no-load, when the cooling water is cooler than the design temperature. Discuss such an application with your ejector manufacturer.

Closed-system design pressure, for strength. If you expect to subject the ejector system to a pressure above the shop hydrostatic test pressure of 20 psig, specify the maximum pressure (and temperature if above 300°F). Otherwise, a hazardous condition could be created, exposing people and equipment to danger. An ejector on a general service distillation column or other equipment may alternate between vacuum and pressure service. Or, you may wish to be able to steam out the system to remove soluble or meltable fouling products. Or, you may have a high-pressure source of water or air which you plan to use for leak testing. Or, you may plan to install isolating valves in locations which could subject ejector stages and condensers (and sealpots?) to full steam pressure.

Allowable noise level. The default noise level is 90 dB. This determines the extent of silencing required for mechanical pumps. You may have a lower allowable noise level, depending on where the equipment will be located. The objectionable noise from the last-stage discharge will be effectively reduced by almost any device added: aftercondenser, silencer, muffler, or discharging to a sealed leg in a hotwell or sealpot.

Hours of operation for this case. For multiple-case specifications in which operating costs are evaluated, this information is helpful to the manufacturer in designing an ejector system which minimizes your total operating costs (investment equivalent). Your attachment describing the multiple-case situation should clearly indicate the procedure for economic evaluation.

Initial evacuation requirement. Chapter 4 gives a method for estimating the time required to evacuate your process system to the desired operating pressure. If that time is excessive or marginal, you may give your requirements to the manufacturer here. The manufacturer will design an ejector which will evacuate the system promptly. Various methods of meeting your evacuation requirement include increasing the size of the last one or two stages, using twin elements for the last one or two stages for steam economy, or offering a completely separate evacuation (hogging, drawdown) ejector for the task. Your specifications and the general specifications may encourage or discourage the manufacturer from offering these options. You may receive alternative designs in the bids to show you the differences in first cost, operating costs, and operating features.

If liquids are present in the system and they flash as the pressure is lowered, that complicates the calculation. The vapors will sweep the air out early in the evacuation process, but they add to the load until all the liquid is gone or cooled by the flashing. Describe the situation in an attachment.

You may prefer to purchase a separate hogging ejector, dedicated to this vacuum system or installed to serve several systems. It may also serve as a low-priced emergency spare to one or more ejectors. A noncondensing three-stage ejector can maintain a pressure as low as 2 torr. Specify a hogging ejector as a separate ejector, on its own form, with no intercondensers and with a silencer or aftercondenser.

If a hogging ejector is not intended as an emergency spare, it need not evacuate the system all the way down to the design pressure. A single-stage ejector can evacuate a system down to about 75 torr, removing more than 90 percent of the air initially in a dry system at atmospheric pressure. A two-stage hogging ejector can evacuate to about 15 torr, removing 98 percent of the air. Then the hogging ejector is isolated from the system, and the continuous-operation ejector is turned on to complete the evacuation quickly.

If a hogging ejector is being specified, observe that larger units will evacuate faster, using less total steam for the evacuation task. This is the result of the reduced effect of a fixed air leakage and the increased efficiency of larger ejectors. Of course, the hourly usage will be greater.

Materials. The materials of construction are usually similar to those in the piping and water-cooled heat exchangers in the process system. The steam nozzles and diffusers must resist the erosive action of very high velocity steam (3000 to 5000 ft/s) and the possible corrosive action of wet gas mixtures at temperatures ranging from the steam supply temperature to below the freezing point of water for pressures below 5 torr.

If even trace quantities of materials would contaminate recycled condensate, then you may specify that those materials are not to be anywhere in contact with the process.

The manufacturers can make ejectors from a wide variety of materials, and may have different recommendations for the same service. If you specify uncommon materials, you may restrict the number of suppliers. Some users have come to prefer porcelain, impervious graphite, Haveg, Teflon, and other special corrosion-resistant materials in spite of the special care required. Other materials from which ejectors have been made are glass, stoneware, rubber-lined steel, titanium, Hastelloys, and various plastics. Some of these materials have temperature limitations. The refractory materials are difficult to manufacture to close tolerances; thus the efficiency is less and the units will use more steam than normal.

Stages. The steam nozzles must be of a hard, erosion-resistant material. The default material here is 316 SS. If extra corrosion resistance is required, I recommend paying the extra for hard, corrosion-resistant metal steam nozzles. If you accept graphite or plastic nozzles, be prepared to replace them regularly.

Manufacturers' standard materials for the rest of the stages and contact condensers are usually cast iron and steel, with corrosion-resistant condenser water nozzles. Note that the steam chest is exposed to the process vapor when the steam is not flowing.

Suction chambers and diffusers are typically cast up to 6-in size and fabricated in larger sizes. In corrosion-resistant metals the diffusers are often bar stock in smaller sizes.

I recommend that you consider specifying a hard, corrosion-resistant material such as 316 SS for the diffuser of the last stage if it uses less than 200 pph steam. It is the smallest stage in the ejector, and the increase in material cost is small. A little extra investment here will yield extra reliability where it is needed most.

Surface condensers. Materials should be selected to resist corrosion and fouling on both the process side and the water side. Tubes contact both sides and usually are of a copper-bearing or stainless steel alloy. Do not use plain carbon steel---it fouls quickly in most water services. I was embarrassed once to recognize my own handwriting on a specification of steel tubes for a water-cooled exchanger that had to be retubed. I knew better, but I must have been dreaming that day. Type 304 SS is given as a default material for baffles, tie rods, and spacers. If they foul severely or corrode away, the tube bundle must be replaced. Select appropriate materials for the tube sheets, channels, and shells.

Economic objective. The economic objective in purchasing your vacuum producer will generally be similar to that of your project, or of similar projects if this is an isolated design task. Usually you want the options you select to be consistent with the lowest total cost of maintaining vacuum over the life of the project. Because that is a fuzzy objective, it can make more sense restated: Avoid the extremes of a low-first-cost unit which is very costly to operate, or an extremely high-first-cost unit which uses the least amount of steam and may have more parts to fail.

The concept of investment equivalents communicates your views of utilities operating costs in a form which combines them directly with the equipment purchase price. The form presents only your conclusions, and does not divulge confidential data such as your specific unit costs or your economic analysis methods.

The investment equivalents answer questions such as, how much more am I willing to pay for a pump that does the same job as the others, but uses one less brake horsepower because of its higher efficiency? You develop the answers using your own cost data and your own economic analysis methods, then communicate the result without elaborating on it. Your analysis data may include incremental costs, fixed costs, and investment transfers. Your analysis method may be return on investment, discounted cash flow, payback, etc., and may assume a high or low interest rate and a long or short project life. The details of such potentially complicated calculations are outside the scope of this book. I will use simple examples to illustrate the general procedure.

As an example, assume your ejector will be operated nearly continuously for several years, say 8000 hours per year. Your plant cost for the motive steam is found to be \$3 per 1000 pounds. The currently desired payback on incremental investment to reduce operating costs is 5 years. You find that investment transfer is not appropriate in this situation. Your investment equivalent of 1 pph steam is

$$\text{IE} = \left(\frac{\$3}{1000\ \text{lb}}\right)\left(\frac{8000\ \text{h}}{\text{year}}\right)(5\ \text{years}) = \$120\ \text{per pph}$$

In a similar manner, you might arrive at investment equivalents of \$200 per gpm for cooling water and \$2000 per brake horsepower (BHP) for electric power.

Please, do not regard these example numbers as recommended values, industry averages, or even accurate representations of the ratio of the relative costs of the three utilities! They are simply numbers within the range of reasonableness. Your individual values may easily vary by a factor of 3 or more above or below these values. I mention them here to establish the credibility of their order of magnitude. Years ago I shocked some fellow engineers by publishing the equivalent results from combining our utilities cost data with the common decision-making rules. I was asked to check the location of the decimal point. It was correct. Equipment manufacturers tended to react the same way, and developed innovative ways to reduce operating costs.

Providing even a rough guess is better than providing no cost information. It gives the manufacturer some incentive and direction to work toward the benefit of your project, and will tend to give you a set of more competitive bids with less spread in design and cost than if you gave no guidance. Whether you give \$50 or \$300 as the investment equivalent for 1 pph steam will not change the true total cost of operation very much. The higher value might encourage you to spend more for an ejector having a larger first condenser, and possibly having one more stage and one more condenser, but the reduction in operating costs would more than pay for the difference over the project life.

Fluids handled. Here is another opportunity to save a lot of money by departing from overconservative tradition. Specify a smaller air load which better reflects current design and maintenance skills and high energy costs. On the other hand, if you overlook a major load component or supply the wrong physical properties, the ejector may not be adequate. I recall an accelerated effort to replace an ejector for which the process engineer forgot to include a large condensable vapor component. He and I were both guilty.

Air load. Air leakage is always present in rough vacuum systems. In addition, air may enter the system dissolved in one or more feed streams. Air or nitrogen is commonly used as blowback to keep pressure measurement or gas sampling lines from plugging.

The air-leakage rate in a vacuum system tends to be established by the air-handling capacity of the vacuum pump. If the air leakage exceeds the design capacity, the resulting pressure rise will prompt the operators to locate and correct enough air leaks to permit normal operation. If the air leakage is less than the design value, there is little evidence of the size of the air leakage, and the subject is seldom of interest to the operators. The system will accumulate air leaks until the pressure rise prompts corrective action.

This reality is contrary to the erroneous concept some inexperienced people have when sizing an ejector. They assume the leakage to be a fixed quantity not subject to control, and so they tend to follow the most conservative guidelines they encounter. They are fearful of undersizing an ejector system because they expect this would prevent the system from operating. If a new plant has three times the capacity of the previous one, then the ejectors are made correspondingly larger. As a result, many vacuum producers are oversized by as much as a factor of 10 or more, consuming costly excessive amounts of steam, cooling water, and electricity.

Even grossly oversized ejectors will not protect you from all air leaks. If someone accidentally opens a 1-in valve to a vacuum system, admitting over 500 pph of air, it will overload all but the very largest of steam-jet air ejectors.

What, then, is the proper size for the design air load? Chapter 7 discusses this subject and gives typical leakage values that Ryans and Roper [2] associate with several hardware components at different system pressures. The amount of air leakage through the metal walls and weld porosity is negligible in comparison to that through discrete leakage paths such as valve-stem and pump shaft packing. Why is that subject discussed there and not here? Because the actual leakage present in a system at a given time depends in part on how well the system is designed and maintained, of course including the quality and intensity of a leak-detection program.

When the total air load in service exceeds the air-handling capacity of the last stage of a multistage ejector, the system pressure will begin to rise. The leak-detection work then begins and continues until the air leakage is reduced to an acceptable amount. Thus, the ejector design air load, in combination with the basic leak-tightness of the system design, determines the level of leak detection required. This consideration puts pressure on the ejector-system designer to specify large air loads. and to design a system which is easy to keep tight. The higher operating costs resulting from larger design air loads are an incentive to keep the design air load small.

Another consideration, which may be significant, is the harm that air leaks may cause within the operating system and environmental pollution they may cause. The oxygen in the air may react unfavorably with process materials, in direct proportion to the air leakage. Because the condensable vapor load is often directly proportional to the noncondensable gas load, the water pollution and air pollution may also be in direct proportion to the air leakage. Thus, off-specification product, lost product, polymerization fouling, corrosion, disruption of natural circulation patterns, and pollution of water and air are specific problems which may be quantitatively related to the amount of air leakage.

To minimize these effects and reduce the ejector first cost and operating costs, some users deliberately specify a small air handling capacity, so that operation cannot continue with a large air leak. The operators are thus forced to monitor air leakage carefully and maintain continual attention to all aspects of air-leakage reduction.

When I first began working with ejectors, I discussed the subject of ejector air-leakage sizing with the other engineers in my car pool, a source of free consulting advice. Art Power (not related to me) offered an attractively simple rule: Use 10 pph for small systems and 20 pph for large systems. Later I found the source of that rule [4], which does not describe how large a system must be to qualify it as large. It does say that when the design pressure is as low as 1 torr and special precautions are made to ensure leak-tightness, the air leakages can be kept as low as 2 to 5 pph. To this must be added any expected noncondensables other than air leakage, plus any associated condensable vapors.

To visualize the tightness of a low-leakage system, 10 pph represents an accumulation of air leaks equivalent to the air which a $^1/_{10}$–in critical-flow orifice will admit from atmospheric pressure into a vacuum system.

The current trend is away from the extravagant, oversized ejectors of the past and toward smaller, better-designed systems that reflect current levels of design and operation quality. Most of the times I have tested an ejector to measure the air load, I have found it to be a small fraction of the design load. Other people have had the same experience. The European ejector standards generally parallel the HEI standards, except that they deliberately recommend smaller air loads for ejectors serving the smaller power plant condensers. They openly rely on improved shaft sealing and leak-detection methods.

Having cautioned you against misinterpreting the guidelines, I now refer you to the HEI chart in Fig. 10.8, which represents a simple, quantitative opinion of what air-leakage rates can be maintained in commercial practice. By definition, if the air leakage in a system falls

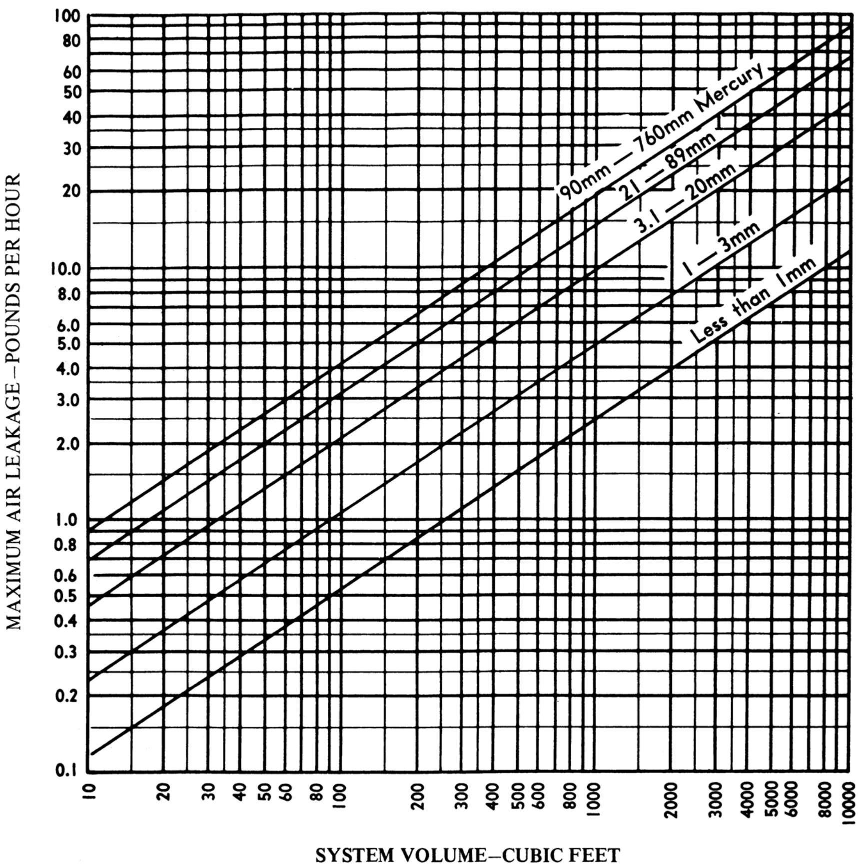


Figure 10.8 Maximum air leakage values for "commercially tight" systems. (Courtesy *Heat Exchange Institute*)

below the value shown for a given system volume and pressure, the system may be considered commercially tight. Obviously, that is only the end result of effective design and operation. The chart does reflect in a quantitative manner that larger systems will generally have more and larger air-leakage paths, that it is more expensive to remove air from lower-pressure systems, and that low-pressure systems are designed and maintained more carefully than higher-pressure systems.

Water vapor. is handled separately from other load components because it is common in ejector loads, and because it is a condensable vapor that has unique properties. Chapter 5 describes the procedures for vapor-liquid equilibrium calculations. Figure 5.4 will be useful in calculating the saturation water vapor load in an air-water mixture.

Other gases and vapors. Give the name, mass flow rate, and molecular weight for each of the other significant components in the load. If the list of known load components exceeds the space on the form, you may lump several of the lesser components under "other gases" and give their combined mass flow and mixture molecular weight.

For condensable vapors you must provide additional physical properties for designing the condensers and calculating the saturation vapor load in the condenser vents. If the liquid condensate is not very soluble in water, then give the liquid physical properties indicated. If the ideal solution and gas laws described in Chap. 5 do not describe the behavior accurately enough to use them in designing the condensers and calculating vent flows to subsequent stages, then you must supply data as described in Chap. 5. If your data are incomplete or ambiguous, the bids may contain a scatter that reflects the manufacturers' uncertainties. The low-bid designs may not work, or they may be undersized. This is especially important if the liquid phase is not very soluble in water, described also as having a high activity coefficient, or if it forms an azeotrope with water.

Do not be misled by the simple label "condensable" which is sometimes given to vapors. Typically, it means that the vapor pressure near normal cooling-water temperatures is less than atmospheric pressure. Pure vapor at atmospheric pressure can be condensed in a water-cooled heat exchanger. Unless you provide all the data requested on the form, the manufacturer is forced to guess how to design the ejector. As explained in Chap. 5, the economics of competition encourage the manufacturer to resolve the ambiguity in a manner that makes its bid most competitive, but reduces the .usable capacity of the ejector.

Add data which identify special problems, such as a material which has a freezing point above the lowest water temperature. Identify the problem material and describe the problem. Some materials are commonly recognized as problems: "traces of HCl."

Comments. This is the catch-all for minor changes that require only a little space. Refer to an attachment for an extended description of a special application situation. Revise the form or the general specifications to represent your standard requirements.

If you wish the manufacturer to supply a package system, as shown in Figure 10.9, indicate that here. You may describe your requirements for such a package here, or in a special attachment, or in an attached general specification. Examine the drawing to decide how much you want included and how much of that you wish to specify: available space, preferred layout, location of inlets, outlets, vents, and drains; valves and piping; painting and insulation; steam separator, strainer, pressure regulator, and pressure gauges; lifting lugs; and even a built-in Piccolo for field testing!

Parts interchangeability is not often a significant consideration for custom-designed ejector systems in a plant which does not have multiple installations of ejectors of similar type, materials, and manufacturer. It can be important, however, if the plant has many duplicate ejector systems, or systems using similar materials of construction, or has standardized on a small number of suppliers. This consideration applies primarily to the last-stage ejectors (Z stages). It can be implemented with care by specifying a limited range of sizes (air-handling capacity), design interstage and discharge pressures, and motive steam pressures. This leads to standard nozzles and diffusers. As a practical

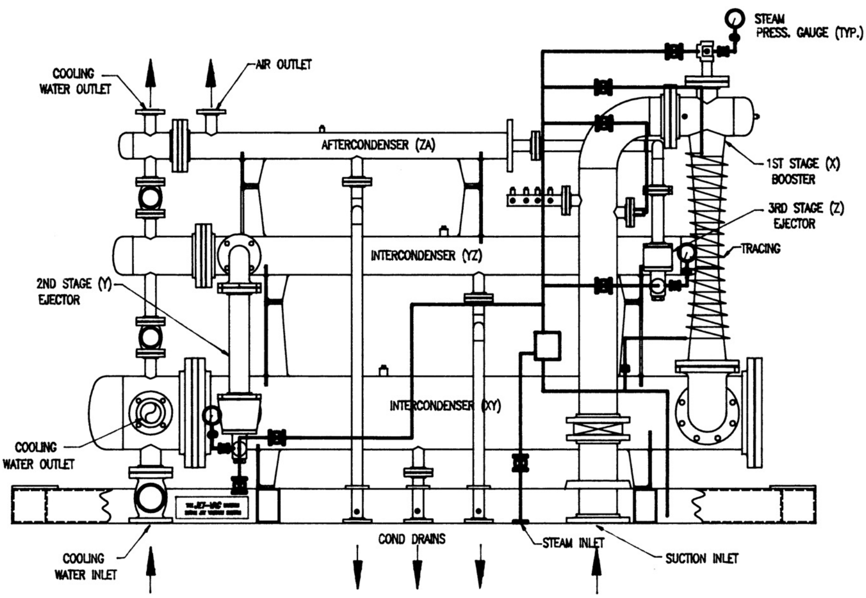


Figure 10.9 Packaged three-stage steam-jet vacuum system with shell and tube intercondensers and aftercondenser, interconnecting piping, external connections, and skid-mounted. *(Courtesy Jet-Vac, Corp.)*

method of implementing such a program, you might select standard air-handling capacities of 10, 20, and 30 pph and discuss your needs with your suppliers to arrive at standard designs for your plant.

Manufacturer's specifications and quotation

The manufacturer fills out Fig. 10.3*b,* form RBP-JS-2. The stage and condenser data are sufficient for you to make a rough check of the adequacy of the design of the most attractive quotations. The liquidring pump data are brief, but sufficient for the operating costs portion of your bid review. Manufacturers may add their standard quotation documents, especially giving more details on any liquid-ring pumps. If you want to see more data, add that request to the form or the general specifications.

10.4 Bid Evaluation

The first step in reviewing a set of bids is the simple part, processing the cost-related data. You add the purchase price, price of spare parts you plan to order with the ejector, estimated shipping cost, cost of witnessing desired tests, and sum of all the steam, water, and electricity usage multiplied by their respective investment equivalent rates. You may add some amount for perceived differences in installation costs associated with different numbers of stages and condensers. Then the real work begins.

"The buyer needs a hundred eyes, the seller not one," according to George Herbert in the 1600s. That. statement nicely summarizes the task of reviewing bids in a competitive bidding situation. Although the specification sheet and general specifications have explained your needs carefully, there is always the possibility that the specifications contained ambiguity, or that the manufacturer has made an error. As described in Chaps. 4 and 5, the errors quickly sort themselves out. The oversized, overpriced designs are not attractive, and the undersized, low-priced bids look good. Your task is to find the lowest total cost bid which also meets your requirements for reliability and performance. The unwary person tends to buy mistakes.

After writing the previous paragraph, I decided to tell you a brief "horror" story to sharpen your defensive senses. A colleague recently described an experience in which his company bought such a mistake and suffered. The details are blurred to protect the parties involved.

An ambiguous specification was written for a small ejector with a design pressure of 7 torr and a significant evacuation requirement. Manufacturers bid in a shotgun pattern ranging from four stages with two condensers, to two stages with one condenser, and even three stages noncondensing! Not surprisingly, the two-stage ejector was the low bidder. During the plant startup it didn't meet the evacuation requirements, and it was unstable near the design suction pressure.

The problem resulted from oversights on the part of the engineer who wrote the specification, and the manufacturer who failed to recognize the importance of the evacuation requirement and the need for stable operation. The problem cost the customer an estimated $1,000,000 in production losses and startup delays. The ejector was shipped back to the ejector manufacturer for a $10,000 refund. A routine bid evaluation design audit would have detected the inadequacy.

Are your senses sharpened?

Reliability assessment is a subjective task. Small stages with compression ratios greater than 10:1 are more easily degraded by moist steam, fouling, and wear. A stage which uses less than 59 pph of 165-psia steam has a nozzle throat less than $^3/_{32}$ in (0.0938 in, 2.4 mm). A Z stage using 59 pph steam will have a diffuser throat diameter of about 0.4 in (10 mm). When stages are added to an ejector to reduce steam usage, the nozzles get smaller. A small stage can achieve compression ratios of 15:1 or more on a test stand in clean condition and supplied with dry steam, but will be less dependable over time under common field conditions than will a larger stage or one with a lower compression ratio.

You review for performance adequacy by checking the design, stage by stage. Checking an ejector design is a straightforward task, as opposed to the manufacturer's design task, which may be a multiple-trial search for the interstage pressures and condenser vent temperatures which will result in the most attractive bids. For your checking, you use the data supplied by the manufacturer on Fig. 10.3*b* (form RBP-JS-2 or equivalent) and follow the stage and condenser calculation methods described in Chaps. 4 and 5.

Some ejector manufacturers prepared quotations based on generalized performance correlations of pressure ratios, mass flow ratios, etc., and sized the stages and condensers approximately for pricing purposes. Price, performance, and utilities usages were guaranteed. If one of these manufacturers received the order, the detailed selection of hardware may have resulted in minor changes in interstage pressures to best match convenient component sizes. The steam and water total usage should not increase, condenser areas should not decrease, performance is guaranteed, and the final design should still be in compliance with the specifications.

Most ejector manufacturers have now integrated their ejector design into the bidding process and have fully computerized it. Nominally, what you see in the bid is exactly what you will get. Even here, some adjustment may be required on the test floor to compensate for occasional anomalies in the data or minor fabrication errors.

Calculate the dry air equivalent (DAE) load to the first stage, then estimate the motive steam required by that stage and compare it to the manufacturer's bid. Note that some manufacturers use multiple steam nozzles in some of the larger stages to reduce the steam consumption. If the manufacturer's design uses less than 90 percent of your estimate, there may be a mistake or the manufacturer may have a high-efficiency design for that stage. If the stage discharges directly into the next stage, add the quoted first-stage motive steam to the suction load and calculate the DAE load to the next stage. Continue for each stage, noting the differences between your estimates and the quoted requirements for each stage. For each condenser, calculate the composition of the vent stream at the quoted pressure and temperatures. That is the basis for calculating the DAE load to each of the stages which follows a condenser. Check each of those stages.

Use the methods in Chap. 5 to estimate the condenser sizes. This check is approximate, because the methods are quite simplified and the geometry data are incomplete. They do accurately predict what is attainable in a surface condenser if the geometry creates a proper flow of vapors and liquids.

Each manufacturer relies heavily on experience with the specific condenser designs used in the bid. Compare the dew points and vent temperatures against the water supply temperatures and temperature rises to identify unusually close (or negative!) approach temperatures. Even if your estimates agree with the sizes in the bid, you are still relying on the manufacturer to design and fabricate the condensers to conform to the flow geometry implicit in the design equations. Some users prefer to have their heat-transfer specialists review the detailed design of surface condensers before fabrication.

If your stage-by-stage estimates and your condenser size estimates agree generally with the bid, then you may regard the design as adequate from a performance standpoint. The estimating data in Chaps. 4 and 5 are offered for rough estimates, and are not adequate for making precise appraisals of a manufacturer's stage-by-stage design. If one or more stages use much less steam than you estimate, check your calculations and discuss your findings with the manufacturer. Neither of you wants you to buy a problem, as both of you would suffer. Be prepared to send your calculations to the manufacturer as a detailed basis for your discussion. If either of you has made a mistake, you will soon identify it.

A source of major errors can be an erroneous treatment of condensable vapor when the liquid phase is immiscible in water. Be especially alert to perform the vapor-liquid equilibrium calculations properly and make sure the manufacturers do so also. For example, the manufacturer might have treated the immiscible condensable vapor as though it completely condensed in the first intercondenser. Proper treatment might show that it would begin to condense only in a later condenser, or possibly not condense at all!

Another source of differences you will encounter is that manufacturers will differ from one another in their predictions of the efficiency which they can attain in a given situation. They will differ in subtle details such as the taper angles in their diffusers, the length of a diffuser throat, the amount of underexpansion or overexpansion in a nozzle, the position of the nozzle, and the use of multiple nozzles. Except for the presence of multiple nozzles, these data are not useful to you for bid evaluation. They are simply some of the behind-the-scenes details involved in designing ejector systems. The differences may reflect true differences in efficiency of designs, or they may reflect differences in the margin of safety each manufacturer uses when bidding in a competitive situation.

As a matter of business policy, you want to avoid a pattern of buying from overoptimistic sellers who deliver either undersized ejectors or ejectors which use more than the quoted steam and water rates, unless your purchase contract contains an equitable price adjustment clause which you enforce.

In most applications an arbitrary air leakage is the basis for the ejector design load, and an ejector which has slightly more or less capacity will perform adequately. "Substantial performance" usually is all that is required to satisfy your immediate design needs.

Ejector capacity and steam usage are very important, however, in some special ejector applications, such as stripping processes using large booster stages to handle well-defined vapor loads, or thermocompressors, or steam-jet refrigeration. If you are concerned about the capacity and steam usage for any reason, you may wish to complicate your purchase contract by including provisions for adjusting the price in an equitable manner if the performance test reveals the system to be undersized, or using excessive steam and water, or both. Your overall objective is to satisfy the capacity requirement of your project and minimize your total long-term cost, in which utilities costs are represented approximately by the economic objectives data in your specifications.

Avoid discussing the details of one manufacturer's bid with another manufacturer. It is unethical and will hurt your reputation, impairing your working relationships. The best reference for a discussion of a manufacturer's design is your own rough design, prepared before you received any bids. Your calculations should explore at least two pressures for the first intercondenser, and more if a load component is immiscible in water. Performing your own design has additional benefits: It improves the cost and size estimates you usually make as you prepare the specifications, it tests whether the specifications contain sufficient data for design, and it yields information useful in the design of the installation. You will anticipate sizes of stages and condensers, steam and water usage, and the size and composition of condensate and vent streams.

After you have done a few of these calculations, you will have a heightened awareness of the internal functioning of all parts of an ejector system.

10.5 Shop Performance Testing

I recommend that you performance test each custom ejector system. Some manufacturers do so routinely. Typically, they test individual stages. Testing assembled ejectors usually costs extra, especially if the ejector has large contact condensers. The results of their tests become a matter of record for them, being a credible basis for the performance data and curves required by the specifications and general specifications. If your purchase contract has a price adjustment provision based upon the results of the tests, then you will want to witness the tests. Even if there is no great concern about performance, you may wish to witness the tests as a learning opportunity for the person who will be responsible for the ejector operation. Some circumstances that argue strongly for witnessing the tests: the first multistage ejector in the plant, more than four stages, suction pressure less than 1 torr, in a very critical service.

If your manufacturer cannot perform a shop test, then I recommend that you perform a field test that is roughly equivalent to the shop test: accurate metering of air and water vapor or other vapor loads, accurate measurement of steam pressure and vacuum pressures, and accurate measurement of water temperatures into and out of each condenser. The hardware added for the field tests will be very useful for troubleshooting and testing in the future. Be prepared for delays if a defect is found by the test.

I am skeptical of the capabilities of a manufacturer who does not test routinely. Although ejector technology is relatively mature, the practitioners will lose their skills without regular hands-on experience. Then the manufacturer will be poorly positioned to design custom systems with confidence and provide effective field support beyond its specific experience.

Inspect the equipment before shipment if the manufacturer is supplying a complete packaged system. You are expecting to save time in the field, and the shop inspection will help you achieve that.

10.6 As-Built Performance and Construction Data

The manufacturer is to supply as-built data by filling out the forms shown in Fig. 10.5 (performance curves, form RBP-MPC) and Fig. 10.6 (performance data, form RBP-MPD). The forms should have your company name and mailing address. Please change the form numbers if you alter the contents.

10.7 Nomenclature

BHP	brake horsepower, measured at the drive coupling at the design point or other specified condition
cp	viscosity, centipoise
DAE	dry air equivalent (as defined by HEI)
fps	velocity, ft/s
FT**2	area, ft^2
gpm	volumetric flow, gallons (U.S.) per minute
MDP	maximum discharge pressure for a stage, torr
pph	lbm/h
PU	pickup pressure, either MDP, in torr, or motive steam, in psig
torr	absolute pressure unit, 1 mmHg absolute

10.8 References

1. J. L. Ryans and Stephen Croll, "Selecting Vacuum Systems," *Chemical Engineering,* Dec. 14, 1981, pp. 72-90.
2. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
3. *Standards for Steam Jet Vacuum Systems,* 4th ed., Heat Exchange Institute, Inc., Cleveland, OH, 1988.
4. S. G. Ketterer and C. G. Blatchley, "Steam Jet Vacuum Pumps," Schutte and Koerting Co., 1963.

ANSWER for **POP QUIZ** Problem 2, page 56

Your first action is to ask whether the nozzles have different bases, in which case they may be assembled however they fit. Most likely the shop did not overlook that possibility, and the nozzles do have interchangeable bases. Ask whether either nozzle has a spacer. If so, ask them to match-mark the nozzles and spacers. If the spacer(s) have been mixed up also, refer to your equipment files or the manufacturer for the answer.

Now to sort out the nozzles. The last stage nozzle is the one with the lowest ratio of outlet diameter to throat diameter, unless the stages are supplied with different steam pressures.

If you want background information, refer to page 72 and examine the plot of nozzle expansion ratio versus steam expansion ratio. If you study this plot for a few minutes to understand the general relationship, you may not need to see the curve again to solve this kind of problem. Let us assign some example pressures and see what that tells us. Let the motive steam pressure be 100 psia, the design suction pressure be 1 psia, and the discharge pressure 15 psia. The steam expansion ratio in the first stage nozzle is 100/1 = 100. The curve shows the outlet diameter to be about 2.9 time the throat diameter.

Stage compression ratios range typically from 3 to 10 with the higher compression ratios in the lower pressure stages. An interstage pressure of 5 psia results in compression ratios of 5:1 and 3:1. The expansion ratio in the second stage is 20:1 and its outlet diameter is 1.5 times the throat diameter. The useful generalization from this is that the nozzles for lower pressure stages have a larger ratio of outlet diameter to throat diameter, provided the same steam pressure is delivered to each stage.

If different ,steam pressures are supplied to the nozzles, see whether the design pressure is stamped on the nozzles, and use the higher pressure nozzle for the second stage. If they are not stamped, call the manufacturer or assemble it with the largest throat diameter in the first stage and plan to performance test it in the field. If it doesn't work, swap nozzles and try again.

Problem number 3
You perform a "garbage bag" test to measure the flow of noncondensable gases from the cool vent of an ejector or vacuum pump. A 3 cubic ft bag placed over the vent inflates in 34 seconds. What is the flow in pph?

Answer Page 292

Part

4

Other Ejector Applications and Upgrading Ejector Usage

Chapter

11

Other Ejector Applications

11.1 Overview

An engineer who has learned about steam-jet ejectors and worked with them will recognize opportunities and be encouraged to work with some of the other types of ejectors, plus some of the rare special applications of vacuum ejectors. Applications within the vacuum area include steam-jet refrigeration, process-vapor-powered vacuum ejectors, and air-jet ejectors. Positive-pressure applications include gas-jet compressors, thermocompressors, steam-jet air compressors, and bulk-solids-conveying ejectors. Actually, the distinction between vacuum and pressure is not a barrier, because vacuum pump systems can be designed to discharge against high positive pressures.

This chapter is written for engineers. The basic principles used in the detailed derivations in Chaps. 4 and 5 will be referred to briefly in this chapter, but the discussions here will move at a more rapid pace.

11.2 Process Vacuum

Boosters for water removal or organic vapor stripping

In some operations, large volumes of water vapor are removed from a process and must be compressed (boosted) to a pressure high enough that they can be condensed using water-cooled condensers. Steam stripping or simple flashing under vacuum may remove organic vapors from a process, and compress the vapors for condensing. Regardless of the details, a large load is to be compressed and condensed. The noncondensable component is of secondary importance, but must be routinely included in the specifications and design considerations.

Sheer size and the corresponding potential for operating-cost reductions are the dominant features in this application. The stage size may be estimated from Fig. 10.1, and the steam consumption at design conditions from Chap. 4. Multiple nozzles will shorten the overall length, multiple elements will permit adjusting the capacity, and the ejector suction chamber may be integrated into the suction vessel head to save space and cost.

If you regard the operating cost as significant, then consider some possible steam-saving strategies. Excess steam pressure may be throttled to the design value by a steam-pressure regulator. Or the throttling may be done more efficiently by integrating the control mechanism into a variable-area steam nozzle ejector. A tapered spindle is advanced into the inlet of the steam nozzle to utilize most of the expansion energy of the steam. When cooling water is cooler than design, the steam flow may be reduced accordingly. The computer control systems of today can execute a control strategy based upon the cooling-water temperature and steam pressure as they both vary.

Another cost-reduction possibility is to use inexpensive low-pressure steam for the booster stage and higher-pressure steam or a liquid-ring pump for the secondary stages. With low-pressure steam the booster stage may be larger, use more steam, and require a larger condenser using more water.

Discuss the application with a manufacturer.

High-back-pressure jets for flare stack header discharge

This application, discussed earlier, may be handled by designing the last stage for a higher-than-normal discharge pressure or by adding a booster stage. Figure 4.8 shows the steam usage for stages with discharge pressures up to 1000 torr, almost 5 psig. It may be extrapolated a little further. Or, a booster stage may be considered to be a gas-jet compressor or thermocompressor, and those methods used for performance estimates. Note that the steam-load estimating methods require that the DAE load be converted by multiplying it by 0.81 to obtain the steam-equivalent load.

Pilot plants: small, low-cost, simple, flexible

Because pilot-plant work is typically on a small scale, steam-jet ejectors for pilot plants are also typically quite small. They are not, however, scaled-down versions of production plant designs. Corrosion-resistant materials are recommended, to be suitable for a variety of services and to preserve the geometry in the small stages. In the small sizes, the cost of special materials adds only a little to the over all price of the unit. Low first cost, simplicity, and versatility are more important than low steam and water usage [1]. Accordingly, noncondensing designs are common, as are cart-mounted low-level packages that use surface condensers followed by liquid-ring pumps.

Figure 11.1 is a laboratory-size three-stage noncondensing ejector sized for about 2 pph air at 13 torr, blanking off at about 2 torr, and using about 75 pph steam. Figure 11.2 is a cart-mounted pilot-plant hybrid ejector system with two ejector stages and surface intercondenser, followed by a liquid-ring pump. Corrosion-resistant materials are used.

Because of their small size, these units require special attention in two areas: steam supply and air leaks. The tiny steam nozzles are more sensitive to moisture in the steam, so special attention should be given to using a good steam separator and carefully insulating short steam lines. Special attention may be required to keep the system air leaks within the small capacity of the units.

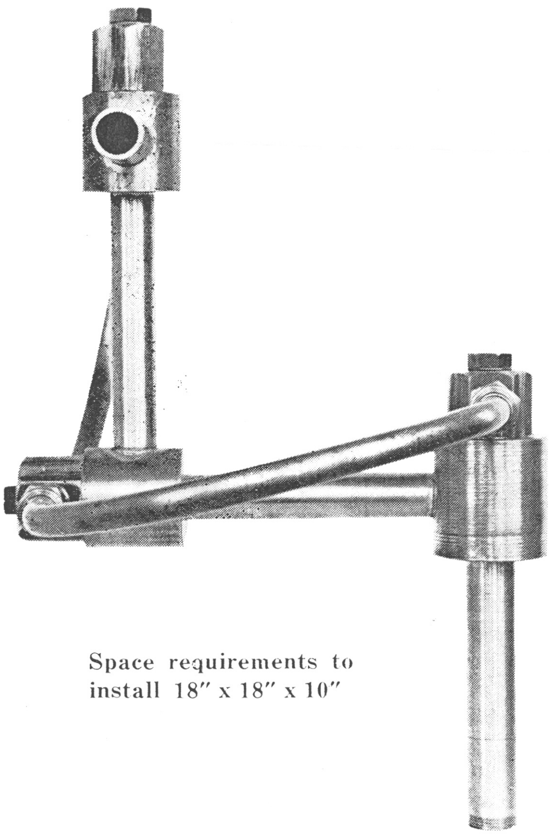


Figure 11.1 Laboratory-size three-stage noncondensing ejector. (*Courtesy J. D. Pedersen)*

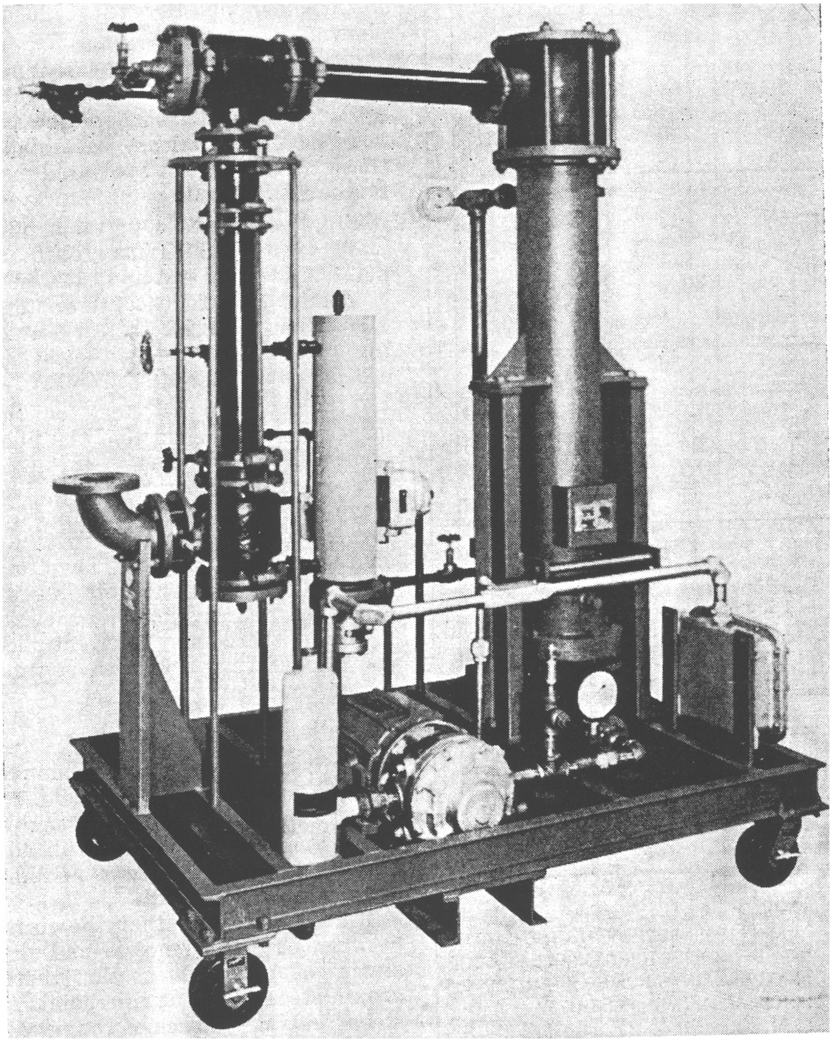


Figure 11.2 Cart-mounted pilot-plant hybrid ejector. *(Courtesy Graham Manufacturing Co., Inc.)*

Installed spares

A spare ejector is defined here as an installed unit that is intended to be used only occasionally, to substitute temporarily for another ejector or other vacuum producer until the other unit can be returned to service. Low price is more important than steam economy, so the spare may have fewer stages and condensers than the units which are operated continuously. The low-price design may be obtained by specifying a low investment equivalent for steam and water.

The spare ejector should be tailored to the needs of your system. It may have a smaller or larger design load than the devices it backs up. It may serve also as an evacuation ejector to help produce the initial vacuum in each of the systems to which it is connected. It may be connected to several systems via a spare-or-evacuation header, so that it may be put on line quickly where it is needed.

Reliability is probably the most important characteristic of this unit, and the greatest contributor to reliability is a properly working last stage. Specify the steam pressure and discharge pressure with care, and consider raising the minimum steam requirement to perhaps 100 pph for the last stage. That stage should be made of corrosion-resistant material to prevent rusting and corrosion while it is shut down in a damp environment.

The estimating methods in this book will give you an approximate idea of the prices and utilities usages. The low-price design will be a little cheaper and use a lot more steam and water than the normal-efficiency design, as the example in the Appendix shows. If in doubt, pay a little more for a design which uses steam economically. Discuss your specific situation with a manufacturer to improve your estimates and design the best system.

Process vapor as motive fluid

Process vapor or a vapor compatible with the process has been substituted for steam as a motive fluid in some ejector applications in which water vapor in the process is unacceptable [2-5]. The ideal application is a situation in which the organic vapor is available at sufficient pressure, and in which the organic condensate can be recycled to an existing process for cleanup. The typical system includes a condensate receiver, feed pump, and vaporizer, as shown in Fig. 11.3. That system also includes a stripper.

Benefits can include energy savings, especially if the process vapor is a high boiler which permits low intercondenser pressures. Other major benefits can include elimination of process and condensate contamination and the recovery of valuable process materials. The elimination of water and the possibility of running the stages hotter may reduce fouling, especially if the motive fluid acts as a solvent for the fouling materials. Erosion by liquid droplets in the motive vapor is eliminated because of the typically lower gas velocities. Corrosion is often reduced or eliminated when water is kept out of the system, sometimes permitting the use of ordinary materials of construction instead of expensive alloys.

Organic vapors used in such systems include monochlorobenzene, orthodichlorobenzene, cyclohexane, methanol, Freon, phenol, ethylene glycol, ethylene, xylene, perchlorethylene, acetone, butanediol, and toluene. As one example of the design differences between such

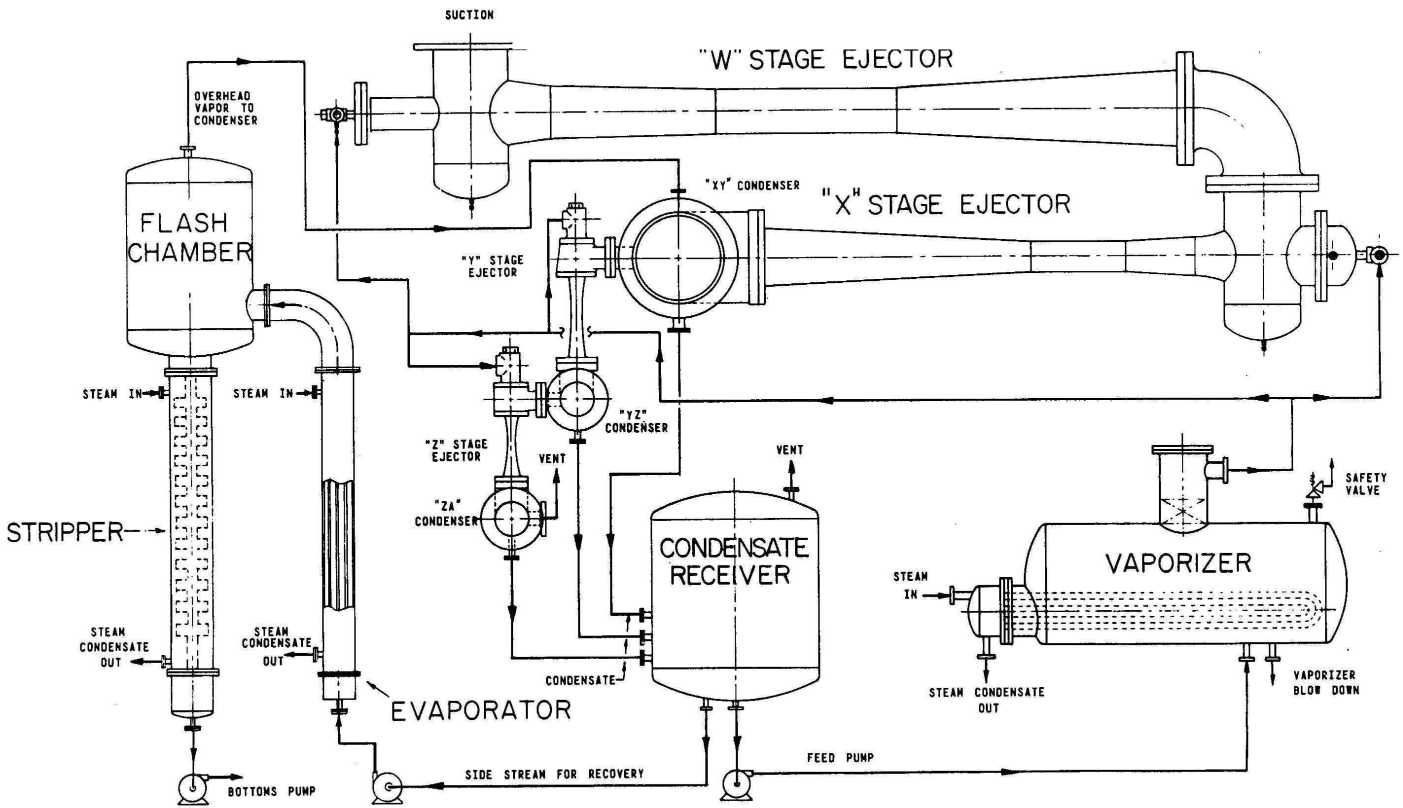


Figure 11.3 Process-vapor-powered ejector system. (*Courtesy The Jet-Vac Corp.*)

systems and a steam-powered ejector, ethylene glycol motive vapor is supplied at a pressure of 300 torr to operate ejector stages having suction pressures of 0.2 torr. Of course, liquid-ring pumps or other vacuum producers are required for the final stages in that application.

Disadvantages of process-vapor-powered systems are high capital costs for the vacuum-producing system, possible patent restrictions, and the possibility that prototype testing may be required if the manufacturer does not have experience with the specific motive fluid. Field tests may be required if the manufacturer's shop is not equipped to handle unusual materials. Thermal degradation and contamination of the motive fluid are additional problems emphasized by Ryans and Roper [2]. Manufacturers continue to accumulate experience, however, so the reliability of this design alternative continues to improve.

Estimates of the amount of motive fluid required to power a stage may be based upon the steam rate for such a stage, adjusted in proportion to the expansion energy available in a steam expansion and a process vapor expansion to the same suction pressure. This will probably be a conservative estimate, useful in testing the feasibility of a proposed design.

As an example, suppose that you wish to compress the load vapors from 5 torr to 50 torr using an unnamed organic vapor. From Fig. 4.8, obtain an R_a value of about 2.6 lb of 165-psia steam per pound of dry air. From Fig. A.l, the steam Mollier diagram in the Appendix, obtain an enthalpy of 1200 BTU/lbm for saturated 165-psia steam. Dropping vertically from that point to the 5-torr line to represent an ideal, isentropic expansion in the steam nozzle, obtain an enthalpy of 770. The ideal expansion energy is thus 1200 – 770 = 430 BTU/lbm for steam. Next, perform the same operation with a Mollier diagram of the organic vapor, or use the ideal gas laws described in Chap. 4 to calculate the ideal expansion energy for the organic vapor. Suppose, for example, that it is 250 BTU/lbm. A conservative estimate of the motive vapor requirement for that stage is thus

$$R_{a,pv} = 2.6\frac{430}{250} = 4.5 \text{ lb process vapor per pound of dry air}$$

Additional stages are treated the same way, and the condensers are calculated as described in Chap. 5. Note that water vapor is not present, so Fig. 5.15 is not applicable, because it is based on water vapor condensation. In general, the overall coefficients will be lower than with steam, the mass flows will be larger than with steam, and the total energy requirements may be less or more than with steam.

These methods are described here primarily to give you confidence in the application of process vapor to power ejectors, and to let you do quick checks of feasibility. You may prefer to let the manufacturer do all the calculations.

Be aware that if your application is outside the experience of the manufacturer and the industry, you may have some start-up problems. Also, you should be prepared for some learning experiences in which close coordination with the manufacturer will be required to arrive at the best design and operating procedures for your application.

Compressed air as motive fluid

In some special applications, compressed air is the preferred motive fluid for vacuum ejectors [6]. Air-operated ejectors are especially attractive when compressed air is readily available and steam is not, and where mechanical pumps are not desired. Multiple-stage ejectors have no intercondensers, because air is a noncondensable gas at normal temperatures and pressures. Thus air-jet ejectors are usually limited to two or three stages. Materials of construction should be suitable for the load gases and the motive air, noting that moisture may not normally be present if the motive air is warm. A discharge silencer is required.

Major disadvantages are the operating cost and the possible increase in air contamination over that produced by other vacuum pumps. All the motive air accompanies the process vapors to the discharge, making it more difficult to condense any vapor in an aftercooler to reduce air contamination. The air consumption may be estimated from Fig. 11.4. Add 50 percent for unheated motive air, and use Fig. 4.13 for a rough correction for pressures other than 100 psig.

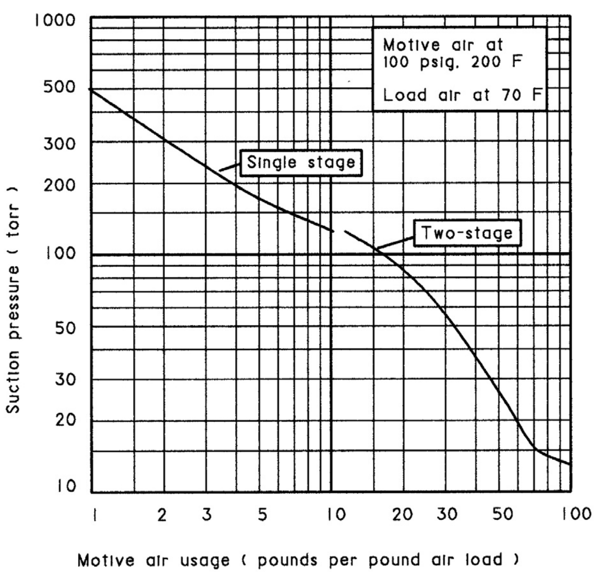


Figure 11.4 Air usage of air-operated ejector. *(Courtesy* Chemical Engineering*)*

Hybrid systems: steam-jet ejector with liquid-ring last stage(s)

The advantages and disadvantages of liquid-ring pumps are discussed in Chap. 9. The hybrid specification for liquid-ring pumps in Chap. 10 is so brief that you may wish to supplement it or replace it with your own specification if you have one. Possible general specifications you may add include specifications for motors, electrical package systems, instruments and controls, electrical grounding, and preferred supplier lists for electrical and instrument components.

For convenience in troubleshooting, locate pressure taps at the pump suction and discharge and arrange that the volumetric flow of the vent to the atmosphere can be conveniently measured.

11.3 Process, Miscellaneous

Injectors pump cold water into pressurized boilers

As described briefly in Chap. 2 and illustrated in Fig. 2.8, the injector is a device used to pump cool water into a boiler against the boiler pressure. It can lift cool water from 3 to 20 ft using 25- to 140-psig steam, or pump pressurized water with steam at pressures to 240 psig. These are representative values, and manufacturers' catalogs [7] may be consulted for proper application and sizing.

A process application for this device is to heat water and deliver it under pressure to pressure nozzles for pressure washing.

My first encounter with ejectors was a Penberthy injector on the steam boiler in my father's creamery plant in Montana in the 1940s. The boiler was salvaged from a retired farm threshing machine. I was allowed to operate the injector a few times.

Use exhaust heat or extra pressure in process

Where the heat given up by steam condensing in intercondensers or aftercondensers can be used in the process, steam-jet ejectors have a high practical efficiency [8]. Where steam is to be added directly to a process, the extra pressure energy in the steam may perform a useful pumping, circulating, or mixing function. Heaters are designed to inject the steam as a high-velocity central jet or annular jet which mixes with the liquid and condenses as small, discrete bubbles away from the walls of the ejector. This avoids the cavitation and noise associated with low-velocity mixing.

Many of these devices are available as preengineered designs, mass-produced and stocked for low price and quick delivery. Some flexibility is introduced in some product lines by the use of interchangeable nozzles and diffuser throats that adapt them to high or low steam or water pressure and different combinations of load and motive fluids. Because of these design compromises, these devices seldom approach the pumping efficiency of custom-designed equipment. Special modifications are sometimes available, but the extra price is accordingly high. If you are confident of your ability to modify a unit to meet your special requirements, you may wish to purchase a stock design with a small nozzle, for example, and modify the nozzle yourself. You save the money---and you take the risk.

The heaters are available in a large variety of sizes and designs for in-line and open tank applications, including plastic-lined and sanitary models designed for quick disassembly and cleaning. Sizing and application data are best obtained from the manufacturers' catalogs [9-13].

Steam-operated liquid pumps are in-line devices that usually resemble steam-jet gas ejectors. See the manufacturers' catalogs [11, 14-16] for sizing and application data.

11.4 Steam-Jet Refrigeration

Jet refrigeration systems are compression refrigeration systems in which the compression is performed by a jet ejector. The most common system uses steam-driven ejectors to chill a stream of water by flashing off water vapor in a vacuum system and condensing the load and motive water vapor in a condenser. The condenser typically is a water-cooled contact condenser or surface condenser, but may be an air-cooled [17] or evaporative-cooled condenser. Other fluids, including Freon refrigerants [18], have been evaluated.

Steam-jet refrigeration may be economically competitive with mechanical systems if steam is available at an unusually low cost or if the operation is infrequent. If excess steam is available in warm weather, this application can help correct the balance between steam production and power generation. Steam pressures as low as 2 psig or even zero psig can be used for the booster stages, with higher-pressure steam for the final stages. Flash cooling of bulk solids such as vegetables and sand or gravel for concrete mix may be performed in batch or continuous-feed operations. For example, vegetables are sprayed with water, which is flashed off under vacuum for quick, uniform chilling prior to storage or shipment. Water which is chilled by flashing is well deaerated, a desirable feature for many applications. In some circumstances the simplicity and reliability of steam-jet refrigeration are valued more than low operating costs. Often there is some design latitude for reducing operating costs by increasing the first cost.

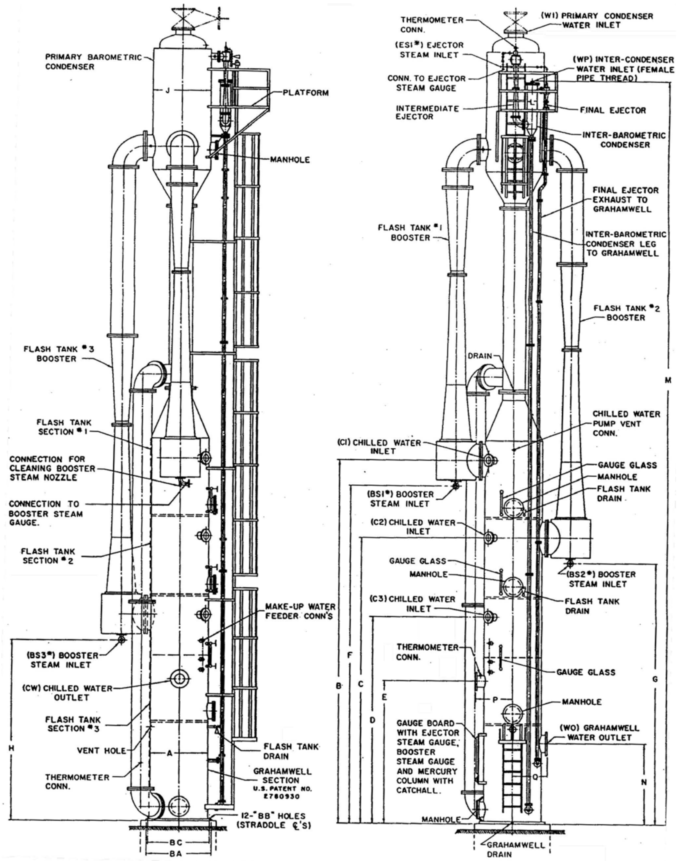
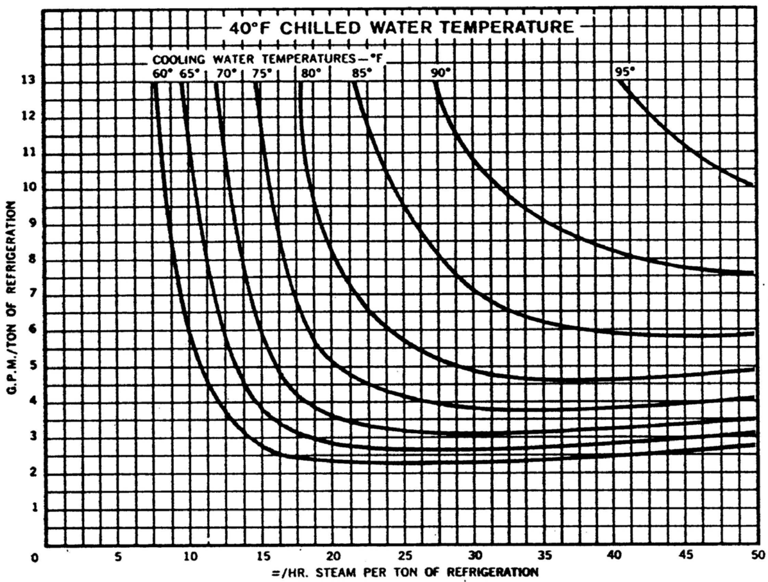


Figure 11.5 Packaged steam-jet refrigeration system, three boosters. *(Courtesy Graham Manufacturing Co., Inc.)*

The design and calculations are a straightforward application of the principles and data in Chaps. 4 and 5. The simplest configuration is a flash tank, booster stage, condenser, and small two-stage air ejector. The user supports them in a structure and provides the necessary circulation pump for the chilled water, liquid-level control, and interconnecting piping. Several manufacturers are prepared to supply complete preengineered, free-standing units as shown in Fig. 11.5, complete with access ladders and platforms. That example has three boosters, a subject which will be discussed shortly.

Quick estimates of steam and water usage

Quick estimates of steam and cooling-water requirements may be made by using the curves in Figs. 11.6 and 11.7, based on 100-psig motive steam and liberal rates, which include a two-stage air ejector



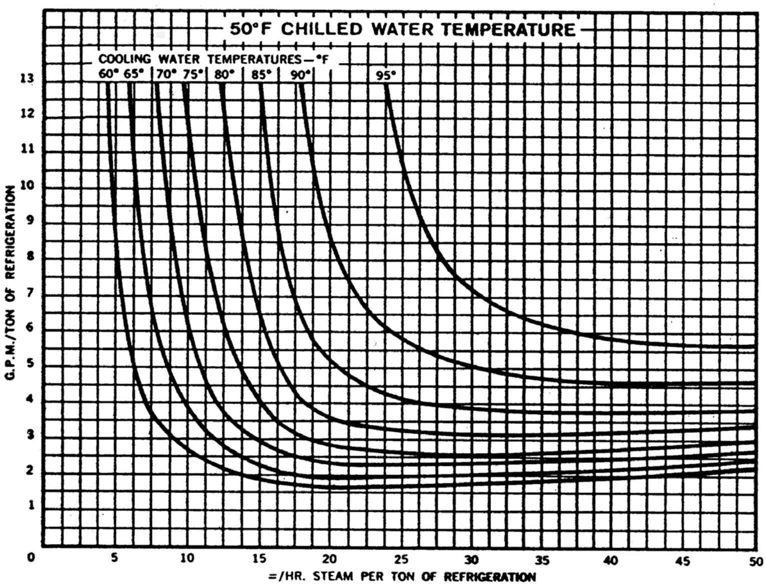
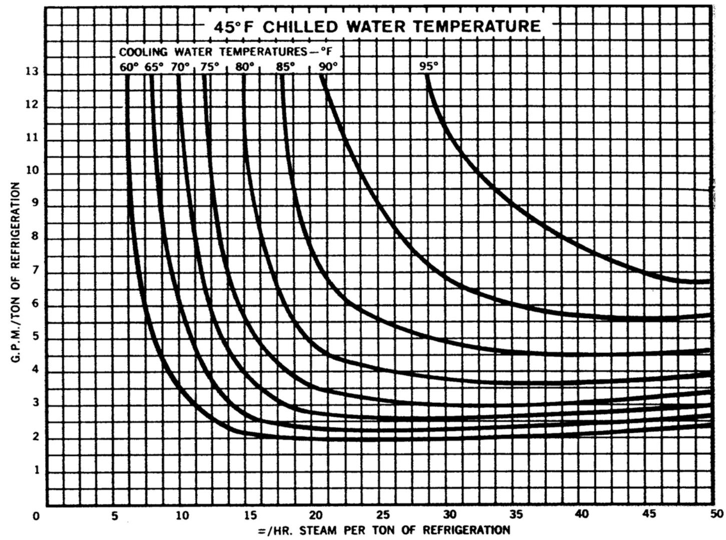


Figure 11.6 Steam-jet refrigeration condensing water usage vs. steam usage, 100-psig steam. (*Courtesy Croll-Reynolds Co., Inc.)*

after the large condenser, Figure 11.6 illustrates the design latitude in adjusting the relative amounts of steam and water for a given application, Figure 11.7 illustrates the benefits of multiple cooling stages, Use Fig, 4,13 to correct for other steam pressures, remembering to adjust the cooling-water rates in direct proportion also. Note



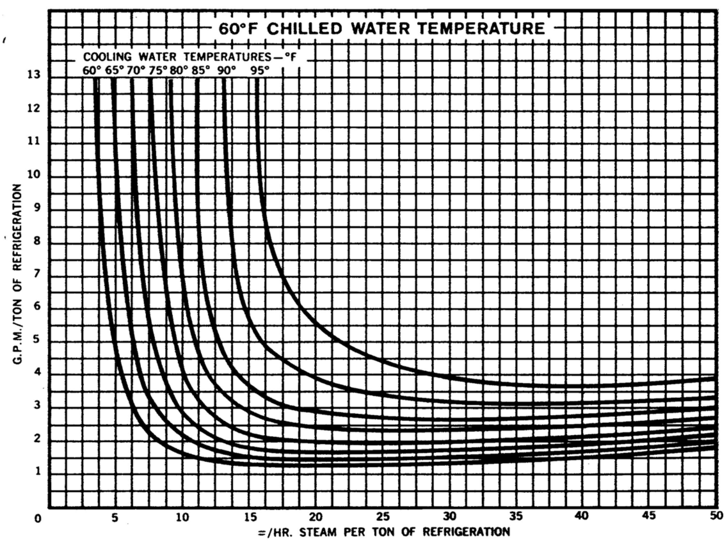
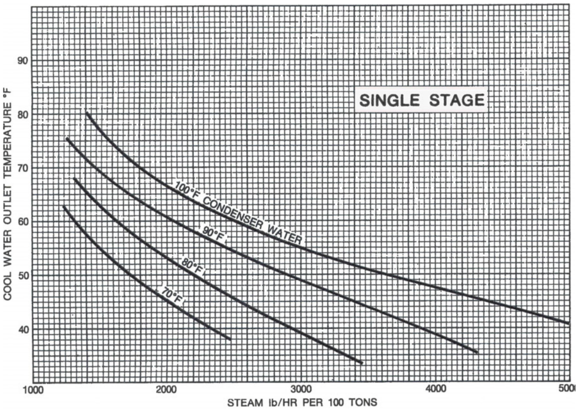


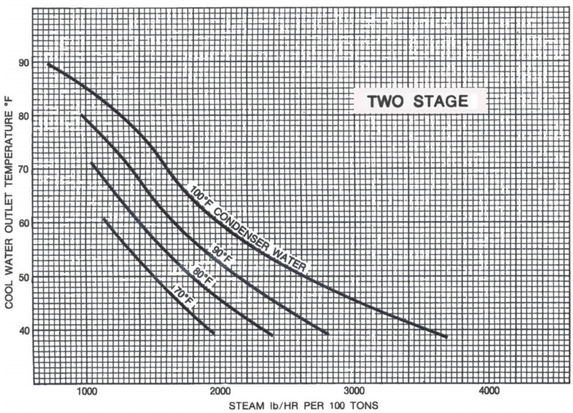
Figure 11.6 (Continued) Steam-jet refrigeration condensing water usage vs. steam usage, 100-psig steam. *(Courtesy CrollReynolds Co., Inc.)*

that the steam and water rates are based on tons of refrigeration, Freezing one ton of ice per day requires a refrigeration effect of 12,000 BTU/h.

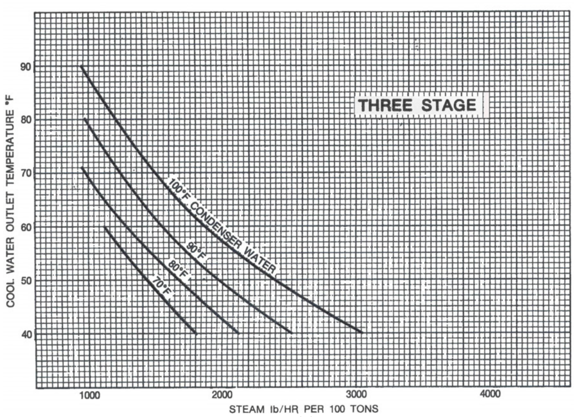
As an example, cool 50 gpm water from 80°F to 50°F using 150-psig steam and 90°F cooling water. Water has a specific heat of about 1.0,



(a)

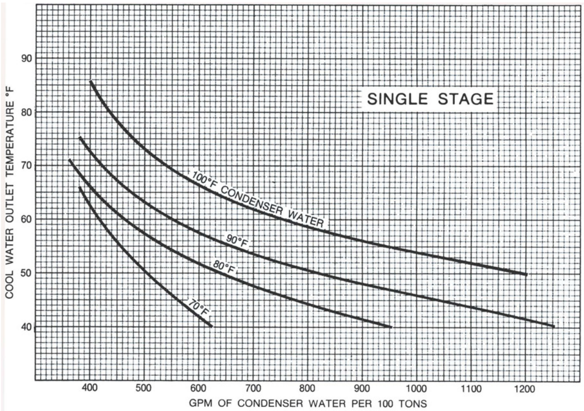


(b)

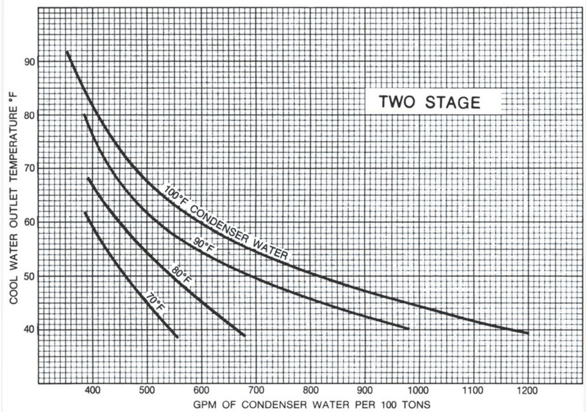


(c)

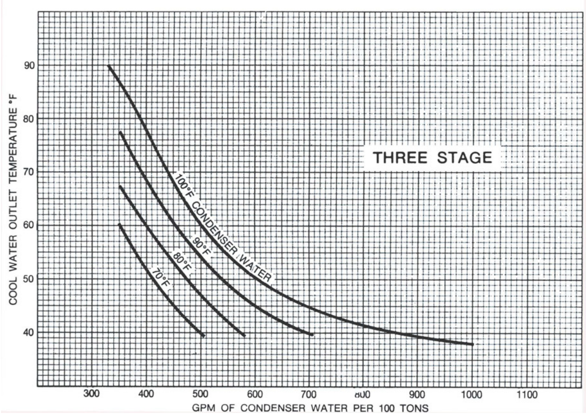
Figure 11.7 Steam-jet refrigeration steam usage, multiple-stage cooling. Contact condensers and two-stage air ejector included. 100 psig steam. *(Courtesy The Jet Vac Corp.)*



(d)



(e)



(f)

Figure 11.7 (Continued) Steam-jet refrigeration condenser water usage, multiple-stage cooling. Contact condensers and two-stage air ejector included. 100 psig steam. *(Courtesy The Jet Vac Corp.)*

The gross cooling load is

$$\text{Cooling} = (50\text{ gpm})(1.0)(80 - 50)\frac{500\text{ pph}}{\text{gpm water}} = 750{,}000\text{ BTU/h}$$
$$= 62.5\text{ tons}$$

From Fig. 11.7*a*, obtain 2900 pph steam per 100 tons, and from Fig. 11.7*b*, obtain 810 gpm water. From Fig. 4.13, obtain 0.94 as the multiplier for 150-psig steam. The estimates of steam and water are

$$\text{Steam} = \left(\frac{2900}{100}\right)(62.5)(0.94) = 1704\text{ pph}$$
$$\text{Water} = \left(\frac{810}{100}\right)(62.5)(0.94) = 476\text{ gpm}$$

To compare the utility rates in Figs. 11.6 and 11.7, find the steam and water rates per ton for 100-psig steam. The steam rate for this example is 29 pph per ton, and the water rate is 8.1 gpm/ton. Entering Fig. 11.6 with a chilled-water temperature of 50°F, cooling-water temperature of 90°F, and water rate of 8.1 gpm/ton, obtain a steam rate of 20 pph/ton. This is in substantial agreement with the other numbers offered as rough estimates for you to establish feasibility. Note that the cooling-water temperature is near the upper limit on the curves, remember that Fig. 11.6 is described as "liberal" and contains an allowance for the air ejector, and realize that differences in the assumptions behind the curves will result in differences larger than any differences in basic design data.

Detailed estimates of steam and water usage

Figure 11.8*a* shows the configuration of this simple system, annotated to identify important parameters. Return water at 80°F is introduced to the top of the flash tank; W_f, pph of water flashes into vapor at a pressure of P_f and is taken away by the booster. The chilled remainder of the liquid exits from the bottom of the flash tank at 50°F. The booster ejector is powered by W_s pph motive steam at 150 psig, which expands through the steam nozzle to the pressure (P_f) then compresses the water vapor to the condenser pressure P_c. The motive steam and water vapor combine in the ejector and emerge as W_c pph vapor to be condensed in the contact condenser. In the contact condenser, the condensing vapor gives up heat to cooling water, which enters at 90°F and leaves at T_o. The condensing temperature T_c is a few degrees above T_o and determines the condensing pressure P_c.

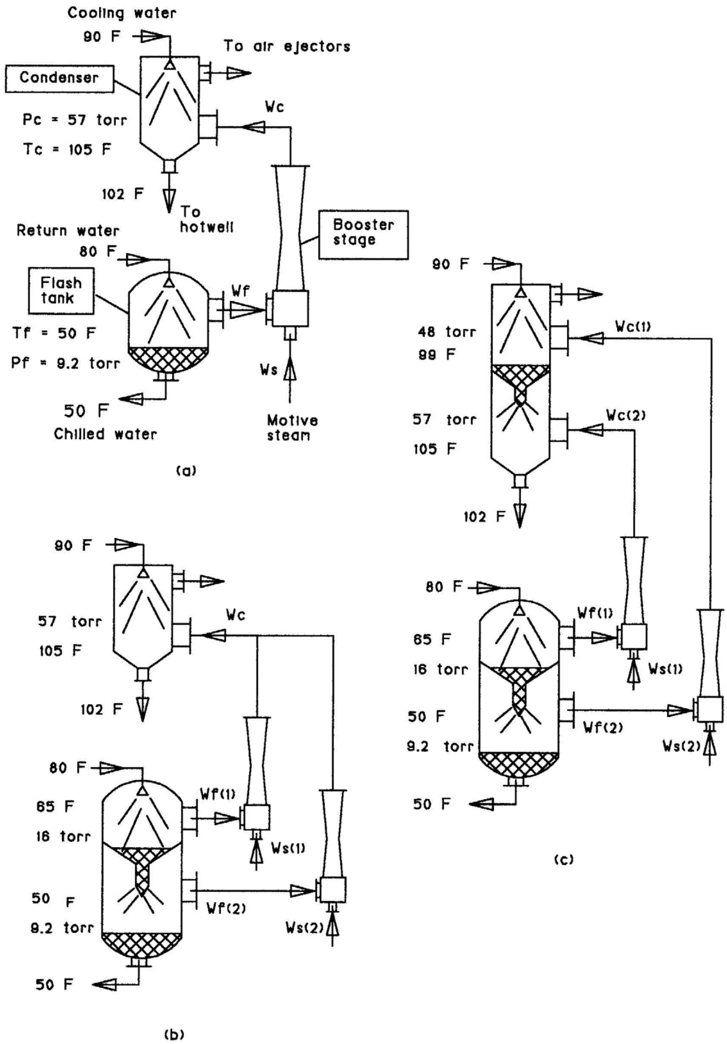


Figure 11.8 Steam-jet refrigeration configurations: (*a*) one flash stage, one condenser; (*b*) two flash stages, one condenser; (*c*) two flash stages, two condenser stages.

The saturation pressure of water at 50°F is 9.2 torr. The water enters the flash tank at 80°F, and it flashes to vapor at 50°F. From steam tables, obtain the enthalpy of the water entering the flash tank and the enthalpy of the vapor and the liquid leaving as 48.02, 1083.7, and 18.07 BTU/lbm. The fraction which flashes as vapor is

$$x = \frac{48.02 - 18.07}{1083.7 - 18.07} = 0.0281, \text{or } 2.81 \text{ percent}$$

The net refrigeration effect, based on the chilled water delivered, is (1.0 – 0.0281)lbm* (1.0) specific heat* (30) F° = 29.16 BTU/lbm. The flashed vapor flow is (0.0281/29.16)(12,000) = 11.6 lbm/ton. That corresponds to 11.3 lbm/ton of gross refrigeration effect. The difference between the two numbers is the result of adding warm makeup water to replace the vapor removed by flashing. Those numbers may be used for all steam-jet refrigeration calculations.

Continuing, the design procedure is to select a condenser pressure, calculate the steam required to compress the vapor to that pressure, then calculate the water required to condense the steam and water vapor. As design alternatives, several condenser pressures are selected, and the steam, water, and first cost are calculated. The design alternatives are compared to find the one that results in the lowest total evaluated cost.

To demonstrate that process, select a condenser pressure of 80 torr. The load vapor is 11.6(62.5) = 725 pph water vapor. The dry air equivalent is 725/0.8 = 906 pph DAE. At the suction pressure of 9.2 torr, Fig. 10.1 shows that a stage with a 16-in suction connection and a length of 13 ft is required. From Fig. 4.8, the steam rate for compressing from 9.2 to 80 torr is 2.5. The motive steam is W_s = 2.5(906) = 2266 pph (36 pph/ton). The total vapor load to be condensed is W_c = 725 + 2266 = 2991 pph. The saturation pressure corresponding to 80 torr is 117°F. Subtracting 3°F as a terminal approach difference yields a water outlet temperature T_o of 114°F. Thus, the water temperature rise is 24°F. Using a latent heat of 1100 BTU/lbm for the mixture of motive steam and cool vapor, the water flow is

$$\text{Water} = \frac{2991(1100)}{24(500)} = 274 \text{ gpm} \quad \text{or } 4.4 \text{ gpm/ton}$$

Next, select a lower condenser pressure to use less steam and more water. Selecting a water temperature rise of 12°F yields a water outlet temperature of 102°F, a condensing temperature of 105°F, and a corresponding condensing pressure of 57 torr. The steam usage is now W_s = 1.55(906) = 1404 pph (22.5 pph/ton). The steam condensed W_c = 725 + 1404 = 2129 pph. The water flow is

$$\text{Water} = \frac{2129(1100)}{12(1500)} = 390 \text{ gpm} \quad \text{or } 6.2 \text{ gpm/ton}$$

For a comparison with Fig. 11.6, divide these steam and water rates by 0.94 to obtain the rates for 100-psig steam, obtaining 24 pph/ton and 6.6 gpm/ton. Enter Fig. 11.6 with 50°F chilled water, 90°F cooling water, and 24 pph/ton, obtaining a water rate of 6.0 gpm/ton. The agreement within 10 percent is probably typical at the "knee" of the curves. The differences may be greater at the extremes of the curves.

Multistage flash. Steam usage may be reduced by cooling the water in a series of flash tanks operating at successively lower pressures, as illustrated in Fig. 11.8*b*. The vapor removed in the higher-pressure stages requires less motive steam to compress it. Staging is often attractive if the water temperature drops more than 10°F.

In this example, consider the result of taking half the temperature drop in each of two flash stages. The outlet temperature of stage 1 is 65°F, and its pressure is 16 torr. Half the vapor will flash in this stage; 725/2 = 363 pph. The dry air load is 906/2 = 453 pph DAE. Steam usage in stage 1 (16 to 57 torr) is 1.0(453) = 453 pph. Steam usage in stage 2 is half the previous rate, 1404/2 = 702 pph. Total steam is now 453 + 702 = 1155 pph. Vapor condensed W_c = 725 + 1155 = 1880 pph. Water usage is

$$\text{Water} = \frac{1880(1100)}{12(500)} = 345\ \text{gpm} \qquad \text{or } 5.5\ \text{gpm/ton}$$

Thus, at the same condensing pressure, steam usage has been reduced 18 percent, and water usage has been reduced 12 percent. The price has increased because two half-capacity stages replace the single stage, and the flash tank now has two compartments and more complex nozzles and flow controls.

Multistage condensers. Steam and water usage may be further reduced by performing the vapor condensation in stages, operating at successively higher pressures as the water warms. The vapor condensed in the lower-pressure stages requires less motive steam to compress it. As Fig. 11.8*c* shows, this example calculation has been extended to a final configuration of two flash stages and two condenser stages. Note that the flashed vapor from the top flash tank is compressed to the bottom condenser, and that from the lower flash tank to the top condenser. That roughly equalizes the compression ratios.

For a preliminary calculation of this configuration, assume that half the cooling-water rise occurs in the first condenser, yielding an outlet temperature there of 96°F, a condensing temperature of 99°F, and a condensing pressure of 48 torr. The first booster compresses from 16 torr to 57 torr as before, using 453 pph steam. The second booster now compresses from 9.2 to 48 torr, using 1.33(453) = 602 pph, a saving of 100 pph. The total steam usage is now 453 + 602 = 1055 pph, and the total condensed is now 725 + 1055 = 1788 pph. The total cooling-water usage is

$$\text{Water} = \frac{1780(1100)}{12(500)} = 326\ \text{gpm} \qquad \text{or } 5.2\ \text{gpm/ton}$$

Checking the water temperature rise in the first condenser, the total vapor condensed is 363 + 602 = 965 pph. The temperature rise is

$$\text{Water rise} = \frac{965(1100)}{326(500)} = 6.5\ °\text{F} \quad \text{(versus 6.0 assumed)}$$

Configuration choice and design optimization. A two-stage flash and two-stage condenser configuration represents or approaches the economic optimum design of these systems. In a stacked configuration such as Fig. 11.5, the overall height may exceed the height which cooling water can reach. Configurations other than the stacked arrangement are limited only by the imagination. It is convenient for multiple flash stages and multiple contact condensers to be stacked, so that the water can flow by gravity from stage to stage. Contact condensers can lift water a short distance from a cooling tower, eliminating a pump at that location.

Because the booster stages typically have a compression ratio greater than 2:1, their capacity cannot be altered by changing the motive steam pressure. They are simply turned on and off to control the chilled-water temperature, relying on mixing in the cold-water receiver to smooth out the temperature swings.

The final choice of number of flash stages and condensers depends upon the size of the unit, the investment equivalents assigned to steam and water, the steam and cooling-water design conditions, and the turndown method and temperature control method. Factors which lead toward more stages of flash and condensation are high utilities costs, low steam pressure, high cooling-water temperature, large refrigeration load, low chilled-water temperature, and high chilled-water return temperature.

Fine-tuning of designs involves shifting the load among the flash and condenser stages for sizing the vessels and meeting user requirements for turndown and control. Sizing the air ejectors is a matter of judgment and experience, strongly influenced by the presence of non-condensables in the steam or dissolved in the cooling water.

Significant reductions in steam costs are possible in situations where normal steam pressure is well above the design value and where cooling-water temperatures are frequently lower than design. Consult your manufacturer to determine the possible savings and the procedure for achieving them. A steam-pressure controller may adjust the motive steam pressure to the lowest value appropriate given the current water temperature or the current condenser pressure. Remember that this steam economy practice applies only to the booster stages. The air ejectors must always be supplied with steam at or above their design pressure.

Performance curves

How much refrigeration effect will a steam-jet system produce when the return water temperature is higher or lower than design, or when the circulation rate is higher or lower? What happens if one or more boosters are turned off?

To answer these questions, it is useful to have system performance curves. One method of preparing and using performance curves is described here. The ejector performance curve shapes were discussed in Chaps. 3 and 4. The interaction between the ejector and the process system was shown in Chap. 6 to depend in part on the system curve. The operating point was shown to be at the intersection of the system curve and the ejector curve. That is also true here, with two differences. Control of steam-jet refrigeration systems is by turning booster stages on and off, with no throttling or addition of condensable or noncondensable load to the ejector, and the system curve is a simple function of the chilled-water flow rate and return temperature.

A simple system curve is shown in Fig. 11.9*a*. The vertical scale here is temperature, which is directly related to absolute pressure, but more convenient for plotting and calculations. The horizontal scale is the load, expressed as tons of refrigeration effect. It could as easily be expressed as BTU/h or as pph flashed vapor, whichever is most convenient for you. The system curve is shown as a straight line here, based on the simplifying assumptions that the latent heat of vaporization is a constant 1060 BTU/lbm and that the vapor lost by flashing is negligible. The slope of the line has units of temperature/load. The temperature change is

$$\text{Temperature change, °F} = \frac{\text{Q}(12000)}{G(1.0)(500)} = 24Q/G \qquad (11.1)$$

where Q is in tons and G is in gpm, with a slope of $24/G$ in units of (°F/ton · gpm).

This leads us to Fig. 11.9*b*, a family of system curves with a common starting temperature and different slopes representing different circulation rates. The starting temperature represents the chilled water return temperature. The set of curves may be shifted up and down to represent different return temperatures.

In Fig. 11.9*c* we have added a single booster curve, also plotted with units of temperature versus load. Unlike the water curves, this curve is fixed. It cannot be shifted up or down. Individual boosters can only be turned on and off. The intersections of the water curves with the booster indicate the gross refrigeration effect and the chilledwater supply temperature for each particular combination of return temperature, water circulation rate, and booster. The 50 percent

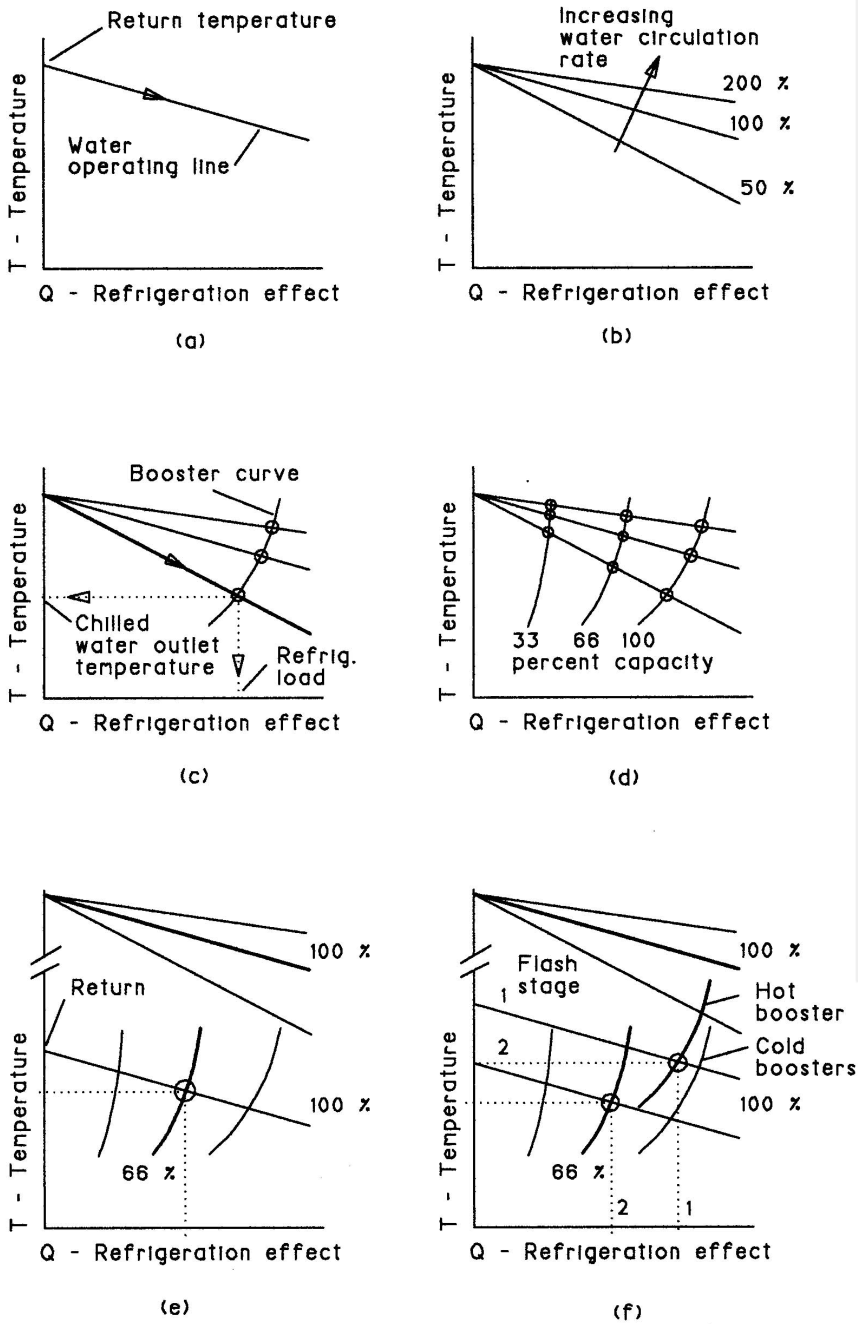


Figure 11.9 Performance curves for steam-jet refrigeration: (*a*) chilled-water cooling curve; (*b*) effect of water flow rate; (*c*) operating points, one booster; (*d*) operating points, two boosters; (*e*) variable return temperature; (*f*) multiple flash stages.

water circulation rate is used to illustrate how to find the chilled-water outlet temperature and the refrigeration load.

In Fig. 11.9*d* the single booster is replaced by two boosters with capacities of 33 percent and 66 percent of design capacity for improved operating flexibility. The three booster curves represent operation with the small booster, the large booster, and both boosters. Observe that the three circulation rates and the three booster curves define nine possible operating points. Interpolation is appropriate to describe intermediate circulation rates. This is looking good, but no provision has been made yet for the effect of varying return temperature.

Figure 11.9*e* further complicates the diagram to simplify its use. The water lines have been moved up to an unscaled portion of the vertical axis, remaining only to provide the proper slope of the liquid operating line. As an example of its use, we find the capacity for a given combination of water circulation rate (100 percent), return temperature, and active boosters (66 percent). Locate the return temperature on the vertical scale. Through that point, draw a liquid operating line parallel to the appropriate circulation reference line above (100 percent here). The intersection of the liquid operating line with the selected booster capacity (66 percent here) yields the refrigeration effect and the chilled-water supply temperature.

Finally, we consider multistage flash designs. As shown in Fig. 11.9*f*, this is not a big step from what we have already done. The curves have been modified to represent a two-stage flash design in which the load is split evenly between the first and second stages, and in which the second stage has the two boosters we have been working with. If the first- (hot-) stage booster is operating, draw the water line as shown to its intersection with that booster, noting the refrigeration load and the temperature of the water leaving that stage. That is also the temperature at which water enters the second (cold) stage, so draw the operating line as shown to the curve representing the active combination of cold boosters (66 percent here). Note the refrigeration effect and water outlet temperature of the second stage. The total refrigeration effect is the sum of the effects in the two stages.

By preparing multiple copies of diagrams similar to Fig. 11.9*f*, one may evaluate many combinations of operating conditions. A simple variation in the procedure is to create a transparent overlay containing the water curves, then lay it over the booster curves to find the operating point for the first flash stage. That operating point is marked on the transparency with a grease pencil. The overlay is then shifted to the left so that the operating point for stage 1 is on the vertical scale, becoming the entering temperature for stage 2, and so forth. The transparency may be cleaned and reused.

Some hydraulic details

Usually only one booster is connected to each flash stage. If more than one are connected to a flash compartment, then all must be turned on and off together. Otherwise, vapor will bypass from the condenser to the flash compartment through the idled booster.

Figure 11.10 shows one all-hydraulic method of handling the distribution of the chilled-water stream to multicompartment flash stages. Two flash compartments are shown in this example. The compartments are separated by a partition that extends down into the common discharge surge tank. Above the flash compartments is a flowdirection chamber that is divided into two flow-switching compartments. The compartments are separated by a vertical partition that extends down into the water, but permits unrestricted flow of water below the compartments. Each flow-switching compartment has a pressure-equalizing pipe and a water feeder pipe that communicate directly with a flash compartment below. The water feeder pipe has a spray nozzle on its lower end. All water feeder pipes have the same elevation at the top.

The operating principle is that the pressure reduction in a flash compartment when its booster operates causes the water to preferentially flow into that compartment. The pressure in each flash compartment is about 10 to 20 torr when the booster is operating, but becomes 40 to 60 torr when the booster is not operating. The pressure-equalizing pipe communicates that pressure to the vapor space in the flow-switching compartment above.

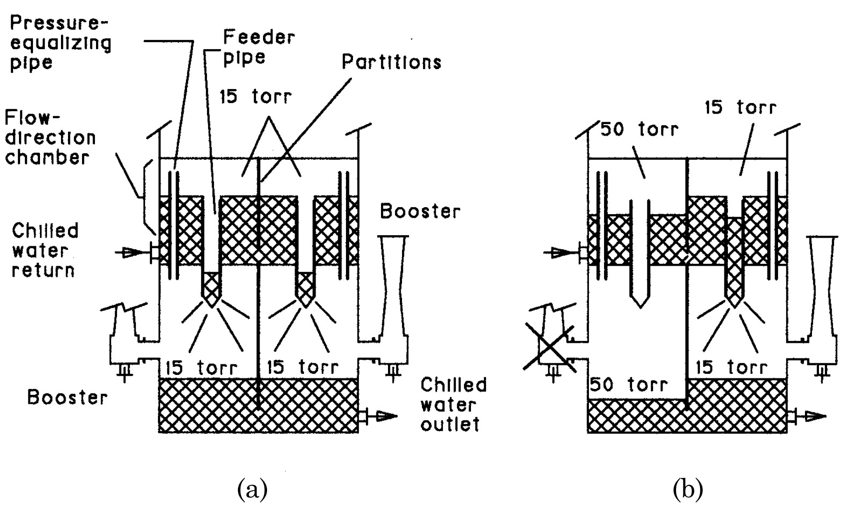


Figure 11.10 Automatic water flow directors: (*a*) all boosters on; (*b*) one booster idled. *(Adapted from sketches, courtesy Graham Manufacturing Co., Inc.)*

If both boosters are on or both boosters are off, the water levels on both sides of the partition in the flow-direction chamber will be the same, as shown in Fig. 11. l0*a*. The flow will divide equally into the feeder pipes and spray into the flash compartments below.

If one booster is off and one booster is on, however, the resultant difference in pressures in the vapor spaces will depress the water level above the inactive flash compartment, as shown in Fig. 11.l0*b*. A differential pressure of 20 torr will result in a water-level difference of 10 in of water. As a result, all the water will flow through the active compartment.

The principle may be extended to three or more flash compartments. Although the example showed a side-by-side arrangement of flash compartments operating in parallel, the principle may also be extended to parallel operation in a stacked configuration by extending the feeder pipes and draining each compartment to a common discharge reservoir. For a series/parallel configuration, the flow-directing chamber may be located in the bottom of an upper flash tank.

The lower condensers in a multiple-condenser configuration must be vented to prevent the accumulation of noncondensable gases. A small pipe from the top of the lower condenser into the vapor space of the condenser above it will accomplish that.

Because the hydraulic switching device is subject to fouling, the manufacturer may prefer to perform the flow switching with external valves, automatically controlled.

11.5 Steam-Jet and Gas-Jet Compressors

These are ejectors that discharge at pressures that are often higher than atmospheric pressure. When high-pressure steam is used to compress low-pressure steam to obtain a desirable thermal effect, the ejector is commonly called a thermocompressor, or sometimes thermal compressor. The operating principles, however, are the same as for the steam-jet or air-jet vacuum ejector.

Application differences are that many thermocompressors have a compression ratio less than 2:1, and thus have more complicated performance curves and permit some capacity control by throttling the motive gas or using a variable-area motive nozzle. In some applications, very high pressure designs utilize the energy present in gases extracted from underground storage.

Although their most common application is to compress low-pressure steam to a usable pressure level, these devices may be used wherever a vapor or gas pumping effect is desired and where the high-pressure motive fluid is already being used or where its pro-posed use is economical. One process-integrated application is to com-press flashed vapor from one portion of a process and deliver it at a higher pressure to another application, performing separate but related cooling and heating effects in one operation. When low-pressure steam is compressed to a usable pressure instead of being vented to the atmosphere, it reclaims waste energy and reduces the nuisance aspects of vented steam. I once helped design and expedite delivery of a "steam stuffer" ejector that used 400-psig steam to boost 185-psig steam and deliver 400,000 pph of 200-psig steam to a marginally sized steam header.

Typically, these ejectors are single-stage units resembling the three shown in Fig. 11.11. Figure 11.11*a* is the simplest of the three, designed for moderate pressures. Figure 11.11*b* is more complex, using an actuator to position a spindle in the nozzle for capacity control. The construction appears to be suitable for higher pressures. Figure 11.11*c* is a multiple-nozzle ejector, of a larger size and low pressure. The .multiple-nozzle design is claimed to have a higher efficiency, leading to reduced steam requirements.

Common variations include heavier construction for high pressures and steam jacketing of nozzles and diffusers for gas-jet applications to prevent ice or hydrate formation.

Once the device is built, the control range for adjustments of indi-

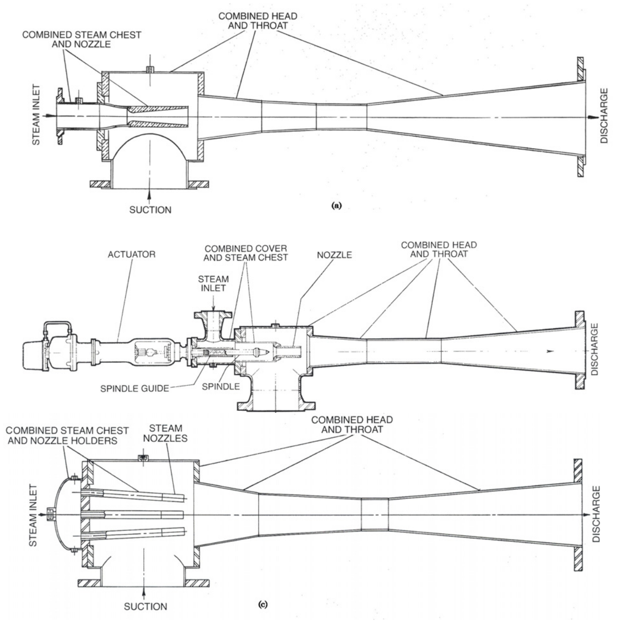


Figure 11.11 Steam-jet and gas-jet ejector: (*a*) single-nozzle; (*b*) variable-area (spindle) nozzle; (*c*) multiple-nozzle. *(Courtesy Croll-Reynolds Co., Inc.)*

vidual units is limited [19-23]. Multiple elements are the most effective way to maintain efficiency over a wide operating range.

Performance estimates

When a potential application is being evaluated, the first requirement is to estimate the performance by answering questions such as, how much high-pressure steam is required? or, what suction pressure or discharge pressure can be created? If the first estimates establish the general feasibility, then design attention is focused on maximizing the effectiveness.

For quick estimates, you must know the pressures of the motive gas and the suction and discharge; the desired mass flow of motive, suction, or discharge gas; and the temperature and molecular weight of both streams.

The manufacturer will also want to know the specific heats and specific heat ratios of the gases, plus the compressibility factors or critical temperatures and pressures if the gases are not common.

Figure 11.12 is extremely useful for estimating the design-point performance of steam-jet compressors. It is based on published data [19-23] that have been smoothed considerably. It is generally accurate to within 20 percent over the range of compression ratios up to 5, expansion ratios up to 1000, and motive/load ratios from 0.25 to 5. It is useful for screening proposed applications for general feasibility and for rough estimates of the economics of feasible applications. A more detailed calculation may then be made to fine-tune the estimate and size the equipment.

Example calculations

It is desired to use 150-psig steam to compress steam from 20 psia to 40 psia, delivering 20,000 pph. Both steam sources are dry and saturated at the inlet conditions. Determine the steam flow rates and size of the unit.

The expansion ratio is (150 + 14.7)/20 = 8.2. The compression ratio is 40/20 = 2.0. Entering Fig. 11.12 with these values, obtain R_s = 1.7 lb of motive steam per pound of load steam. The motive steam requirement is [1.7/(1 + 1.7)](20,000) = 12,600 pph. The load steam is 20,000–12,600 = 7400 pph. Thermocompressor stages should be sized at the suction and discharge. Commercial units often have equal-size connections. Considering both the molecular weight and common steam temperatures, use a factor of 1.33 to convert to DAE for using the sizing charts, Figs. 10.1 and 10.2. Also, remember that 1.0 psia = 51.7 torr.

The suction DAE is 7400(1.33) = 9842 pph; the size factor is 9842/[(20)(51.7)] = 9.5; and the size is 6 in. The discharge DAE is

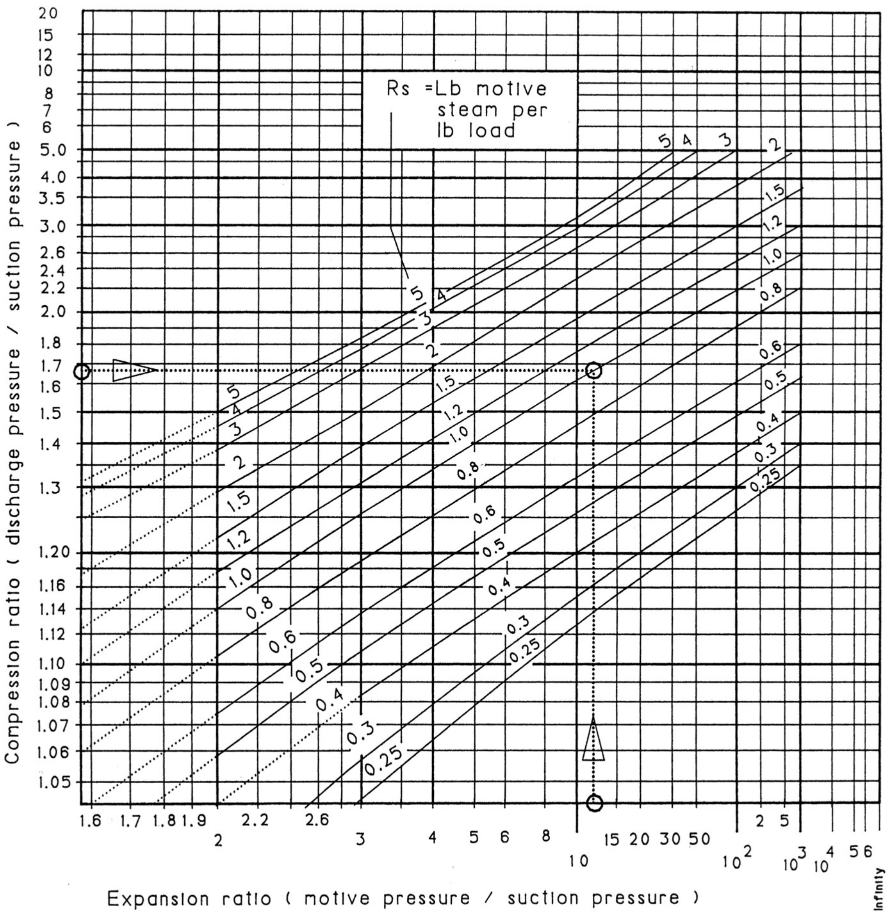


Figure 11.12 Steam rates for thermocompressors and ejectors. Motive steam and load steam are saturated. Example: Use 150-psig steam to compress load steam from -1 psig to 8 psig. Expansion ratio = 164.7/13.7 = 12; Compression ratio = 22.7/13.7 = 1.66. From chart, R_S = 1.0 lb motive per pound load.

12,600(1.33) = 16,758 pph; the size factor is 16,758/[(40)(51.7)] = 8.1; and that size is also 6 in. The length is 5 ft, the weight is 500 lb, and the 1992 price is $12,000 U.S. As noted on Fig. 10.1, the sizing velocity may be as low as 50 ft/s, which would make this a 10-in or 12-in unit. Low velocities are used with compression ratios less than 1.2.

If the process and cost numbers make this application look attractive, try different combinations of flow and pressures to find the one that is best for your application. Then do a detailed calculation to confirm the performance. An adequate check involves using a theoretical

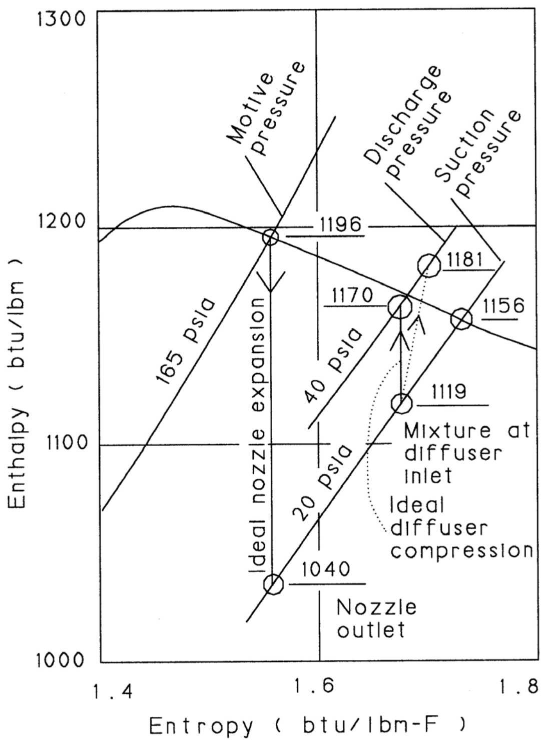


Figure 11.13 Numerical example of Mollier chart use for ejector calculations.

model described in Chap. 4. Briefly, the nozzle expansion is ideal (isentropic), the motive and load gases mix at the suction pressure with no loss of momentum (negligible wall friction), and the remaining kinetic energy of the mixture is converted to pressure with a diffuser efficiency of 60 to 90 percent. The energy values may be calculated using ideal gas laws or steam tables and charts. For this example, I will use a Mollier chart.

Consult a Mollier chart to obtain enthalpy data. Figure A.l in the Appendix is a Mollier chart which shows the range of interest for ejector calculations, although it is not intended for precise calculations. Figure 11.13 illustrates the use of the Mollier chart for the example calculations following. The enthalpy of saturated motive steam is 1196 BTU/lbm, and the suction steam enthalpy is 1156 BTU/lbm. The motive steam expands isentropically (along a vertical line) from the starting point down to the 20-psia pressure line, where it has an enthalpy of 1040 BTU/lbm. The kinetic energy available at the nozzle outlet is 1196 - 1040 = 156 BTU/lbm of motive steam. After mixing, the kinetic energy of the mixture is

$$\mathrm{KE}_{\mathrm{mix}} = \frac{\mathrm{KE}_{\mathrm{nozzle}}}{(1 + 1/R_s)^2} \tag{11.2}$$

$$= \frac{156}{(1+1/1.7)^2} = 62 \text{ BTU per pound of mixture}$$

Next, calculate the enthalpy of the mixture emerging from the diffuser. Kinetic energy is again zero, and the overall process is assumed to be adiabatic, so the enthalpy at the discharge is

$$h_{discharge} = \frac{R_s(h_{motive}) + h_{suction}}{R_s + 1} \tag{11.3}$$

$$= \frac{1.7(1196) + 1156}{1.7 + 1} = 1181 \text{ BTU/lbm}$$

Diffuser efficiency. The diffuser efficiency is defined as the ratio of the ideal (isentropic) enthalpy increase during compression divided by this actual energy available.

$$E_d = \frac{\text{ideal enthalpy increase}}{KE_{mix}} \tag{11.4}$$

The enthalpy of the mixture entering the diffuser equals the discharge enthalpy minus the kinetic energy of the mixture, 1181 – 62 = 1119 BTU/lbm. To determine the ideal (isentropic) enthalpy increase during compression, locate the point on the Mollier diagram where the 1118-BTU/lbm enthalpy line crosses the 20-psia suction-pressure line, then proceed up (isentropically) to the 40-psia line, where the enthalpy is 1170 BTU/lbm. Thus, the ideal compression work is 1170 – 1119 = 51 BTU/lbm. Finally, the diffuser efficiency E_d = 51/62 = 82 percent. Is this reasonable? Consider the subject of diffuser efficiency.

Shapiro [24] shows the best diffuser efficiency to decrease with increasing Mach numbers for supersonic diffusers. Following that lead, Fig. 11.14 shows a range of diffuser efficiencies plotted versus mixture kinetic energy entering the diffuser, a parameter available during these calculations. The "high" and "low" curves enclose most of the single-nozzle data sets I examined. Multiple-nozzle efficiencies may be higher. This figure should be regarded as preliminary, subject to improvement.

As a matter of interest, I discovered some catalog data points which showed indicated efficiencies exceeding the theoretical limit – diffuser efficiency exceeded 100 percent! That has probably embarrassed that manufacturer. It is also an object lesson for all of us that anyone can make mistakes, and some simple checking on your part can prevent your buying some mistakes.

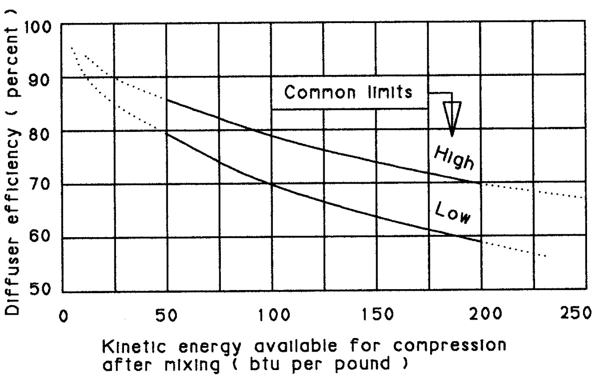


Figure 11.14 Diffuser efficiency versus kinetic energy. Bases: single-nozzle, ideal nozzle expansion, mixing at constant pressure with conservation of momentum.

In defense of manufacturers (since I just poked fun at one), they have mixed incentives as they decide what data to publish. Some of the useful design data are available to anyone: water vapor pressure, specific heat, and latent heat. Other data begin to disclose the manufacturer's individual practices: design guidelines such as approach temperatures in condensers, choice of single versus multiple nozzles, and stage stability criteria. And finally, some data collectively represent any technological edge the manufacturer may have: details of nozzle and diffuser design and nozzle placement, best steam rates or diffuser efficiencies, highly effective condenser designs, and costs.

If manufacturers publish their best steam rates, they may have trouble meeting that performance on all applications, and they will encourage their competitors to do more research to catch up. If they publish low efficiency data, they may be viewed as technically deficient, and may discourage users from some potential ejector applications. They don't want to be constantly defending their published data, and they do not want to be the source of flagrant errors.

With that background in mind, look at Fig. 11.14 and decide whether a diffuser efficiency of 82 percent is reasonable with a mixture kinetic energy of 62 BTU/lbm. The efficiency range for that kinetic energy is 76 to 84 percent. If we want to have more confidence in the design we are proposing, we may increase the amount of steam. A little change in the steam rate makes a big change in the efficiency, so let us simply increase R_s from 1.7 to 1.8 and repeat the calculations.

The mixture kinetic energy is now 156/(1 + 1/1.8)2 = 64 BTU/lbm. Discharge enthalpy is [1.8(1196) + 1156]/(1.8 + 1) = 1182 BTU/lbm. The mixture enthalpy entering the diffuser is 1182 − 64 = 1118

BTU/lbm. The ideal compression work is 1168 – 1118= 50 BTU/lbm. E_d = 50/64 = 78 percent. This falls well within the high/low band and is more likely to be confirmed by ejector manufacturers.

If you have checked the example calculations and traced the processes on a large Mollier chart, you will appreciate the contributions to error made by roundoff and chart reading uncertainties.

The choice of methods for estimating steam rates is sometimes arbitrary-whether one uses the stage steam rates in Chap. 4 or the methods described here. I lean toward using these methods and confirming with detailed calculations. When exploring an application that has no comforting precedent, the detailed calculations provide a welcome reality check. Your discussions with manufacturers and your specifications will be improved by your grasp of the subject.

Thermocompressor and gas-jet ejector applications

One simple guideline for identifying potential applications is to be alert for any application in which a high-pressure fluid is introduced at a pressure higher than the minimum which would be acceptable. Then look for a useful pumping effect which could be accomplished by a jet compressor at that location [25]. Sometimes the economic benefits are very large.

Thermocompressor heating example. As an example, suppose that a process requires heating by steam at a pressure not lower than 75 psig, and steam is available in the area at 50 psig and 200 psig. The normal practice would be to use the 200-psig steam and let the extra temperature be throttled away or used to reduce the heater area. As Fig. 11.12 quickly shows, an alternative is to install a thermocompressor using 1.3 lb of 200-psig steam to compress each pound of 50-psig steam, delivering 2.3 lb at 75 psig. This reduces the use of 200-psig steam by 43 percent, substituting low-cost 50-psig steam.

If there is no nearby source of low-pressure steam, look for a water stream that must be cooled and from which flashed vapor can be combined with the steam to produce a useful effect. The vapor flashing calculations and design methods are described in the subject of steam-jet refrigeration in this chapter.

Flash cooling of process liquids. One such application is a hot-water solution containing a nonvolatile solute. By flashing the water vapor in a series of flash chambers, the compression work is minimized and the steam economy is maximized. The configuration is shown

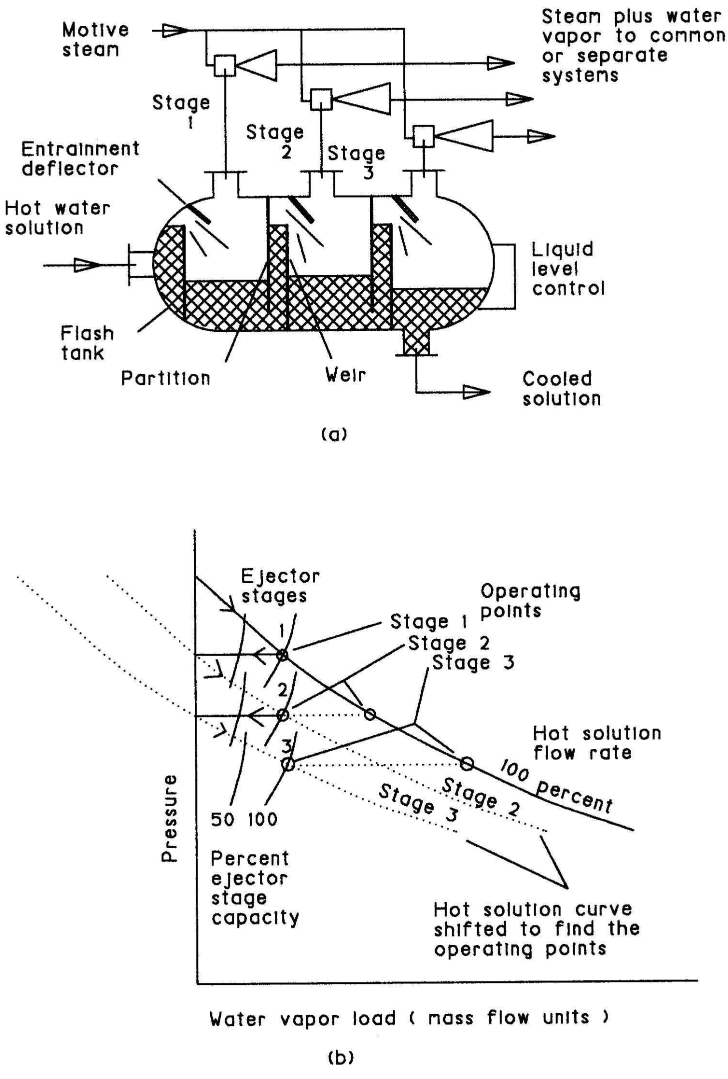


Figure 11.15 Process flash cooling and vapor recompression: *(a)* flash chambers and ejectors; *(b)* performance curves.

schematically in Fig. 11.15*a*. For simplicity, three flash stages are shown, with one ejector per flash stage.

In this application the pressure differential between stages is large enough that it will pump the liquid through the seal and weir between stages. Thus, the stages may be side by side in a horizontal tank. The size and configuration of the weirs and vapor outlet are chosen to minimize entrainment of liquid droplets. The liquid operating line is shown curved here, reflecting the effect of the change in solute concentration. The methods for calculating these curves will be familiar to people familiar with the process. The performance curves here plot pressure versus water vapor mass flow, a preference of the operator. The ejector curves show multiple ejectors for each stage for capacity control.

Observe the similarities and differences between this operating line and that used in Fig. 11.9. Because this liquid line cannot be approximated accurately by a straight line, it cannot be shifted up and down for graphic construction convenience. Instead, it is shifted horizontally so that the liquid exiting one stage becomes the feed to the next stage. Transparent overlays may be useful, especially if multiple liquid rates and varying feed conditions are to be evaluated.

Process-integrated thermocompressors. One example of a complex thermocompressor application [26] illustrates the energy-saving benefits which can be attained by a well-integrated use of thermocompressors. A sketch would describe it more completely, but I would rather leave you with a fuzzy picture which you will tend to interpret as potential applications in your processes. All the basic theory and application concepts have been covered previously in this chapter.

The flow diagram begins with a four-stage flash system which cools a liquid stream from 115°C to 60°C, yielding flashed low-pressure steam at pressures of 1010 mbar, 590 mbar, 312 mbar, and 195 mbar (1 mbar = 0.75 torr). The first three streams of flashed steam are used as motive steam for three thermocompressors.

The three thermocompressors chill a second liquid stream from 18°C to 12°C in a three-stage flash system at pressures of 18, 16, and 14 mbar.

The discharge vapors from the three thermocompressors are used to heat a third stream from 26°C to 32°C in a three-stage contact condenser at pressures of 38, 42, and 46 mbar.

And *finally*, some of the 32°C liquid is sent back to condense the vapor from the last flash stage in the first operation. It is complicated, but you have the tools to evaluate it accurately enough to establish the feasibility, perform a preliminary heat and material balance, adjust the design to achieve the best economy, and estimate sizes and prices.

Gas-jet compressor calculations. Gas-jet compressor performance predictions may be found in a manufacturer's catalog [22] if it happens to have your gas combination and to match the other specifications of your application. If not, use Figs. 11.12 and 11.14 with some modifications and cautions.

The load gas should be converted to equivalent motive gas, using the molecular weight entrainment ratio curve, Fig. 4.6. Although the data in Fig. 4.6 were obtained in vacuum tests, it is common practice to use the curve for pressures above atmospheric pressure. To avoid correcting in the wrong direction, remember that high-density gases are easier to pump than low-density gases. Temperature effects are usually secondary, but if motive and load temperature ratios are significantly different from the corresponding values for steam, then Fig. 11.12 may be less accurate. Other factors to consider are the gas compressibility factor (departure from ideal gas behavior) and the low specific heat ratios characteristic of high-molecular-weight gases.

Mollier charts for the motive gas are highly recommended. You will soon develop judgment in adjusting the results from Fig. 11.12 for your gases. Discuss your application with a manufacturer before you spend a lot of time on performance calculations to optimize your system design. Diffuser efficiency is an important variable to establish early, and one good example in the vicinity of your operating conditions will be a useful benchmark.

Steam-jet air compressor example. A brief example of a steam-jet air compressor will illustrate this. Assume it is desired to use 150-psig steam to compress 200 scfm of atmospheric air to 20 psig for use in an instrument air system. The mass flow of air is

$$W_{air} = \left(\frac{200\text{ scf}}{\text{min}}\right)\left(\frac{1\text{ lbm air}}{13.3\text{ scf}}\right)\left(\frac{60\text{ min}}{\text{h}}\right) = 902\text{ pph}$$

Converting to equivalent steam, refer to Fig. 4.6 to obtain molecular entrainment ratios of 1.0 for air and 0.81 for water vapor. The steam equivalent of the air load is

$$W_{se} = 902\left(\frac{0.81}{1.0}\right) = 731\text{ pph steam}$$

Applying a reasonableness check, observe that the steam equivalent is less than the air load, which agrees with the expectation that air is easier to compress than steam because it has a higher molecular weight.

The expansion ratio is 164.7/14.7 = 11.2, and the compression ratio is 34.7/14.7 = 2.4. From Fig. 11.12, obtain R_s = 2.1. The required steam is 731(2.1)=1535 pph. The discharge DAE is 902 + 1535(1.33) = 2944 pph. The size factor is 2944/[34.7(51.7)] = 1.64, which calls for a 2-in stage with a price of $1300 (U.S., 1992). The cost estimate has only order-of-magnitude accuracy. The application requires an aftercondenser to remove the steam and a subsequent dryer to achieve instrument air dryness. A two-stage design will use 5 to 10 percent less steam.

Jet blowers (blast nozzles, ventilators)

An extreme application of the ejector principle, this type of ejector uses compressed air or steam to pump a large load through a very small differential pressure. The performance characteristics fall to the left and below the scales in Fig. 11.12. Special mixing nozzles are sometimes used to mix the high-velocity jet with the load gas in a manner that achieves a uniform velocity profile. Unlike that in a critical-flow ejector stage, the mixture velocity here is well below the velocity of sound, and a significant pressure rise occurs before the diffuser diverges. Constant-area mixing models are sometimes used to describe the behavior.

Consult the manufacturers' catalogs [27, 28] for descriptive information. See Fig. 11.16 for performance estimates. For conversion to other units of pressure, 1 mbar = 0.75 torr and 14.7 psia = 1013 mbar = 1.013 bar.

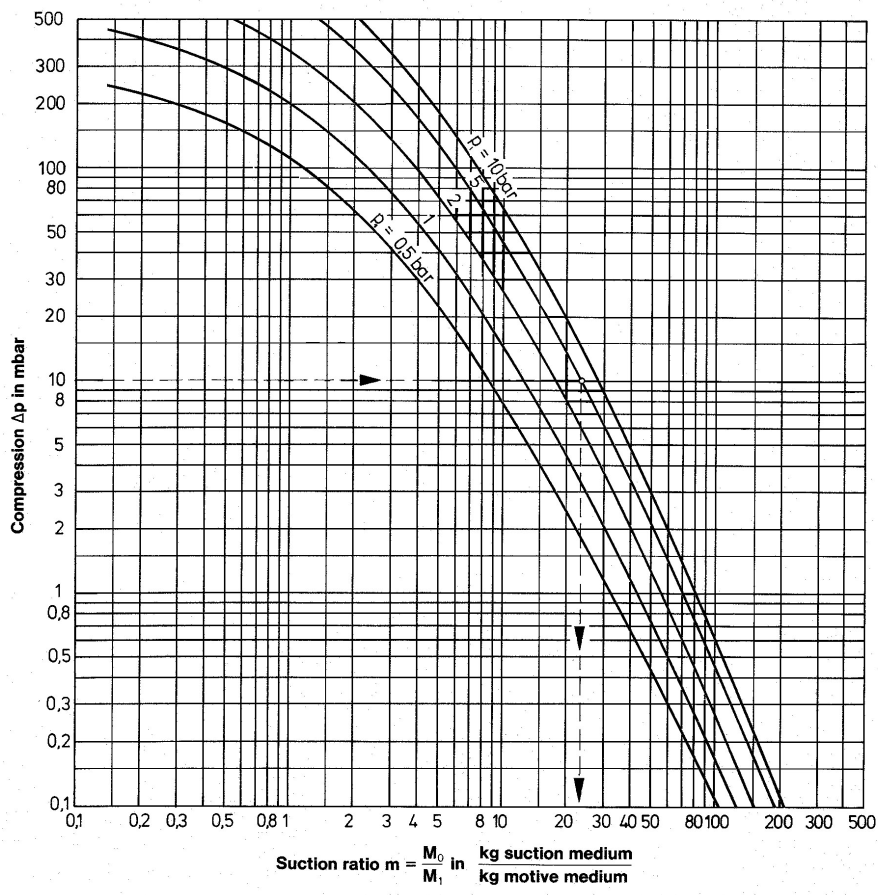


Figure 11.16 Steam-jet ventilator performance. *(Courtesy GEAWiegand, GmbH)*

Efficiencies, ethics, and performance testing

Because thermocompressors are often justified on a cost-reduction basis, the choice among competitive bids can often be influenced by small differences in vendor quotations. Unlike steam-jet vacuum ejectors, thermocompressors have only one stage, with no interstage condensers to offer tradeoffs of price versus steam economy. Each manufacturer simply presents the best design available. Diffuser efficiency is everything. In turn, diffuser efficiency is affected by the diffuser geometry and by the motive steam nozzle geometry and position.

The situation is complicated by the industry perception that most users know very little about ejectors, many users appear unwilling to spend more money for better operating economy, and most users do little or no performance testing. Occasionally, mutterings are heard about manufacturers who claim unattainable efficiencies for some of their designs. In a situation where the only ethical constraint is self-monitoring, some unethical situations are likely to develop. Who would know the difference without performance testing?

I recommend the strategy of defining pragmatically what you are willing to pay for performance, selecting a bid that appears realistic and offers a low evaluated cost, adding a price-performance clause to your purchase contract if it seems appropriate, witnessing the performance test to assure yourself that the performance is correct, and adjusting the price if performance is significantly different from the contract performance. The cost of witnessing the test is not trivial, but it assures you that you get what you are paying for. Once, when I witnessed the test of a batch of thermocompressors and examined the results, I discovered that one was undersized. The manufacturer had neglected to make a conversion from equivalent air to equivalent steam for that unit. It was replaced with a properly sized unit.

The flip side of this discussion of ethics is to encourage you to treat the ejector manufacturers even-handedly, respect the confidentiality of the information they share with you, and make sure that you purchase true performance.

11.6 Liquid-Jet Liquid Ejector (Eductor)

In sheer numbers, this is one of the most common ejector applications. An above-ground centrifugal pump is combined with an ejector placed at the bottom of a well to form a jet-pump system that is able to draw water from wells deeper than 34 ft. Several hundred thousand such units are sold each year in the United States alone. For that reason, and because water-jet experiments are ideally suited to low-budget research, much testing has been done and reported in the public literature [29-36].

The treatment here is brief. This chapter refers briefly to the fundamental calculations, reviewed in Chap. 4, then illustrates a method for predicting the performance of an example application. The method uses Fig. 11.17, which is a nomograph version of Eq. (4.11) with a correction for net positive suction head (NPSH). It predicts the performance of a custom-designed ejector designed for best efficiency at that operating condition. It does *not* predict performance away from that condition.

For many applications the performance of a standard-design unit is adequate, and such a unit will be much cheaper than a custom-designed ejector that requires custom engineering by the manufactur-

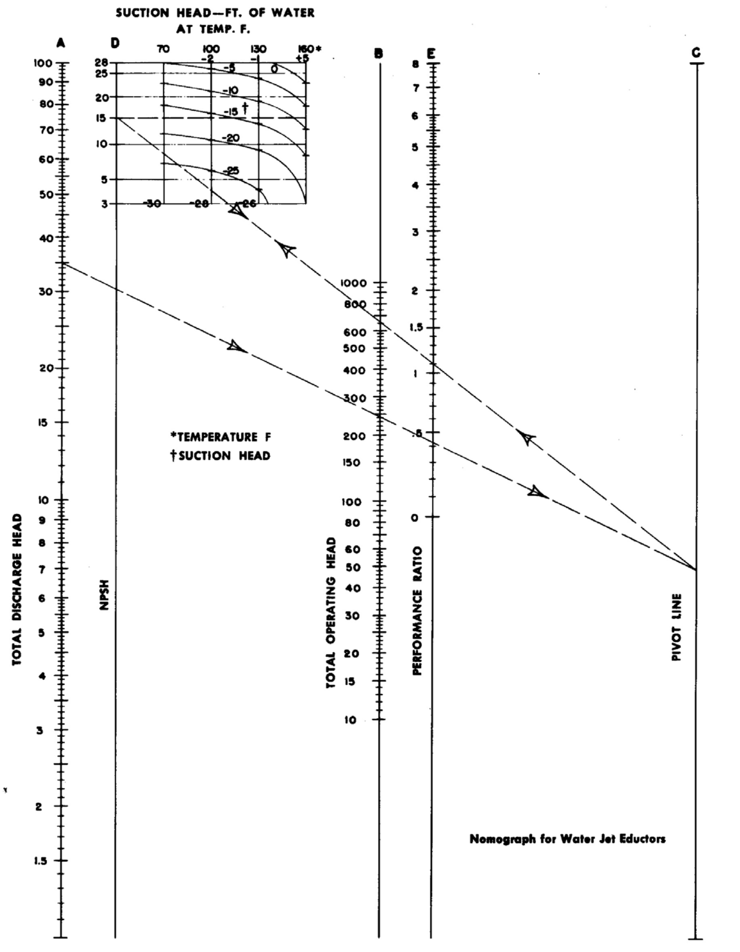


Figure 11.17 Water-jet eductor performance nomograph. *(Courtesy Schutte & Koerting Division, Ketema)*

er. The price of a screwed-connection eductor may be 25 percent of the price of a flanged custom steam-jet ejector stage of the same size. The price of standard flanged units may be 50 percent of that of custom steam jets. For descriptive information and sizing data for standard units, refer to the manufacturers' literature [14, 23, 37-41]. Most ejector manufacturers are prepared to supply water-jet ejectors, even if they are not listed in these references.

Example liquid-jet ejector calculations

Figure 11.18 shows the configuration for an example application, taken from a manufacturer's catalog [37]. Warm liquid toluene is to be lifted from a low-level storage vessel at atmospheric pressure and pumped into an elevated vessel at atmospheric pressure, using water at 100 psig and 70°F. The lift is initially 4 ft, increasing to 17 ft at the end of the transfer. The delivery level is 25 ft above the ejector. Five thousand gallons are to be transferred in 15 min. The toluene is at a temperature of 160°F, the specific gravity is 0.87, and its vapor pressure is given as 4.3 psia.

Can this be done, how much water is required, and what are the approximate size and price of the ejector?

My first reaction to this application would be to question whether the designer was really willing to contaminate a large water stream with toluene! In spite of the fact that it is nominally insoluble in water, toluene is an environmental contaminant, and expensive treatment of the water would be required if the water were to be separated

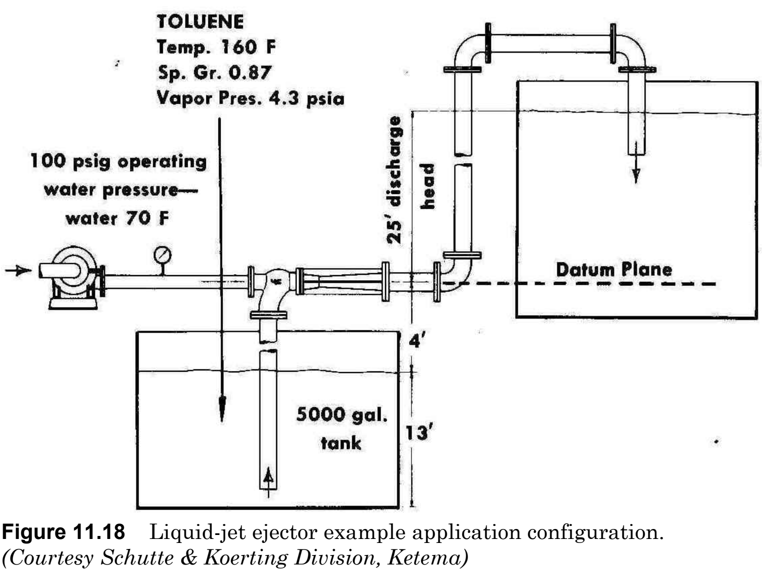


Figure 11.18 Liquid-jet ejector example application configuration. *(Courtesy Schutte & Koerting Division, Ketema)*

and discharged from the plant. For that reason, this is a good example of the need to consider all aspects of a proposed application, not just the fluid mechanics. Assuming that this question has been addressed properly, I continue with the example.

Use of the nomograph in Fig. 11.17 requires that the total discharge head and the total operating head be determined. The scale units are not given, but any measure of energy will be adequate. Because this analysis is complicated by the load fluid having a different specific gravity from the motive fluid, the convenient units would be "feet of fluid" (actually, ft·lbf / lbm).

The total discharge head equals the lift head plus the discharge head. The average lift is (4 + 13/2) = 10.5 ft of toluene. Note here that the average is used to determine the approximate average performance. At the end of the transfer, however, the lift will be 17 ft, so we must verify that the unit will continue to perform adequately with that lift. The lift in equivalent water is 10.5(0.87) = 9.1 ft. The discharge is a mixture of water and toluene, so we guess that the ratio is about 1:1 and use a mixture specific gravity of 0.93. The discharge head is 25(0.93) = 23.3 ft. And the total discharge head is (9.1 + 23.3)/0.93 = 34.9 ft of mixture.

The total operating head is the motive head plus the suction lift,

$$\text{Total operating head} = \frac{100(2.31\ \text{ft/psi}) + 9.1}{1.0} = 240\ \text{ft of water}$$

On the nomograph, construct a line from 34.9 on the total discharge head scale A through 240 on the total operating head scale B, and mark a pivot point on the pivot line C. Pause a moment to consider what has happened so far and refer to Eqs. (4.9) and (4.11), which is reproduced here:

$$R_e = e(R_h^{1/2}) - 1 \tag{4.11}$$

So far, we have determined the value of R_h, the head ratio. Here, R_h = 240/34.9 = 6.88, and its square root is 2.62. The pivot point is at a location that corresponds to 2.62 on the pivot line C. Observe that we would have reached the same pivot point had we entered the chart with heads of 24 and 3.49, or 68.8 and 10.

The final step is to determine the proper value of *e*, which is a measure of overall efficiency. In Chap. 4, *e* is given as 0.9 for a (large) ejector with a central nozzle and cold water. It was also stated that *e* is less than 0.9 if the liquids flash into vapor as they mix violently. The nomograph scale D is an adjustment of *e* as a function of the net positive suction head (NPSH).

Before calculating the NPSH correction, use the cold-water value of 0.9 for e to find the best performance: the highest possible value of the entrainment ratio R_e ("performance ratio" on the nomograph). $R_{e,max}$=(0.9)(2.62) – 1 = 1.36 gal water/gal water, or 1.36/0.87 = 1.68 gal toluene/gal water. The actual value may be less for this hot toluene. Next, examine the NPSH effect.

NPSH = total head – vapor pressure head. Because the friction loss in the inlet line is negligible or included in the lift, the total head is atmospheric pressure minus the lift, 34 – 9.1 = 24.9 ft. The vapor pressure is 4.3(2.31) = 9.9 ft. NPSH = 24.9 – 9.9 = 15 ft of water.

Finally, back on the nomograph, construct a line from the marked pivot point to 15 on scale D, and find a performance ratio of 1.1 on scale E. Correcting for specific gravity ratios, the performance ratio is 1.1/0.87 = 1.26 gal toluene/gal water. This ratio is close enough to the assumed value of 1.0 that it is not necessary to repeat the calculations to use a more accurate specific gravity for the mixture.

The first question has been answered; it will work at the average conditions. Will it also work at the maximum lift conditions? Repeating the calculations at the maximum lift shows the performance ratio reduced about 20 percent. The performance of an ejector designed for the average lift will be lower than this. The specifications for this ejector should describe the entire operation if you buy on a performance specification. If you pick a unit from a catalog, consider the extremes as well as the average conditions.

The suction flow to the delivery tank is 5000 gal/15 min = 333 gpm. The motive water flow is load/performance ratio = 333/1.26 = 264 gpm. The total discharge flow is 333 + 264 = 597 gpm.

To size the unit, use the rule that a 1-in unit will discharge 20 gpm of water when the motive pressure is 100 psig, that the delivery is proportional to the square root of the motive pressure, and that the capacity is proportional to the square of the discharge diameter.

$$Q_d\,(\text{gpm for 1 in}) = 2(P_m, \text{psig})^{0.5} \tag{11.5}$$

$$\text{Size} = \left[\frac{\text{discharge, gpm}}{2(P_m, \text{psig})^{0.5}}\right]^{0.5} = \tag{11.6}$$

$$= \left[\frac{597, \text{gpm}}{2(100\ \text{psig})^{0.5}}\right]^{0.5} = 5.5 \qquad \text{say 6 in}$$

From Fig. 10.2, a steam-jet ejector this size has a price of \$5000 U.S. in 1992, so a stock flanged eductor will be about \$2500.

Internal dimensions and modifications

The methods of Chap. 4 can be used to estimate the nozzle and diffuser throat diameters of standard units from the tabulated performance data and by consulting the manufacturer. The manufacturer may increase the price significantly for even small modifications because of the special handling required for the order. Accordingly, you may elect to make small changes yourself to save time and money by assuming responsibility for the performance.

The only modification which it normally is appropriate for you to make is a slight enlargement of the motive nozzle to a diameter which better matches the flow rates you desire. The performance will probably not be seriously affected by the nozzle's being in a nonoptimum position. Remember to round and polish the nozzle entrance and remove the burr at the exit to retain fair nozzle efficiency.

11.7 Conveying Solids

Other than a brief overview of all ejector types in Chap. 2, this is our first look at ejectors in which two phases interact. Previously, gas pumped gas and liquid pumped liquid. Now, we are looking at ejectors which use a liquid or gas jet to pump bulk solids.

If the solids are small, nonsticky particles, they will mix well with the fluid and behave almost as if they were in solution. The performance calculations may be made by considering the solid to behave as though it were a fluid, mixing with the jet at the suction pressure and flowing with it through the diffuser. The mixture density is used in sizing the diffuser throat as described in Chap. 4 for liquids with different densities. If you stick to the fundamental model of an ideal nozzle, conservation of momentum in the mixing process, and pressure recovery in a diffuser having an efficiency E_d of 90 to 95 percent, your estimates will be realistic. With liquids, a discharge velocity between 3 and 5 ft/s will keep most solids in suspension without grinding the solids.

The model works best with liquid as the motive fluid, but is also useful with gas as the motive fluid. When the solids consist of large particles or tend to agglomerate, however, you must rely on experience or tests to establish the performance. Ejector manufacturers have test facilities. Consult the catalogs for application and performance information [14, 23, 37, 38, 40, 42, 43].

11.8 Miscellaneous Utility Ejectors

For a variety of ejector applications not covered by the previous material, see the manufacturers' catalogs [11, 44-46]. These are often small devices using available steam, compressed air, or water to perform useful tasks such as priming pumps, lifting water out of a low spot, sampling from an open flume, or mixing two streams intimately.

An effective device for creating an intimate contact between gas and liquid while pumping the gas is the venturi scrubber (fume scrubber). It is effective at removing very small particulate solids.

11.9 Desuperheaters

Desuperheating is often required when high-pressure steam is throttled into a low-pressure steam header. Several ejector manufacturers offer desuperheaters, for which consult their general catalogs. A practical limit in controlling desuperheater operation is the accuracy of temperature measurement. A controller adjusts the flow of desuperheating water to maintain the desired steam temperature. If the control temperature is set below about 10°F superheat, there is the danger that the measurement error will cause the controller to dump a large flow of water into the steam attempting to reduce its temperature below the saturation temperature.

An internal thermal sleeve is sometimes used to protect the downstream steam pipe from the damaging effect of cycling between steam and water temperatures. See the manufacturer's literature for application details.

11.10 Special Design Situations

Several special design situations deserve some mention. They occur so infrequently that I have not previously discussed them at much length. Each situation has some special considerations that must be handled properly to make the application both reliable and safe.

Low-level-condenser condensate removal methods

As condensate leaves a condenser drain, it is subcooled only a few degrees below its saturation temperature at the condenser pressure, and hence may be considered a boiling liquid. The simplest way to remove the condensate is to elevate the condenser about 36 ft above a sealpot and allow gravity to pump the warm liquid into a system which is nominally at atmospheric pressure. The net positive suction head (NPSH) is high enough there that a conventional pump can remove it for delivery to a disposal or recovery system.

Low-level condensate systems are in extensive use, however, in spite of several disadvantages [47, 48]. Typically the condensate outlet area is enlarged to form a hotwell. If a centrifugal pump is used, a liquid-level controller must be added to the hotwell to maintain the boiling liquid level high enough above the pump to meet the NPSH requirements of the pump. If the removal pump is a liquid-ring pump, which is capable of handling two-phase loads, the condenser is lowered and the liquid-level controller omitted. Another option is to use a steam-operated ejector to remove the boiling water.

If the condenser is a small surface condenser, the condensate flow rate may be so low that two condensate receiver tanks may be used, with one accepting condensate while the other is being drained or pressured empty. The draining and switching operation may be automated. For batch operations, a single receiver may be sized to hold the condensate from one operating cycle. For small flows, a dual-element automatic float trap may serve as the receiver, using air or steam to empty one chamber while the other chamber is accepting condensate.

See Chap. 7 for ideas on collecting condensate from multiple condensers into the lowest-pressure condenser.

The obvious benefit of the low-level installation is that no elevated structure is required, and a very compact system can result. The major disadvantage is that proper operation is now dependent on one or two more devices. When the devices fail, water may be sucked back into the process because there is no barometric leg seal on each condenser.

Sublimers (freeze-out precondensers)

The triple point of a pure material is that temperature below which the liquid state does not exist in a stable form. The triple point of water is 0 °C, so below that temperature the water vapor which condenses becomes ice. Conversely, ice evaporates directly into water vapor without becoming wet. The same is true for other vapors which condense below their triple-point temperature. If the vapor has a high triple-point temperature, the solid may condense in the piping, the stages, or the condensers. The accumulation of solids may interfere with proper operation of the ejector system.

Several strategies are used to avoid or cope with this fouling problem. Ejector systems may be installed in pairs, so that one can be cleaned while the other is on-line. Precondensers may freeze out much of the solids, using cooling water or refrigerant [4]. Surface precondensers may have extended surfaces to accumulate a larger volume of solids during an operating cycle. The stages and piping may be heated with full-body steam jackets or electric heaters.

Solids may be washed out periodically, using spray nozzles in the stages and condensers, if the solids are soluble in water or a solvent. Steam pressure may be applied to the ejector system to melt or dissolve the solids, provided that the entire ejector system has been designed to contain the steam pressure safely.

Any surface condensers must be designed to be undamaged by the thermal cycling. One practice that can damage a surface heat exchanger is to alternate cooling and heating by cycling from water to steam inside the tubes. Electric heaters should be carefully installed to prevent burnout and dangerous hot spots.

Low-pressure or high-pressure motive steam

As described previously in the discussion of steam-jet refrigeration, low-pressure steam is suitable for the booster stages ahead of the first condenser. Even subatmospheric-pressure steam can be used for vacuum thermocompressors as described in the discussion of thermo compressors. Steam at pressures of 5 to 30 psig can be used as motive steam in ejector vacuum systems [49]. Compared to an ejector designed for 100-psig steam, the low-pressure-steam vacuum ejector will typically have one more stage, possibly one more condenser, and a higher price. The total operating cost will be lower, however, if the low-pressure steam has a much lower cost.

Another possibility is to use higher-pressure steam or a liquid-ring vacuum pump for the stages after the first condenser.

Often the low-pressure steam has excessive moisture. A good-quality steam separator, followed by a small pressure drop through a pressure-reducing station, will deliver dry steam for best operation.

Problems associated with high-pressure steam are different. For stages requiring small steam flows, the high pressure results in tiny orifices that are easily blocked by moisture droplets or debris. High-pressure steam is often superheated, a factor which must be accurately known for proper sizing of the steam nozzles, design of stages in a non-condensing sequence, and design of condensers. High-pressure steam may require costly and bulky high-pressure steam lines and may pose a hazard to people accustomed to working with lower-pressure steam.

If the disadvantages of using high-pressure steam outweigh the advantages of lower steam usage and cleaner steam, the high-pressure steam may be throttled to a lower pressure for the ejectors and partially desuperheated.

Multiple-element ejectors

Twin-element ejectors are common for maximum reliability in steam turbine exhaust condenser service. If the ejector condensers are sized for all stages operating, then the increased capacity may permit operation to continue in spite of a large air leak or a vapor overload, or both. See Chap. 7 for safety precautions if the stages may be isolated by block valves at both the suction and discharge.

An alternative to the twin-element approach to reliability is a completely separate full-capacity standby ejector without the complexity of isolating valves. The separate unit is completely accessible for testing and maintenance of the condensers as well as the stages, and one unit may serve as a common spare for several ejectors.

If multiple elements are used to match varying or unknown loads, the elements may split the load in a variety of ways [50]. The maximum load may be split by elements of equal size or unequal size (1/2-1/2, 1/3-2/3, etc.). Extra elements can be added as installed spares to improve the reliability, but the large stages are regarded as highly reliable.

11.11 Designing Your Own Systems

I approach this final technical subject with very mixed feelings. First is the fear that you may "shoot yourself in the foot," wasting time and money learning about ejector design details which are more economically purchased from ejector manufacturers. The major benefit most users can get from this book is to learn how to specify, install, operate, troubleshoot, and maintain steam-jet vacuum ejectors. That work is straightforward and challenging, and results in reliable ejector systems. They work well.

If you have followed me all the way through this book, however, you may be ready for some additional advanced work.

The kind of design work I do recommend is system design, not hardware design. A complex example of an integrated system design is the three-stream application described in the discussion of thermocompressors. When you start with an intimate knowledge of your process system, add a basic understanding of thermocompressors and an ability to estimate their performance, and combine that with the ejector manufacturers' ability to confirm and expand on performance details, you may discover and achieve some significant operating-cost reductions.

You are also partially prepared to design vacuum ejector systems in the rough vacuum range, relying on the ejector manufacturers to supply the stages and another source to design and supply the condensers. Normally that is not a cost-effective procedure, but it can make sense if the condensers are available at a special price and are designed or selected by a competent heat-transfer specialist familiar with the design of vacuum condensers. You have the coordinating responsibility, however, and if the system doesn't work, you are it! You might find a cooperative ejector manufacturer willing to help you make it work, for a price.

If you attempt to replace contact condensers with surface condensers from someone other than the ejector manufacturer, you will have the design and coordinating responsibilities as above. As described in Chap. 12, for several reasons it may be cost-effective to retire the old ejectors and specify new ones.

If you or someone else obtains an ejector from a salvage yard or assembles one from salvage components, it may be a testimonial to ingenuity, but a monument to inefficiency. After giving the mandatory warning, I will agree that some bargains may be found in a salvage yard. But, I have seldom found them.

Stage design

A strong need for secrecy may motivate you to design your own ejector stages. You may achieve some success with incompressible-flow ejectors (liquid-jet liquid ejectors) and less success with low-compression-ratio (noncritical) gas-jet compressors and thermocompressors. If you are going to design a high-compression-ratio vacuum ejector, however, you will be traveling a costly road traveled for many years by the ejector manufacturers. Shock waves, diffuser taper angles and lengths, nozzle expansion and nozzle placement, and prediction of performance curves are complex subjects that must be understood.

In preference to starting from scratch, I recommend that you approach a manufacturer with a secrecy contract. Or, you may consult or hire a trustworthy, knowledgeable ex-employee of an ejector company. A problem here is that the sort of person you are looking for may not be interested in what may be a short-term, limited-scope project.

If anyone is still reading – perhaps some research students or some ejector nuts who will read anything about ejectors – here are some references that may be useful to you. They are of differing quality, and I offer only a few observations about them. Please remember, there is a world of difference between "designing" and "designing well." In general, the subject of operating cost-effectiveness is noticeably absent.

These references, in turn, contain many references – enough to keep you busy sorting out the many approaches to design. If you are at all concerned about efficiency (low operating cost), I urge you to look for the most efficient features and plan on tests to establish the final dimensions and performance. Motive nozzles should be machined and polished, with smoothly rounded entrances, short throat, and proper outlet taper. Multiple spacers or a continuously adjustable nozzle holder will allow you to find the best nozzle position relative to the diffuser.

Kroll [51] published at a time when energy costs were low, secrecy was prevalent, and everyone was willing to experiment. He offers empirical correlations, minimum theory, a worked example, and references.

Shapiro [24] analyzes the behavior of ideal gases in a supersonic diffuser, describing the compression shock phenomenon and how it limits the attainable diffuser efficiency.

Dotterweich and Mooney [52] discuss the application of ejectors to pumping high-pressure natural gas using higher-pressure natural gas as the motive fluid. They use ideal gas laws, idealized constant pressure mixing, idealized nozzles, diffuser efficiency of 65 percent, and diffuser flow coefficient of 85 percent. They discuss performance curves and steam jacketing to prevent hydrate formation. Example calculations are included.

Harris and Fischer [53] describe the importance of nozzle location, discuss stage efficiency, and discuss the combination of stages into multiple-stage systems.

Huang, Jiang, and Hu [18] describe the design and analysis of a Freon-113 jet refrigeration system, including off-design performance. They provide equations for calculating the saturation pressure, enthalpy, and entropy for that material.

Hedges and Hill [54] develop a general-purpose finite-difference model of axisymmetric mixing of parallel, compressible streams in a converging-diverging mixing section (diffuser). The model is limited to weak shock waves and does not accommodate curvature effects.

Hasinger [55] studies various mixing modes: constant-area, constant-pressure, and defined pressure distribution. He discusses efficiencies and losses, and offers simplified calculation methods. Many design and performance curves are included.

Emanuel [56] uses a constant-pressure mixing model and compares his results with the HEI entrainment ratio correlations.

Kurtz [57] uses an isothermal-flow numerical analysis which models the shear effects at the diffuser wall and between the nozzle and secondary stream. It is implemented via a FORTRAN program (not included).

Jeelani et al. [58] briefly present the design methods of Kroll and others in a nomograph, with an example design.

Dutton and Carroll [59, 60] use a constant area mixing model for their analysis, which produced a FORTRAN program (not included).

11.12 Nomenclature

BHP	brake horsepower, measured at the drive coupling at the design point or other specified condition
cp	viscosity, centipoise
DAE	dry air equivalent (as defined by HEI)
e	ejector overall efficiency factor, equal to the square of the overall efficiency
E_d	diffuser efficiency, sometimes representing the overall ejector efficiency
fps	velocity, ft/s
FT**2	area, ft^2
G	chilled-water circulation rate, gpm
gpm	volumetric flow, gallons (U.S.) per minute
h	enthalpy, BTU/lbm
KE	kinetic energy, ft .lbf / lbm or BTU / lbm
lbf	pound force
lbm	pound mass
MDP	maximum discharge pressure for a stage, torr or psia
NPSH	net positive suction head, ft ·lbf / lbm
P_c	pressure in condenser, torr or psia
P_f	pressure in flash chamber, torr or psia
P_m	motive fluid pressure, units as appropriate
pph	lbm/h
PU	pickup pressure, either MDP or motive steam
Q	refrigeration heat load, tons
R_a	mass ratio, motive steam to load air, dimensionless
R_e	mass entrainment ratio, load/motive, dimensionless
R_h	head ratio, motive head/load head, dimensionless
R_m	mass motive ratio, motive/load, dimensionless
R_s	mass ratio of motive steam to load steam, dimensionless
scf	standard cubic foot, an implicit mass unit, the quantity of a gas occupying one cubic foot at standard temperature and pressure
T_c	condensing temperature, °F or °C
T_f	flash temperature, °F or °C
T_o	condenser cooling-water outlet temperature, °F or °C
ton	refrigeration effect, 12,000 BTU/h
torr	absolute pressure unit, 1 mmHg
W_e	flow of vapor to a condenser, pph
W_f	flow of vapor from a flash chamber, pph

W_s flow of motive steam, pph

W_{se} flow of steam-equivalent load, pph

11.13 References

1. W. D. Mains and R. E. Richenberg, "Steam Jet Ejectors in Pilot and Production Plants," *Chemical Engineering Progress*, 63 (3): 85-88 (1967).
2. J. L. Ryans and D. L. Roper, *Process Vacuum System Design and Operation*, McGraw-Hill, New York, 1986.
3. Harald Grave, Frerich Hinrichs, and Walter Hummel, "Steam and Other Vapors as Working Medium for Jet Pumps," *Chem.-Ing.-Tech.*, 63 (3): 248-250 (1991) (in German).
4. Herbert Gehring, "Process-Integrated Environmental Protection in the Use of Steam Jet Vacuum Pumps," *Chem.-Ing.-Tech.*, 63 (5): 471-483 (1991) (in German).
5. "Process Vapor Power," Bulletin No. JVA-23, The Jet-Vac Corp.
6. F. Duncan Berkeley, "Air Ejectors Cheaper than Steam," *Chemical Engineering*, November 1954, 164-168.
7. "Technical Data: Automatic Injector," Section 1000, Bulletin 1500, Penberthy Division of Stanwich Industries, Inc.
8. D. H. Jackson, "Steam Jet Ejectors: Their Uses and Advantages," *Chemical Engineering Progress*, 77-79 (1976).
9. "Jet Pump Technical Data: Heating Liquids," Section 1000, Bulletin 1400, Penberthy Division of Stanwich Industries, Inc.
10. "Steam Jet Heaters," Bulletin 3A, and "Steam Heaters for Liquids: Direct Contact High Capacity," Bulletin 3A-HC, Schutte and Koerting Division, Ketema.
11. "Jet Pumps and Gas Scrubbers," catalog, GEA Wiegand GmbH, Germany.
12. "Steam Jet Heaters," Nash-Weineck Bulletin 86.4.1, Nash-Kinema, Inc.
13. "Series 640 Pipeline Heaters," Fox Valve Development Corp.
14. "Jet Pump Technical Data: Pumping Liquids," Section 1000, Bulletin 1200, Penberthy Division of Stanwich Industries, Inc.
15. "Steam Jet Syphons," Bulletin 2A, Schutte and Koerting Division, Ketema
16. "Series 270-SYP Steam Jet Syphon," Fox Valve Development Corp.
17. J. Christman, "Vacuum Steam Condensers: Air vs. Water Cooling," *Hydrocarbon Processing*, September 1980.
18. B. J. Huang, C. B. Jiang, and F. L. Hu, "Ejector Performance Characteristics and Design Analysis of Jet Refrigeration Systems," *Transactions of the ASME*, 107: 792-802 (1985).
19. "Thermocompressors," Bulletin TC-V80-A, Croll-Reynolds Co., Inc.
20. Philip Freneau, "Steam-Jet Thermocompressors Supply Intermediate-Pressure Processes," *Power*, January-February 1945.
21. "Ejector Manual," Graham Manufacturing Co., Inc.
22. "Jet Compressors," performance data supplement to Bulletin 4F, Schutte and Koerting Division, Ketema.
23. "Jet Pumps and Gas Scrubbers," catalog, GEA Wiegand GmbH, Germany.
24. Ascher H. Shapiro, *The Dynamics and Thermodynamics of Compressible Fluid Flow*, I, Ronald Press, New York, 1953.
25. C. G. Blatchley and H. J. Stratton, "The Performance of Thermocompressors as Related to Paper Machine Dryer Drainage Systems," *Pulp Paper Magazine*, 65 (7): 301-306 (1964).
26. Herbert Baier, "Vakuumerzeugung durch Dampfstrahl-Vakuumpumpen," (Vacuum production by steam-jet vacuum pumps), *Vakuum in der Praxis*, 2S: 127-135 (1989). Distributed as a reprint by GEA Wiegand GmbH, Germany.
27. "Jet Blowers and Blast Nozzles," Bulletin 4-AB (qualitative descriptions), Schutte and Koerting Division, Ketema.
28. "Steam Jet Ventilators," catalog, Jet Pumps and Gas Scrubbers, C9-C10, GEA Wiegand GmbH, Germany.
29. James E. Gosline and Morrough P. O'Brien, *The Water Jet Pump*, University of

California Publications in Engineering, 3(3), University of California Press, Berkeley, 1942, 167-190.
30. Arthur G. Hansen and Roger Kinnavy, "The Design of Water-Jet Pumps, Part 1: Experimental Determination of Optimum Design Parameters," ASME Paper 65-WA/FE-31, November 1965.
31. Arthur G. Hansen, Roger Kinnavy, and A. Victor Jett, "The Design of Water-Jet Pumps, Part 2: Jet-Pump System Characteristics," ASME Paper 65-WA/FE-32, November 1965.
32. Nelson L. Sanger, "Cavitating Performance of Two Low-Area-Ratio Water Jet Pumps Having Throat Lengths of 7.25 Diameters," NASA Technical Note TN D-4592, Lewis Research Center, Cleveland, Ohio, May 1968.
33. Nelson L. Sanger, "Noncavitating and Cavitating Performance of Two Low-Area-Ratio Water Jet Pumps Having Throat Lengths of 5.66 Diameters," NASA Technical Note TN D-4759, Lewis Research Center, Cleveland, Ohio, 1968.
34. Nelson L. Sanger, "An Experimental Investigation of Several Low-Area-Ratio Water Jet Pumps," *Transactions of the ASME*, Journal of Basic Engineering, March 1970, 11-20.
35. Alex M. Jumpeter, "Jet Pumps," Sec. 4, in Igor J. Karassik, William C. Krutzsch, Warren H. Fraser, and Joseph P. Messina, *Pump Handbook,* McGraw-Hill, New York, 1976.
36. J. Stepanoff, *Centrifugal and Axial Flow Pumps: Theory, Design, and Application,* 2d ed, John Wiley & Sons, New York (1957), chap. 18.
37. "Water Jet Eductors," Bulletin 2M, and "Performance Data on Water Jet Eductors," supplement to Bulletin 2M, Schutte & Koerting Division, Ketema.
38. "Hydro-Jet Ejectors," Bulletin No. JVA-18, The Jet-Vac Corp.
39. "Vacuum/Pressure Producing Machines and Associated Equipment," Hick Hargreaves & Co. Ltd, England, January 1989.
40. "Instructions for Installing and Operating Croll-Reynolds Single Stage Aqua-ductor," Instruction No. 152, Croll-Reynolds Co., Inc.
41. "Water Jet Eductors: Peri-Jet," Catalog 790, Derbyshire Machine & Tool Co.
42. "Solids Handling Eductors," Bulletin 310, Fox Valve Development Corp.
43. "Evactor-Conveyor Systems," Bulletin EC83A, Croll-Reynolds Co., Inc.
44. "Ejectors, Condensers, Vacuum Equipment," Bulletin JVA-24, Jet-Vac Division, Artisan Industries.
45. "Jet Ejector Selection Guide," Bulletin J-1, Schutte & Koerting Division, Ketema.
46. "Jet Pump Application Guide," Section 1000, Bulletin 1100, Penberthy Division, Stanwich Industries, Inc.
47. R. M. Arrowsmith, "Special Chemical and Power Engineering Pumping Systems," *Chemical and Process Engineering,* July 1970, 77-78.
48. "Instructions for Installing and Operating Multi-Stage Ejectors with Surface Intercoolers for Vacuum Service," Bulletin JVS-12, The Jet-Vac Corp.
49. John Mathews, "Low-Pressure-Steam Ejectors," *Chemical Engineering,* June 22, 1987. Reprint CRV 87-2, Croll-Reynolds Co., Inc.
50. D. B. Birgenheier and L. Wetzel, "Vacuum System Design, Crude Vacuum Tower Service," ASME Paper, Hydrocarbon Processing Symposium, PD-Vol. 13, Book No. 100255, 1988; reprinted by Graham Manufacturing Co., Inc.
51. Edgar Kroll, "The Design of Jet Pumps," *Chemical Engineering Progress,* 1 (2): 21-24 (1947).
52. Frank H. Dotterweich and C. V. Mooney, "How to Design and Operate Gas Jet Compressors," *Petroleum Refiner,* 34 (10): 104-109 (1955).
53. L. S. Harris and A. S. Fischer, "Characteristics of the Steam-Jet Vacuum Pump," ASME paper 63-WA-132 and *Transactions of the ASME,* Journal of Engineering for Industry, November 1964, 358-364.
54. K. R. Hedges and P. G. Hill, "Compressible Flow Ejectors: Part I, Development of a Finite-Difference Flow Model, and Part II, Flow Field Measurements and Analysis," *Transactions of the ASME,* September 1974, 272-288.
55. Siegfried H. Hasinger," Performance Characteristics of Ejector Devices," Wright-Patterson Aerospace Research Laboratories, ARL TR 75-0205, June 1975.

56. George Emanuel, "Optimum Performance for a Single-Stage Gaseous Ejector," *AIAA Journal,* 14 (9): 1292-1296 (1976).
57. E. F. Kurtz, Jr., "Theoretical Model for Predicting Steam-Ejector Performance," *ASME Journal of Engineering for Industry,* May 1976, 645-661.
58. S. A. K. Jeelani, K. V. Kasipathi Rae, and G. R. Balasubramanian, "Designing Steam-Jet Ejectors," *Chemical Engineering,* Apr. 9, 1979, 135-136.
59. J. C. Dutton and B. F. Carroll, "Optimal Supersonic Ejector Designs," *ASME Journal of Fluids Engineering,* 108: 414-420 (1986).
60. J. C. Dutton and B. F. Carroll, "Limitations of Ejector Performance due to Exit Choking," *ASME Journal of Fluids Engineering,* 110: 91-93 (1988).

Chapter 12

Upgrading Existing Ejector Procedures and Hardware

12.1 Overview

In beginning this last chapter, I feel that I am launching a protégé, much as I felt when my children left home for college. I am searching for special words of wisdom. In their situations I had nearly 20 years of preparation. In your situation, this might be the first chapter you are reading. You may be a manager who is merely looking at selected parts of this book to get its flavor. Accordingly, I will attempt to make some brief, useful generalizations and will recommend specific policies that lead to successful operations.

In attempting to deliver a package of useful technology to people who appear to have a use for it, I am reminded of the story of a new county agricultural agent who arrived with his new degree, full of information he wanted to share with the farmers in the county. He approached the first farmer and explained how he had studied soil conservation and other wonders of the science of agriculture, and was prepared to share his discoveries with the farmer. The old farmer looked him up and down and said, "I don't believe there's much a young boy like you can teach me. Why, I've worn out three farms in my lifetime, and this here's the fourth."

Dejected, the agent returned to his car and drove to the next farm. There, he regained most of his enthusiasm and knocked on the door. Again he presented his story. The second farmer meditated a few moments, then said, "It all sounds pretty good, and probably is something I could learn, but I'm not sure I can benefit from all that extra know-how. You see, even now I'm not farming half as well as I already know how to!"

People who react like the first farmer generally will not read this. A few might be interested in whatever helps them continue to operate an aging plant until it is shut down. If this describes your situation, you will find Chaps. 3 and 8 helpful in keeping your ejectors alive and operating with a minimum of attention.

If you think like the second farmer, you represent the majority of managers and other people working with ejectors, in that you are struggling to set priorities in the midst of an overload of work, information, and, especially, advice. I will address that need, attempting to define a realistic set of priorities.

Brief guidelines are given below, directed at people either involved in an all-out upgrade program or using a minimum-budget approach. Most people will be somewhere between these two extremes. The split in subjects between the two circumstances is one of emphasis. You should read both sets of guidelines and adopt the combination that seems right for you.

Upgrade now, across the board

I assume you have identified many of the deficiencies in your use of ejectors and are ready to make changes, provided you are given credible directions for the changes. I am including the situation where design, operation, and maintenance are all in the same company, or at least in a continuing relationship. For that receptive situation, I recommend that managers of engineering, production, and maintenance coordinate their efforts by selecting at least one engineer or technician representing each area to serve on a team responsible for planning and coordinating the upgrading program. In a small plant, this might be one person. Toward that end, I have tried to present the information in this book in a form usable with a minimum of adaptation effort.

For example, the design group can immediately use the specification forms, manufacturers' curves and design data forms, and general specifications by simply pasting your company name and address on the forms. The instructions in Chaps. 9 and 10 tell how to use the forms to specify ejectors to get reliable, economical designs, and how to evaluate bids to detect and avoid buying the occasional error.

The production and maintenance people can use Chap. 8 to upgrade their instructions for operating, testing, troubleshooting, and maintaining ejectors. They can decide how testing and troubleshooting are to be done and what test instrumentation and facilities are needed.

Collectively, they may define a training program. If the plant is just starting an in-depth look at ejectors, the program might begin by sending one or two engineers to witness the shop test of a multistage ejector purchased for the plant. Or, if the plant has several people with some ejector experience, the program might begin with an onsite half-day ejector seminar given by an ejector manufacturer.

Record keeping is another important subject, especially for the testing, troubleshooting, and maintenance people. Decide how many copies of purchase specifications and manufacturers' curves and data forms are to be distributed to different files. Who will be responsible for maintaining the files in the plant, how will the reports of tests and maintenance inspections be filed, and what freedom will people have to borrow (and lose!) entire files? A convenient quick-copy machine may be part of the answer.

After the team has established working procedures and acquired the necessary test facilities, it is ready to work on the list of process and hardware upgrades and cost reductions presented later in this chapter.

Limited-budget (staged) upgrading

If you are in a situation where your budget and staff are quite limited, you will have to upgrade more slowly, spending available money and time where they will do the most good, to demonstrate that the expenditures are productive.

The first-priority item is to be able to perform field tests of vacuum ejectors. The ability to measure sub-atmospheric pressure with sufficient accuracy and confidence is the highest-priority item in any vacuum ejector testing activity. A small lab bench with an oil-sealed vacuum pump and associated hardware to maintain sealed-leg absolute-pressure mercury manometers and check other gauges with equivalent or better accuracy is essential. The bench and adjacent locked storage areas and paper files should be identified for the ejector work, so that the key hardware and records will not be damaged or lost between ejector test activities.

Do not be misled by the fact that an experienced person can often make do with makeshift pressure measurements. The experienced person knows the accuracy required for each measurement, and how to get the maximum accuracy from each observation. The inexperienced person needs more help, and a reliable pressure measurement is essential for her or him.

Reliable steam-pressure gauges are the next requirement; these are usually available in a processing plant.

The third hardware item required for field tests is a multiple-orifice critical-flow air meter (flute, piccolo), described in Chap. 8 and the Appendix. It is cheaper to buy a commercial device than to make one, unless you have idle machine shop time. You may wish to fabricate a few individual air orifices for field tests as a supplement to or substitute for the flute, using the methods described in the Appendix.

Once the basic hardware is available, the next task is to train at least one person to test and troubleshoot ejectors. Give the job as a temporary assignment to a person who has demonstrated learning skills, curiosity, and a willingness to work outdoors under adverse conditions when necessary. Make the distinction between specialist and expert. This person will immediately be a specialist, by assignment. It takes time and talent to become an expert.

Agree to a mutual appraisal of the assignment in a year, so that you can redefine the assignment or give it to another person if that is appropriate. If the ejector specialist is a technician, it will help to assign a process engineer as a technical resource. When the technician needs help in understanding a mathematical or physical concept, or in locating needed technical resources, the engineer can help.

Training is best begun by doing an ejector test, ideally by witnessing and participating in a test in an ejector manufacturer's shop on an ejector being built for your plant. The ejector will work right, the instruments will all be calibrated and working, and many skilled people will be available. Or, your specialist may test one of your ejectors in your plant, preferably with the guidance of an experienced person from among your employees or on loan for the day from a nearby friendly plant. Plan such a test carefully to make sure that all the required instruments are available and working, that the ejector is working, and that ejector performance curves are on hand. Many field tests are delayed because critical hardware is missing or because steam or water is not available.

When the initial training period is complete, begin the work of catching up. Prepare a paper file for each ejector in the plant, and for each file obtain a copy of the relevant process flow diagram, purchase specifications, the manufacturer's specifications, performance curves and performance data, and spare parts lists. Also obtain any records of testing, troubleshooting, inspections, maintenance, and modifications. Manufacturers can supply missing data.

Prepare an ejector test kit as suggested in the Appendix. You will be tempted to load it up with lots of extra fittings to save time and trips up to the jet level in the structures. But you will find the kit heavy to carry up the stairs. Look for an elevator.

Now you are ready for business. It will tend to be "steady, by jerks." Problems will tend to arrive in clusters, especially if they are caused by low steam pressure, dirty steam, low water pressure, high water temperature, or upsets caused by process changes. Evenings and weekends are likely times, and holidays are highly preferred by failure gremlins.

Read this book, then read it again later. Be tolerant of your forgetfulness, because most people only partially absorb what they read each time. As you gain experience, you will get more out of the book and may pursue some of the many references by ordering magazine article reprints, getting manufacturers' catalogs, and buying or borrowing some of the books.

Critique each installation you work with. Note the real and potential trouble spots or inconveniences. Offer to give the plant or unit engineer or manager a sketch and written summary of your suggestions. Keep a copy in your file. If you find an error in this book, or if you think of a fact or a topic that would improve the book, please write and tell me about it. Share! If you have studied a problem and this book does not give you the answer, and you cannot find the answer in other convenient references or get it from other people in your plant, call the manufacturer and discuss the problem or write me. Grow!

Ejectors can give reliable, trouble-free performance when properly specified, installed, operated, and maintained. When carelessly specified and used, however, they can become "lemons." Even a properly designed ejector will not work properly if there are major defects in the installation or in the operating conditions. My personal horror story involves a five-stage ejector with many problems. Three problems were contributed by the manufacturer, and we managed to contribute an additional seven over a period of two years. It took a lot of patience and cooperation to solve all those problems without yielding to the temptation to label the ejector a "lemon." Everyone involved carries mental scars from that experience. I continue to respect that manufacturer, and check the design of all bids.

12.2 Ejector Economics

One brief note here, for managers who do not intend to read much of this book. I recommend that you skim over Appendix G, "Ejector Operating Costs and Design Optimization." It examines why operating costs typically constitute more than 80 percent of the total cost of owning an ejector, shows by an example the very large latitude that ejector manufacturers have in tradeoffs between first cost and operating costs, and concludes with two strong recommendations:

1. Always give a manufacturer some realistic utilities cost guidelines in your inquiry specifications. Even very approximate values will help you avoid the costly extremes.

2. Do not spend too little when buying your ejector. Spending too little, to gain a short-term savings, can result in buying a "stretched" design or an inefficient design.

   The stretched design may accomplish the design task on the test stand by use of fewer stages that operate at conditions approaching the limits of ejector performance. The performance of a small stage operating at a compression ratio higher than 10:1 is easily degraded by wet steam, fouling, or wear.

   The inefficient design may be reliable, but uses fewer stages and smaller surface condensers to reduce the purchase price by increasing the steam and water usage. This results in outrageous operating costs over the project life and contributes to early economic obsolescence.

   On the other hand, extra money spent for a more effective design that uses less steam is at worst earning a marginal return on the incremental investment.

12.3 Operating, Testing, and Troubleshooting

Many operating engineers and managers do not realize how valuable field testing and the interpretation of ejector performance curves and specifications can be in diagnosing problems in the ejector system or the process system. Ejector operation and troubleshooting is a very learnable skill. It can become a routine technical task instead of a murky art that some people have and some do not have. Once the test procedure is learned and an ejector installation is equipped with the proper test connections, a very effective performance test can be conducted in 15 min. Managers whose people have learned how to test ejectors are enthusiastic about the results.

Train at least one person, and define a target typical condition, such as "test a three-stage ejector to determine the first-stage performance from no-load to its limiting air capacity, then determine the steam pickup pressure for the last stage at 80 percent of limiting air capacity." A trained engineer and a trained technician make a good team for working on the ejector systems in a plant. Ejector manufacturers may be prepared to conduct training seminars at their shop (an excellent hands-on environment) or in your plant or offices. Witnessing a test of one of your new ejectors at the manufacturer's shop is a good start. You can read this book on the plane there, and again on the trip back to understand better what you just did and saw.

Conduct a performance test of all new ejector systems in the field as soon as possible, using air as a load and obtaining a performance curve for the first stage from no-load to the load at which the first-stage curve "breaks" upward. Then determine the steam pickup pressure at 80 percent of maximum air load, noting the actual discharge pressure seen by the last stage.

See Chap. 8 for testing and troubleshooting information and suggestions.

In plants that have several ejectors, it is useful to have a small indoor test stand for testing the smaller atmospheric stages. These are the stages that cause most ejector problems. Many atmospheric stages are 1- to 3-in size, so the test stand can be quite modest. It should have a steam supply at a pressure matching or exceeding the design pressure for the plant ejectors. The suction line should have an air-load test flute. High-quality vacuum and steam pressure gauges should be used.

A surface or contact aftercondenser should be configured to permit a back pressure of at least 2 psig, or higher to match the highest in the plant, with provisions for throttling the condenser vent and measuring the discharge pressure at the ejector discharge flange. Because that throttling valve could be accidentally closed, the system should be protected against overpressure by a large relief valve or a static liquid blowout seal.

12.4 Maintenance

Upgrade any inadequate pressure measurement hardware, instruments, and procedures. Obtain missing manufacturers' data such as performance curves, design and maintenance data, and spare parts lists. Create and upgrade stage performance curves when testing and troubleshooting. Keep useful records of maintenance inspections and repairs, using the results of inspections and testing to schedule subsequent testing and inspections. Maintain a "do list" of hardware improvements to be made during future shutdowns or inspections.

12.5 Design and Purchasing

A person who has worked with ejectors and tested a few is thereby better prepared to specify and design ejector systems. The paperwork acquires a three-dimensional reality, giving the person a clearer mental image of what. is behind the flat paper specifications. Even a walk-through inspection of several installations and an introductory talk about ejectors will be helpful to a process engineer, a detail designer, or a purchasing agent working with ejectors. A purchasing agent should be generally familiar with the contents of Appendix G, "Ejector Operating Costs and Design Optimization."

Chapters 9 and 10 guide you in selecting a vacuum pump type and specifying ejectors, Chaps. 4 and 5 are useful when performing preliminary design estimates and evaluating ejector quotations for general correctness, and Chap. 7 contains useful installation guidelines. Chapter 8 has information useful to a person witnessing a performance test in the manufacturer's shop.

12.6 Upgrading Vacuum Systems and Other Process Systems

The information in this book will enable you to upgrade an existing process system in almost any way that does or can involve ejectors of any type. Improved safety and reliability and compliance with environmental regulations lead the list of upgrade priorities, but cost reduction and improved operating convenience follow closely. Improved operating convenience leads directly to improved reliability, and releases operator attention for improvements elsewhere.

These suggestions are brief summaries of previous discussions, mostly in Chaps. 7, 8, and 10.

If you have a list of several improvements you want to make, but are hesitant because of your limited experience with ejectors, look for a strategy that allows you to start with a simple task and learn by doing before you get to the more complex tasks. A bad early experience may be a wonderful learning opportunity, but it will spoil your enthusiasm and may lose your support for a more important complex task. Keep in touch with more than one ejector manufacturer at first to minimize the effects of blind spots in your understanding.

Safety

If low-pressure equipment can easily be subjected to full steam pressure, remove the block valves, or add properly sized pressure relief valves, or replace the equipment with components designed for the full pressure and temperature.

Insulate steam lines and operator-level steam jackets well to prevent burns. The ejectors also work better and wear less when the motive steam is dry. Make sure electric superheaters have overheating protection built in. I know of one unprotected electric superheater that exploded!

Provide access platforms or rearrange equipment so that people are not in hazardous positions when doing common operating, testing, or maintenance tasks.

Reliability

The last stage of an ejector system is the first stage to look at for improving reliability. It is the smallest and easiest to modify, and it has the greatest impact on operation. If the design discharge pressure is marginal, look for ways to reduce the discharge pressure and consult the manufacturer to see whether a new nozzle or some other simple modification would improve the discharge-pressure capability. If initial evacuation time is longer than desired, you might also replace the last stage with a larger unit. If corrosion or wear is noticeable, upgrade at least this stage to a hard, corrosion-resistant material.

If erosion by wet steam is noticeable, add a good separator and trap to the steam supply. If pipe scale and other solid debris plugs nozzles and you do not have strainers, add strainers to the steam supply after the separator.

If the steam supply pressure frequently drops just low enough to cause ejector failure, you may wish to ask the manufacturer to renozzle your ejector stages for a slightly lower steam pressure. You will use more steam and may lose a little capacity in exchange for the improved reliability.

If you do not have spare nozzles for all stages and a spare diffuser for the last stage, order them. You may relax this requirement for large nozzles on the first few stages if you inspect them regularly and know the corrosion and fouling rate to be low. Most fouling can be removed without damaging the nozzles and diffusers..

If fouling is a continuing problem, consider some fixes. If low-boiling vapors condense in the stages, perhaps steam-heating the stages will help. Check with the manufacturer to discuss ways to do this and to estimate how much the capacity may be reduced. Or, you may place a two-element pre condenser ahead of the first stage so that a fouled unit can be cleaned by steaming or washing while the other unit protects the ejector. If surface condensers foul and a contact condenser would not, consider the design alternative of using contact condensers and recycling cooled condensate to the contact condensers. This reduces water pollution, and perhaps transfers the cleanup problem to a pair of plate-and-frame water/condensate coolers near the hotwell. The modification requires routine coordination of the design task between the process engineer and the ejector manufacturers.

Protecting the environment

An ejector system discharges condensate and a vent gas stream to the atmosphere. Both streams contain some of the "condensable" vapor present in the load to the first stage. If the condensate must be treated before the water can be discharged from the plant, then that treatment cost must be included in the total cost of operating the ejector. The same logic applies to the vent gas stream.

Condensate treatment costs may be reduced considerably by replacing any contact condensers with surface condensers. The manufacturer should do the redesign to ensure that the calculations of the condenser pressure drops and the approach temperatures are done properly. If the design load contains a large quantity of condensable vapor which is soluble in water, then the reduction in mass flow of water on the process side of the condenser may permit a much larger flow of condensable vapor to the subsequent stage. You may find it simpler and cheaper to replace the old system with a new design, especially if you have improved knowledge of the desired design pressure, process conditions, physical properties, air load, steam and water conditions, and operating costs.

Vent gas emissions may be reduced by avoiding the use of noncondensable gas to control system pressure, and by the use of an aftercondenser with a vent temperature that closely approaches the cooling-water temperature. A refrigerated vent condenser may also be useful in reducing condensable vapor losses.

Protecting the Process

Chapter 6 describes the conditions that cause an ejector to "backfire" steam into the process system, and how to test for it.

Energy savings

Fondrk [1] offers several excellent energy-saving suggestions in his 16-page gem on ejectors. I have adopted most of them here.

If the total operating cost for an ejector or a collection of ejectors is large enough to justify some attention, then a survey may be in order. Even a brief survey of existing ejectors will identify the grossly inefficient or wrong-sized ejectors.

A more detailed survey is required to identify those special opportunities for cost reductions by installing thermocompressors or throttling motive steam.

Replacing inefficient or wrong-sized ejectors. If you have a four-stage ejector on a system that operates at 75 torr, something is wrong! Perhaps the system is performing a different process than it was designed for, or perhaps the ejector was salvaged from an idle unit as a quick source of a low-cost ejector. You may have an ejector designed for 100 pph air plus some condensable vapor that is maintaining vacuum in a small, tightly designed system for which you might normally consider 10 pph air to be an appropriate load.

One virtue of ejectors can result in a long-term economic loss to the operator: They can be so durable, when supplied with dry steam in a noncorrosive application, that they can outlive their economic useful-

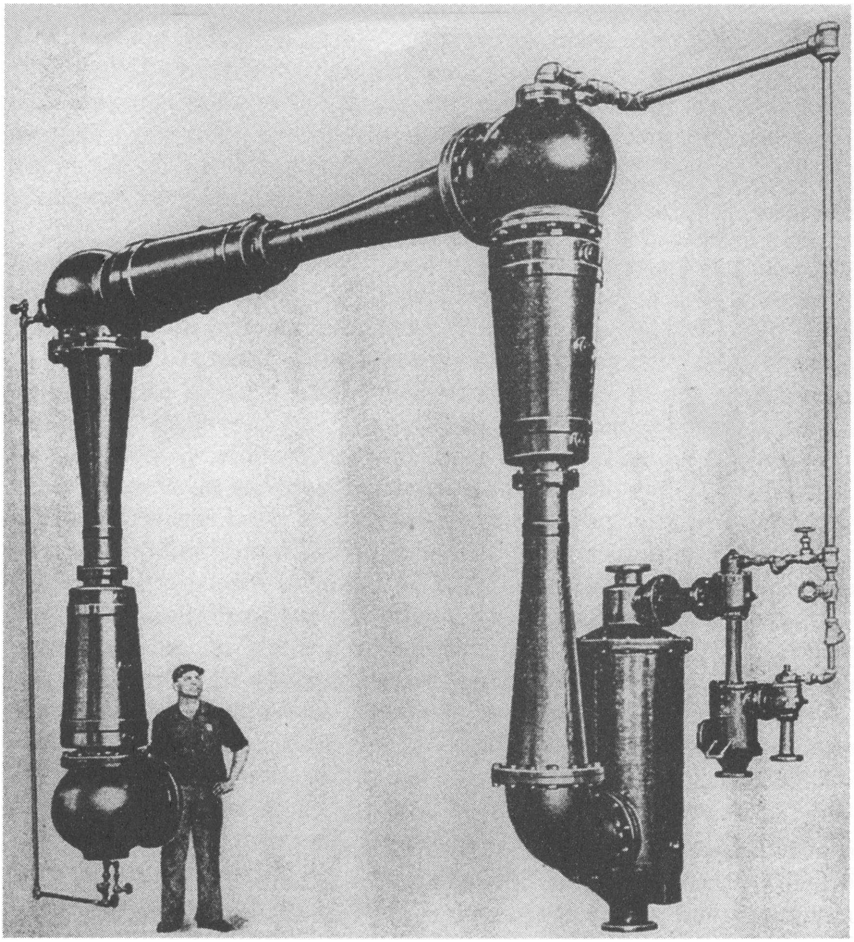


Figure 12.1 A classic five-stage steam-jet ejector. (Courtesy Unique Systems, Inc.)

ness. When I see an "old timer" like the ejector in Fig. 12.1, I react with a mixture of respect and suspicion. When built, it had a larger capacity at 100 μm (0.1 torr) than any previous vacuum pump. If it were in operation after many years, however, I would routinely question whether the process had changed, and whether a modern design might make more efficient use of steam while creating less water pollution. The old ejector might prove to be outdated, or it might still be an effective installation.

Figure 12.2 is another interesting design from years gone by. It contains the functional equivalent of two noncondensing stages in series within a body that externally looks like a single stage. The cluster. of small nozzles surrounding the central nozzle acts as an

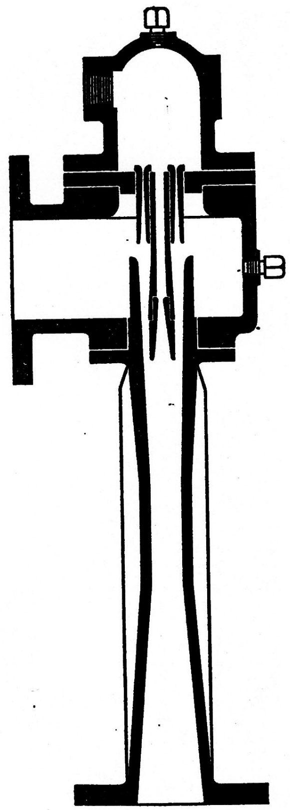


Figure 12.2 Two noncondensing stages in one body. *(Courtesy Unique Systems, Inc.)*

annular first stage, precompressing the load gas in the annulus formed by the central nozzle body and the extended diffuser inlet. The precompressed gas is again compressed as it encounters the steam from the central nozzle. Again, marvel at this, but be realistic. Replacement parts may not be available. If performance data or curves are available, compare the performance with that of modern designs. If data are not available, a test will reveal its performance. Nozzle throat dimensions will reveal how much steam it uses.

Once I assisted an engineer expanding a production unit that was originally installed as a pilot unit. The two-stage noncondensing ejector used more than 1000 pph steam, and a casual approach to the task would have called for a simple duplication of the existing ejector. When the process engineer saw the economy of a condensing ejector, he also conducted some garbage-bag tests to measure the actual air leakage at the aftercondenser vent. He then specified a reduced design air load and a condensing design. In the next expansion, the two-stage ejector was duplicated. Then a duplicate two-stage unit replaced the old noncondensing unit. Thus, a single inquiry led to a steam usage reduction of 2000 pph, worth more than $1 million U.S. investment equivalent in many locations today.

A quick estimate of the steam and water requirements of a modern ejector, sized using present-day guidelines and designed with today's emphasis on operating costs, may show you a payback that quickly recovers your new investment while it reduces pollution problems.

Survey for thermocompressor applications. I believe there is a huge backlog of undeveloped, highly profitable applications for thermocompressors in the process industries. They are easy for the trained eye to find and evaluate, although they are not obvious to someone not familiar with ejectors. The information in Chap. 11 removes the constraints of inconvenience and ignorance. For many process industry plants, those now operating and those being designed, investment of a few hours in an energy conservation survey will yield a big payoff.

This survey is best done by an engineer who is familiar with the process and familiar with the steam-jet refrigeration and thermocompressor calculation methods in Chap. 11. Guidelines are to look for large irreversibilities such as high-pressure steam being throttled to lower pressure for heating or injection into a process, large temperature differential in a steam-heated heat exchanger, large temperature drop in a boiling liquid as it flashes, and cooling a liquid stream in a heat exchanger when it could be cooled by flashing to a lower pressure.

The throttled steam might perform a useful pumping effect if it were to flow through an ejector, compressing low-pressure waste steam to a usable pressure level. The flashing liquid might yield useful low-pressure steam for heating, or for use as motive steam in a low-pressure thermocompressor.

The natural tradeoff is that investment is increased and operating flexibility may be reduced. Thermocompressor costs are modest, but larger heat exchangers (because of lower temperature differentials), added flash chambers, and added control systems all increase the investment. Multiple elements and variable-nozzle thermocompressors will improve operating flexibility.

Throttling excess steam pressure. Savings in steam usage are possible where large swings occur in the pressure of motive steam for ejectors, or where the cooling-water temperature to a large booster condenser is usually much lower than the maximum for which the system was designed.

Throttling the motive steam to a pressure just above the design pressure for the ejectors will reduce steam usage in proportion to the absolute-pressure reduction. This requires no coordination or special information from the manufacturer.

To save steam usage by large boosters on any ejector system, whether it be vacuum refrigeration or a process application, it is necessary to obtain additional information from the manufacturer or by test. A curve showing maximum discharge pressure versus motive steam pressure will permit you to design and set up a control system which will measure the pressure in the booster condenser and throttle the motive steam to a pressure that is just adequate for reliable operation, with no loss in capacity. Some ejector manufacturers are prepared to supply the control system.

Improved operator convenience. Hardware features that make it easier for operators to do their work translate directly into improved operations. An isolating valve between the ejector first stage and the process system permits the operator to localize the source of vacuum troubles, speeding the identification and correction of the problem. In those systems where a brief rise in vacuum pressure is acceptable, it also permits quick preventive maintenance performance tests without shutting the system down. Dual water valves at the inlet to each contact condenser, one for flow adjustment and one for on/off operation, simplify the adjustment of water flow to the proper value. Thermometers in the drainlegs of contact condensers are also helpful for that adjustment.

Water valve handles with several calibration marks (tape, paint, crayon, or filed notches) can represent the varied opinions of operators who are forced to guess at the proper settings.

Chapter 7 discusses these and many other details of an ejector installation.

One retrofit convenience is shown in Fig. 12.3. It is a patented quick-access steam nozzle assembly that replaces the existing nozzle assembly. Unique features include a large inspection plug at the top (rear) of the steam chest and a slender spacer and steam nozzle assembly which screws into the top of the nozzle plate instead of the bottom. After the inspection plug is removed with a spanner wrench, the nozzle can be visually inspected and checked with the "nozzle" end of a go/no-go gauge rod made specifically for that stage. The gauge rod has a go/no-go assembly on each end: a small pair on one end for the nozzle, and a large pair on the other end for the diffuser. Removal of the nozzle assembly permits visual inspection of the diffuser and checking for wear and fouling with the "diffuser" end of the gauge rod.

Use of this device is a useful supplement to performance testing and regular disassembly and inspection. It may permit a longer interval between full inspections. If the "go" tip of the "nozzle" end of the gauge rod cannot pass through the nozzle, the nozzle needs cleaning.

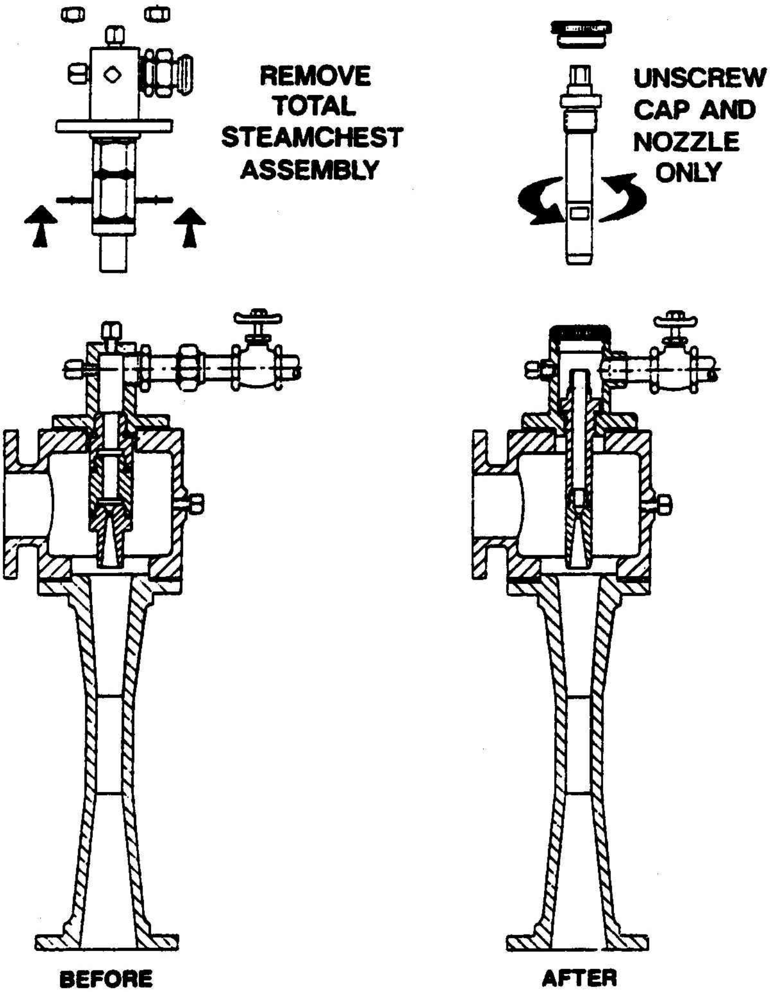


Figure 12.3 Retrofit quick-inspection steam nozzle assembly. *("Quickcheck Steamchest"(TM), Courtesy Unique Systems, Inc.)*

If the "no-go" section passes, the nozzle needs replacement. If the big "no-go" section on the "diffuser" end of the gauge rod passes through the diffuser, the diffuser needs to be replaced. Such a partial-disassembly procedure may not detect diffuser grooves or ovalling from moisture erosion, and may not detect harmful fouling.

Survey for other special ejector applications. Someone who has become familiar with ejector applications by reading this book, or in some other way, may identify potential "utility" ejector applications in a process plant. The problem is to match the specialized knowledge with the diverse applications. One method is to have your ejector specialist or a manufacturer's representative conduct an ejector seminar for several people representing the various plant areas. Maintaining a set of ejector catalogs for easy reference is a useful follow-up to such a seminar.

12.7 Final Words of Encouragement, a Challenge for a Revolution

With the information in this book, you are able to completely alter the ways you work with ejectors. I would love to have had a book like this when I first started to work with ejectors. I am pleasantly surprised by the range of applicability and accuracy of several of the topics I originally thought might emerge a little fuzzy. My parents would have been shocked by my immodesty, but I want to level with you as I give you some parting words of encouragement.

You are now liberated from the lack of reliable technical information describing most steam-jet ejectors. This area is a straightforward application of basic process engineering technology to a special class of equipment.

You are now prepared to make preliminary estimates, economic studies, and bid evaluations for steam-jet vacuum ejectors, process-vapor-driven vacuum ejectors, thermocompressors, gas-jet compressors, steam-jet refrigeration, and liquid-jet liquid ejectors. For steam-jet ejectors you have detailed, cradle-to-grave assistance, including detailed forms and instructions for purchase specifications, manufacturers' data, and field tests. Your only constraint now is your willingness and ability to challenge your priorities and change them where appropriate. .

If you look around you for comfort in precedence, you will not change. Most ejector users do not do testing or analysis, because they do not have the know-how. Some will not change, even with the know-how. I am encouraging you to step forward, now. If you hesitate because you are unsure of the credibility of this book or your own ability to apply what is in it, please discuss that uncertainty with an ejector expert you respect. You CAN do it!

Your primary benefit will be improved ejector reliability. A significant secondary benefit will be a big reduction in operating costs: reduced steam and water usage and reduced product spoilage and loss. You will have to invest some time and money in training, testing, engineering, and hardware to achieve these benefits.

Here are some pictures in my mind of the plant or company that fully profits by this book. Part-time or full-time ejector technicians have a dedicated work area for maintaining and checking vacuum-pressure gauges, testing atmospheric stages, and keeping files on the plant ejectors. On a routine schedule and on demand, they test ejector systems in place, using air loads to check the performance curves of the first and last stages and determine the pickup steam pressure of the last stage. Test results are reported to production, are used to schedule maintenance inspections, and are filed. Troubleshooting is a systematic application of the strategies and guidelines in this book, utilizing hardware features in the ejector installation that simplify the task. During shutdowns for maintenance, hardware upgrades are added to the ejector systems for improved operations, troubleshooting, and testing.

In the engineering offices, ejectors are specified carefully, based on modest air loads appropriate to each application, and emphasizing reliability by careful attention to defining the steam and water conditions and the discharge pressure for the last stage. All specifications contain investment equivalent costs that realistically represent the long-term financial objectives of the company. Quotations are evaluated carefully for reasonableness, and the successful bid is seldom the one with the lowest price. Quality is the objective. Installations are designed with respect for the day-to-day needs of plant production and maintenance people, and the layout detail is reviewed by a plant representative or other person with ejector experience.

A sleeper which became apparent to me as I worked on Chap. 11 is the application of thermocompressors to process systems. The correlation I developed is surprisingly convenient and broad in its application. It permits quick evaluation of a potential application in a process system, regardless of whether the pressures are vacuum or above atmospheric pressure. There are many applications waiting to be discovered in existing and new plants. A process engineer can pick this up and apply it without having to first learn all about multistage vacuum ejectors. $ave a few million?

My last word? Do not spend too little on ejectors.

12.8 Nomenclature

pph lbm/h

torr absolute pressure unit, 1 mmHG

12.9 Reference

1. Victor V. Fondrk, "Installation, Operation, Maintenance, and Troubleshooting of Ejector Systems," Unique Systems, Inc.

Answer to Problem 4

Potential water sources here are the spray in the contact condenser, the steam line, the process system, and the hotwell. A slip blank at the 2d-stage suction showed that stage was OK and the steam line was dry. A slip blank at the intercondenser vapor inlet excluded the process system, and the hotwell had no other connections. A leak in the drainleg anywhere above the overflow level would suck air into the drainleg. Air bubbles from a tiny leak might be washed into the hotwell. But a large leak would create a bubble large enough to obstruct the condenser drain. A leak in the drainleg was the culprit. See page 224 for pros and cons of drainleg features

Problem Number 5

A 5-stage ejector was tested in the field, and it produced the zero-load pressure shown on its performance curve. However, it "broke" to a much higher pressure when even a tiny air load was added. Troubleshooting started with a slip blank ahead of the last stage and zero-load. It maintained a zero-load pressure of 15 torr (!?), versus 60 torr on the performance.curve What must be happening, and what do you do?

Answer on Page 34

Appendixes

Appendix

A

Physical Properties of Common Fluids

TABLE A1 Physical Properties of Some Gases, Vapors, and Liquids. Estimating quality only, use accurate data for design and purchase.

Name	Molecular weight	Name	Molecular weight
Air	29	Hydrogen chloride	36
Acetic anhydride	102	Hydrogen sulfide	34
Acetylene	26	Krypton	84
Ammonia	17	Neon	20
Argon	40	Nitrogen	28
Bromine	160	Nitrous oxide	44
Carbon dioxide	44	Methane	16
Carbon disulfide	76	Methyl chloride	50
Carbon monoxide	28	Oxygen	32
Chlorine	71	Ozone	48
Ethane	30	Phosgene	99
Ethylene	28	Propane	45
Fluorine	38	Sulfur dioxide	64
Helium	4	Water vapor	18
Hydrogen	2	Xenon	131

(a) Assorted gases and vapors, some soluble/reactive with water

TABLE A1 Physical Properties of Some Gases, Vapors, and Liquids (Continued)

(b) Solvents whose liquid phase is soluble in water

Name	Molecular weight	Latent heat, BTU/lb	Vapor pressure, torr		Solubility in water, %
			20°C	50°C	
Acetone	58	228	186	600	100
Ethanol	46	377	44	220	100
Ethylene glycol	62	381	«0.1	<0.1	100
Isopropanol	60	305	33	175	100
Methanol	32	490	97	402	100
Propanol	60	330	15	88	100

(c) Solvents whose liquid phases are only slightly soluble in water

Name	Molecular weight	Latent heat, BTU/lb	Vapor pressure, torr		Solubility in water %	Specific heat, BTU/lb.oF	Conductivity BTU/h.ft.oF	Viscosity cp	Specific gravity
			20°C	°50C					
Benzene	78	178	75	270	<1	0.42	0.092	0.49	0.88
Butanol	74	269	4	34	8	0.69	0.105	1.80	0.81
Carbon Tetr.	154	87	92	315	0	0.20	0.060	0.74	1.60
Cyclohexane	84	161	78	280	0	0.44	0.062	0.70	0.78
Heptane	100	145	35	135	0	0.51	0.057	0.34	0.68
Hexane	86	152	121	415	0	0.53	0.080	0.26	0.66
Isobutanol	74	261	8	50	9	0.72	0.109	2.20	0.80
Styrene	104	160	5	24	0	0.42	0.067	0.62	0.91
Toluene	92	163	22	90	0	0.44	0.086	0.46	0.87

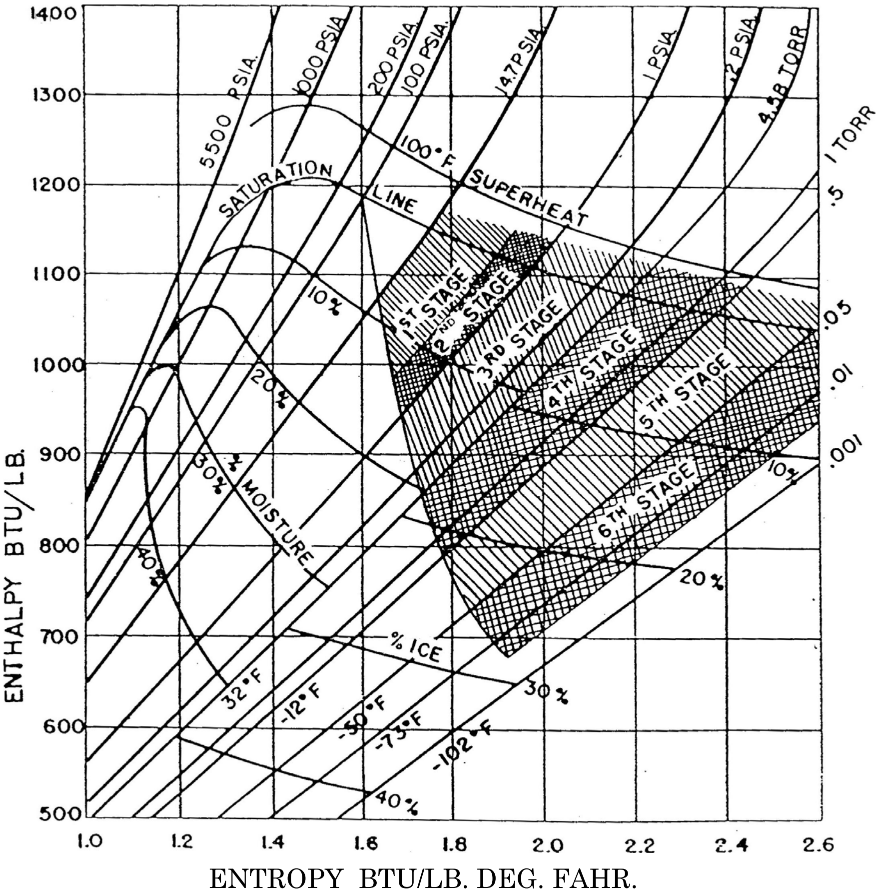


Figure A.1 Mollier diagram, steam-jet ejector range. In this isolated example, the stage numbering sequence is reversed from the standard convention. *(From L. S. Harris and A. S. Fischer, "Characteristics of the Steam-Jet Vacuum Pump," ASME paper 63-WA-132, and Transactions of the ASME,* Journal of Engineering for Industry, *November 1964, pp. 358-364.)*

VAPOR PRESSURE OF WATER

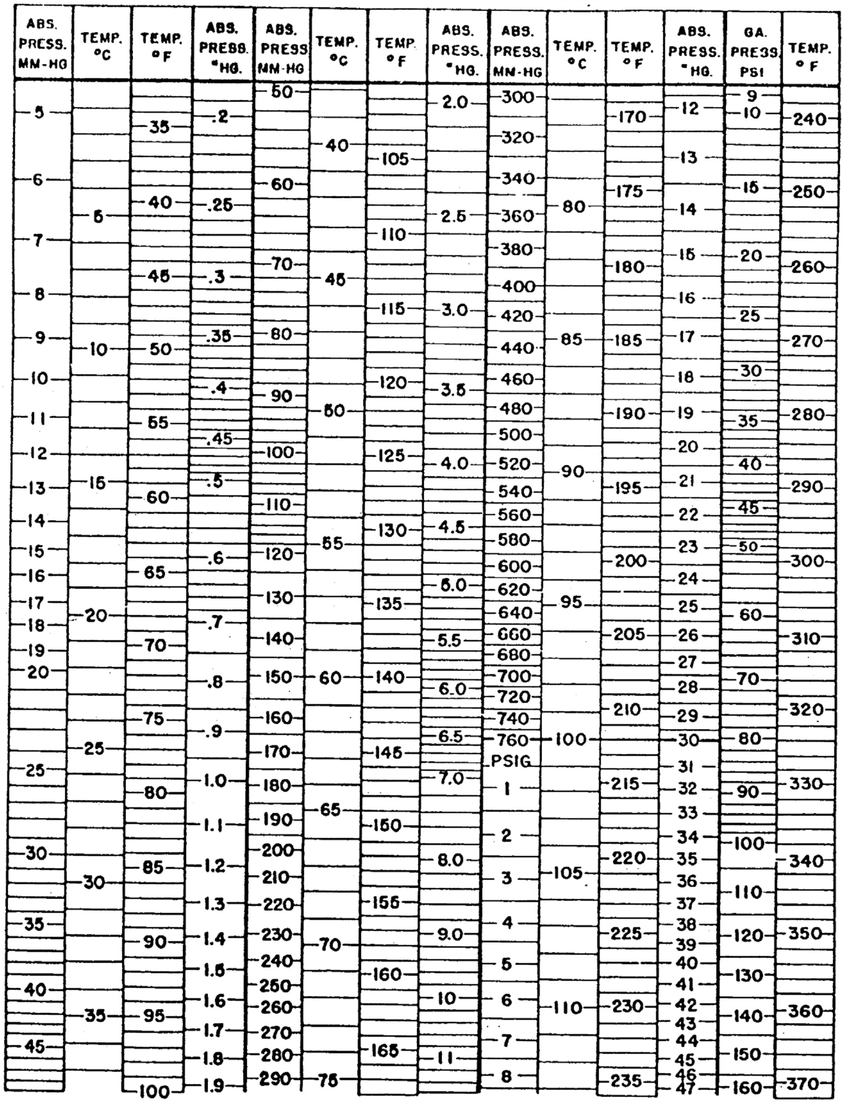


Figure A.2 Vapor pressure of water. *(Courtesy the Jet-Vac Corporation.)*

Appendix

B

Handy Vacuum Engineering Data and Rules of Thumb

Here are some engineering data and rules of thumb which are useful for the person doing vacuum engineering calculations. Some are precise, and some have obviously been rounded for convenience.

Gravitational acceleration = 32.2 ft/s^2

= 9.806 m /s^2

Universal gas constant = 1546 ft.lbf / lbm.mole.oR

= 1.987 BTU / lbm.mole.oR

= 1.987 g.cal / g.mole.oK

1.0 BTU = 778 ft.lbf

Standard atmospheric pressure = 760 torr

= 14.7 psia

= 1013 mbar

= 30 inHg absolute

= 34 ft of cold water

Mercury has a specific gravity of 13.6.

1.0 psi = 2.31 ft of water = 51.7 torr = 2 inHg

Air has a specific heat of 0.24 BTU / lbm.^{o}F at normal conditions, a specific heat ratio k of 1.4, and a specific volume of 13.3 ft^3 / lbm at standard conditions.

Water has a specific heat of 1.0 BTU / lbm.°F, has a density of 62.4 Ibm / ft³, freezes at 4.6 torr (0°C, 32°F), and has a vapor pressure of 49 torr at 100°F. One gpm = 500 pph.

Rounded-entrance critical-flow orifices used for ejectors have a flow coefficient of about 0.97. A 1-in rounded-entrance critical-flow orifice will admit 956 pph dry air from standard atmospheric conditions into a vacuum. A 0.10-in orifice will pass 10 pph air into a vacuum. A 1-in orifice will also pass 4159 pph steam from 100 psia, dry and saturated, and 48 pph steam from 1 psia, dry and saturated. A more precise estimate for saturated steam is

$$W_s = 50\left(D^2\right)\left(P_m\right)^{0.96}$$

where W_s is pph steam, D is in inches, and P_m is in psia.

The molecular weight entrainment ratio of a gas is roughly equal to the square root of its specific gravity relative to air. Figure B.1 is sufficiently compact that it deserves inclusion here. Note that the DAE factor here is the reciprocal of the entrainment ratio.

A vacuum ejector sized for 1 pph DAE at 1 torr has a suction size of 1.6 in (40 mm). The discharge size is 75 percent of the suction, and the overall length is 10 times the suction for single-nozzle stages.

The steam required (pounds of steam per pound of dry air) by a multistage condensing ejector in the rough vacuum range is $Ra = 15/(P_s^{0.333})$ for 100-psig steam, where Ps *is* in torr. Dropping the design suction pressure 10 percent increases the design steam 5 percent.

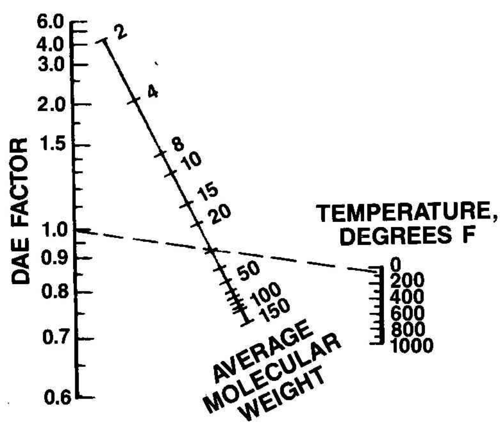


Figure B.1 Combined molecular weight and temperature entrainment ratio. *(Courtesy Nash-Kinema, Inc.)*

A mixture of motive and load steam condensing under vacuum has a latent heat of about 1100 BTU / lbm. One pound of motive steam has a DAE of 1.33 lb at the stage discharge.

In an air/water vapor mixture which is cooled 10°F below the condensing temperature, 2 lb of water vapor accompany each pound of air from the condenser vent.

The shell diameter of a surface condenser is twice the shell-side vapor inlet diameter, which matches the previous-stage discharge size. The vent diameter is half the vapor inlet diameter.

A 36-in diameter contact condenser will handle 1000 gpm water, and will condense 100 pph steam per torr with less than 2 percent noncondensable gases.

Appendix

C

SI Unit Conversions

An example in familiar units will demonstrate or Confirm the use of Table C.1. Let us convert 7.3 yards into centimeters. In the table,

$$1.0 \text{ m} = 1.0936 \text{ yd} = 100 \text{ cm}$$

from which

$$1.0 \text{ yd} = \frac{100}{1.0936} \text{ cm} = 91.44 \text{ cm}$$

and

$$7.3 \text{ yd} = 7.3(91.44) = 667.5 \text{ cm}$$

TABLE C.1 SI Unit Conversions

Quantity	SI base units	Conversions to other common units multiply by factor shown)				
Area	m^2	1.1960 yd^2	10.764 ft^2	1550.0 in^2	10^4 cm^2	10^6 mm^2
Density	kg/m^3	$3.6127 \cdot 10^{-5}$ lb/in^3	10^{-3} g/cm^3	$8.3454 \cdot 10^{-3}$ lb/US gal	0.062428 lb/ft^3	1.0 g/L
Energy, heat	$kg \cdot m^2/s^2$ (J)	$2.7777 \cdot 10^{-7}$ kWh	$9.4782 \cdot 10^{-4}$ Btu	0.23885 cal	0.73756 ft-lbf	3.7251 hp-hr
Heat capacity	$(kg \cdot m^2/s^2)/(kg \cdot {}^oK)$	$2.3885 \cdot 10^{-4}$ Btu/(lb • °F)	$2.3885 \cdot 10^{-4}$ cal/(g • °C)	$2.3885 \cdot 10^{-4}$ kcal/(kg • °C)	$2.3885 \cdot 10^{-4}$ pcu/(lb • °C)	$4.2992 \cdot 10^{-4}$ Btu/(lb • °C)
Heat-transfer coefficient	$(kg \cdot m^2/s^2)/(s \cdot m^2 \cdot {}^oK)$	$2.3885 \cdot 10^{-5}$ cal/(s • cm^2 • °C)	$6.9883 \cdot 10^{-5}$ hp/(ft^2 • °F)	10^{-4} W /(cm^2 • °C)	0.17610 Btu/(hr • ft^2 • °F)	0.85985 kcal/(hr • m^2 • °C)
Length	m	$6.2137 \cdot 10^{-4}$ mile	1.0936 yd	3.2808 ft	39.370 in	100 cm
Mass	kg	10^{-3} metric ton	$1.1023 \cdot 10^{-3}$ ton	2.2046 lb	35.274 oz (avdp)	1000 g
Mass flow	kg/s	86.4 metric ton/day	95.240 ton/day	132.28 lb/min	7936.6 lb/hr	60,000 g/min
Power, heat flow	$kg \cdot m^2/s^3$ (W)	$2.8435 \cdot 10^{-4}$ ton refr	10^{-3} kW	$1.0347 \cdot 10^{-3}$ hp	0.23885 cal/s	3.4121 Btu/hr
Pressure	kg/m^2 (Pa)	$9.8692 \cdot 10^{-6}$ atm	$1.4504 \cdot 10^{-4}$ lbf/in^2	$2.9530 \cdot 10^{-4}$ in Hg abs	$7\,5006 \cdot 10^{-3}$ torr	10^{-2} mbar
Throughput or,	$kg \cdot m^2/s^3$ $Pa \cdot m^3/s$ (W)	1.7184 (lb-mol • °R)/hr	7.5006 torr • L/s	9.8692 atm • cm^3/s	10.282 SCFM • °R	15.893 torr • ft^3/min
Velocity	m/s	0.037282 mile/min	2.2369 mile/hr	3.2808 ft/s	3.6 km/hr	196.85 ft/min
Viscosity (dynamic)	$kg/m \cdot s$	0.67197 lb/ft • s	10 g/cm • s (P)	1000 cP	2419.1 lb/ft • hr	3600 kg/m • hr
Volume	m^3	35.315 ft^3	219.97 Imp gal	264.17 US gal	1000 L	$6.1024 \cdot 10^4 in^3$
Volumetric flow	m^3/s	2118.9 ft^3/min	3600 m^3/hr	13,198 Imp gal/min	15,851 US gal/min	60,000 L/min

(From J. L. Ryans and D. L. Roper, Process Vacuum Systems Design and Operation, *McGraw-Hill, NewY ork, 1986.)*

Appendix D

Glossary

These definitions are primarily for steam-jet ejectors and their installations. Generic terms are used wherever they appear well defined. The list does not include promotional names created by some manufacturers to label related product lines.

Absolute pressure Pressure measured relative to absolute zero. Air

Chamber See Suction Chamber.

Assembly (Stage Assembly) The hardware collection that forms a complete ejector-stage pumping device, commonly called a stage. Components of a stage assembly include a nozzle, suction chamber, diffuser, possible nozzle spacer, and gaskets plus fasteners. A functional stage may consist of one or more assemblies in parallel. Element is another name for a stage assembly, used when considering the stage assembly in reference to other stage assemblies in parallel.

Atmospheric Pressure Loosely, standard atmospheric pressure at sea level, 760 torr. More precisely, the current value of the atmospheric pressure as measured by a barometer at the specific elevation. Be careful to distinguish between pressure at the site elevation and pressure corrected to sea level.

Backfiring A flow reversal condition in which some motive steam flows back into the suction system. When the stage discharge pressure exceeds the maximum discharge pressure (see MDP), the suction flow is reduced and may become negative.

Barometric Condenser See Contact Condenser.

Barometric Leg See Drainleg.

Basic Performance Curve The characteristic curve of an ejector stage designed for a compression ratio greater than 2:1, a "critical" stage. The curve is typically plotted as suction pressure in torr versus suction load expressed as equivalent air. The shape of the curve near the design point is insensitive to modest changes in motive steam pressure, and completely describes the behavior of the stage if its maximum discharge pressure (MDP) is not exceeded.

Blind, Blind Suction (blind flange) See No- Load.

Booster A term applied to ejector stages with suction pressures below the lowest practical pressure for condensers, especially when the load is largely condensable vapor.

Bootleg See Drainleg.

Bore See Throat.

Break Pressure Either of two limiting pressure conditions in a critical ejector stage, defined with respect to a specified suction load. The break discharge pressure, for a specified motive steam pressure, is that discharge pressure above which the suction pressure departs upward from the basic performance curve. The break motive pressure, for a specified load and discharge pressure, is that motive pressure below which the suction pressure departs upward from the basic performance curve.

Capacity The vapor-handling capability of an ejector system or ejector stage in mass flow units, typically pounds per hour of dry air equivalent (DAE) at the design suction pressure.

Cavitation The development of vapor bubbles in a pumped liquid, degrading the pumping effectiveness and usually resulting in mechanical erosion in the pumping device and piping. Where the local pressure is less than the vapor pressure, the liquid partially flashes into vapor. Subsequent collapse of vapor bubbles in the vicinity of equipment surfaces causes microscopic hammering damage.

Cold Trap A refrigerated device used to minimize the condensable vapors to a vacuum pumping device or a vacuum-pressure gauge.

Compression Ratio The ratio of the absolute pressure at the ejector discharge to the absolute pressure at the ejector suction. Compression ratios as high as 15 and 20 are sometimes achievable, but 10 is a normal maximum. Ratios from 3 to 6 are usually the most economical, especially after the first condenser.

Condensing Ejector An ejector which has one or more condensers between stages to reduce the steam usage of subsequent stages.

Contact Condenser A condenser in which the coolant directly contacts the hot vapors, usually as a spray or cascading sheets, and entrains the condensate. Alternatively called a barometric condenser because the large volume of liquid at the drain of a vacuum condenser is most economically removed by a long gravity pipe (barometric leg) to a hotwell or sealpot.

Critical (Critical Flow, Critical Stage) A condition in which sonic or supersonic velocities exist, identified by gas expansion and compression ratios which exceed about 2:1. The geometry and behavior of critical nozzles and diffusers differ significantly from those of devices in which velocities are subsonic. Critical flow will exist in atmospheric airflow nozzles discharging to a vacuum system when the vacuum system pressure is less than 53 percent of atmospheric pressure.

Diffuser The heart of the ejector. Its internal shape is similar to that of a venturi, to mix the motive and load fluids, then convert high-velocity kinetic energy into pressure energy. It is designed for a particular suction and discharge pressure and capacity, and requires a definite minimum quantity of steam to operate properly at those conditions. Other names: venturi and tail.

Discharge (Outlet) The stage connection from which the mixture of motive steam and load vapor emerges.

Discharge Pressure The pressure at a stage discharge connection, usually in the same units as the suction pressure.

Drainleg The pipe which descends from a condenser drain to a hotwell or sealpot. If the condenser is sufficiently elevated, gravity will "pump" the condensate from a vacuum condenser to the sealpot, which operates at atmospheric pressure.

Drop Test The method of measuring air leakage into a vacuum system by measuring the rate of rise of the pressure (vacuum "drop") caused by air leakage.

Eductor A jet pump using liquid to pump liquid. Loosely, an ejector.

Ejector A pumping device in which a high-pressure motive fluid passes through a nozzle, acquiring a high velocity, before it mixes with the load fluid at a low pressure. The medium-velocity mixture of fluids passes through a venturi-shaped diffuser in which the pressure rises to the discharge pressure. A steam-jet vacuum pump uses high-pressure steam to remove air and other vapors from subatmospheric-pressure equipment, discharging them at atmospheric pressure. Functionally, an ejector may be described by the motive fluid and the load material: steam-jet air ejector, nitrogen-jet solids ejector, water jet water ejector, etc.

Element One of the stage assemblies which accomplish a compression task. Most ejector systems are single-element, having one assembly at each stage. If the compression task for a stage is apportioned among more than one assembly, that stage is said to have multiple elements. See Assembly, Multiple-Element Ejectors.

END (End of Curve) See No-Load.

Equivalent Air The flow in pph of dry air at 70°F which will produce the same suction pressure as the load mixture for which an ejector stage was designed. HEI standard correction factors for temperature and molecular weight are used to convert gas and vapor load mixtures to equivalent air (DAE) loads. Air and steam are the standard gases used for testing ejectors.

Evacuation (Hogging, Priming, Pull-Down, Pump-Down) The initial removal of air from a system before starting a vacuum operation. The evacuation capability of an ejector may be given as the time in minutes required to evacuate 100 ft^3 of system from atmospheric pressure to a given vacuum pressure.

Exhauster A jet pump using liquid or gas to pump gas. See Jet Blower.

Extension See Nozzle Extension.

Fume Scrubber See Gas Scrubber.

Gas-Jet Ejector (Jet Compressor, Thermocompressor) An ejector using gas or steam to pump a gas or vapor, for a purpose other than removing air and gases from a vacuum system. Where the purpose of the pumping is to create a chilling effect, to compress low-pressure or waste steam to a usable level, or both, it is called a thermocompressor.

Gas (Fume) Scrubber A low-differential-pressure ejector in which a motive liquid jet spray washes a gas stream to remove vapors and particulate matter.

Hogging See Evacuation.

Hotwell See Sealpot.

Injector An ejector using steam as the motive fluid, mixing it with cool water, and delivering hot water at a pressure which exceeds the steam pressure. Used as a boiler feed pump or hot-water pump.

Inlet See Suction.

Jet Blower (Ventilator, draft ejector) An air- or steam-operated ejector used to pump large volumes of air against very low differential pressures.

Jet Compressor See Gas-Jet Ejector.

Load The vapor mixture entering the suction of an ejector or an ejector stage or element, usually expressed in mass flow units such as pounds per hour (pph).

Manometer An instrument used to measure pressure. The pressure may be measured relative to atmospheric pressure, a near-zero reference, or some other reference.

MDP (Maximum Discharge Pressure) In critical ejector stages only, the stage discharge pressure below which the ejector is always stable, operating on its basic performance curve. This limiting pressure varies with both the motive steam pressure and the load.

Millimeter of Mercury The pressure required to sustain a differential of one millimeter in a mercury column at standard conditions; one torr.

Multiple-Element An ejector system that contains two or more stage assemblies configured in parallel at one or more locations in the system. A common example is the two-stage twin-element ejector serving a steam power condenser. A variation has multiple elements for the first stages only, with each element serving a separate process system and designed for different loads and suction pressures. Another variation has a two-element last stage, with one element being a large hogging ejector operated only during startup. Multiple-booster steam-jet refrigeration systems have multiple elements for load control in the first stage, but only a single element after the large primary condenser. Some thermocompressor applications have multiple elements for load control.

Multistage Ejector An ejector system with two or more stages arranged in series. The stages are numbered in sequence, starting with the lowest pres sure stage. The industry standard (HEI) naming convention is alphabetical in reverse order, starting with Z for the atmospheric stage. Successively lower-pressure stages are named Y, X, etc. The alphabetical naming convention is not followed by all manufacturers. The ejector system may have condensers between stages (condensing), or it may not (noncondensing).

No-Load (Blind Suction, Shutoff) The condition in which there is noload to an ejector stage because the ejector has been isolated by a blind flange, slip blank, or isolating valve at the suction. Also, the suction pressure and maximum discharge pressure associated with zero flow. If the ejector is connected to a large system, this flow condition becomes difficult to achieve because some load reversal can occur: Some motive steam may flow back to the suction system through a combination of bulk flow and diffusion.

Noncondensable Gas A gas at a temperature higher than its critical temperature, and thus having no liquid state. A gas or vapor which does not condense significantly within an ejector system.

Noncondensing Ejector A multistage ejector with no intercondensers.

Nozzle The device that efficiently converts pressure energy into velocity energy, producing highly supersonic steam flows in vacuum ejectors. Such supersonic nozzles have a converging/diverging interior geometry. Liquid-jet nozzles or gas-jet nozzles for non supersonic flow do not have a diverging outlet geometry. Nozzle variations include annular geometry, multiple-nozzle arrays, and variable area, involving a spindle advanced into the nozzle throat.

Nozzle Extension (Spacer) A device used to position the nozzle for best operation. The nozzle screws into one end, and the other end is screwed into the nozzle plate. Many ejector stages do not have nozzle extensions.

Nozzle Plate The back portion of the standard L-shaped suction chamber, drilled and tapped for the nozzle(s). It may be a separate plate, or it may be integral with the back of the suction chamber.

Outlet See Discharge.

Overload The condition in which a combination of load gases and vapors exceeds the design load to one or more stages in an ejector, causing those stages to operate above their design suction pressures.

Pickup (Recovery) Pressure The pressure, of either the motive steam or the discharge, at which an ejector stage recovers to a condition of stable operation at a given load. For the Z stage, the steam pickup pressure at design load is a good measure of the operating safety margin, useful in planning maintenance. See MDP.

Pressure (Gauge, Absolute) The force with which a gas or liquid presses against a unit area: pounds per square inch, torr, etc. Measured relative to atmospheric pressure (gauge), or relative to a near-zero reference (absolute).

Priming, Pull-Down, Pump-Down See Evacuation.

Recovery Pressure See Pickup Pressure.

Sealpot (Hotwell) The vessel or well located below the ejector condensers which receives liquid that drains by gravity from the condensers. Those drainlegs which discharge beneath the liquid surface are sealed against an influx of air. The seal pot is sized to fill all drainlegs with water to 34 or more ft and still maintain the seal. Immiscible condensate may be recovered by decanting in this vessel.

Secondaries The relatively small last one or two stages of an ejector that has a large first-stage or booster stages. Where the large boosters may handle mostly condensable vapor, as in a refrigeration system, the secondaries handle only the small quantity of air and vapors of saturation emerging from the first intercondenser.

Shutoff See No-Load.

Spacer See Nozzle Extension.

Stable Operation Operation of an ejector stage on its basic performance curve without fluctuation of the suction pressure. This definition has significance only for critical stages.

Stage Loosely, a stage assembly. More specifically, a pumping or compression operation performed by one or more stage assemblies in parallel. Operationally, a stage is characterized by the suction pressure and load, the discharge pressure, and the motive fluid conditions.

Steam Chest The small chamber immediately behind the steam nozzle in some nozzle assemblies. Typically, it has a steam inlet at one side and an inspection plug in line with the nozzle axis for nozzle inspection. It may be integral with the nozzle plate. It may have a connection for a pressure gauge, and it may have a small strainer to protect the nozzle from scale.

Steam-Jet Refrigeration A vacuum ejector system with one or more large boosters that chill a stream of water by removing flashed vapor in a flash tank.

Suction (Inlet) The stage connection into which the load vapors are drawn.

Suction (Air) Chamber The large chamber that directs the load vapor toward the diffuser inlet to mix with the high-velocity steam. Mechanically, it maintains the position of the nozzle with respect to the diffuser. It is usually drilled and tapped for test pressure connections.

Suction Pressure The pressure at the ejector suction, usually in torr for vacuum ejectors. Usually measured in the suction chamber.

Surface Condenser A condenser in which heat is transferred from the condensing vapors through tubes to a coolant liquid, usually cooling water. The process materials and water do not mix. Examples are shell-and-tube and helical-coil exchangers.

Syphon A steam-jet liquid ejector. Steam or other condensable vapor condenses as it mixes with the pumped liquid.

Tailpipe See Drainleg.

Thermocompressor See Gas-Jet Ejector.

Throat (Bore) The minimum diameter of the nozzle or diffuser. In the critical-flow nozzle, it is the location where the velocity is sonic. In the critical diffuser at design operating conditions, it is where the compression shock occurs, with the quick transition from supersonic to subsonic flow. Nominally, the velocity is sonic in the diffuser throat. In the sub-critical diffuser, the throat velocity is subsonic.

torr After Torricelli; a unit of pressure defined as 1/760 of a standard atmosphere, 1 mmHg abs, and 133.3 pascals (SI pressure).

Trap See Cold Trap.

Ultimate Pressure (Vacuum) See No-Load.

Vacuum Subatmospheric pressure measured relative to atmospheric pressure. Or, the condition of being at a pressure less than atmospheric. Precisely, the absence of anything, empty.

Vacuum System A process system or other apparatus, surrounded by atmospheric pressure, from which a vacuum pump extracts gases.

Ventilator See Jet Blower.

Answer to problem number 7

Water vapor load = 25 pph (0.75 lb water vapor/lb air) [p 112]
= 18.75 pph

Total DAE load = $25 + \dfrac{18.75}{0.99(0.8)}$ [pp 79,80]
=48.7 pph

Problem number 8

A 2-stage ejector is to be designed for a load of 100 pph air and 300 pph water vapor at 60 torr and 100 °F. The terminal (condensing) difference is 3 °F and the vent (approach) difference is 3 °F, cooling water 90 °F, steam 150 psig D&S. Find the intercondenser pressure for lowest operating cost. Given, equivalent investment costs are $300 per pph steam and $400 per gpm of cooling water.

Answer on page 294

Appendix

E

Sizing Air and Steam Metering Orifices for Testing

Performance-testing vacuum ejectors in the field with a dry air load is simple. A critical-flow multiple-orifice meter (flute, piccolo) is quickly adjusted to admit the desired load, and the pressure is measured at the first, last, and other stages.

The test procedures are described in Chap. 8. Procedures for adding steam as a condensable vapor load are also described, although this is complicated for field testing. As a rough guide, the mass flow of atmospheric-pressure saturated steam through a critical-flow orifice will be 67 percent of the standard air flow. The Heat Exchange Institute Standards for Steam Jet Vacuum Systems has much helpful information for such tests. A brief summary will be given here.

The quickest and usually the cheapest way to obtain a multiple-orifice device is to buy it. They are available with a I-in NPT screwed outlet and air capacities of up to 128 and 256 pph atmospheric air. Larger sizes are also available.

Occasionally, however, it is appropriate to prepare your own test orifice or set of orifices for metering air or for metering steam. You may be in a hurry, you may want to have a test orifice sized for each ejector and installed at the ejector, you may have idle machine shop time or low machine shop costs, or you may want a configuration that is not commercially available. Examples of special configurations include a ½ -inch unit, metric-load orifice sizes and metric connections, and large-orifice assemblies into which you can insert a 1-in unit for precise adjustment of the load. These examples will be discussed later, after describing how to size and fabricate the orifices.

Appendix B gave rules of thumb for sizing orifices. The information here is a little more accurate, and the HEI standards are the industry reference.

Air orifice sizing

The throat diameter of a critical-flow orifice is the basis for designing the orifice. Other dimensions are small multiples of the throat diameter. The orifice size is a function of the desired mass flow and the ambient temperature and pressure. The HEI standard has tables, charts, and equations for applying the orifices. One chart shows the variation in nozzle flow coefficient with nozzle size for an air orifice: from 0.956 for a $^1/_{16}$ -in orifice to 0.993 for a 1-in orifice at standard atmospheric conditions.

Assuming that a 2 percent error in flow measurement is well within the requirements of most users in a field testing situation, I fit the following equation to the HEI data, exactly at 60 pph ($^1/_4$ -in orifice) at standard conditions of 30 inHg (762 torr) and 70°F (21°C). The equation implicitly assumes a constant flow coefficient over the range, leading to 0.7 percent mass flow undersizing of a 4 pph orifice, and 1.6 percent mass flow oversizing of a 1000 pph nozzle. If you need a 1000 pph orifice, you probably should refer to the HEI standard for more accuracy.

$$d = 0.0326 \frac{\left[(460 + T_a)/(460 + 70)\right]^{0.25}}{(P_a/762)^{0.5}} (W_a)^{0.5} \tag{E.1}$$

where d = orifice throat diameter, in
T_a = atmospheric temperature, °f
P_a = atmospheric pressure, torr
W_a = mass flow of air, pph

As an example, size an air orifice for 45 pph air at standard atmospheric temperature and pressure.

$$d = 0.0326 \frac{\left[(460+70)/(460+70)\right]^{0.25}}{(762/762)^{0.5}} (45)^{0.5} = 0.219 \text{ in}$$

Calculating airflow in a given orifice size

When you need to calculate the airflow in a given orifice size, use this rearrangement of Eq. (E. 1):

$$W_a = 941d^2 \frac{(P_a / 762)}{[(460 + T_a)/(460 + 70)]^{0.5}} \tag{E.2}$$

As an example, find the airflow in a 0.25-in orifice at ambient conditions of 50°F and 31 inHg (787 torr).

$$W_a = 941(0.25)^2 \frac{(787/762)}{[(460+50)/(460+70)]^{0.5}} = 61.9 \text{ pph}$$

Flow adjustment for nonstandard conditions. When using air orifices in field tests, use the following equation to determine the appropriate adjustment factor for the flow under actual ambient conditions.

$$\frac{W_a}{W_{a,std}} = \frac{(P_a / P_{a,std})}{(T_a / T_{a,std})^{0.5}} \tag{E.3}$$

As an example, find the correction factor for ambient conditions of 100°F and 29 inHg absolute, for an orifice set sized for 70°F and 30 in Hg. Use the factor to introduce a load of 75 pph air into a vacuum system.

$$\frac{W_a}{W_{a,std}} = \frac{(29/30)}{[(460+100)/(460+70)]^{0.5}} = 0.94$$

The orifice combination to use should have a combined "standard" flow capacity of 75/0.94 = 80 pph.

Sizing an orifice set for nonstandard plant conditions. Assume that we wish to size a set of air orifices for a plant at Denver, Colorado, the "mile-high" city. Let the plant also be at that elevation, and use the altitude-pressure table, Table E.1. By interpolation, find 40°F and 24.7 inHg (627 torr) for that altitude. You may wish to adjust the temperature to represent the ambient temperature at your plant site during typical tests. The equation for air orifice sizes at the Denver standard conditions is

$$d = 0.0326 \frac{[(460+40)/(460+70)]^{0.25}}{(627/762)^{0.5}} (W_a)^{0.5}$$

$$d = 0.0354 (W_a)^{0.5}$$

Any orifice devices which are marked to indicate the nominal air-metering capacity should also be marked to identify the temperature and pressure bases, unless these are very close to 760 torr and 70°F.

TABLE E.1 Altitude Pressure

Feet Altitude	Temp F	Pressure In. Hg	Feet Altitude	Temp F	Pressure In. Hg
0	59.00	29.92	55,000	-69.70	2.712
50			60,000	-69.70	2.135
100	58.64	29.81	65,000	-69.70	1.682
200	58.28	29.71	70,000	-67.42	1.325
300	57.93	29.60	75,000	-64.70	1.046
400	57.57	29.49	80,000	-61.98	8.273-1
500	57.21	29.38	85,000	-59.26	6.553-1
600	56.86	29.28	90,000	-56.53	5.200-1
700	56.50	29.17	95,000	-53.82	4.133-1
800	56.15	29.07	100,000	-51.10	3.290-1
900	55.79	28.96	110,000	-41.29	2.098-1
1,000	55.43	28.86	120,000	-26.09	1.358-1
1,500	53.65	28.33	130,000	-10.91	8.921-2
2,000	51.87	27.82	140,000	4.25	5.947-2
2,500	50.09	27.32	150,000	19.40	4.018-2
3,000	48.30	26.82	160,000	27.50	2.746-2
3,500	46.52	26.33	170,000	27.50	1.880-2
4,000	44.74	25.84	180,000	18.88	1.2843-2
4,500	42.96	25.37	190,000	8.10	8.7026-3
5,000	41.17	24.90	200,000	-2.67	5.8457-3
7,500	32.26	22.66	210,000	-21.96	3.8790-3
10,000	23.36	20.58	220,000	-43.46	2.5229-3
15,000	5.55	16.89	230,000	-64.94	1.6046-3
20,000	-12.26	13.76	240,000	-86.40	9.9546-4
25,000	-30.05	11.12	250,000	-107.84	6.0065-4
30,000	-47.83	8.903	260,000	-129.26	3.5127-4
35,000	-65.61	7.060	270,000	-134.50	2.0046-4
40,000	-69.70	5.558	280,000	-134.50	1.1433-4
45,000	-69.70	4.375	290,000	-134.50	6.5244-5
50,000	-69.70	3.444	300,000	-126.77	3.7368-5

Note: A number (preceded by a plus or minus sign) following the entry in each block indicates the power of ten by which that entry should be multiplied.

SOURCE OF DATA: NASA, U.S. Standard Atmosphere, 1962

The correction factor for nonstandard conditions for this Denver orifice set becomes

$$\frac{W_a}{W_{a,std}} = \frac{(P_a/627)}{[(460+T_a)/(460+40)]^{0.5}}$$

Sizing for other engineering units. As an example, adapt the sizing and adjustment equation to a system of millimeters, kilograms, millibar, and degrees Celsius for a plant site having standard conditions of 950 mbar and 15°C. Noting that 1.0 in = 25.4 mm, 1.0 kg = 2.2 Ibm, absolute zero temperature is -273°C, and 1.0 torr = 1.3332 mbar, convert 762 torr to 1016 mbar, and convert 70°F to 21°C:

$$d(mm) = 0.0326(25.4)\frac{\left[\left(273 + T_a(^\circ C)\right)/(273+21)\right]^{0.25}}{\left[P_a(mbar)/1016\right]^{0.5}}\left[2.2W_a(kg/hr)\right]^{0.5}$$

$$d = 1.26\frac{\left[(273+T_a)/(273+21)\right]^{0.25}}{\left[P_a/1016\right]^{0.5}}\left[W_a\right]^{0.5}$$

Which, for this plant location standard conditions, becomes

$$d = 1.26(W_a)^{0.5}$$

And the correction for departure from the location standard conditions becomes

$$\frac{W_a}{W_{a,std}} = \frac{(P_a/950)}{\left[(273+T_a)/(273+15)\right]^{0.5}}$$

Steam orifice sizing

Equation (4.25) gives the mass flow of dry and saturated steam through a nozzle having an orifice flow coefficient of 0.97:

$$W_s = 50d^2P_m{}^{0.96}$$

where W_s is in pph, d is in inches, and P_m is in psia.

See Chap. 4 for the range, accuracy, and superheat adjustments. Rearranging to solve for the diameter,

$$d = 0.141\frac{(W_s)^{0.5}}{(P_m)^{0.48}} \tag{E.4}$$

Sizing for other engineering units. Translate to a system of millimeters, kilograms per hour, and bar, noting that 1.0 bar = 14.5 psia,

$$W_s(kg/h) = \frac{50}{2.2}\left[\frac{d(mm)}{25.4}\right]^2\left[P_m(bar)14.5\right]^{0.96}$$

which reduces to

$$W_s = 0.459(d)^2(P_m)^{0.96} \tag{E.5}$$

and

$$d = 0.148\frac{(W_s)^{0.5}}{(P_m)^{0.48}} \tag{E.6}$$

Caution: If you use any of these equations out of the context of this book, remember to identify the proper units for all variables.

Designing and machining air and steam orifices

The design and use of critical-flow orifices is shown in Fig. E.1. The important dimensions of a critical-flow orifice are shown in Fig. E.1a. Those dimensions are small multiples of the throat diameter.

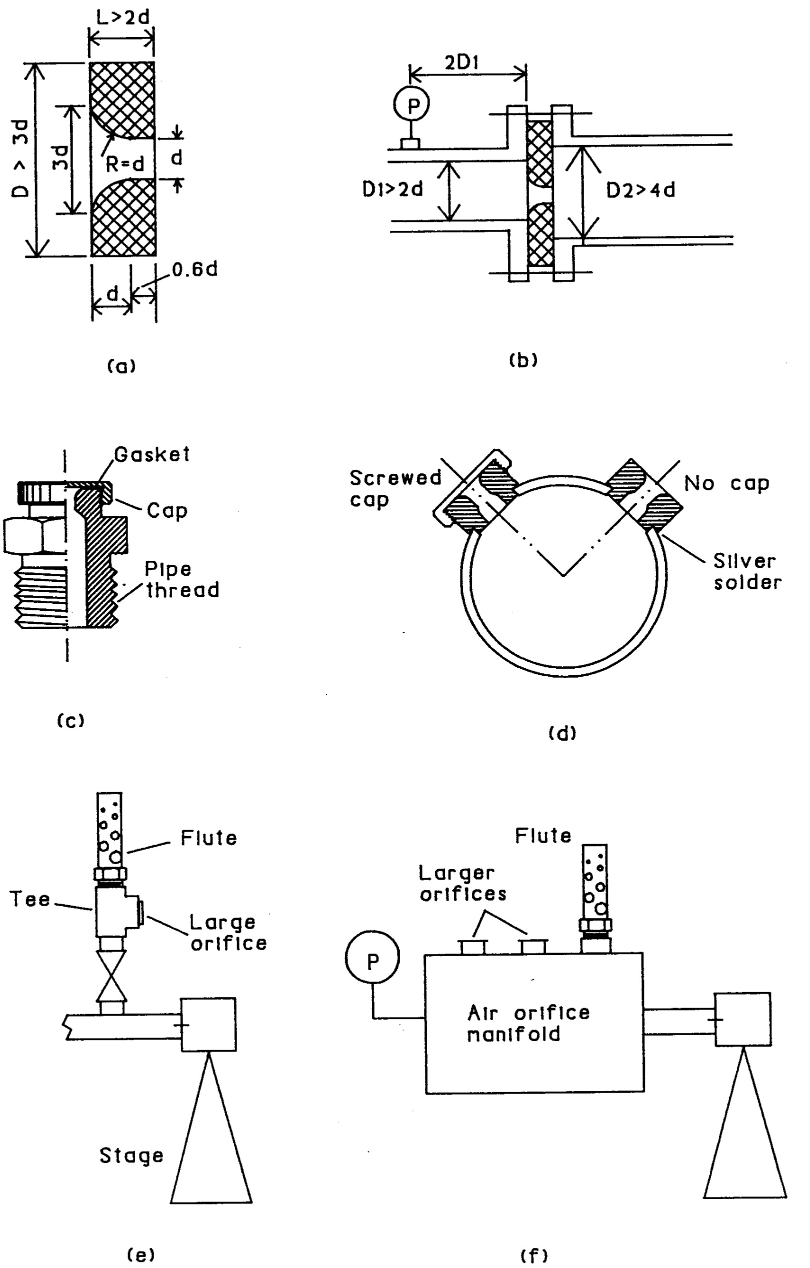
For precision, drill the orifice slightly undersize, then size it with a straight-sided reamer or with a twist drill bit adapted by honing the corners slightly round. Gauge pins may be used to check the size of small orifices and steam nozzles. The entrance should be smoothly rounded, there should be no ridge at either end of the throat, and the whole interior should be polished.

The external design may be any size and shape which is convenient for positioning the orifice, provided it does not interfere with the flow. Figure E.1b shows an orifice plate, with minimum upstream and downstream diameters. My upstream diameter is more lenient than the HEI recommendation.

Figure E.1c shows an orifice in a screwed plug assembly with an optional screwed cap. The discharge side is back-drilled to keep the orifice throat short. This is a versatile device. The orifice may be sized for routine testing of a specific ejector, sized for about 80 percent of the design air load. It may be left installed at the ejector site, or it may be installed only for the test. The cap protects the orifice from damage when not in use. A valve below the orifice protects it from process corrosion or fouling, and permits removal without disrupting ejector operation. Another use of this device is to extend the range of a convenient sized flute, as shown in Fig. E.1e.

Figure E.1d shows a small multiple-orifice device for field testing. The orifices are typically in rows 90° apart and staggered along the axis for convenience in marking the sizes and avoiding flow interference between orifices. Avoid any arrangement in which the jet of air from one orifice is aimed directly at another orifice. The largest orifices are located near the outlet. The example is shown with and without caps. Caps protect the orifices from damage and fouling, and they tightly seal the inactive orifices. On the other hand, they add to the cost, the bulk, and the weight of the flute, and they may be dropped and lost.

Figure E.1e shows typical field testing with a flute in combination with a large orifice to extend the air test load range. Figure E.1f shows a typical shop test configuration. In shop tests, small pieces of soft rubber sheet may be used to close inactive orifices. The vacuum will hold them in place.



igure E.1 Design and use of critical flow orifices: (a) basic dimensions, polish interior and round corners; (b) in-line orifice for steam or gas metering; (c) orifice plug; cap is optional; (d) flute, with caps and without caps; (e) field test using flute and large additional orifice; (j) shop test air load setup

Use of multiple-orifice devices

The first such device I used was one I made of ½ in pipe, with several different-sized orifices. It was small and fit conveniently into the ½ in connection that was available on most of the ejectors I tested at the time. I used tape to cover the inactive orifices. When I learned that a 1-in size was commercially available, I bought one and carried pipe fittings to adapt it to the ½-in test connections. Thereafter, I recommended a 1-in NPT test load connection in all ejector installations, for testing with the flute. I prefer a capless design and cover the inactive orifices with tape. I do have minor misgivings about this practice, and I intend to conduct a leak test sometime to determine how much air leaks through the tape.

If I were testing ejectors with design air loads less than 64 pph, I would prefer the more convenient ½-in size, capless, if a good one were available. Some 1-in units have loads up to 256 pph. Larger-size units are cumbersome to carry and adapt to the smaller test openings commonly available. If you wish to provide air loads to large and small ejectors, you may wish to buy a large and a smaller unit. Or, you may wish to buy a 1-in unit and supplement it with one or two additional orifices as shown in Figs. E.1e and f.

Appendix

F

Drill Sizes: Fractional, Wire, Letter, and Metric

TABLE F.1 Drill Sizes: Fractional, Wire, Letter, and Metric

Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.
0.0059	97		0.0413		1.05	0.1065	36		0.1960	9	
0.0063	96		0.0420	58		0.1083		2.75	0.1969		5.10
0.0067	95		0.0430	57		0.1094	**7/64"**		0.1990	8	
0.0071	94		0.0433		1.10	0.1100	35		0.2008		5.10
0.0075	93		0.0453		1.15	0.1102		2.80	0.2010	7	
0.0079	92	0.20	0.0465	56		0.1110	34		0.2031	**13/64”**	
0.0083	91		0.0469	**3/64"**		0.1130	33		0.2040	6	
0.0087	90	0.22	0.0472		1.20	0.1142		2.90	0.2047		5.20
0.0091	89		0.0492		1.25	0.1160	32		0.2055	5	
0.0095	88		0.0512		1.30	0.1181		3.00	0.2067		5.25
0.0098		0.25	0.0520	55		0.1200	31		0.2087		0.23
0.0100	87		0.0531		1.35	0.1220		3.10	0.2090	4	
0.0105	86		0.0550	54		0.1250	**1/8"**		0.2126		5.40
0.0110	85	0.28	0.0551		1.40	0.1260		3.20	0.2130	3	
0.0115	84		0.0571		1.45	0.1280		3.25	0.2165		5.50
0.0118		0.30	0.0591		1.50	0.1285	30		0.2188	**7/32"**	
0.0120	83		0.0595	53		0.1299		3.30	0.2205		5.60
0.0125	82		0.0610		1.55	0.1339		3.40	0.2210	2	
0.0126		0.32	0.0625	**1/16"**		0.1360	29		0.2244		5.70
0.0130	81		0.0630		1.60	0.1378		3.50	0.2264		5.75
0.0135	80		0.0635	52		0.1405	28		0.2280	1	
0.0138		0.35	0.0650		1.65	0.1406	**9/64"**		0.2283		5.80
0.0145	79		0.0669		1.70	0.1417		3.60	0.2323		5.90
0.0156	**1/64"**		0.0670	51		0.1440	27		0.2340	A	
0.0157		0.40	0.0689		1.75	0.1457		3.70	0.2344	**15/64"**	
0.0160	78		0.0700	50		0.1470	26		0.2362		6.00
0.0177		0.45	0.0709		1.80	0.1476		3.75	0.2380	B	
0.0180	77		0.0728		1.85	0.1495	25		0.2402		6.10
0.0197		0.50	0.0730	49		0.1496		3.80	0.2420	C	
0.0200	76		0.0748		1.90	0.1520	24		0.2441		6.20
0.0210	75		0.0760	48		0.1535		3.90	0.2460	D	
0.0217		0.55	0.0768		1.95	0.1540	23		0.2461		6.25
0.0225	74		0.0781	**5/64"**		0.1562	**5/32"**		0.2480		6.30
0.0236		0.60	0.0785	47		0.1570	22		0.2500	**1/4" E**	
0.0240	73		0.0787		2.00	0.1575		4.00	0.2520		6.40
0.0250	72		0.0807		2.05	0.1590	21		0.2559		6.50
0.0256		0.65	0.0810	46		0.1610	20		0.2570	F	
0.0260	71		0.0820	45		0.1614		0.17	0.2598		6.00
0.0276		0.70	0.0827		2.10	0.1654		4.20	0.2610	G	
0.0280	70		0.0846		2.15	0.1660	19		0.2638		6.70
0.0292	69		0.0860	44		0.1673		4.25	0.2656	**17/64"**	
0.0295		0.75	0.0866		2.20	0.1693		4.30	0.2657		6.75
0.0310	68		0.0886		2.25	0.1695	18		0.2660	H	
0.0312	**1/32"**		0.0890	43		0.1719	**11/64"**		0.2677		6.80
0.0315	Fract.	0.80	0.0906		2.30	0.1730	17		0.2717		6.90
0.0320	67		0.0925		2.35	0.1732		4.40	0.2720	I	
0.0330	66		0.0935	42		0.1770	16		0.2756		7.00
0.0335		0.85	0.0938	**3/32"**		0.1772		4.50	0.2770	J	
0.0350	65		0.0945		2.40	0.1800	15		0.2795		7.10
0.0354		0.90	0.0960	41		0.1811		4.60	0.2810	K	
0.0360	64		0.0965		2.45	0.1820	14		0.2812	**9/32"**	
0.0370	63		0.0980	40		0.1850	13	4.70	0.2835		7.20
0.0374		0.95	0.0984		2.50	0.1870		4.75	0.2854		7.25
0.0380	62		0.0995	39		0.1875	**3/16"**		0.2874		7.30
0.0390	61		0.1015	38		0.1890	12	4.80	0.2900	L	
0.0394		1.00	0.1024		2.60	0.1910	11		0.2913		7.40
0.0400	60		0.1040	37		0.1929		4.90			
0.0410	59		0.1063		2.70	0.1935	10				

(SOURCE: Cleveland Twist Drill Company)

TABLE F.1 Drill Sizes: Fractional, Wire, Letter, and Metric (Continued)

Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.	Decimal	Fract. Wire Letter	mm.
0.2950	M		0.4688	**15/32"**		0.9843		25.0	1.4961		38.0
0.2953		7.50	0.4724		12.0	0.9844	**63/64"**		1.5000	**1-1/2"**	
0.2969	**19/64"**		0.4844	**31/64"**		1.0000	**1"**		1.5156	**1-33/64"**	
0.2992		7.60	0.4921		12.5	1.0039		25.5	1.5157		38.5
0.3020	N		0.5000	**1/2"**		1.0156	**1-1/64"**		1.5312	**1-17/32"**	
0.3031		7.70	0.5118		13.0	1.0236		26.0	1.5354		39.0
0.3051		7.75	0.5156	**33/64"**		1.0312	**1-1/32"**		1.5469	**1-35/64"**	
0.3071		7.80	0.5312	**17/32"**		1.0433		26.5	1.5551		39.5
0.3110		7.90	0.5315		13.5	1.0469	**1-3/64"**		1.5625	**1-9/16"**	
0.3125	**5/16"**		0.5469	**35/64"**		1.0625	**1-1/16"**		1.5748		40.0
0.3150		8.00	0.5512		14.0	1.0630		27.0	1.5781	**1-37/64"**	
0.3160	O		0.5625	**9/16"**		1.0781	**1-5/64"**		1.5938	**1-19/32"**	
0.3189		8.10	0.5709		14.5	1.0827		27.5	1.5945		40.5
0.3228		8.20	0.5781	**37/64"**		1.0938	**1-3/32"**		1.6094	**1-39/64"**	
0.3230	P		0.5906		15.0	1.1024		28.0	1.6142		41.0
0.3248		8.25	0.5938	**19/32"**		1.1094	**1-7/64"**		1.6250	**1-5/8"**	
0.3268		8.30	0.6094	**39/64"**		1.1220		28.5	1.6339		41.5
0.3281	**21/64"**		0.6102		15.5	1.1250	**1-1/8"**		1.6406	**1-41/64"**	
0.3307		8.40	0.6250	**5/8"**		1.1406	**1-9/64"**		1.6535		42.0
0.3320	Q		0.6299		16.0	1.1417		29.0	1.6562	**1-21/32"**	
0.3346		8.50	0.6406	**41/64"**		1.1562	**1-5/32"**		1.6719	**1-43/64"**	
0.3386		8.60	0.6496		16.5	1.1614		29.5	1.6732		42.5
0.3390	R		0.6562	**21/32"**		1.1719	**1-11/64"**		1.6875	**1-11/16"**	
0.3425		8.70	0.6693		17.0	1.1811		30.0	1.6929		43.0
0.3438	**11/32"**		0.6719	**43/64"**		1.1875	**1-3/16"**		1.7031	**1-45/64"**	
0.3445		8.75	0.6875	**11/16"**		1.2008		30.5	1.7126		43.5
0.3465		8.80	0.6890		17.5	1.2031	**1-13/64"**		1.7188	**1-23/32"**	
0.3480	S		0.7031	**45/64"**		1.2188	**1-7/32"**		1.7323		44.0
0.3504		8.90	0.7087		18.0	1.2205		31.0	1.7344	**1-47/64"**	
0.3543		9.00	0.7188	**23/32"**		1.2344	**1-15/64"**		1.7500	**1-3/4"**	
0.3580	T		0.7283		18.5	1.2402		31.5	1.7520		44.5
0.3583		9.10	0.7344	**47/64"**		1 .2500	**1-1/4"**		1.7656	**1-49/64"**	
0.3594	**23/64"**		0.7480		19.0	1.2598		32.0	1.7717		45.0
0.3622		9.20	0.7500	**3/4"**		1.2656	**1-17/64"**		1.7812	**1-25/32"**	
0.3642		9.25	0.7656	**49/64"**		1.2795		32.5	1.7913		45.5
0.3661		9.30	0.7677		19.5	1.2812	**1-9/32"**		1.7969	**1-51/64"**	
0.3680	U		0.7812	**25/32"**		1.2969	**1-19/64"**		1.8110		46.0
0.3701		9.40	0.7874		20.0	1.2992		33.0	1.8125	**1-13/16"**	
0.3740		9.50	0.7969	**51/64"**		1.3125	**1-5/16"**		1.8281	**1-53/64"**	
0.3750	**3/8"**		0.8071		20.5	1.3189		33.5	1.8307		46.5
0.3770	V		0.8125	**13/16"**		1.3281	**1-21/64"**		1.8438	**1-27/32"**	
0.3780		9.60	0.8268		21.0	1.3386		34.0	1.8504		47.0
0.3819		9.70	0.8281	**53/64"**		1.3438	**1-11/32"**		1.8594	**1-55/64"**	
0.3839		9.75	0.8438	**27/32"**		1.3583		34.5	1.8701		47.5
0.3858		9.80	0.8465		21.5	1.3594	**1-23/64"**		1.8750	**1-7/8"**	
0.3860	W		0.8594	**55/64"**		1.3750	**1-3/8"**		1.8898		48.0
0.3898		9.00	0.8661		22.0	1.3780		35.0	1.8906	**1-57/64"**	
0.3906	**25/64"**		0.8750	**7/8"**		1.3906	**1-25/64"**		1.9062	**1-29/32"**	
0.3937		10.0	0.8858		22.5	1.3976		35.5	1.9094		48.5
0.3970	X		0.8906	**57/64"**		1.4062	**1-13/32"**		1.9219	**1-59/64"**	
0.4040	Y		0.9055		23.0	1.4173		36.0	1.9291		49.0
0.4062	**13/32"**		0.9062	**29/32"**		1.4219	**1-27/64"**		1.9375	**1-15/16"**	
0.4130	Z		0.9219	**59/64"**		1.4370		36.5	1.9488		49.5
0.4134		10.5	0.9252		23.5	1.4375	**1-7/16"**		1.9531	**1-61/64"**	
0.4219	**27/64"**		0.9375	**15/16"**		1.4531	**1-29/64"**		1.9685		50.0
0.4331		11.0	0.9449		24.0	1.4567		37.0	1 9688	**1-31/32"**	
0.4375	**7/16"**		0.9531	**61/64"**		1.4688	**1-15/32"**		1.9844	**1-63/64"**	
0.4528		11.5	0.9646		24.5	1.4764		37.5	1.9882		50.5
0.4531	**29/64"**		0.9688	**31/32"**		1.4844	**1-31/64"**		2.0000	**2"**	

Answer to problem number 10

Here we are to use the curves on page 93. For two stages with intercondenser, 90 °F water, and 100 psig steam, $R_a = 3.3$. The multiplier for 165 psia steam is 0.92, giving an adjusted R_a of 3.3(0.92) = 3.0. The load of 100 pph air plus 300 pph water vapor was previously calculated to total 480 pph dae. The uncorrected steam usage is 3.0(480) = 1440 pph. This calculation treats the load as noncondensable. We need a rational method for crediting the ejector for the vapor load removed by the intercondenser.

From our recent experience we saw the optimum condenser pressure to be near 150-175 torr. An ejector designed for air load only would have a higher condenser pressure, about 250 torr. Testing this assumption for reasonableness, we see compression ratios of 250/60=4.2 for the first stage and 813/250=3.3 for the second stages. This looks well balanced and near optimum design for a noncondensable load.

Most of the vapor was condensed and less than 0.3 pound of vapor per pound of air remained in the vent. The load to stage 2 is thus about 110 pph dae. Had the load to stage 1 been 480 pph of dry air, the second stage would have seen a load of 528 pph dae. Thus, a credit of 528-110=418 pph may be taken for reducing the load to the second stage.

Finally, for the balanced design here the second stage R_a=half the overall value, or 1.5. The credit for removing most of the vapor load to the second stage is 1.5(418)=627 pph. The net steam use is 1440 – 627 = 813 pph at 150 psig.

This result is uncomfortably close to the 850 pph from the detailed estimate in problem 8. It uses data from an optimized noncondensables ejector to predict he design of one handling a significant condensable vapor load. I need more experience with this method before I would trust it much. Beware!

Appendix G

Ejector Operating Costs and Design Optimization

G.1 The High Importance of Operating Costs

The high importance of ejector operating costs is the major reason I was assigned to study ejectors years ago. The operating costs of most ejector systems, when considered over the life of the ejector, are an order of magnitude higher than the purchase price. A specific assignment was to find out how to evaluate bids: "Will the higher-priced ejectors really use less steam and water, and will they work as well as the cheaper ones?" The general answer to both questions is "yes."

The first step in designing an ejector for economy or in evaluating bids is to define the utilities costs. As described in Chap. 10, the most useful basis is investment equivalents. They answer questions typified by, how much more am I willing to spend for an ejector that uses 1 pph less steam to do the same job? If this question is not answered with a realistically high value, the buyer will tend to buy the lowest-priced ejector, which will use the most steam and water. The accumulated result of many such decisions is a "creeping expansion" of utilities consumption, which is not controlled at its source, but must be accommodated at a plant level. In turn, this may contribute to reduced profits and premature economic obsolescence.

As a starting point, consider the incremental cost per unit of consumption. For example, suppose that this is $3.00 per 1000 lb of steam. An ejector in nearly continuous service will be running about

8000 h per year. If your economic guidelines require a payback period of no more than 5 years, then 40,000 operating hours will be your payback usage time. The investment equivalent for steam is (40,000)($3/1000) = $120 per pph steam, based on incremental costs only. What more must be considered?

The period cost and investment cost per unit should also be considered. They represent the fixed costs of operations, and investment in shared facilities. The period cost includes such "fixed" costs as labor. Investment transfers represent a share of the plant facilities required to deliver steam to the ejector-water treatment, steam generators, and distribution headers-which may be increased by a policy that leads to buying an ejector which uses 1 pph more steam. Both costs are usually based on capacity requirements, often expressed as usage per year.

The general policy or practice which directs this choice may in the future lead to a need to expand the facilities sooner or by a greater amount, or may prevent retirement of a marginal utilities facility, leading to higher future labor costs. Because each plant situation is unique, no blanket rule is appropriate here. You may find that including period and investment costs increases the investment equivalent by 200 percent, 50 percent, or nothing. As an example of how to use these data, assume that the plant fixed cost and investment transfer are $2 and $10 per 1000 lb per year. This might represent a plant with a healthy future, including plans for expansion or modernization of the steam facilities. The addition to the investment equivalent for 1 pph steam is 8000($2 + $10)/1000 = $96 per pph steam. The total investment equivalent is $120 + $96 = $216 per pph steam. Round this off to $200 per pph.

If the condensate must be treated in a waste treatment system, a portion of the cost-charging formula will probably be based on the volumetric flow. This may be a small extra for steam condensate, but will be increased by a factor of 50 or more if contact condensers are used. For that reason, surface condensers are usually preferred, and contact condensers are the exception.

The investment equivalent per gpm cooling water will be similar to that for steam, although possibly varying by a factor of 2 or more above or below that for steam. The absolute value of both utilities may vary by a factor of 3 or more above or below the values in the above example, depending on the location of the plant and the economic objectives of the project.

G.2 Design Optimization Considerations, Example Calculations, and Bid Comparisons

When preparing a bid for a steam-jet ejector, the manufacturer can make tradeoffs in the price versus utilities usage, especially in multi stage systems with surface condensers. Important adjustable variables are the number of stages and condensers, the interstage pressures, condenser vent temperatures, and cooling-water usage. The application engineer will have basic data similar to those given in Chaps. 4 and 5, but in much more detail and in the form of convenient computer programs. Given the ejector specification data, the programs quickly yield the configuration and conditions resulting in lowest total cost.

Your task in evaluating bids is to check the attractive bids for general correctness, using cruder methods and motivated by the desire to avoid buying a mistake. The required stage and condenser calculations were described in detail in Chaps. 4 and 5, and bid evaluation was discussed in general terms in Chap. 10.

The calculations which follow will demonstrate the stage-by-stage design of an ejector, using simplified methods. You may find these calculations to be tedious at first, then routine as you learn the procedures. The results of a number of such design calculations are summarized in Table G.1. Each of those designs satisfies the same performance specifications, but they differ in the (implicit) importance assigned to steam and water costs. You will gain perspective on the importance of several of the adjustable design variables, and will learn about the interrelationships among several of these variables.

Example ejector design optimization

Provide an ejector to handle 100 pph air and 215 pph water vapor at 15 torr and 70°F, using 150-psig steam and 90 °F cooling water, and discharging through an aftercondenser at 812 torr. Maximum cooling-water outlet temperature is 122°F, surface condensers are required, and all materials are to be 304 SS. Investment equivalents are not given, so a variety of options will be explored and evaluated.

The dry air equivalent (DAE) load is calculated as described in Chap. 4.

$$\text{Load} = \frac{100}{(1.0)(1.0)} + \frac{215}{(0.8)(1.0)} = 368 \text{ pph DAE}$$

As a start, consider the simplest configuration, a two-stage noncondensing ejector, as potentially the cheapest. As seen in Fig. 4.12, this uses 24 pounds of 100-psig steam per pound of air. Including the steam usage multiplier for 150-psig steam from Fig. 4.13, the total steam usage is

$$\text{Steam} = 368(24)(0.94) = 8800 \text{ pph}$$

TABLE G.1 Ejector "Bid Comparison" Summary

Bases: 100 pph air, 215 pph water vapor, at 15 torr and 70°F
150-psig steam, 90 °F cooling water, 815 torr discharge
Aftercondenser, surface condensers, all 304 SS
Steam rates, Chap. 4, Condensers, Chap5

Case	Number		Approach		Utilities			Intercondensers		Condenser
No.	Stages	Intercon-densers	temperature, °F	Price $K	Steam pph	Water gpm	Total (N/A)	Vents °F	Pressure Torr	areas ft^2
1	2	0	NA	38	8,800	1,800	10,600	NA	NA	--/471
2			NA	30	8,800	(S)	8,800	(S)=silencer		
3	2	1	20	23	2,411	564	2,975	110	120	225/95
4			20	27	2,712	181	2,893	110	100	220/130
5			10	28	2,500	180	2,680	100	100	286/124
6			10	29	1,910	137	2,047	100	125	356/77
7			5	33	1,844	139	1,983	95	110	490/76
8	3	1	40	29	2,088	480	2,568	130	-/175	152/81
9	3	2	30	25	1,784	342	2,166	120/130	120/250	154/52/34
10			20	27	1,264	192	1,456	110/120	100/250	203/85/20
11			10	30	1,206	126	1,332	100/120	70/200	252/92/39
12			10	33	1,195	110	1,305	100/120	80/250	365/102/23
13			5	42	1,069	77	1,144	95/110	75/250	616/144/23
14			3	54	1,029	80	1,109	93/110	75/225	1,037/139/28
15			"0"	NA	962	86	1,050	90/90	80/250	(infinite)
16	4	3	10	33	1,047	124	1,171	100/105/120	75/140/275	311/168/39/19
17			5	35	1,069	100	1,169	95/110/120	75/125/275	430/92/51/24
18			3	51	1,027	80	1,107	93/107/120	67/120/280	853/176/76/24

From Fig. 10.2, the first stage has an 8-in suction, a 1992 price of \$6000, and price multipliers of 2.0 for two-stage noncondensing and 2.0 for stainless steel.

$$\text{Stage price} = \$6000(2.0)(2.0) = \$24{,}000$$

To size the aftercondenser, select a 10°F rise in cooling-water temperature to minimize the condenser size, and a vent temperature of 150°F at 812 torr. The noncondensable gases are less than 1 percent of the entering vapors, so the mixture may be treated as pure water vapor with a saturation temperature of 215°F. The terminal temperature difference at the inlet is thus 215 - 100 = 115°F, and the vent temperature difference is 150 - 90 = 60°F. The log mean temperature difference (LMTD) for the condenser is

$$\text{LMTD} = \frac{115 - 60}{\ln(115/60)} = 85°\text{F}$$

The latent heat given up by each pound of steam condensing at atmospheric pressure is about 1000 BTU, and the overall condensing coefficient U from Fig. 5.15 is 220 BTU/(h.°F.ft^2). The area is

$$\text{Area} = \frac{W_c\,(\text{latent heat})}{U(\text{LMTD})} = \frac{9000(1000)}{20(85)} = 481\ \text{ft}^2$$

To simplify the analysis, a flat cost of \$30 per square foot was used for all condensers, leading to a price of \$14,000 for this one. The total cost of stages plus condenser for this ejector is \$24,000 + \$14,000 = \$38,000. .

As will soon be seen, adding an intercondenser will reduce the total price. Thus, the only reason for considering this unit might be that it does not require an elevated structure because it has no vacuum condensers. Also, if the steam may be discharged to the atmosphere, a \$6000 silencer may be used to replace the condenser. Basically, this design is not normally suitable for this application.

Add an intercondenser. A condenser between the stages will reduce the load to the second stage, reducing the steam usage in the second stage and reducing the size of the aftercondenser. To start, you select an interstage pressure and vent temperature. Experience is your best guide. If you do not have experience, just start with a pressure and temperature combination and evaluate it. Try another combination, and soon you will have experience! I used an inlet pressure of 120 torr and vent conditions of 110 torr and 120°F (30°F above the water inlet). For all intercondensers I used a pressure drop of 10 percent or 10 torr, whichever was smaller.

For the steam usage of stage 1, obtain $R_a = 2.5$ from Fig. 4.8 (for P_s = 15 torr and P_d = 120 torr) and calculate the usage:

$$W_{S1} = 2.5(368) = 920 \text{ pph steam}$$

The water vapor in the first condenser vent is related to the airflow and the vent temperature and pressure. Enter Fig. 5.4 with the vent temperature of 120°F and pressure of 110 torr, obtaining a ratio of 0.9 lb water vapor per lb air.

$$\text{Water vapor} = 0.9(100) = 90 \text{ pph}$$

Neglecting a modest temperature correction, the DAE of the water vapor is 90/0.81 = 113 pph. The total load to stage 2 is now 100 + 113 = 213 pph DAE. Compression from 110 torr to 812 torr requires an R_a of 7.0, resulting in a steam usage of 7.0(213) = 1491 pph. The total steam used by the two stages is now 920 + 1491 = 2411 pph, considerably less than for the noncondensing configuration.

To size the intercondenser, first find the dew point at the inlet. The molar flows entering are 100/29 = 3.4 moles of air and (215 + 920)/18 = 62.2 moles of water vapor, totaling 65.6 moles. The noncondensable fraction is 3.4/65.6 = 5 percent. The water vapor partial pressure is (1.0 - 0.05)(120 torr) = 114 torr, and the corresponding dew point from Fig. 5.3 or a steam table is 129°F. Use a 10°F water rise as before to calculate the flow rate. Note that the vapor condensed in this condenser is 200 + 920 - 90 = 1030 pph and that the latent heat per pound has increased.

$$\text{Flow}_1 = \frac{1030(1100)}{1\ 10(500)} = 227 \text{ gpm}$$

For 5 percent noncondensables, $U = 210$. The temperature differences are now (129 - 100) = 29°F at the inlet and 30°F at the vent; LMTD = 29.5°F. The intercondenser area is

$$\text{Area}_1 = \frac{1030(1100)}{210(29.5)} = 183 \text{ ft}^2\text{, for a price of \$5500}$$

In a similar way we find the aftercondenser to be 95 ft^2, priced at $2800, and using 343 gpm water. The second stage is now much smaller, with a price of $3000. In summary, the two-stage design has a price of $23,000 and uses 2411 pph steam and 564 gpm cooling water. It is superior to the noncondensing design unless no elevated structure is available and steam and water are almost free.

Add more stages. Now add an additional stage and find that a design with only one intercondenser, after the second stage, is a little higher priced than a two-stage condensing design, but uses less steam and water. Adding a second intercondenser brings us to the typical design for these specifications. Trying a few combinations of interstage pressures soon reveals the best combination for a given set of vent temperatures.

Add condenser area to reduce steam usage. Larger intercondensers lower the vent temperatures, reducing the vapor load to the subsequent stages and reducing the steam usage. As the condensers increase in size, however, they also increase in complexity, with special baffles and counterflow of water and vapor to cool the vent with the incoming water. A common economy feature is to use a series flow arrangement where the water from the first surface condenser flows through the intercondensers and aftercondensers. Occasionally, there are practical limits to this economy: a tiny condenser following a large condenser, for example.

A systematic way to explore the effect of larger condensers is to vary the temperature difference between the vent gas and the inlet water to the first condenser. Differences of 20°, 10°, 5°, 3°, and even 0°F are used here. The zero temperature difference represents an unattainable design, but is useful in identifying the lower limit of steam usage, approached by ejectors designed for extreme economy.

In designing for the very lowest steam usage, it is possible that a four-stage design with three intercondensers might be desirable if steam and water costs are very high. Limitations on the accuracy of the basic data and the methods used for this analysis prevent me from offering a conclusion on that subject. The four-stage design may or may not be superior here. With cooler water, it probably would be.

Comparing the total costs of each design

The results of the above "bid preparation" actions are summarized in Table G.1. Keep in mind that each design is a semioptimized design that was adjusted until it looked right for the approach temperature being used. You will not expect to see such a scatter of designs if you provide understandable (and believable) economic guidelines in your specifications. However, if you suddenly begin to include a high steam cost in your bid inquiry specifications, some manufacturers may question whether you are really prepared to pay the extra price for the improved economy you claim to want. They may not offer their most economical design, or they may offer several alternative designs so you can select the best for you. Be prepared for a credibility test.

Notice that the "Utilities total" column is the sum of the steam usage in pph and the water usage in gpm. These are combined to simplify the analysis, justified on the bases that their investment equivalents tend to be similar, there is room in intercondenser design to trade off water for steam, and water usage is roughly proportional to steam usage, about 8 to 20 percent. This reduces the economic analysis to a straightforward comparison of first cost to a simple total utilities operating cost.

The remaining problem is that every design and procurement situation has its own unique cost considerations. Are there, then, any useful guidelines that can be learned from this exercise? Yes: Always provide some quantitative economic guidelines (investment equivalents) in your specifications, and do not spend too little when you select the successful bidder. Do not expect that a qualitative objective such as "low steam usage is desirable" will yield the designs you need. Have the courage of your convictions, and assume that the manufacturers will still bid competitively when you specify what may appear to be high investment equivalents for steam and water. They know that their competitors see the situation as clearly as they do.

To confirm the validity of the above guidelines, and to quantify them, examine Fig. G.1. It displays the purchase price and total evaluated cost of each of the reasonable designs, plotted versus the utilities totals. The total evaluated cost is shown for a range of utilities investment equivalents from zero to \$1000 per pph or .gpm. The bottom curve shows the purchase price for each of the designs in Table G.1, noting the vent temperature difference (approach) for each. Several of the designs are simply not competitive. A smooth curve is drawn through the lowest points to represent what the ejector market can present to you as the answer to your specifications. That curve is used to create the total evaluated cost curve for the range of investment equivalents considered.

Additional features to note are the "zero approach" limit and the dotted lines through the optimum designs. The "zero approach" vertical line represents the absolute lower limit on utilities usage. It is approached by ejectors with large condensers and corresponding large prices. The dotted lines trace the location of the optimum design on each of the total evaluated cost curves. Two dotted lines are used, delimiting an optimum band. The width of the band is a reminder that it is only necessary to be in the general vicinity of the theoretical optimum, and that the curves themselves are merely approximations.

A very clear pattern, however, is that the normal utilities costs lead to low approach temperatures, low steam and water usage, and high ejector prices. For this example, steam investment equivalents of more than \$100 per pph result in approach temperatures as low as 3°F. Even \$1000 per pph may be exceeded in situations where energy costs are high, project life is long, tax investment incentives are present, and marginal interest rates are low (capital is available).

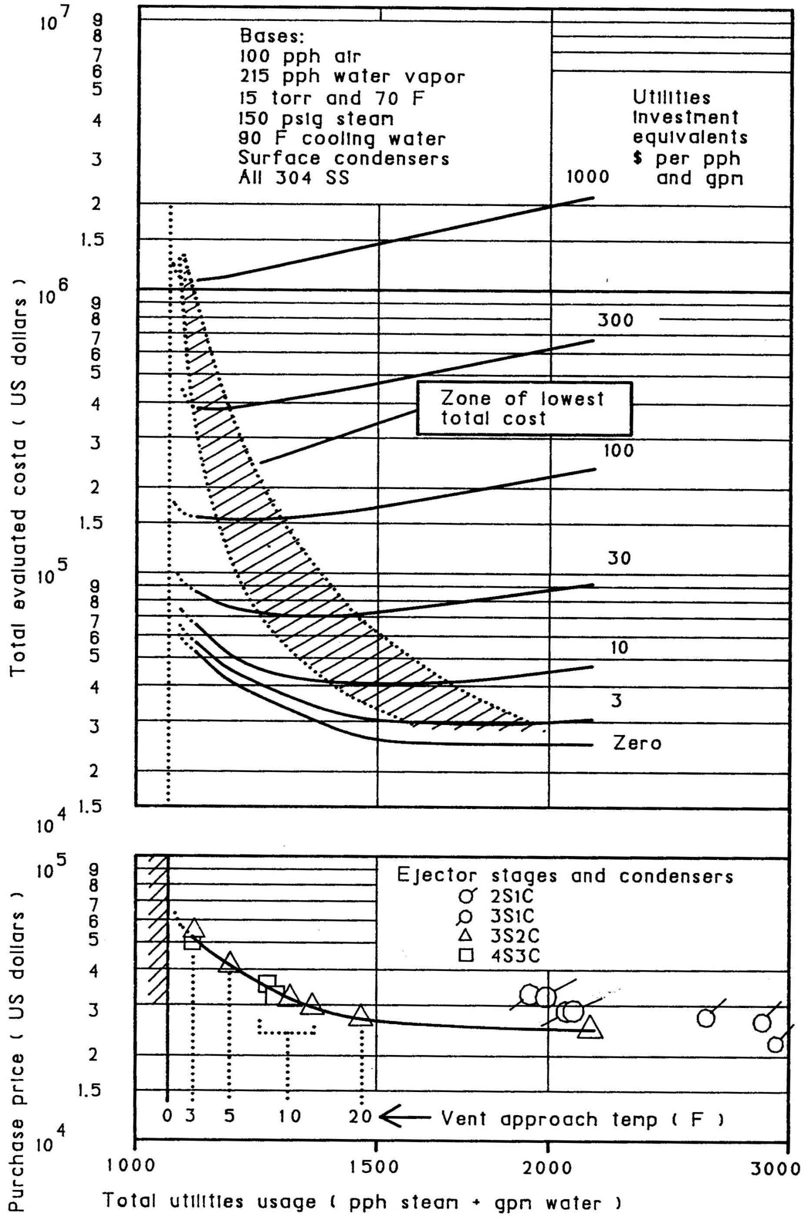


Figure G.1 Comparison of ejector "bids" on a total cost basis.

Are the differences in utilities usage real? Where is a practical limit? How does this example compare to what you will see? These questions will be answered in succession.

With computer programs to .prepare their bids, the manufacturers can perform detailed designs for their bids, down to the drill bit and reamer sizes for each steam nozzle. They can go to their shop test floor with these sizes. Differences in basic steam rates for ejector stages should differ by no more than 5 percent among the major manufacturers, who have years of test data and good test facilities. They can design closely and make the design work on the test floor. Occasionally, a larger difference may appear if one manufacturer has a high-efficiency design that is well matched to an application. This is unlikely to occur in the rough vacuum range with low-steam-usage designs, however, because the low-compression-ratio stages typically used are an area which has been explored intensively by many manufacturers for years.

Differences in condenser areas and costs will reflect differences in the manufacturers' condenser design experience and practices, resulting in a much larger spread than for the steam rates.

With this preparation, consider cases 11 and 13 in Table G.1. Case 11 has a price of \$30,000, requires 1332 total utilities, and has a first intercondenser area of 203 ft^2. Case 13 costs \$42,000, requires 1144 total utilities, and has a 616-ft^2 condenser. The utilities difference is 16 percent of the lower utilities usage, a realistic difference. The \$12,000 price difference divided by the 188 pph utilities usage difference yields \$64 as the breakeven investment equivalent cost. If you value your utilities higher than this, the higher-priced ejector is the place to invest your money. If you draw a \$64 investment equivalent curve on Fig. G.l, you will see that these two cases have equal total evaluated costs.

The differences are real, especially if they represent alternative bids from one manufacturer. An approach of 5°F is close, but not uncommon. If your utilities investment equivalents are well above the \$64 breakeven value, your choice of the higher-priced option represents a win-win move. Your total cost of running the ejector for the life of the project will be reduced, and the manufacturer will make more money by selling a higher-priced unit. Because most ejectors are operated with steam pressure about 10 percent higher than design, your savings will be increased proportionally.

What about an even lower steam user-case 14, for example? It has a price of \$54,000, utilities of 1109, a 3°F approach, and a first condenser of 1037 ft^2. You reduce your steam usage by 35 pph for a price increase of \$14,000, a breakeven value of \$400 investment equivalent per pph. The utilities difference of 3 percent is less than the difference between manufacturers and may represent a different amount of conservatism. You can not go wrong either way, provided the other aspects of each design check out.

The four-stage design with three intercondensers was economically comparable to the three-stage unit, and might look a little better upon closer inspection. If the cooling-water temperature were lower, it might look attractive. The increased installation and maintenance costs, however, plus the reduction in reliability associated with the extra stage and condenser, tend to offset any operating cost reduction.

Practical restraints on optimization

This example was specified large enough to be free of size effects. If the air and water vapor loads were 20 percent as large, 20 pph air and 43 pph water vapor, then size effects would be significant. Some of the stages would be small enough that reduced efficiency would require size correction factors. Minimum nozzle guidelines might also raise the steam consumption. The condensers would be enough smaller that close approach temperatures might be impractical. The lowerpriced equipment would generate less enthusiasm in manufacturers for innovating and optimizing, and you also would be reluctant to invest more time in the design and procurement beyond that needed to ensure high reliability.

Appendix H

Forms for Ejector Calculations, Tests, and Inspections

1. Vapor-liquid equilibrium calculations, ideal solution
2. Vapor-liquid equilibrium calculations, immiscible liquid
3. Vapor-pressure paper
4. Ejector test report-field test form
5. Stage dimension data sheet, field measurements

```
VAPOR-LIQUID EQUILIBRIUM CALCULATIONS

BASES:  Ideal solution (Raoult's law), ideal gas (Dalton's law)
                       o                          | For non-ideal solutions,
        ( p  = x  P   ,    P = p  + p  + ... )    | add activity coeff.
           i    i  i          1    2              | Refer to Chapter 5 for
                                                  | suggestions and leads
Total pressure, P  ______ Torr                    | to more information.

Temperature       ________  F / C
Steam,          TERs  _______       input(L/V)/
  Other gases,  TERg  _______                /
                                            V
                              o    K =   ||     |     |     |     |     |
           w                  P     o    ||------------------------------ Vapor
Component  PPH    MW   n    Torr   P /P  ||* v1 | v2  | v3  | v4  | v5  | PPH
==============================================================================
Water     |     | 18|    |    |        ||     |     |     |     |     |Ws
  Vapor
--------- Other Gases and Vapors---------------------------------------------
Air + NC  |     |   |    | Inf|  Inf   ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
          |     |   |    |    |        ||     |     |     |     |     |
------------------------------------------------------------------------------
Totals W ______  N ______             V _____ _____ _____ _____ _____Wg_____

                                N - V = L _____ _____ _____ _____ _____Vg_____

                            output (L/V)  _____ _____ _____ _____ _____MWg____

                                                                 MWERg____
------------------------------------------------------------------------------
                                             n
* calculation of vapor moles:   v = -----------
                                        (L/V)
                                        ----- + 1
                                          K
------------------------------------------------------------------------------
Dry Air Equivalent of vapor load

 Water vapor:          _________PPH
                   __________________________  =  ________  PPH DAE

                   0.8 MWERs (__________TERs)

 Other gases:          _________PPH
                   __________________________  =  ________  PPH DAE

                 ______MWERg (_________ TERg)

                                           Total  ________  PPH DAE
```

Figure H.1 Worksheet calculation of vapor-liquid equilibrium for ideal solutions.

```
VAPOR-LIQUID EQUILIBRIUM CALCULATIONS FOR IMMISCIBLES + WATER

BASES:  Ideal solution (Raoult's law), ideal gas (Dalton's law)
                    o                             | For non-ideal solutions,
        ( p  = x  P    ,   P = p  + p  + ... )    | add activity coeff.
            i    i  i          1    2             | Refer to Chapter 5 for
Immiscibles                                       | suggestions and leads
Partial pressure, P ______ Torr <----------      | to more information.
                   im                        \
Temperature  ________ F / C      Immiscibles  \_|  P  = P     - P
  Steam,        TERs  _______    input(L/V)/    |   im   total   water vapor
  Other gases, TERg  _______              /
                                         V
                        o   K =  ||     |     |     |     |     |
           w            P    o   ||------------------------------ Vapor
Component  PPH   MW  n   Torr  P /P ||* v1  | v2  | v3  | v4  | v5  | PPH
===========================================================================


--------- Other Gases and Vapors-------------------------------------------
Air + NC |     |   |    | Inf|  Inf ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
         |     |   |    |    |      ||     |     |     |     |     |
---------------------------------------------------------------------------
Totals W ______   N ______          V _____ _____ _____ _____ _____Wg_____

                             N - V = L _____ _____ _____ _____ _____Vg_____

                        output (L/V)  _____ _____ _____ _____ _____MWg____

                                                               MWERg____
---------------------------------------------------------------------------
                           n          ||  For Immiscible liquids, base
* calculation of    v = ----------    ||  water vapor flow on air.flow.
  vapor moles:           (L/V)        ||
                         ----- + 1    ||  See Chapter 5, page 118 for
                           K          ||  an example.
---------------------------------------------------------------------------
Dry Air Equivalent of vapor load

 Water vapor:            _________PPH
                     __________________________  =  ________  PPH DAE

                     0.8 MWERs ( _________TERs)

 Other gases:            _________PPH
                     __________________________  =  ________  PPH DAE

                   ______MWERg ( _________ TERg)

                                              Total ________  PPH DAE
```

Figure H.2 Worksheet calculation of vapor-liquid equilibrium for ideal immiscible mixtures.

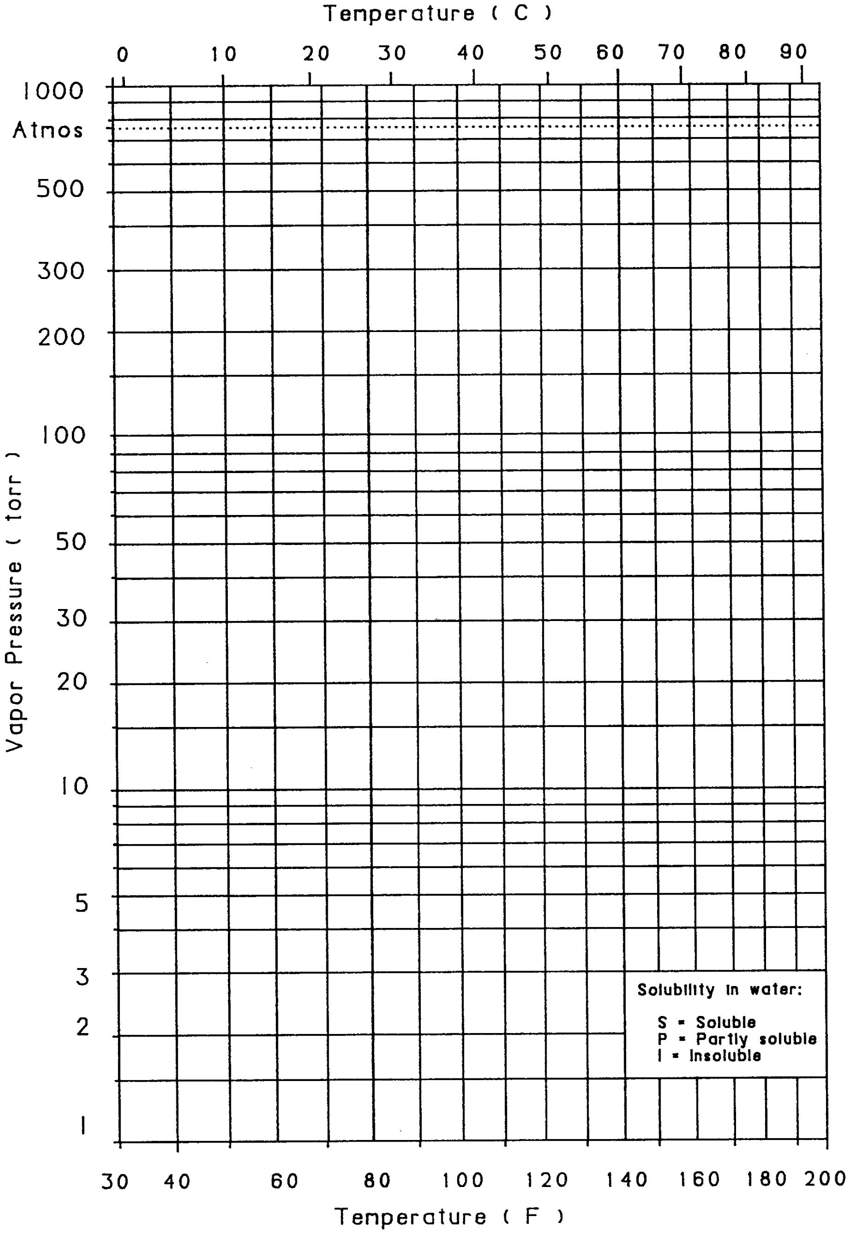


Figure H.3 Vapor-pressure graph paper.

Ejector Test Report – Field Test Form

CUSTOMER ________________ P.O. NUMBER ________________ MANUFACTURER ________________

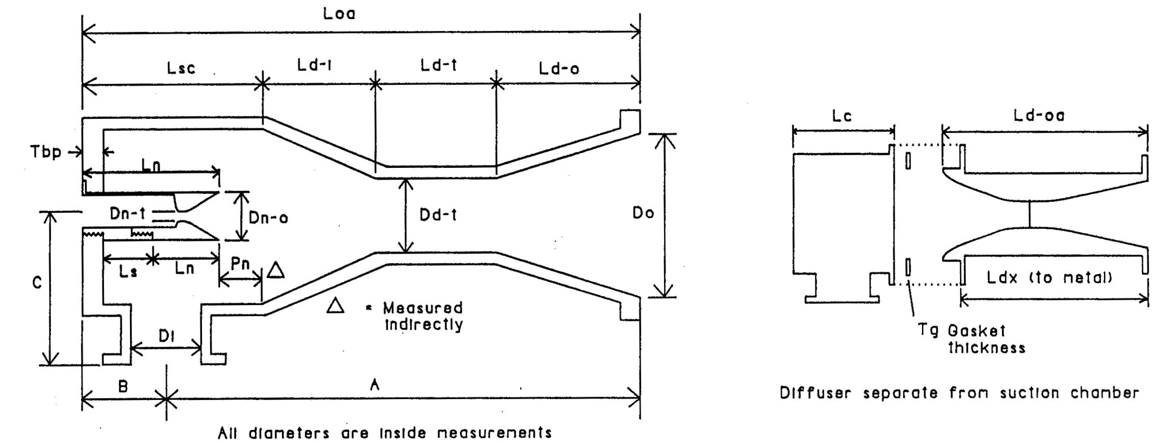
EQUIPMENT DESCRIPTION ________________ SERIAL NUMBER ________________ TEST NUMBER: ________________

LOAD		UTILITIES		AMBIENT DATA	
70°F Air Equivalent ________	#/Hr	Steam Pressure ________	PSIG	Barometric Pressure ________	In. HgA
70°F Water Vapor Equiv. ________	#/Hr	Steam Temp. ________	°F	Ambient Temp. ________	°F.
Design Suction Temperature ________	°F.	Water Temp. ________	°F		
Design Suction Pressure ________	mm or Inches HgA				

Air Load		Steam Load					Ejector Suction Pressure mm or Inches HgA					Ejector Discharge Pressure mm or Inches HgA					Water Flow GPM		Temperatures Out of Intercondensers – °F			
																			Water		Vent	
Orifice Dia.	#/Hr	Orifice Dia.	PSIG Upstream	#/Hr	°F Downstream	Corr Load	1st	2nd	3rd	4th	5th	1st	2nd	3rd	4th	5th	1st I.C.	2nd I.C.	1st	2nd	1st	2nd

CERTIFIED BY: Signature ________________ Title ________________ Date ________________

Figure H.4 Ejector test report -- field test form. *(Courtesy Heat Exchange Institute)*



Dimensions in ☐ inches ☐ mm

	Nozzle						Overall						Diffusers: Integral					Diffusers: Separate			
Stage No.	Tbp	Dn-t	Dn-o	Ln	Ls	Pn	A	B	C	Loa	Di	Do	Lsc	Ld-i	Ld-t	Ld-o	Dt	Lc	Tg	Ldx	Ld-oa

Item: Data recorded by: Date:

Description:

Figure H.5 Stage dimensions data sheet, field measurements.

Appendix

I

Instructions for Preparing Ejector Specifications

INSTRUCTIONS FOR COMPLETING STEAM JET EJECTOR BUYER'S SPECIFICATIONS

(Form RBP-JS-1) See Chap. 10 of *"Steam Jet Ejectors for the Process Industries"*,
either: first edition, McGraw-Hill, New York, 1994
or, second edition, Robert Power, Charleston, WV, 2004

Use the Comments area for changes and additions which do not fit in the upper portions of the page.

HEADER - Use one sheet for each case if an ejector is to be designed for operation at different conditions, or cases.

SYSTEM CONFIGURATION

TYPE: Select ejector, or hybrid if a combination of ejector and liquid-ring pump is desirable, or select both if a liquid-ring pump is acceptable.

NUMBER OF ELEMENTS: Usually one. In extremely critical services, use a second element as a built-in spare, or consider a complete spare ejector. If load is known to vary predictably, steam may be saved during low-load operation.

NUMBER OF STAGES: Usually determined by the seller, unless you have a preference and know the consequences of your choice.

PRECONDENSERS: Not wanted, unless you prefer the seller to supply an item that is usually included in the plant process design.

INTERCONDENSER AND AFTERCONDENSER TYPES: Usually surface, to minimize waste-water treatment costs and product loss. Intercondensers reduce steam consumption. If you are tempted to omit intercondensers to reduce the price of an ejector which you expect to be operated infrequently, please consider that the conservative strategy is to use intercondensers. The very worst outcome is to buy a noncondensing ejector which ends up being operated continuously. An aftercondenser acts as a silencer and reduces product loss, air pollution, and the nuisance of a steam plume. If the last stage is to be discharged to air, specify "Silencer" here or in the Comments area.

CONDENSER DRAINS: Barometric leg, unless you have a low-level installation and require special treatment. Describe a low-level application in the Comments area.

SURFACE CONDENSERS: Let seller choose horizontal vs vertical unless you have a preference. Select straight vs U-tube, and fixed tube sheet vs removable bundle. Give minimum acceptable tube O.D. and tube wall gage, and the maximum acceptable tube length. The maximum acceptable length affects the layout, and consideration of the estimated size of the first stage is useful in specifying this. If you prefer construction standards other than TEMA "B", change it. Condense vapors on the shell side, with cooling water in the tubes for higher water velocities and lower water side fouling. Most condensers are mounted horizontally. If the process side is expected to foul, condense the vapors in the tubes and consider a removable tube bundle with an outside packed head. Consult your heat exchanger specialists or plant maintenance people to determine removable bundle requirements. If water is on the shell side of a fixed tube sheet condenser, specify "no tubes in the baffle window".

Figure I.1

FOULING FACTOR (Surface condensers only): Use 0.002 as a moderate fouling factor. If heavy fouling is expected, consult your ejector specialist, or your heat exchanger specialist, or your ejector manufacturer, or all three.

MOTORS (only for hybrid systems): Similar to motors for centrifugal pumps.

GENERAL SPECIFICATIONS: List any general specifications which will accompany your inquiry specifications, to guide the seller in preparing the bid and designing the system.

OPERATING CONDITIONS

CASE: 100 percent if there is only one case. For multiple cases, complete this section of the form for each case. For multiple cases, the percent of operating time represented by each case is used by the seller in combination with the utilities cost data to arrive at the most economical design which handles all cases.

SUCTION PRESSURE AND TEMPERATURE: The design value of the absolute pressure in Torr at the ejector suction is equal to the pressure in the process system less the pressure drop in the equipment and piping to the ejector. Size vapor lines economically, a 10 percent drop in design pressure results in a 5 percent increase in operating cost. Let the design load contain the safety factor, not the pressure. Specify the design suction temperature. The normal temperature range has only a secondary effect on ejector design.

DISCHARGE PRESSURE AND TEMPERATURE: The design discharge pressure is extremely important to reliability. Specify the highest pressure against which the ejector must discharge, at blind (no-load) and at the design load, WHEN NEW AND CLEAN! The default value of 812 Torr (1 psig) includes an allowance for short vent piping or 6-inches submergence in a hotwell, plus a small allowance for wear and fouling. If in doubt, add a little more so that your ejector starts with a 1.0 psi allowance for wear and fouling. The design discharge temperature has no effect on vacuum performance, but does determine how much water vapor and condensable vapor will accompany the noncondensable gases to the atmospheric vent.

MOTIVE FLUID: The default is steam. If another gas is to be used, note it prominantly here and in the Comments. Discuss it with one of your ejector suppliers. Specify 5 percent below the lowest pressure at which the ejector is expected to operate, and the highest pressure for mechanical strength. For temperature, give "D&S" if there is less than 30 ^{O}F superheat, or give the design temperature in F (not degrees superheat) if it is superheated more than 30 ^{O}F. The default minimum acceptable nozzle throat is 0.0938 (3/32 inch). Some people specify 0.125 (1/8 inch) for the Z-stage. Larger sizes result in more reliability, by using more steam.

CONDENSING WATER: Specify the highest inlet temperature at which the ejector must operate, and the highest outlet temperature, above which you would get excessive fouling or corrosion. Give the desired range of shell-side and tube-side velocities, based on your experience with silting, fouling, erosion, and vibration.

Figure I.2

LIQUID RING PUMP SEALANT: Identify completely if other than water, and give the supply maximum temperature and minimum pressure.

SITE BAROMETRIC PRESSURE: Specify the average pressure at your site. This is used as a reference for gage pressure, and serves as a check on the adequacy of the discharge pressure.

STABLE RANGE: The default is 0 to 100 percent of design load. See the book if you consider changing it for control or economy reasons.

CLOSED-SYSTEM DESIGN PRESSURE: Usually not applicable (NA). If the ejector system is to be subjected to a pressure higher than the shop hydrostatic test pressure of 15 psig, then indicate the pressure here and describe the circumstances in the Comments area.

ALLOWABLE NOISE LEVEL: Default is 90 db. This is a reminder to the seller that a silencer is required if the last stage discharges directly to the air. It is very uncomfortable working near an unsilenced ejector discharge.

HOURS OPERATION THIS CASE: Give hours per day and per year. This further describes the multicase operation, as a guide for the seller and the person evaluating bids.

INITIAL EVACUTION REQUIREMENT: Usually "No" when a conservative estimate (Chap. 4) indicates that the evacuation time of a normal ejector is acceptable. Otherwise, "Yes", and give the requested data. The assumed air leakage may be given as "zero", or a nominal value such as 20 percent of the design air load. The normal interpretation of this requirement is that the hogging function is integral to the ejector design, usually a larger Z-stage, or Y- and Z-stages.

If the evacuation time requirements increase the ejector steam usage significantly, consider a twin-element last stage, or a separate single stage or multistage noncondensing hogging ejector. A separate hogging ejector may be on a hogging/spare header connected in parallel with this ejector and other ejectors, and providing both hogging and emergency spare service to several ejector systems.

MATERIALS

EJECTORS AND CONTACT CONDENSERS: Manufacturer's standard for all except nozzles is cast iron and steel. Conditions inside the ejectors include process gases and vapors, water and water vapor, high velocities, and temperatures ranging from cool to hot. A stainless steel diffuser for a small last stage is money invested well. If you want it, specify it here or under Comments.

SURFACE CONDENSERS: Materials should be similar to those in the process system, resistant to process-side and water-side corrosion and fouling. The process-side temperatures will be in the range of 100 °F to over 200 °F, with dewpoints below 200°F. Corrosion-resistant shell-side hardware is to preserve the shellside geometry, if you lose it, you lose the bundle.

Figure I.3

LIQUID RING VACUUM PUMP (hybrid systems only): Similar to materials for centrifugal pumps.

INVESTMENT EQUIVALENTS

Give the value for each of the utilities used by the system, including refrigeration costs if you have a refrigerated precondenser or aftercondenser vent. Specifying "Low Price" may lead to outrageous operating costs. See Chapter 10 and Appendix G for the importance of this data. If you are uncertain, estimate a little high. That is actually the conservative approach economically. Do not spend too little!

FLUIDS HANDLED

DESIGN FLOWS: Specify the maximum flows which the ejector must handle at the design suction pressure. Allow a "reasonable" amount for air leaks, plus dissolved gases, reaction products, instrument blowback, and saturated load of condensable vapors at the condenser vent temperature and pressure, if the ejector serves a condenser vent. A mixture of noncondensable gases may be identified as a mixture, together with the total mass flow and mixture molecular weight.

CONDENSABLE VAPORS: The manufacturer uses this information to design the stages and condensers. Give the vapor pressure at the two temperatures if the vapor pressure curve is nearly a straight line on vapor-pressure paper (Cox chart), otherwise, submit vapor pressure vs temperature curves on the Cox chart from Appendix H. If the condensable forms a non-ideal solution with water, use vapor-pressure paper to prepare plots of partial pressures of condensable and water versus temperature for lines of constant solution concentration. Reference the attachments here or in Comments.

If more than one condensable is present in significant quantities in a nonideal solution, have a knowledgeable process engineer discuss the specification with an ejector manufacturer to decide how to describe the behavior. If condensable liquids are only partly soluble (mutually) in water, treat them as immiscible (0 percent solubility) and supply the liquid phase physical properties.

COMMENTS

In addition to items previously mentioned, use this space to specify items not covered in the previous specifications or the General Specifications. Refer to special attachments and departures from the General Specifications. Examples of special topics include installations where the ejector first stage suction connection is integral with the head of a vessel, where one or more stages have an in-line suction, space available for a package unit, provision of a "Flute" or other special orifices for field testing, and a list of materials which must be excluded from the system.

Figure I.4

Answer to problem number 9

Estimate steam and cooling water requirements of a 2- stage ejector with a load of 100 pph air and 300 pph water vapor at 100 °F and 60 torr. Steam pressure is 150 psig and water design temperature is 90 °F. Use curves and methods on page 95.

Air (noncondensable) is 25 percent of the load, and we find Ra = 2.0 where the 25 percent curve crosses 60 torr.

Next, we must apply several multipliers to match our system with the system which the curves represent.

steam pressure,	**M--P_m =0.92**
water temperature,	**M—cwt=1.02**
contact condensers,	**M---ctcd= 0.9**
stable to shutoff,	**M---stab-0=1.05**
discharge 813 torr,	**M----disch813=1.03**

adjusted steam flow = 2.0(400)0.92(1.02)0.9(1.05)1.03
= 739 pph

Which is acceptably close to the previous analysis for rough sizing purposes.

Problem number 10

This is a guess-by-golly session to conclude this Pop Quiz series.

Work problem 8 again, but this time using the curves on page 93.

Answers on page number 454

Appendix J

Test Kit Contents List

These items are useful to carry in an ejector field test kit. Adapt the list to your needs and standard fitting sizes.

Quantity	*Item*
6	Adapter, ½ in NPT to ¼ in OD polyethylene tubing
3	Bushing, ½ in NPT to ¼ in NPT
3	Plug, ¼ in NPT
3	Plug, ½ in NPT
3	Globe or needle valve, ¼ in NPT male INLET to ¼ in OD polyethylene tubing
1	Roll Teflon tape, for making screwed connections
2	Polyethylene tubing, with spare fittings, 6 ft, ¼ in OD
1	Slip blank, sheet metal, for 1-in to 2-in flanges
1	Slip blank, sheet metal, for 2.5 in to 4-in flanges
1	Adjustable end wrench, 10 in
1	Slip-joint pliers, 10 in
1	Multiple-orifice device with 1-in NPT male connection, plus additional large orifices and fittings as needed for large air-load tests, or larger multiple-orifice device and adapters.
1	Roll masking tape, to cover inactive small orifices
1	Reducing adapter fitting: 1-in female NPT to ½ in male NPT
1	Thin polyethylene garbage bag, or tube 6 ft long by 6 in diameter knotted one end: as air meter for ejector vents
1	Stop watch or watch indicating seconds, for air meter or pressure rise rate tests.
1	Pair gloves, for handling hot metal.

Test kit contents, continued

Quantity	*Item*
1	Wire, light coat hanger or thinner, for probing fouled nozzles
1	Flashlight, halogen, small, for inspecting steam nozzles and reading nameplates.
1	Set of fresh batteries for flashlight.
1	Millimeter scale, metal, 150 mm long
1	Clipboard with weather cover
-	Performance curve paper, log-log, for plotting multistage jet tests, Fig. 10.5. Rectangular grid is OK for single-stage tests.
-	Performance test data forms, Fig. H.4
-	Stage dimensional data forms, Fig. H.5
-	Maintenance inspection data forms or checklists, if available
-	Notebook to log procedures and key findings
1-	Technical reference with testing guidelines and vacuum engineering data: HEI standards or this book.

Pressure measurement equipment:

1 Electronic pressure measurement device: capacitance diaphragm or piezoelectric.

Or, sealed-leg mercury manometer, 150 mm, preferably protected with good liquid trap.

Or, atmospheric reference mercury manometer, 30+ in

Appendix K

Ejector Manufacturers and Suppliers of Referenced Hardware and Information

The following lists identify several of the organizations who manufacture ejectors or ejector-related hardware, or supply referenced standards information.

Ejectors

Croll-Reynolds Co., Inc., P.O. Box 668, 751 Central Avenue, Westfield, NJ 07091-0668, (908) 232-4200. Fax (908) 232-2146. Web: croll.com

Derbyshire Machine & Tool Co, 5100 Belfield Avenue, Philadelphia, PA 19144-1788, (215) 844-3200, (800) 220-3717. Fax (215) 849-8680. Web: derbyshiremachine.com

.Fox Valve Development Corp., Hamilton Business Park, Unit 6A, Dover, NJ 07801, (973) 328-1011. Fax (973) 328-3651
. Web: foxvalve.com

Graham Manufacturing Co., Inc., 20 Florence Avenue, Batavia, NY 14021-0719, (585) 343-2216. Fax (585) 343-1097. Web: graham-mfg.com

Hick Hargreaves & Co. Ltd., Bolton, Lancashire, BL5 3SL, England, Tel: (0204) 23373. Web: hickhargreaves.co.uk or, bocedwards.com

Hibon International S.A., 38 Boulevard de Reims, F-59058 Roubaix CEDEX 1, France. +33 3 20 45 3940, Fax: +33 3 20 45 3937 Web: hibon.com

The Jet-Vac Co., a Division of Artisan Industries, 73 Pond Street, Waltham, MA 02451-4594, (781) 893-6800. Fax: (781)647-0143 Web: artisanind.com

Koerting Hannover, Badensledter St. 56, D-W-3000 Hannover, Germany. +49(0) 511/2129-0 Fax: +49(0) 511/2129-223

Mazzai Injector Corp., 500 Rooster Dr., Bakersfield, CA 93307-9555 (661) 363-6500 Fax: (661) 363-7500 Web: mazzei.net

Nash-Kinema, Inc., 700 Glassport-Elizabeth Road, P.O. Box 176, Elizabeth, PA 1537-1864, (412) 384-3610.

NItech Inc., 64 Horse Hill Road, Cedar Knolls, NJ 07927, (973) 5381940. Fax: (973) 538-3511 Web: nitech-vac.com

Penberthy Division, Tyco Valve & Controls, P.O. Box 112, 320 Locust St., Prophetstown, IL 61277, (815) 537-2311. Fax: (815) 537-5764 Web: penberthy-online.com

Schutte & Koerting Division, Ketema, 2233 State Road, Bensalem, PA 19020. (215) 639-0900. Fax: (215) 639-1597 Web: s-k.com

Unique Systems, Inc., 1 Saddle Road, Cedar Knolls, NJ 07927 (973) 455-0440. Fax: (973) 455-7214 Web: uniquesystems.com

GEA Wiegand GmbH, Einsteinstrasse 9-15, Postfach D-7505 Ettlingen, Germany. Tel: (+49) 7243-7050 Fax: (+49) 7243-7053 Web: gea-wiegand.com

Ejector standards

Heat Exchange Institute, 1300 Summer Ave, Cleveland, OH 44115-2851 (216) 241-7333 Fax: (216) 241-0105 Web: heatexchange.org

Multiple-orifice air test device

Croll-Reynolds (see above)

Unique Systems, Inc. (see above)

Capacitance diaphragm vacuum pressure measurement

MKS Instruments, Inc., 6 Shattuck Road, Andover, MA 01810, (800) 227-8766 or (978) 975-2350.

The VirTis Co., Inc., 815 State Route 208, Gardiner, NY 12525 (845) 255-5000.

Oil-sealed near-zero mechanical vacuum pumps, for rough vacuum gauge checking and calibration

Edwards High Vacuum International, One Edwards Park, 301 Ballardvale Street, Wilmington, MA 01887-0868, (800) 848-9800 (978) 658-5410. F ax: (978) 658-7969 Web: bocedwards.com

Kinney Vacuum Div., Tuthilll Vacuum Systems, 495 Turnpike St., Canton, MA 02021, (781) 828-9500. Fax: (781) 828-5612 Web: kinney@tuthill.com

Leybold Vacuum Products, Inc., 5700 Mellon Road, Export, PA 15632-8990, (724) 327-5700 Fax: (724) 733-1217 Web: leyboldvacuum.com.

Stokes Vacuum, Inc., 5500 Tabor Road, Philadelphia, PA 19120-2124 (215) 831-5400. Fax: (215) 831-5420 Web: stokesvacuum.com

Vacuum supplies, including gauges and high-vacuum systems

CVC Products, Inc., 525 Lee Road, Box 1886, Rochester, NY 14603, (800) 448-5900 or (716) 458-2550. Fax: (716) 458-0424

Leybold Inficon, Inc., Two Technology Drive, East Syracuse, NY 13057, (315) 434-1126 Fax: (315) 437-3803.

Stokes Vacuum, Inc., (see above)

The VirTis Co., (see above)

Wallace & Tiernan, Inc., 25 Main Street, Belleville, NJ 07109-3057, (973) 759-8000. Fax: (973) 751-6589

POP QUIZ

Table of Contents

Appendix L

Failure Modes and Symptoms

Vacuum failure occurs when the vacuum producing system can no longer maintain the desired pressure in the process system under the load conditions existing. The problem may originate in the process system, or the vacuum producing system, or both. The troubleshooting task is to identify the problem sources so they can be corrected and the process system can be returned to operation.

This portion of the Appendix is a description of several sources of vacuum system failure, showing how the failures propagate from the origin and cause the system pressure to rise. The intent here is to familiarize a reader with the failure patterns, so that he or she will be able to match up the failure patterns with the failure symptoms in a troubleshooting situation.

High discharge pressure: Exceeds last-stage MDP at current steam and load conditions. The result is a rise in suction pressure at the last stage. This creates the same problem for the Y stage, and all stages fail in succession, like dominoes. This failure sequence is so common that it will not be described again, other than identifying the rise in suction pressure at the offending stage. Note that at small loads some Z stages are slightly unstable because their MDP curve falls below the discharge pressure. There is so much "lap" at small loads that a modest rise in suction pressure usually does not affect the Y stage. The instability may, however, thwart testing efforts by filling the manometer with water.

Low steam pressure reduces the MDP curve for each affected stage. There is not much effect on load near design point, and it actually improves part-load vacuum. In the Z stage, it results in the MDP becoming less than the discharge pressure, etc.

Debris in steam nozzle produces the same effect as low pressure. If it completely stops the flow, the stage will be cold.

High steam superheat which exceeds design superheat will reduce the steam mass flow, with the same effect as low pressure.

Wet steam will degrade performance of all stages, but is worst for small nozzles and for low-pressure stages. Stages with small nozzles will stop operating when slugs of water pass through the nozzle, causing a steam-pressure gauge at that stage to flutter. Low-pressure stages cannot develop the high expansion energy required and just will not work well. Cracking a steam blowdown or drain valve to the air will show an all-white plume for wet steam, instead of the short transparent zone for dry steam. A sturdy shielded thermometer in the discharge of a Z stage may show less than 240°F, indicating moisture.

Steam leak at nozzle adds vapor load to that stage and noncondensing stages up to the next condenser, where it drops out. Loaded stages have high suction pressure; stages after the next condenser have low pressure. A large steam leak may make the suction chamber of an X or lower-pressure stage warmer than normal.

Fouled diffuser will have degraded suction and MDP curves.

Worn diffuser, if it is worn uniformly, will reduce the MDP curve, possibly below the current discharge pressure, etc. If the stage is not broken, the larger throat may actually increase the suction curve capacity a little. Usually the wear degrades the suction and discharge curves, especially if the diffuser inlet cone and throat are grooved and pitted from wet steam or corrosion. Because the last stage is smallest, it is most affected by a given amount of wear.

Worn nozzle will increase the steam flow, often enough to compensate for the poor nozzle geometry and also conceal some diffuser degradation. Replacing the worn nozzle will reduce the steam flow to the design value, exposing the degraded diffuser.

Wrong nozzle position, caused by omitting a required spacer or gasket, or by using a spacer or gasket from another stage, will degrade the performance curves.

Swapped steam nozzles usually results in two bad stages. Most nozzles and diffusers are stamped with stage numbers or letters.

Large air leak from process is present in all stages and is especially evident in the last stage, raising the suction pressure above the design value, etc. Airflow can be measured with a flow meter or garbage bag at the aftercondenser vent if the drainleg is sealed in the hotwell, or at the seal pot vent if the discharge all goes to a sealpot. A vacuum drop (pressure rise) test will quantify the air leak when the system is down.

Air leak in ejector system will load all stages following the point of entry. This is detectable by isolating the ejector from the process, then measuring last-stage suction pressure, or vent flow. An air leak in a drainleg may cause that condenser to flood from air-lift effect.

Hot load mixture reduces ejector capacity.

Loop seal or trap open between intercondensers will permit air to recycle back to the low-pressure condenser, loading up the Y stage, but not the Z stage. From an aftercondenser, it will also load the Z stage.

Large condensable vapor load loads all stages up to the first condenser, then usually drops out. Immiscible low boiler vapors (toluene, for example) may condense in the second condenser, or may not condense until the aftercondenser, or may not condense at all.

Ice formation in stages operating below 5 torr produces the same effect as fouling, and is prevented by a steam-jacketed diffuser inlet or by superheating the steam. No steam in the steam jacket may permit ice. Frost outside the diffuser indicates ice inside.

Hot cooling water above the design water temperature will permit extra water vapor to the next stage, loading it up and reducing the total noncondensable load capacity. If that stage suction pressure exceeds the MDP of the preceding stage (the design interstage pressure), the preceding stage fails, etc.

Low water flow as a result of low water pressure, obstructions, or misguided attempts at economy will have the same effect as hot water.

Excessive water flow in contact condensers results in flooding, large pressure drop, condenser body cold all over, and next stage suction also cold.

Fouled condenser will result in a high pressure drop on process side, raising the discharge pressure on the preceding stage, etc. On the water side, it results in low. water flow. In contact condensers, this may break up water sheets or spray, permitting vapor overload to the next stage and detectable by hot spots on the side of the condenser where water flow is lacking. If the drain is obstructed, the condenser may flood. Liquid flowing into the next stage will cause it to run cool, and intermittent slugs will cause suction pressure fluctuations there.

Fouled process piping results in high pressure drop. Ejector suction pressure may be much lower than in the process. At discharge from the last stage, fouling will raise discharge pressure, etc. In drainlegs to hotwell, fouling will raise the level of condensate, and may flood condensers.

Mud in hotwell will raise the liquid level and flood the condenser.

Butterfly throttle control valve too close to first stage resulted in fluctuating disruption of motive steam flow and caused a loud "whistle" tone. Rotating the valve body 90° in its flange stopped the noise. Thank you, Vic Fondrk, for this anecdote, which permits me to end my book on an interesting note.

Index

Post Script

Here is where I explain, brag and apologize, encourage and warn, and suggest remedies for some shortcomings that I am aware of. As the next page explains, I have a health problem for which an improved medication and exercise regimen have given me a new lease on life. The Print On Demand (POD) technology and advancements in personal computers permitted me to reconstruct the book in digital electronic form in my computer system and put it on a CD-ROM. From that CD-ROM a commercial POD printer can produce small runs economically. Amazon.com provides a worldwide electronic bookshelf ideal for the small niche replacement market for this book.

To simplify the organization of this book, I made every page nominally **identical to a first edition page**. That makes it useful as a common reference, when discussing a technical detail by telephone, for example. This book must be classed as a new edition because I am now the publisher. I have labored long over some of the "busy" figures to make the tiny symbols more readable. The paper is thinner and the pages are sewn together, making the book easier to use and more durable. The list of manufacturers has updated names, addresses, telephones and faxes, and now---websites.

On page 276 I now emphasize the **limitations of the pressure-rise measurement** to determine the air-leakage rate, especially when the system contains volatile liquid and/or water vapor.

Fig 5.2 on page 135 gives the **tubeside heat transfer coefficient** for water, based on a 1930 paper (the year I was born!). Jack Burns recommends a Rabas-Cane 1970 published correlation, "An Update of Intube Forced Convection Heat Transfer Coefficient With Variable Physical Properties", Advances in Heat Transfer, Vol. 6,pp 503-564, 1970. Improved accuracy led to adoption in PTC-12.2 Condenser Code.

A major new factor in the ejector field is the computerization of the tedious calculations for predicting performance and evaluating quotations. I anticipated the trend in the first edition but have not advanced the subject in this book. An application that would yield a big reduction in process energy costs is a program that predicts the performance of gas jet gas ejectors (thermo-compressors) with accuracy appropriate for rough appraisals. I plan to search for **programs and algorithms useful to ejector users**, and make them available through my website as "freebies" or purchases when ethically appropriate.

Hardware **cost estimates** are still based on 1993 cost indices, so be sure to adjust each estimate using an appropriate cost index.

Robert B. Power grew up in small towns in North Dakota and Montana. He worked summers in his father's creamery processing plant in Plentywood, Montana. Receiving a scholarship to Carnegie Mellon University, he obtained a B.S. in Mechanical Engineering. He spent 33 years with Union Carbide Central Engineering in Charleston, WV. He earned his M.S. in Mechanical Engineering from the WVU Graduate Center in evening classes.

In 1960 he was assigned to some steam jet ejector problems. That led to overhauling Carbide ejector practices beginning with process design specifications, bid evaluations (including operating costs), and layout design. Field support followed for startup and trouble-shooting.

Other specialized experiences include

- Fluid machinery: process applications and mechanical design audits (critical speeds, bearings, deflections, stresses, gears)
  Finite Element Analysis (FEA) to find stresses, deformation, and frequency responses in machine components, process vessels, and civil engineering structures.
- Supervisory assignments: Heat Transfer and Fluid Dynamics, Industrial Engineering, and Machinery Technology
- Teaching FEA and ME review at Carbide and WVU Grad. Center
- Participation in plant reliability surveys
- Developing utility cost factors for major utilities at major plant locations, for routine use in design choices and optimization studies

He retired in 1986 as a part time consultant. Then some encouraging field experiences troubleshooting ejectors prompted him to write the ejector book he never found. The book was published in 1993, with a second printing in 1994 to correct a technical error in the illustrations.

In May 1994 he was diagnosed as having Parkinson's Disease, a non-contagious progressive movement disorder. It led him to retire from active consulting. In 1999 the book was declared 'out of print' and all copyrights reverted to him.

In December 2000 he began an improved medication and exercise regimen that greatly improved his physical and mental well-being. In January 2003 he challenged himself to celebrate his renewed well-being by self-publishing this book, doing all the scanning, proofreading, keyboarding, and cover design to create an electronic version on a CD-ROM in a PDF format suitable for a book printer.

His improved condition enables him to welcome discussions of ejector problems or anecdotes. Any serious consulting effort would not involve travel, and might be limited by his Retired P.E. status.

www.jetwords.com

Steam Jet Ejectors for the Process Industries, by Robert B. Power. Second edition. ISBN: 978-0-9755706-4-1. Second Printing, 2008.

This second "printing" of the book is published exclusively in electronic format on our website: www.jetwords.com. It is made available under a shareware-like arrangement:

- You are free to download and examine the electronic version of the book for an unlimited period of time.
- You may also make free copies of the electronic version to distribute to your friends and colleagues, as long as you include the entire unaltered work (all files in the **JetWords** folder, whether compressed or uncompressed).
- If you find this book useful, please pay the license fee to the author, using the guidelines on the book's website to determine the amount that is appropriate for your situation. This license fee also entitles you to print one copy of the entire book (at your expense) for your own use.
- If you wish to print one or more additional copies of all or portions of the book, an additional amount is due the author, as described in the fee schedule on the website.

See the website (www.jetwords.com) for

- Contact information
- The most current copy of the book
- Errata list for earlier editions and printings
- Pricing guidelines, including special pricing for academic use and for residents of developing countries
- Payment instructions

While you are at the website, check out the errata listings for all the printings of this book. Although I have reviewed carefully to correct errors in the previous printings, I have certainly introduced new errors during the massive task of reviving this book as a second edition and converting it to electronic format. I ask for your alertness and tolerance as you use this book, and I invite your participation in the process of identifying and correcting these pesky errors. I just can't seem to avoid making misteaks.