57.1.5 Gas Turbine Operation

Like other internal combustion engines, the gas turbine requires an outside source of starting power. This is provided by an electrical motor or diesel engine connected through a gear box to the shaft of the gas turbine (the high-pressure shaft in a multishaft configuration). Other devices can be used, including the generator of large electric utility gas turbines, by using a variable frequency power supply. Power is normally required to rotate the rotor past the gas turbine's ignition speed of 10–15% on to 40–80% of rated speed where the gas turbine is self-sustaining, meaning the turbine produces sufficient work to power the compressor and overcome bearing friction, drag, and so on. Below self-sustaining speed, the component efficiencies of the compressor and turbine are too low to reach or exceed this equilibrium.

When the operator initiates the starting sequence of a gas turbine, the control system acts by starting auxiliaries such as those that provide lubrication and the monitoring of sensors provided to ensure a successful start. The control system then calls for application of torque to the shaft by the starting means. In many industrial and utility applications, the rotor must be rotated for a period of time to purge the flow path of unburned fuel that may have collected there. This is a safety precaution. Thereafter, the light-off speed is achieved and ignition takes place and is confirmed by sensors. Ignition is provided by either a sparkplug type device or by an LP gas torch built into the combustor. Fuel flow is then increased to increase the rotor speed. In large gas turbines, a warmup period of one minute or so is required at approximately 20% speed. The starting means remains engaged, since the gas turbine has not reached its self-sustaining speed. This reduces the thermal gradients experienced by some of the turbine components and extends their low cycle fatigue life.

The fuel flow is again increased to bring the rotor to self-sustaining speed. For aircraft engines, this is approximately the idle speed. For power generation applications, the rotor continues to be accelerated to full speed. In the case of these alternator-driving gas turbines, this is set by the speed at which the alternator is synchronized with the power grid to which it is to be connected.

Aircraft engines' speed and thrust are interrelated. The fuel flow is increased and decreased to generate the required thrust. The rotor speed is principally a function of this fuel flow, but also depends on any variable compressor or exhaust nozzle geometry changes programmed into the control algorithms. Thrust is set by the pilot to match the current requirements of the aircraft, through takeoff, climb, cruise, maneuvering, landing, and braking.

At full speed, the power-generation gas turbine and its generator (alternator) must be synchronized with the power grid in both speed (frequency) and phase. This process is computer-controlled and involves making small changes in turbine speed until synchronization is achieved. At this point, the generator is connected with the power grid. The load of a power-generation gas turbine is set by a combination of generator (alternator) excitement and fuel flow. As the excitation is increased, the mechanical work absorbed by the generator increases. To maintain a constant speed (frequency), the fuel flow is increased to match that required by the generator. The operator normally sets the desired electrical output and the turbine's electronic control increases both excitation and fuel flow until the desired operating conditions are reached.

Normal shutdown of a power-generation gas turbine is initiated by the operator and begins with the reduction of load, reversing the loading process described immediately above. At a point near zero load, the breaker connecting the generator to the power grid is opened. Fuel flow is decreased and the turbine is allowed to decelerate to a point below 40% speed, whereupon the fuel is shut off and the rotor is allowed to stop. Large turbines' rotors should be turned periodically to prevent temporary bowing from uneven cool-down that will cause vibration on subsequent startups. Turning of the rotor for cool-down is accomplished by a ratcheting mechanism on smaller gas turbines, or by operation of a motor associated with shaft-driven accessories, or even the starting mechanism on others. Aircraft engine rotors do not tend to exhibit the bowing just described. Bowing is a phenomenon observed in massive rotors left stationary surrounded by cooling, still air that, due to free convection, is cooler at the 6:00 position than at the 12:00 position. The large rotor assumes a similar gradient and, because of proportional thermal expansion, assumes a bowed shape. Because of the massiveness of the rotor, this shape persists for several hours, and could remain present when the operator wishes to restart the turbine.

57.2 GAS TURBINE PERFORMANCE

57.2.1 Gas Turbine Configurations and Cycle Characteristics

There are several possible mechanical configurations for the basic simple cycle, or open cycle, gas turbine. There are also some important variants on the basic cycle: intercooled, regenerative, and reheat cycles.

The simplest configuration is shown in Fig. 57.15. Here the compressor and turbine rotors are connected directly to one another and to shafts by which turbine work in excess of that required to drive the compressor can be applied to other work-absorbing devices. Such devices are the propellers and gear boxes of turboprop engines, electrical generators, ships' propellers, pumps, gas compressors, vehicle gear boxes and driving wheels, and the like. A variation is shown in Fig. 57.16, where a jet

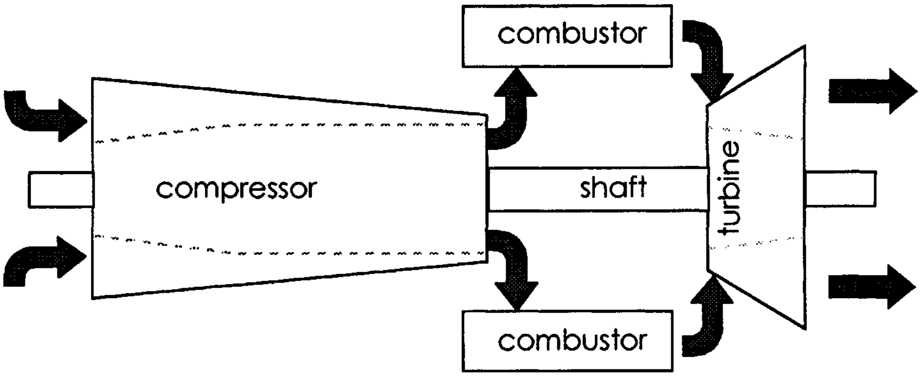

Fig. 57.15 Simple-cycle, single-shaft gas turbine schematic.

nozzle is added to generate thrust. Through aerodynamic design, the pressure drop between the turbine inlet and ambient air is divided so that part of the drop occurs across the turbine and the remainder across the jet nozzle. The pressure at the turbine exit is set so that there is only enough work extracted from the working fluid by the turbine to drive the compressor (and mechanical accessories). The remaining energy accelerates the exhaust flow through the nozzle to provide jet thrust.

The simplest of multishaft arrangements appears in Fig. 57.17. For decades, such arrangements have been used in heavy-duty turbines applied to various petrochemical and gas pipeline uses. Here, the turbine consists of a high-pressure and a low-pressure section. There is no mechanical connection between the rotors of the two turbines. The high-pressure (h.p.) turbine drives the compressor and the low-pressure (l.p.) turbine drives the load—usually a gas compressor for a process, gas well, or pipeline. Often, there is a variable nozzle between the two turbine rotors that can be used to vary the work split between the two turbines. This offers the user an advantage. When it is necessary to lower the load applied to the driven equipment—for example, when it is necessary to reduce the flow from a gas-pumping station—fuel flow would be reduced. With no variable geometry between the turbines, both would drop in speed until a new equilibrium between l.p. and h.p. speeds occurs. By changing the nozzle area between the rotors, the pressure drop split is changed and it is possible to keep the h.p. rotor at a high, constant speed and have all the speed drop occur in the l.p. rotor. By doing this, the compressor of the gas turbine continues to operate at or near its maximum efficiency, contributing to the overall efficiency of the gas turbine and providing high part-load efficiency. This two-shaft arrangement is one of those applied to aircraft engines in industrial applications. Here, the h.p. section is essentially identical to the aircraft turbojet engine or the core of a fan-jet engine. This h.p. section then becomes the *gas generator* and the free-turbine becomes what is referred to as the *power turbine.* The modern turbofan engine is somewhat similar in that a low-pressure turbine drives a fan that forces a concentric flow of air outboard of the gas generator aft, adding to the thrust provided by the engine. In the case of modern turbofans, the fan is upstream of the compressor and is driven by a concentric shaft inside the hollow shaft connecting the h.p. compressor and h.p. turbine.

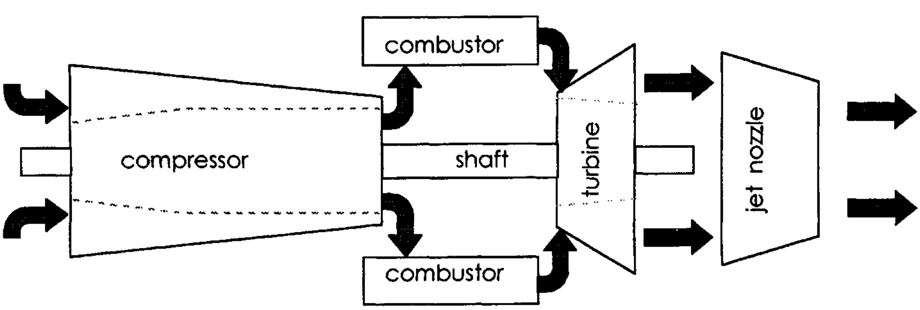

Fig. 57.16 Simple-cycle single-shaft, gas turbine with jet nozzle; simple turbojet engine schematic.

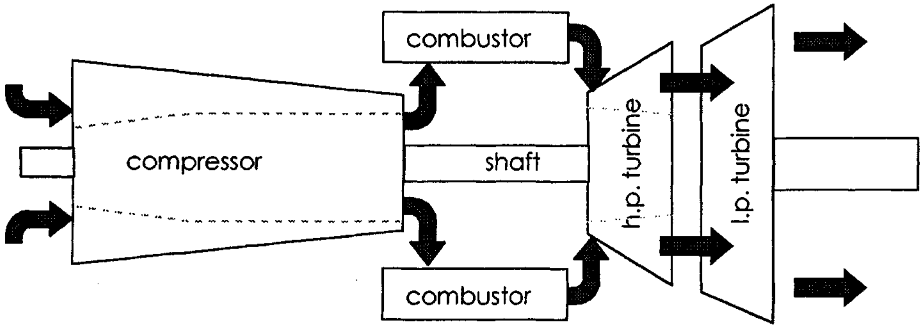

Fig. 57.17 Industrial two-shaft gas turbine schematic showing high-pressure gas generator rotor and separate free-turbine low-pressure rotor.

Figure 57.18 shows a multishaft arrangement common to today's high-pressure turbojet and turbofan engines. The h.p. compressor is connected to the h.p. turbine, and the l.p. compressor to the l.p. turbine, by concentric shafts. There is no mechanical connection between the two rotors (h.p. and l.p.) except via bearings and the associated supporting structure, and the shafts operate at speeds mechanically independent of one another. The need for this apparently complex structure arises from the aerodynamic design constraints encountered in very high-pressure-ratio compressors. By having the higher-pressure stages of a compressor rotating at a higher speed than the early stages, it is possible to avoid the low-annulus-height flow paths that contribute to poor compressor efficiency. The relationship between the speeds of the two shafts is determined by the aerodynamics of the turbines and compressors, the load on the loaded shaft and the fuel flow. The speed of the h.p. rotor is allowed to float, but is generally monitored. Fuel flow and adjustable compressor blade angles are used to control the l.p. rotor speed. Turbojet engines, and at least one industrial aero-derivative engine, have been configured just as shown in Fig. 57.18. Additional industrial aero-derivative engines have gas-generators configured as shown and have power turbines as shown in Fig. 57.17.

The next three configurations reflect deviations from the basic Brayton gas turbine cycle. To describe them, reference must be made back to the temperature–entropy diagram.

Intercooling is the cooling of the working fluid at one or more points during the compression process. Figure 57.19 shows a low-pressure compression, from points a to b. At point b, heat is removed at constant pressure. The result is moving to point c, where the remaining compression takes place (line c–d), after which heat is added by combustion (line d–e). Following combustion, expansion takes place (line e–f). Finally, the cycle is closed by discharge of air to the environment (line f–a), closing the cycle. Intercooling lowers the amount of work required for compression, because work is proportional to the sum of line a–b and line c–d, and this is less than that of line

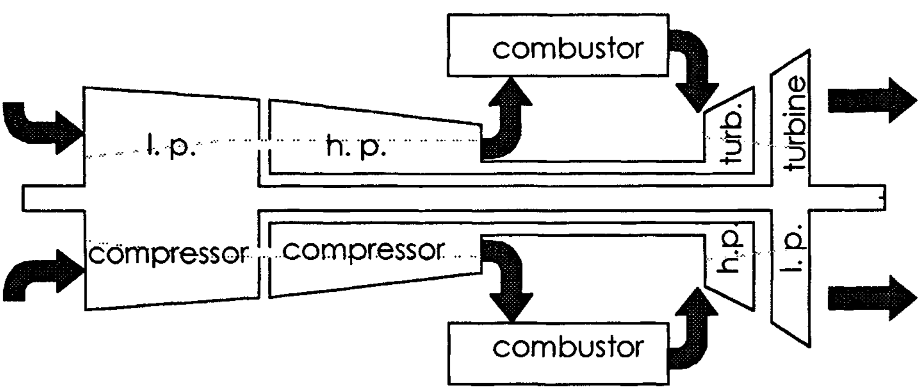

Fig. 57.18 Schematic of multishaft gas turbine arrangement typical of those used in modern high-pressure-ratio aircraft engines. Either a jet nozzle, for jet propulsion, or a free power turbine, for mechanical drive, can be added aft of the l.p. turbine.

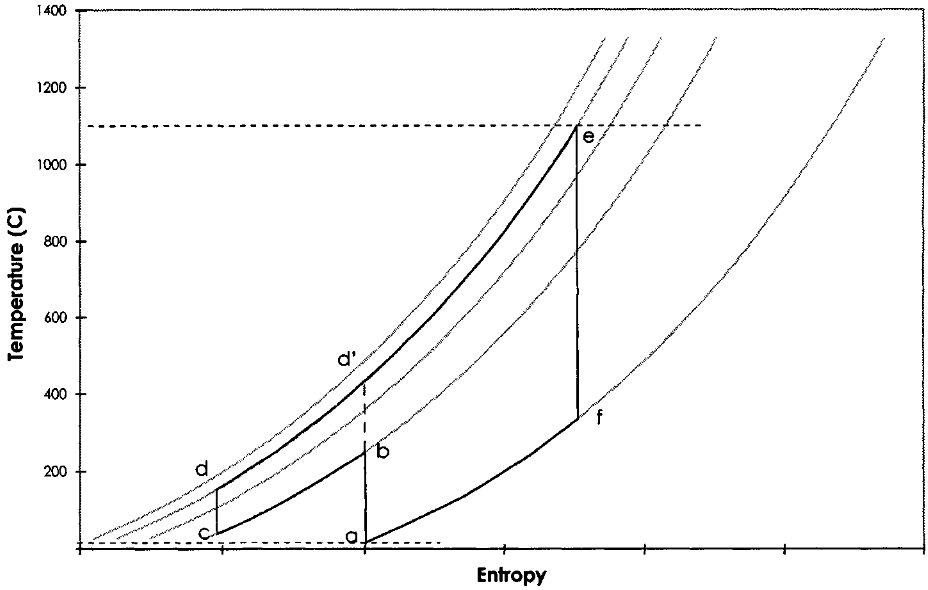

Fig. 57.19 Temperature–entropy diagram for intercooled gas turbine cycle. Firing temperature arbitrarily selected at 1100°C and pressure ratio at 24:1.

a–d', which would be the compression process without the intercooler. Lines of constant pressure are closer together at lower temperatures, due to the same phenomenon that explains higher turbine work than compressor work over the same pressure ratio. Although the compression process is more efficienct with intercooling, more fuel is required by this cycle. Note the line *d–e* as compared with the line *d'–e*. It is clear that the added vertical length of line *d–e* versus *d'–e* is greater than the reduced vertical distance achieved in the compression cycle. For this reason, when the heat in the partially compressed air is rejected, the efficiency of an intercooled cycle is lower than a simple cycle. Attempts to use the rejected, low-quality heat in a cost-effective manner are usually not successful.

The useful work, which is proportional to *e–f* less the sum of *a–b* and *c–d*, is greater than the useful work of the simple *a–d'–e–f–a* cycle. Hence for the same turbomachinery, more work is produced by the intercooled cycle—an increase in power density. This benefit is somewhat offset by the fact that relatively large heat-transfer devices are required to accomplish the intercooling. The intercoolers are roughly the size and volume of the turbomachinery and its accessories. Whether the intercooled cycle offers true economic advantage over simple-cycle applications depends on the details of the application, the design features of the equipment, and the existence of a use for the rejected heat.

An intercooled gas turbine is shown schematically in Fig. 57.20. A single-shaft arrangement is shown to demonstrate the principal, but a multishaft configuration could also be used. The compressor is divided at some point where air can be taken offboard, cooled, and brought back to the compressor for the remainder of the compression process. Combustion and turbine configurations are not affected.

The compressor-discharge temperature of the intercooled cycle (point *d*) is lower than that of the simple cycle (point *d'*). Often, cooling air, used to cool turbine and combustor components, is taken from, or from near, the compressor discharge. An advantage often cited for intercooled cycles is the lower volume of compressor air that has to be extracted. Critics of intercooling point out that the cooling of the cooling air only, rather than the full flow of the machine, would offer the same benefit with smaller heat exchangers. Only upon assessment of the details of the individual application can the point be settled.

The temperature–entropy diagram for a reheat, or refired, gas turbine is shown in Fig. 57.21. The cycle begins with the compression process shown by line *a–b*. The first combustion process is shown by line *b–c*. At point *c*, a turbine expands the fluid (line *c–d*) to a temperature associated with an intermediate pressure ratio. At point *d*, another combustion process takes place, returning the fluid to a high temperature (line *d–e*). At point *e*, the second expansion takes place, returning the fluid to ambient pressure (line *e–f*), whereafter the cycle is closed by discharge of the working fluid back to the atmosphere.

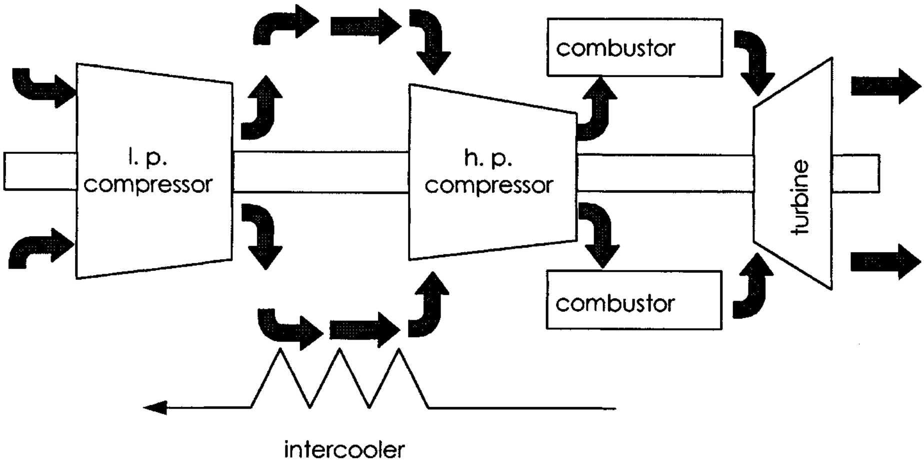

Fig. 57.20 Schematic of a single-shaft, intercooled gas turbine. In this arrangement, both compressor groups are fixed to the same shaft. Concentric, multishaft, and series arrangements are also possible.

An estimate of the cycle efficiency can be made from the temperatures corresponding to the process end points of the cycle in Fig. 57.21. Dividing the turbine temperature drops, less the compressor temperature rise, by the sum of the combustor temperature rises, one calculates an efficiency of approximately 48%. This, of course, reflects perfect compressor, combustor, and turbine efficiency and pure air as the working fluid. Actual efficiencies and properties, and consideration of turbine cooling produce less optimistic values.

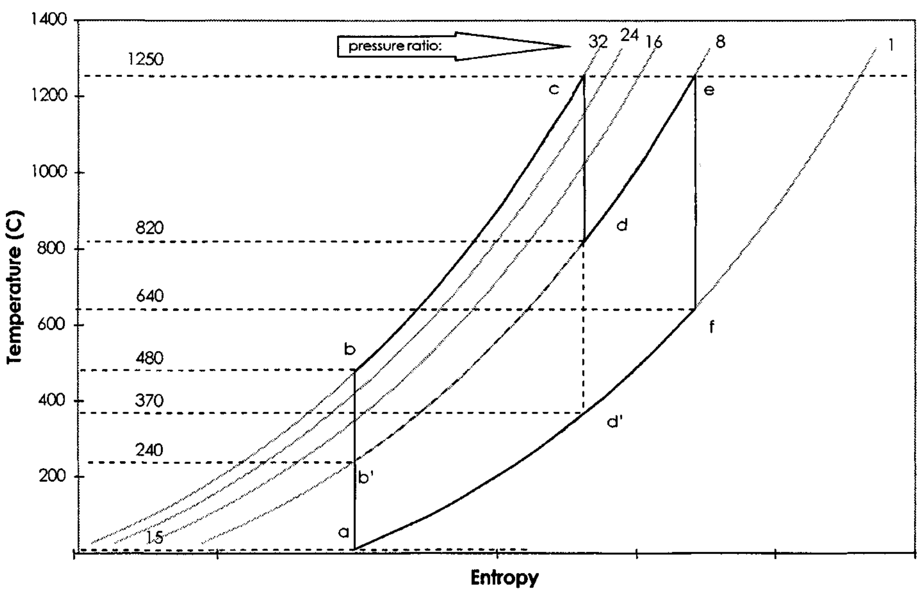

Fig. 57.21 Temperature–entropy diagram for a reheat, or refired, gas turbine. Firing temperatures were arbitrarily chosen to be equal, and to be 1250°C. The intermediate pressure ratio was chosen to be 8:1 and the overall pressure ratio to be 32:1. Dashed lines are used to illustrate comparable simple gas turbine cycles.

$$\text{Eff.} = \frac{(T_c - T_d) + (T_e - T_f) - (T_b - T_a)}{(T_c - T_b) + (T_e - T_d)}$$

A simple cycle with the same firing temperature and exhaust temperature would be described by the cycle a–b'–e–f–a. The efficiency calculated for this cycle is approximately 38%, significantly lower than for the reheat cycle. This is really not a fair comparison, since the simple cycle has a pressure of only 8:1, whereas the refired cycle operates at 32:1.

A simple-cycle gas turbine with the same pressure ratio and firing temperature would be described by the cycle a–b–c–d'–a. Computing the efficiency, one obtains a value of approximately 54%, more efficient than the comparable reheat cycle. However, there is another factor to be considered. The exhaust temperature of the reheat cycle is 270°C higher than for the simple cycle gas turbine. When applied in combined cycle power plants (these will be discussed later) this difference is sufficient to allow optimized reheat cycle-based plants more efficient than simple-cycle based plants of similar overall pressure ratio and firing temperature. Figure 57.22 shows the arrangement of a single-shaft reheat gas turbine.

Regenerators, or recuperators, are devices used to transfer the heat in a gas turbine exhaust to the working fluid, after it exits the compressor but before it is heated in the combustor. Figure 57.23 shows the schematic arrangement of a gas turbine with regenerator. Such gas turbines have been used extensively for compressor drives on natural gas pipelines and have been tested in road vehicle-propulsion applications. Regeneration offers the benefit of high efficiency from a simple, low-pressure gas turbine without resort to combining the gas turbine with a steam turbine and a boiler to make use of exhaust heat. Regenerative gas turbines with modest firing temperature and pressure ratio have comparable efficiency to advanced, aircraft-derived simple-cycle gas turbines.

The temperature–entropy diagram for an ideal, regenerative gas turbine appears in Fig. 57.24. Without regeneration, the 8:1 pressure ratio, 1000°C firing temperature gas turbine has an efficiency of ((1000–480)–(240–15))/(1000–240) = 38.8% by the method used repeatedly above. Regeneration, if perfectly effective, would raise the compressor discharge temperature to the turbine exhaust temperature, 480°C. This would reduce the heat required from the combustor, reducing the denominator of this last equation from 760°C to 520°C and thereby increasing the efficiency to 56.7%. Such efficiency levels are not realized in practice because of real component efficiencies and heat transfer effectiveness in real regenerators. The relative increase in efficiency between simple and regenerative cycles is as indicated in this example.

Figure 57.24 has shown the benefit of regeneration in low-pressure ratio gas turbines. As the pressure ratio is increased, the exhaust temperature decreases and the compressor discharge temperature increases. The dashed line a–b'–c'–d'–a shows the effect of increasing the pressure to 24:1. Note that the exhaust temperature d' is lower than the compressor discharge temperature b'. Here regeneration is impossible. As the pressure ratio (at constant firing temperature) is increased from 8:1 to nearly 24:1, the benefit of regeneration decreases and eventually vanishes. There is, of course, the possibility of intercooling the high-pressure ratio compressor, reducing its discharge temperature to where regeneration is again possible. Economic analysis and detailed analyses of the thermodynamic cycle with real component efficiencies are required to evaluate the benefits of the added costs of the heat transfer and air handling equipment.

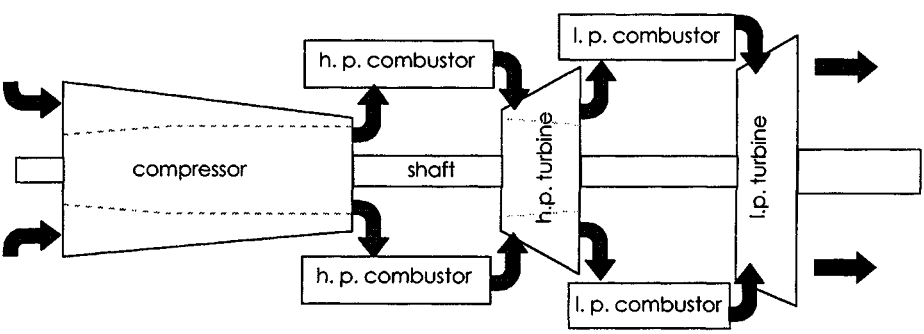

Fig. 57.22 Schematic of a reheat, or refired, gas turbine. This arrangement shows both turbines connected by a shaft. Variations include multiple shaft arrangements and independent components or component groups arranged in series.

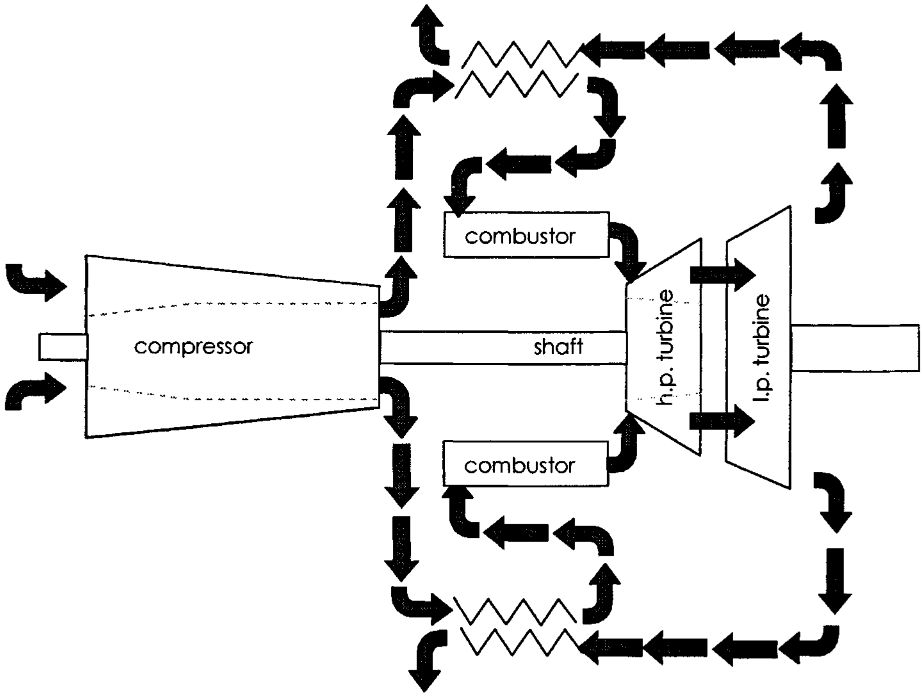

Fig. 57.23 Regenerative, multishaft gas turbine.

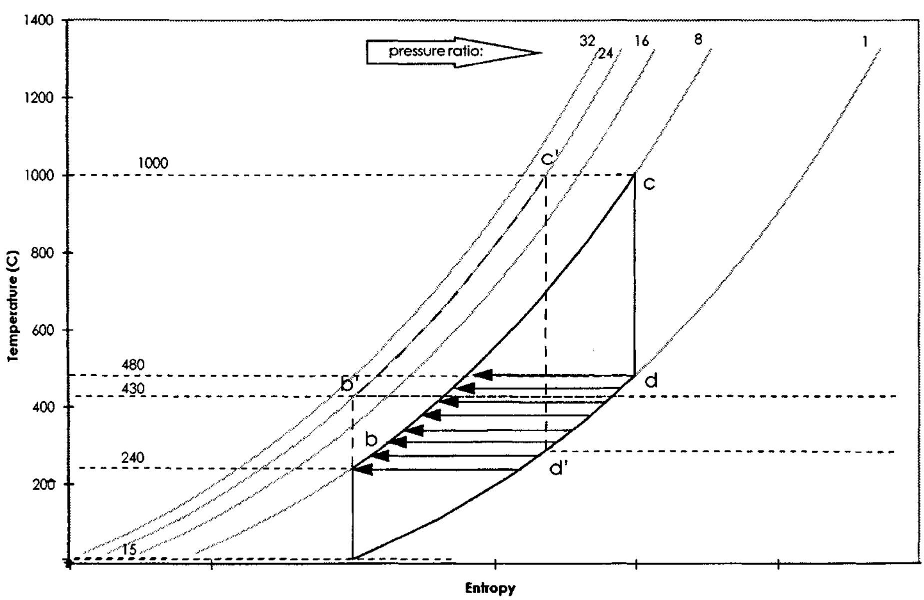

Fig. 57.24 Temperature–entropy diagram comparing an 8:1 pressure ratio, ideal, regenerative cycle with a 24:1 pressure ratio simple cycle, both at a firing temperature of 1000°C.

57.2.2 Trends in Gas Turbine Design and Performance

Output, or Size

The need for power in one location often exceeds the power produced by individual gas turbines. This is true in aircraft applications as well as power generation, and less true in gas pipelines. The specific cost (cost per unit power) of gas turbines decreases as size increases, as can be shown in Fig. 57.25. Note that the cost decreases, but at a decreasing rate; the slope remains negative at the maximum current output for a single gas turbine, around 240 MW. Output increases are accomplished by increased mass flow and increased firing temperature. Mass flow is limited by the inlet annulus area of the compressor. There are three ways of increasing annulus area:

1. Lowering rotor speed while scaling root and tip diameter proportionally. This results in geometric similarity and low risk, but is not possible in the case of synchronous gas turbines, where the shaft of the gas turbine must rotate at either 3600 rpm or 3000 rpm to generate 60 Hz or 50 Hz (respectively) alternating current.
2. Increasing tip diameter. Designers have been moving the tip velocity into the trans-sonic region. Modern airfoil design techniques have made this possible while maintaining good aerodynamic efficiency.
3. Decreasing hub diameter. This involves increasing the solidity near the root, since the cross section of blade roots must be large enough to support the outer portion of the blade against centrifugal force. The increased solidity interferes with aerodynamic efficiency. Also, where a drive shaft is designed into the front of the compressor (cold end drive) and where there is a large bearing at the outboard end of the compressor, there are mechanical limits to reducing the inlet inner diameter.

Firing Temperature

Firing temperature increases provide higher output per unit mass flow and higher combined cycle efficiency. Efficiency is improved by increased firing temperature wherever exhaust heat is put to

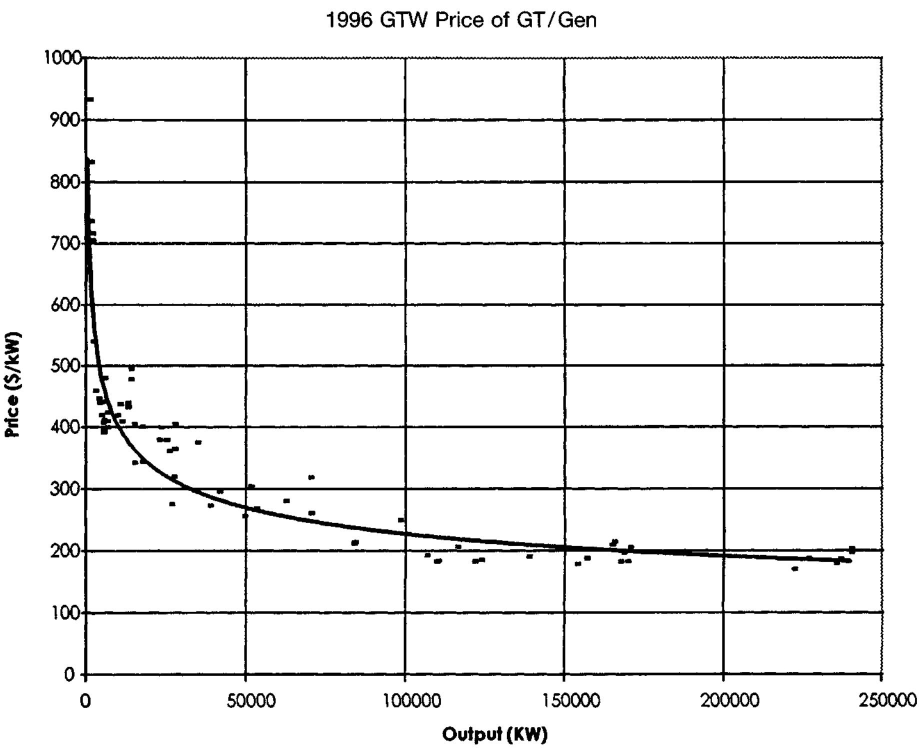

Fig. 57.25 Cost of simple cycle, generator-drive electric power generation equipment (plotted from data published by Gas Turbine World Magazine[15]).

use. Such uses include regeneration/recuperation, district heating, supplying heat to chemical and industrial processes, Rankine bottoming cycles, and adding a power turbine to drive a fan in an aircraft engine. The effect of firing temperature upon the evolution of combined Brayton–Rankine cycles for power generation is illustrated in Fig. 57.26.

Firing temperature increases when the fuel flow to the engine's combustion system is increased. The challenge faced by designers is to increase firing temperature without decreasing the reliability of the engine. A metal temperature increase of 15°C will reduce bucket creep life by 50%. Material advances and increasingly more aggressive cooling techniques must be employed to allow even small increases in firing temperature. These technologies have been discussed previously.

Maintenance practices represent a third means of keeping reliability high while increasing temperature. Sophisticated life-prediction methods and experience on identical or similar turbines are used to set inspection, repair, and replacement intervals. Coupled with design features that reduce the time required to perform maintenance, both planned and unplanned down time can be reduced to offset shorter parts lives, with no impact on reliability.

Increased firing temperature usually increases the cost of the buckets and nozzles (through exotic materials or complicated cooling configurations). Although these parts are expensive, they represent a small fraction of the cost of an entire power plant. The increased output permitted by the use of advanced buckets and nozzles is generally much higher, proportionally, than the increase in powerplant cost; hence, increased firing temperature tends to lower specific powerplant cost.

Pressure Ratio

Two factors drive the choice of pressure ratio. First is the primary dependence of simple-cycle efficiency on pressure ratio. Gas turbines intended for simple-cycle application, such as those used in aircraft propulsion, emergency power, and power where space or weight is a primary consideration, benefit from higher pressure ratios.

Combined-cycle power plants do not necessarily benefit from high pressure ratios. At a given firing temperature, an increase in pressure ratio lowers the exhaust temperature. Lower exhaust temperature means less power from the bottoming cycle and a lower efficiency bottoming cycle. So, as pressure ratio is increased, the gas turbine becomes more efficient and the bottoming cycle becomes less efficient. There is an optimum pressure ratio for each firing temperature, all other design rules held constant. Figure 57.27 shows how specific output and combined cycle efficiency are affected by gas turbine firing temperature and pressure ratio for a given type of gas turbine and steam cycle. At each firing temperature, there is a pressure ratio for which the combined cycle efficiency is highest. Furthermore, as firing temperature is increased, this optimum pressure ratio is higher as well. This fact means that, as firing temperature is increased in pursuit of higher combined cycle efficiency, pressure ratio must also be increased.

Pressure ratio is increased by reducing the flow area through the first-stage nozzle of the turbine. This increases the pressure ratio per stage of the compressor. There is a point at which increased

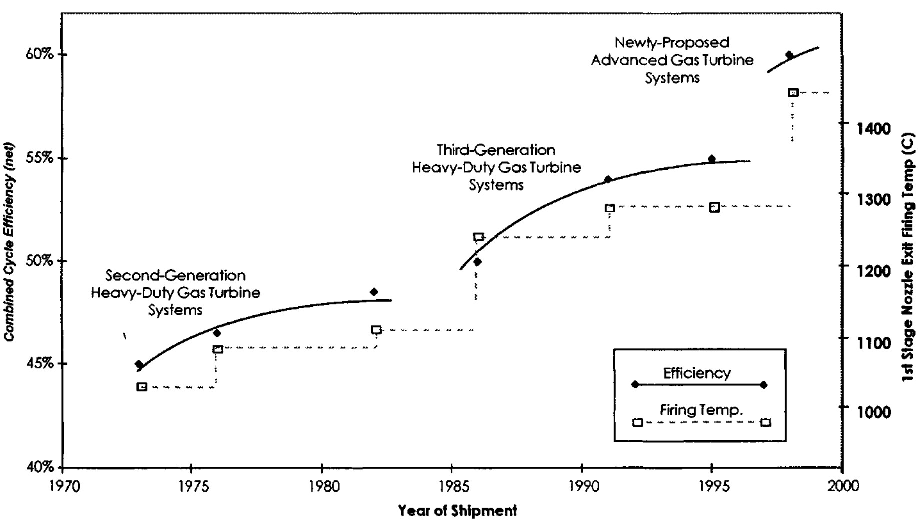

Fig. 57.26 History of power-generation, combined-cycle efficiency and firing temperature, illustrating the trend to higher firing temperature and its effect on efficiency.

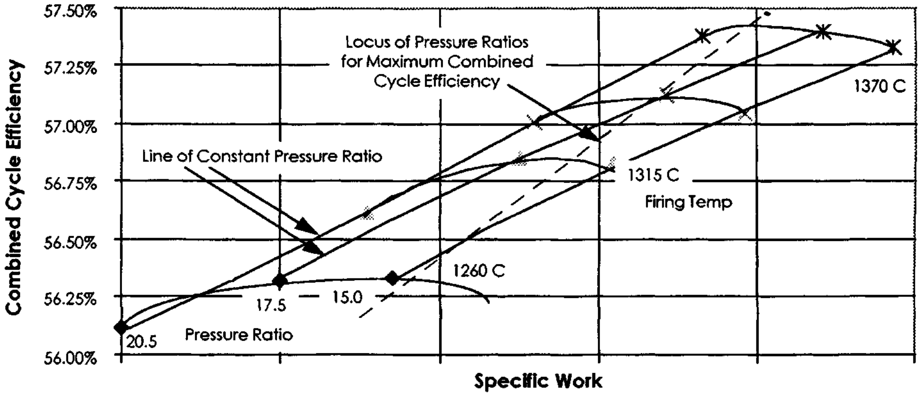

Fig. 57.27 Effect of pressure ratio and firing temperature on combined cycle efficiency and specific work.

pressure ratio causes the compressor airfoils to stall. Stall is avoided by either adding stages (reducing the pressure ratio per stage) or increasing the chord length, and applying advanced aerodynamic design techniques. For significant increases in pressure ratio, a simple, single-shaft rotor with fixed stationary airfoils cannot deliver the necessary combination of pressure ratio, stall margin, and operating flexibility. Features required to meet all design objectives simultaneously include variable-angle stationary blades in one or more stages; extraction features that can be used to bleed air from the compressor during low-speed operation; and multiple rotors that can be operated at different speeds.

Larger size, higher firing temperature, and higher pressure ratio are pursued by manufacturers to lower cost and increase efficiency. Materials and design features evolve to accomplish these advances with only positive impact on reliability.

57.3 APPLICATIONS

57.3.1 Use of Exhaust Heat in Industrial Gas Turbines

Adding equipment for converting exhaust energy to useful work can increase the thermal efficiency of a gas turbine-based power plant by 10 to over 30%. The schemes are numerous, but the most significant is the fitting of a heat-recovery steam generator (HRSG) to the exhaust of the gas turbine and delivering the steam produced to a steam turbine. Both the steam turbine and gas turbine drive electrical generators.

Figure 57.28 displays the combining of the Brayton and Rankine cycles. The Brayton cycle *a–b–c–d–a* has been described already. It is important to point out that the line *d–a* now represents heat transferred in the HRSG. In actual plants, the turbine work is reduced slightly by the backpressure associated with the HRSG. Point *d* would be above the 1 : 1 pressure curve, and the temperature drop proportionately reduced.

The Rankine cycle begins with the pumping of water into the HRSG, line *m–n*. This process is analogous to the compression in the gas turbine, but rather than absorbing 50% of the turbine work, consumes only about 5%, since the work required to pump a liquid is less than that required to compress a gas. The water is heated (line *n–o*) and evaporated (*o–p*). The energy for this is supplied in the HRSG by the exhaust gas of the gas turbine. More energy is extracted to superheat the steam, as indicated by line *p–r*. At this point, superheated steam is delivered to a steam turbine and expanded (*r–s*) to convert the energy therein to mechanical work.

The addition of the HRSG reduces the output of the gas turbine only slightly. The power required by the mechanical devices (like the feedwater pump) in the steam plant is also small. Therefore, most of the steam turbine work can be added to the net gas turbine work with almost no increase in fuel flow. For combined-cycle plants based on industrial gas turbines where exhaust temperature is in the 600°C class, the output of the steam turbine is about half that of the gas turbine. Their combined-cycle efficiency is approximately 50% higher than simple-cycle efficiency. For high-pressure ratio gas turbines with exhaust temperature near 450°C, the associated steam turbine output is close to 25% of the gas turbine output, and efficiency is increased by approximately 25%. The thermodynamic cycles of the more recent large industrial gas turbines have been optimized for high combined-cycle efficiency. They have moderate to high simple-cycle efficiency and relatively high exhaust temperatures. Figure 57.28 has shown that net combined-cycle efficiency (lower heating value) of

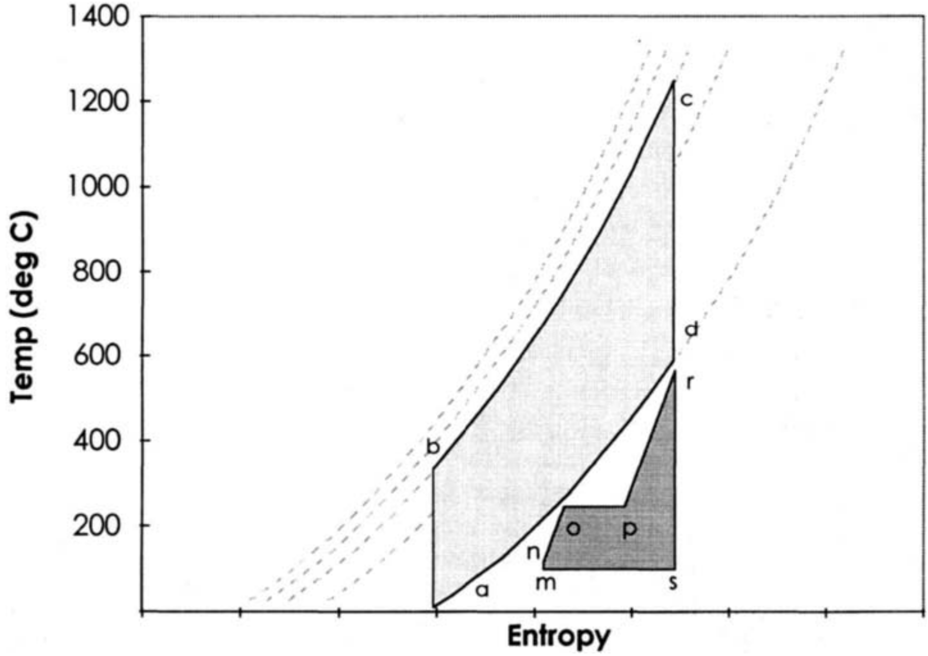

Fig. 57.28 Temperature–entropy diagram illustrating the combining of a gas turbine (*a–b–c–d–a*) and steam turbine cycle (*m–n–o–p–r–s–m*). The heat wasted in process *d–a* in simple-cycle turbines supplies the heat required by processes *n–o, o–p,* and *p–r.*

approximately 55% has been realized as of this writing, and levels of 60% and beyond are under development.

Figure 57.29 shows a simple combined-cycle arrangement, where the HRSG delivers steam at one pressure level. All the steam is supplied to a steam turbine. Here, there is neither steam reheat nor additional heat supplied to the HRSG. There are many alternatives.

Fired HRSGs, where heat is supplied both by the gas turbine and by a burner in the exhaust duct, have been constructed. This practice lowers overall efficiency, but accommodates the economics of some situations of variable load requirements and fuel availability. In other applications, steam from the HRSG is supplied to nearby industries or used for district heating, lowering the power generation efficiency but contributing to the overall economics in specific applications.

Efficiency of electric power generation benefits from more complicated steam cycles. Multiple pressure, non-reheat cycles improve efficiency as a result of additional heat transfer surface in the HRSG. Multiple pressure, reheat cycles, such as shown in Fig. 57.30 match the performance of higher exhaust temperature gas turbines (600°C). Such systems are the most efficient currently available, but are also the most costly. The relative performance for several combined-cycle arrangements is shown in Table 57.1.[16] The comparison was made for plants using a gas turbine in the 1250°C firing temperature class.

Plant costs for simple-cycle gas turbine generators is lower than that for steam turbines and most other types of powerplant. Since combined-cycle plants generate 2/3 of their power with the gas turbine, their cost is between that of simple-cycle gas turbine plants and steam turbine plants. Their efficiency is higher than either. The high efficiency and low cost combine to make combined-cycle plants extremely popular. Very large commitments to this type of plant have been made around the world. Table 57.2 shows some of the more recent to be put into service.

There are other uses for gas turbine exhaust energy. Regeneration, or recuperation, uses the exhaust heat to raise the temperature of the compressor discharge air before the combustion process. Various steam-injection arrangements have been used as well. Here, an HRSG is used as in the combined-cycle arrangements shown in Fig. 57.30, but instead of expanding the steam in a steam turbine, it is introduced into the gas turbine, as illustrated in Fig. 57.31. It may be injected into the combustor, where it lowers the generation of NO_x by cooling the combustion flame. This steam increases the mass flow of the turbine and its heat is converted to useful work as it expands through

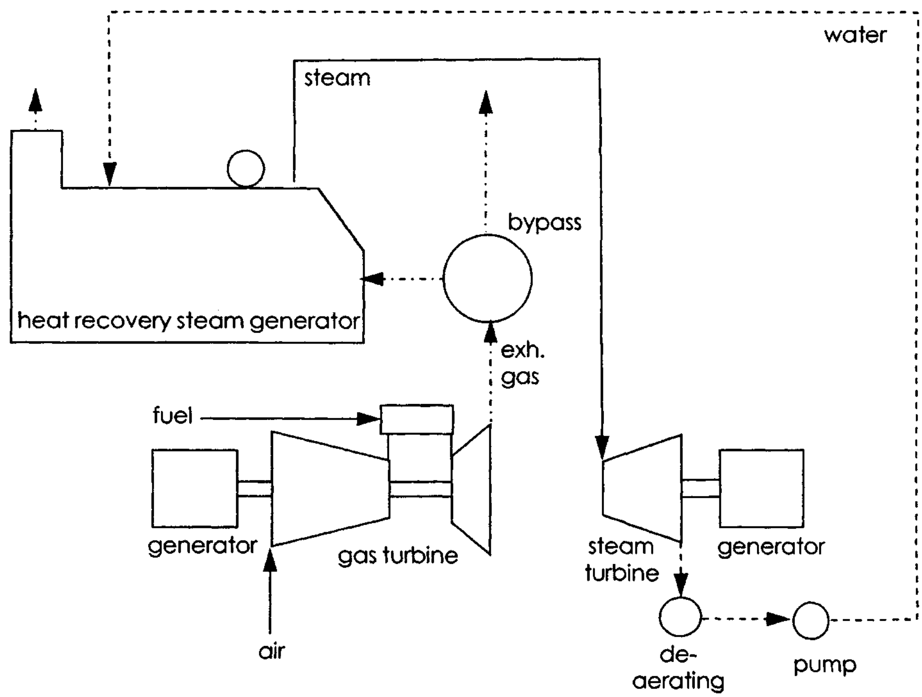

Fig. 57.29 Schematic of simple combined cycle power plant. A single-pressure, nonreheat cycle is shown.

the turbine section of the gas turbine. Steam can also be injected downstream of the combustor at various locations in the turbine, where it adds to the mass flow and heat of the working fluid. Many gas turbines can tolerate steam-injection levels of 5% of the mass flow of the air entering the compressor; others can accommodate 16% or more, if distributed appropriately along the gas path of the gas turbine. Gas turbines specifically designed for massive steam injection have been proposed and studied. These proposals arise from the fact that the injection of steam into gas turbines of existing designs has significant reliability implications. There is a limit to the level of steam injection into combustors without flame-stability problems and loss of flame. Adding steam to the gas flowing through the first-stage nozzle increases the pressure ratio of the machine and reduces the stall margin of the compressor. Addition of steam to the working fluid expanding in the turbine increases the heat-transfer coefficient on the outside surfaces of the blading, raising the temperature of these components. The higher work increases the aerodynamic loading on the blading, which may be an issue on latter-stage nozzles, and increases the torque applied to the shafts and rotor flanges. Design changes can be made to address the effects of steam in the gas path.[17]

Benefits of steam injection are an increase in both efficiency and output over those of the simple-cycle gas turbine. The improvements are less than those of the steam turbine and gas turbine combined-cycles, since the pressure ratio of the steam expansion is much higher in a steam turbine. Steam turbine pressures may be 100 atmospheres; gas turbines no higher than 40. Steam-injection cycles are less costly to produce since there is no steam turbine. There is, of course, higher water consumption with steam injection, since the expanded steam exits the plant in the gas turbine exhaust.

57.3.2 Integrated Gasification Combined Cycle

In many parts of the world, coal is the most abundant and lowest-cost fuel. Coal-fired boilers raising steam that is expanded in steam turbine generators are the conventional means of converting this fuel to electricity. Pulverized coal plants with flue gas desulfurization operate at over 40% overall efficiency and have demonstrated the ability to control sulfur emissions from conventional boiler systems. Gas turbine combined-cycle plants are operating with minimal environmental impact on natural gas at 55% efficiency, and 60% is expected with new technologies. A similar combined-cycle plant that could operate on solid fuel would be an attractive option.

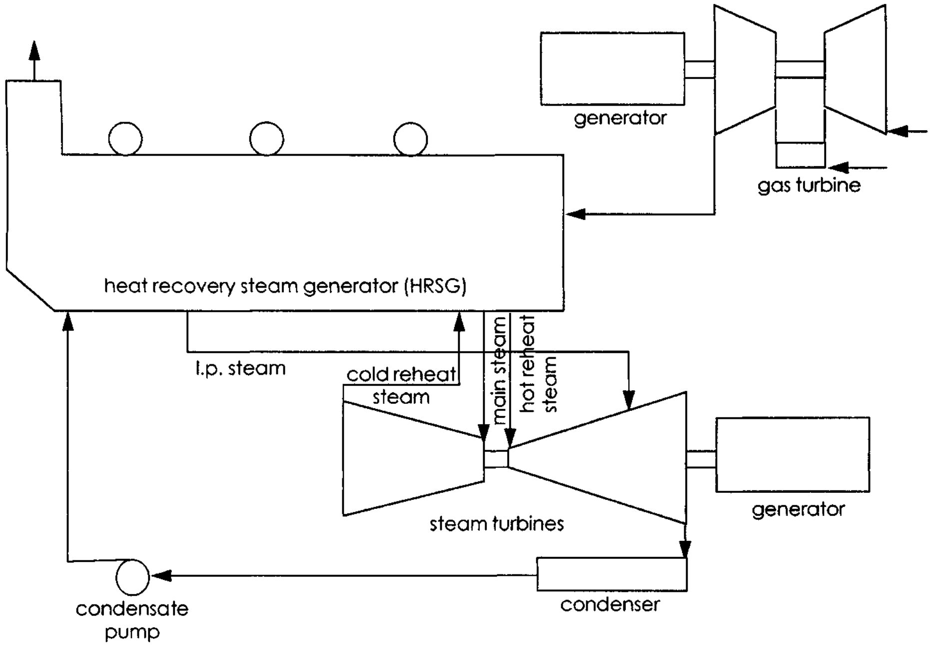

Fig. 57.30 Three-pressure, reheat combined cycle arrangement. The highest power generation efficiency is currently achieved by such plants.

Competing means of utilizing coal with gas turbines have included direct combustion, indirect firing, and gasification. Direct combustion in conventional, on-engine combustors has resulted in rapid, ash-caused erosion of bucket airfoils. Off-base combustion schemes, such as pressurized fluidized bed combustors, have not simultaneously demonstrated the high exit temperature needed for efficiency and low emissions. Indirect firing raises compressor discharge temperature by passing it through a heat exchanger. Metal heat exchangers are not compatible with the high turbine inlet temperature required for competitive efficiency. Ceramic heat exchangers have promise, but their use will necessitate the same types of emission controls required on conventional coal-fired plants. Power plants with the gasification process, desulfurization, and the combined-cycle machinery integrated have been successfully demonstrated, with high efficiency, low emissions, and competitive first cost. Significant numbers of integrated gasification combined-cycle (IGCC) plants are operating or under construction.

Fuels suitable for gasification include several types of coal and other low-cost fuels. Those studied include:

- Bituminous coal
- Sub-bituminous coal
- Lignite
- Petroleum coke
- Heavy oil
- Orimulsion
- Biomass

Fuel feed systems of several kinds have been used to supply fuel into the gasifier at the required pressure. Fuel type, moisture content size, and the particular gasification process need to be considered in selecting a feed system.

Several types of gasifiers have been designed to produce fuel with either air or oxygen provided. The system shown in Fig. 57.32 features a generic oxygen-blown gasifier and a system for extracting some of the air from the compressor discharge and dividing it into oxygen and nitrogen. An oxygen-blown gasifier produces a fuel about 1/3 of the heating value of natural gas. The fuel produced by the gasifier, after sulfur removal, is about 40% CO, 30% H_2. The remaining 30% is mostly H_2O and

Table 57.1 Comparison of Performance for Combined Cycle Arrangements Based on Third Generation (1250°C Firing Temperature) Industrial Gas Turbines

Steam Cycle	Relative Net Plant Output (%)	Relative Net Plant Efficiency (%)
Three pressure, reheat	Base	Base
Two pressure, reheat	−1.1	−1.1
Three pressure, non-reheat	−1.2	−1.2
Two pressure, non-reheat	−2.0	−2.0

CO_2, which are inert and act as diluents in the gas turbine combustor, reducing NO_x formation. A typical lower heating value is 1950 K-Cal/m^3. The fuel exits the gasifier at a temperature higher than that at which it can be cleaned. The gas is cooled by either quench or heat-exchange and cleaned. Cleaning is done by water spray scrubber or dry filtration to remove solids that are harmful to the turbine and potentially harmful to the environment. This is followed by a solvent process that absorbs H_2S.

Some gas turbine models can operate on coal-gas without modification. The implications for the gas turbine relate to the volume of fuel—three times higher, or more, than that of natural gas. When the volume flow through the first stage nozzle of the gas turbine increases, the backpressure on the compressor increases. This increases the pressure ratio of the cycle and decreases the stall margin of the compressor. Gas turbines with robust stall margins need no modification. Others can be adapted by reducing inlet flow by inlet heating or by closing off a portion of the inlet (variable inlet stator vanes can be rotated toward a closed position). The volume flow through the turbine increases as well. This increases the heat transfer to the buckets and nozzles. To preserve parts lives, depending on the robustness of the original design, the firing temperature may have to be reduced. The increased flow and decreased firing temperature, if required, result in higher gas turbine output than developed by the same gas turbine fired on natural gas.

57.3.3 Applications in Electricity Generation

Worldwide shipments of industrial gas turbines before 1965 were below 2 gW total output capacity per year. In 1992, more than 25 gW of capacity was shipped, and deliveries continue at this rate. Of the 1992 volume, nearly 90% was for electric power generation. Approximately 9% of all the world's current generating capacity is by gas turbines, either in simple-cycle or combined-cycle. Current electric power generation additions are increasingly provided by gas turbines. Over 10% of additions are by simple-cycle gas turbines, and over 25% by combined-cycle plants, which derive 2/3 of their capacity from gas turbines. Thus, between 25 and 30% of additions are gas turbine generators. This compares to 40 to 50% by steam turbine generators alone and in combined-cycle. The remaining additions are provided by hydroelectric plants, nuclear, and other means.

The tenfold increase in the volume of gas turbines shipments between 1965 and the present was due to several factors. First, in the late 1960s and early 1970s, there was a need for peaking power in the United States. Gas turbines, because of their low cost, low operating crew size, and fast installation time, were the engine of choice. Because of the seasonal and daily variations in the demand for electric power, generating companies could minimize their investment in plant and equipment by installing a mixture of expensive but efficient base load plants (steam and nuclear), run over 8000 hours per year, and far less expensive—but less efficient—plants that would operate only a few hundred hours per year.

Existing industrial gas turbines and newly designed larger units whose operating speed was chosen to match the requirements of a directly coupled alternator met the demand for peaking power. The experience on these early units resulted in improvements in efficiency, reliability, and cost-

Table 57.2 Recent Large Multiunit Gas Turbine-Based Combined-Cycle Power Plants

Plant	Country	Number of Gas Turbines	Plant Output (mW)
TEPCO—Yokohama	Japan	8	2800
TEPCO—Futtsu	Japan	14	2000
KEPCO—Ildo 1 & 2	Korea	8	1910
PLN—Gresik 1–3	Indonesia	9	1796
Chubu—Kawagoe 1–7	Japan	7	1695
Enron—Wilton on Teeside 1 & 2	U.K.	8	1644
Midland Cogen—Michigan	U.S.A.	12	1470

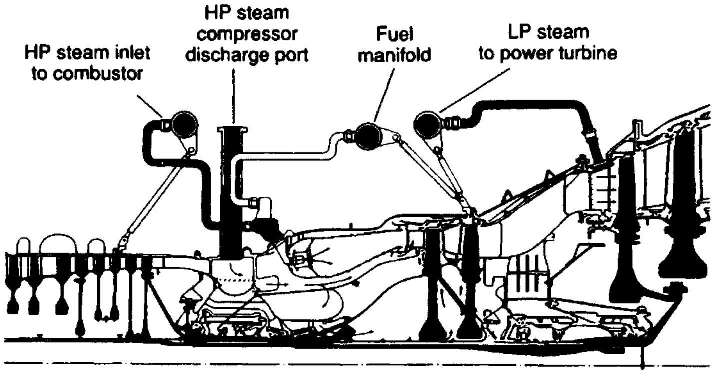

Fig. 57.31 LM500 aero-derivative gas turbine with steam injection (courtesy of General Electric Company).

effectiveness. Much of the technology needed to improve the value of industrial gas turbines came from aircraft engine developments, as it still does. Beginning in the 1970s, with the rapid rise in oil prices and associated natural gas prices, electric utilities focused on ways of improving the efficiency of generating plants. Combined-cycle plants are the most thermally efficient fuel-burning plants. Furthermore, their first-cost is lower than all other types of plants except simple-cycle gas turbine plants.

The only drawback to gas turbine plants was their requirement for more noble fuels; natural gas and light distillates are usually chosen to minimize maintenance requirements. Coal is abundant in many parts of the world and costs significantly less than oil or gas per unit energy. Experiments in the direct firing of gas turbines on coal have been conducted without favorable results. Other schemes

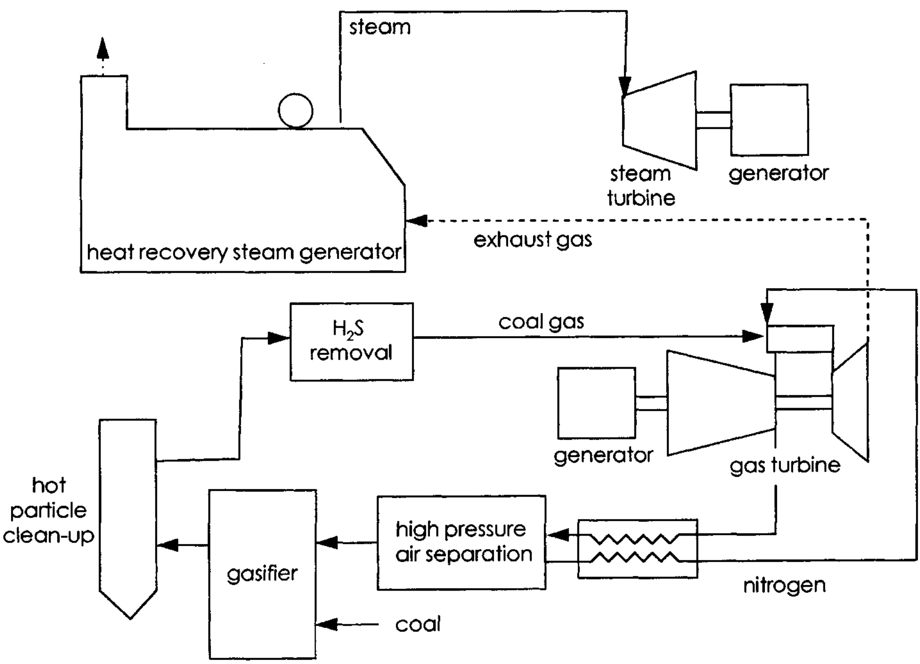

Fig. 57.32 Integrated gasification combined-cycle diagram. Air compressed in the gas turbine is cooled, oxygen separated and fed to the gasifier, nitrogen to gas turbine for NO_x control. Coal is partially burned in the gasifier. The gas produced is cleaned and flows to the gas turbine as fuel.

for using coal in gas turbines include indirect firing, integrating with a fluidized bed combustor, and integrated gasification. The last of these offers the highest efficiency due to its ability to deliver the highest-temperature gas to the turbine blading. Furthermore, integrated gasification is the most environmentally benign means of converting coal to electricity. The technology has been demonstrated in several plants, including early technology demonstration plants and commercial power-generating facilities.

57.3.4 Engines for Aircraft

Aircraft engines exert a forward thrust on the airframe in reaction to the rearward acceleration imparted to part of the passing airflow. This is achieved by means of a propeller or fan or by generating a high-pressure jet that emerges through a nozzle. Propellers can be driven by gas turbines (these are turboprop engines) as well as reciprocating engines. The term *fan* is used to describe a low-pressure compressor of one or more stages; part of its discharge flow bypasses the core compressor, combustor, and turbine of the engine. The acceleration of the air bypassing the core contributes to the engine's thrust. Variations include the duct-burner configurations. The choice of engine type is determined by the required flight speed, since propulsion efficiency of each engine type has different efficiency characteristics.

The appearance of aircraft engines and stationary gas turbines differ considerably. The premium placed on weight for all aircraft equipment results in different economic tradeoffs. This is particularly evident in the casings. Industrial engines, for the most part, are designed with thick cast casings of inexpensive material that are bolted together both horizontally and axially. The horizontal split line allows convenient field disassembly—a necessity in large equipment. Industrial engine casings are left as-cast on the outer surface, while aircraft engine casings are machined to close tolerances and have features that may be costly to machine, but are compensated for by the value placed on low weight. Aircraft engine casings are made of higher-strength material and can be thinner. Disks, especially in the compressor, are thinner than in industrial gas turbines. This is made possible by the use of higher strength materials, light alloy blades, and wheel-to-wheel attachment methods that may not be economical in larger sizes. Hot section (turbine and combustor) parts, however, are quite similar in appearance and material.

Aircraft engine performance is described in terms of thrust, fuel consumption rate, and engine weight.

- Net thrust:

$$F_n = W1(V_j - V_o) + Ae(P_j - P_o) \qquad \text{for turbojet}$$

$$F_n = W_c(V_j - V_O) + W_d(V_d - V_o) + Ae(P_j - P_o) \qquad \text{for turbofan}$$

- Specific thrust:

$$\mathrm{ST} = F_n / W1$$

- Specific fuel consumption:

$$\mathrm{SFC} = W_f / F_n \qquad \text{(general definition)}$$

$$\mathrm{SFC}_{tp} = W_f/\mathrm{HP/hr} \qquad \text{(sometimes applied)}$$

to turboprop and shaft engines

where $W1$ = total inlet mass flow rate
W_c = portion of $W1$ passing through the core engine
W_d = portion of $W1$ passing through bypass duct
V_o = flight velocity
V_j = exhaust jet velocity
Ae = exhaust area
P_j = exhaust jet static pressure
P_o = ambient pressure
W_f = fuel flow rate

Thrust multiplied by flight velocity is a measure of the work done by the propulsion system. Propulsion system efficiency is the ratio of this work to the fuel supplied per unit time. Hence,

$$\eta_{ps} = \text{const} \times (F_n V_o)/W_f$$

or

$$\eta_{ps} = \text{const} \times V_o / \text{SFC}$$

Efficiency is inversely proportional to specific fuel consumption and is a function of both SFC and aircraft velocity.[18,19]

Thrust characteristics and SFC characteristics vary as a function of aircraft velocity, and vary differently for each type of engine. This is because efficiency improves as the aircraft velocity approaches the engine exit velocity. The engine exhaust velocity must be above the aircraft velocity to generate thrust, but large differences in velocity relate to inefficient propulsion. Figures 57.33 and 57.34 show the specific thrust and specific fuel consumption for various engines as functions of aircraft velocity.

The figures show that at lower Mach numbers, the turbofan engines have relatively high propulsion efficiency (low SFC). By employing a large-diameter fan, a very large mass of air can be accelerated to relatively low discharge velocities, avoiding high mismatch between exhaust and aircraft velocity. The need for improved efficiency in the high-subsonic speed regime has produced a focus on turbofan engines rather than turbojets. At lower speeds, turboprop engines are preferred.

Figures 57.35*a* and *b* compare the engines selected for, or competing for, recent applications. All of the larger commercial transport applications and newer military applications are met by turbofan engines.[20]

The range of ratings for each engine designation is due to the practice of fine-tuning engine performance for particular applications, incremental performance gains over time, and optional features. This comparison is a snapshot of performance over a particular time. Relative ratings change often as manufacturers continue to apply new technologies and improve designs. One of the newest and most powerful turbofan engines is shown in Fig. 57.36. It is a two-rotor engine. The one-stage fan and three-stage low-pressure compressor are joined on one shaft connected to a six-stage low-pressure turbine. The 10-stage high-pressure compressor is driven by the two-stage high-pressure turbine, both joined on another shaft that can rotate at a higher speed. The ratio of the air mass flow through the duct to the air flowing through the compressor, combustor, and turbines is 9:1. The overall pressure ratio is 40:1 and the rated thrust is in the 85,000 pound class. The engine is over 3 m in diameter. A new feature for aircraft engines is the double-domed, lean-premixed, fuel-staged dry low-NO_x combustor. The GE 90, PW4084, and 800-series RB.211 Trent high-bypass-ratio turbofan engines have been built for use on the Boeing 777 aircraft.

Supersonic flight requires a considerably increased jet velocity. The afterburner reheats the exhaust gas after the turbines, permitting it to accelerate to an appropriate level above the flight velocity and boosting the thrust to overcome the increased drag.

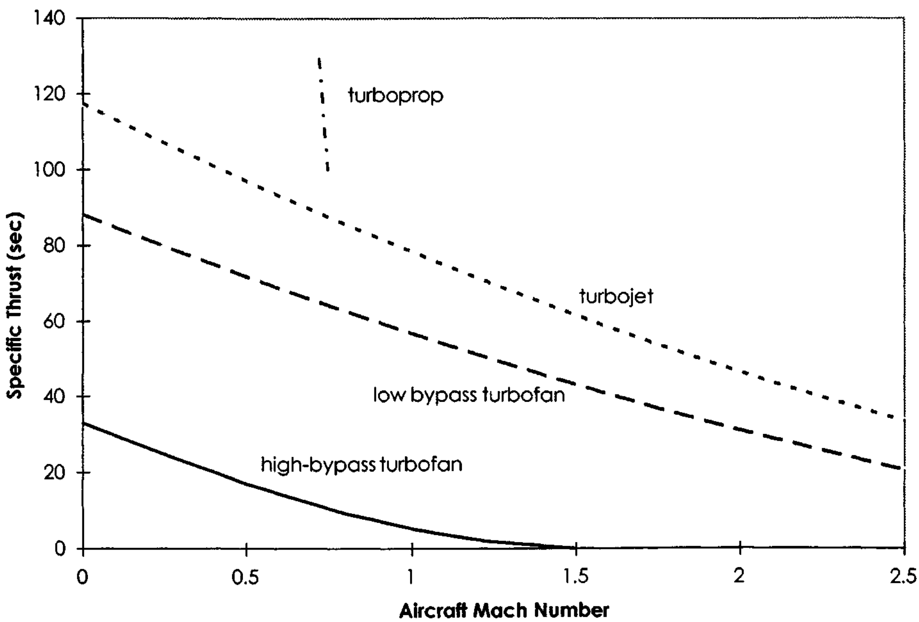

Fig. 57.33 Low values of specific thrust give higher propulsive efficiency at low Mach numbers (redrawn from a figure in Ref. 18).

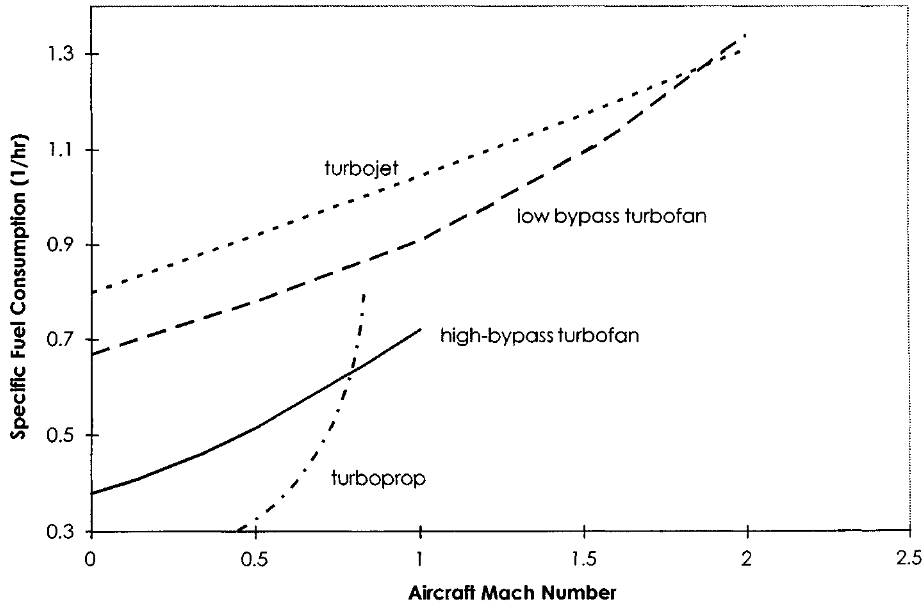

Fig. 57.34 As flight Mach number increases, higher specific thrusts are necessary to maintain high propulsive efficiency and reduce SFC (redrawn from a figure in Ref. 18).

Another aircraft engine type is the auxiliary power unit (APU). It is a small turboshaft engine that provides air-conditioning and electric and hydraulic power while the main engines are stopped, but its main function is to start the main engines without external assistance. The APU is usually started from batteries by an electric motor. When it reaches its operating speed, a substantial flow of air is bled from its compressor outlet and is ducted to drive the air turbine starters on the main engines. The fuel flow is increased when the turbine air supply is reduced by the air bleed, to provide the energy required for compression. These engines are also found on ground carts, which may be temporarily connected to an aircraft to service it. They may also have uses in industrial plants requiring air pressure at 3 or 4 bar.

57.3.5 Engines for Surface Transportation

This category includes engines for rail, road, offroad, and overwater transport. The low weight and high power density of gas turbines are assets in all cases, but direct substitution for Diesel or Otto cycle engines is unusual. When the economics of an application favor high power density, high driven-device speed, or when some heat recovery is possible, gas turbines become the engines of choice. Surface vehicle engines include the array of turboshaft and turboprop derivatives, free-turbine aero-derivative and industrial gas turbines, and purpose-built gas turbines. Applications exist for engines of from around 100 horsepower to nearly 40,000.

Truck, bus, and automobile gas turbine engines are, for the most part, in the development stage. Current U.S. Department of Energy initiatives are supporting development of gas turbine automobile engines of superior efficiency and low emissions. Production cost similar to current power plants is also a program goal. Additional requirements must be met, such as fast throttle response and low fuel consumption at idle. The balancing of efficiency, first-cost, size, and weight have led to different cycle and configuration choices than for aircraft or power-generation applications. Regenerative cycles with low pressure ratios have been selected. Parts count and component costs are addressed through the use of centrifugal compressors, integral blade-disk axial turbines, and radial inflow turbines. Low-pressure-ratio designs support the low stage count. It is possible to achieve the necessary pressure ratio with one centrifugal compression stage and in one turbine stage, or one each high pressure and power turbine. The small size of parts and the selection of radial inflow or integral blade-disk turbines make ceramic materials an option. Single-can combustors are also employed to control cost. Prototypes have been built and operated in the United States, Europe, and Japan.[21]

The most successful automotive application of gas turbines is the power plant for the M1 Abrams Main Battle Tank. The engine uses a two-spool, multistage, all-axial flow gas generator plus power turbine. The cycle is regenerative. Output and cost appear too high for highway vehicle application.

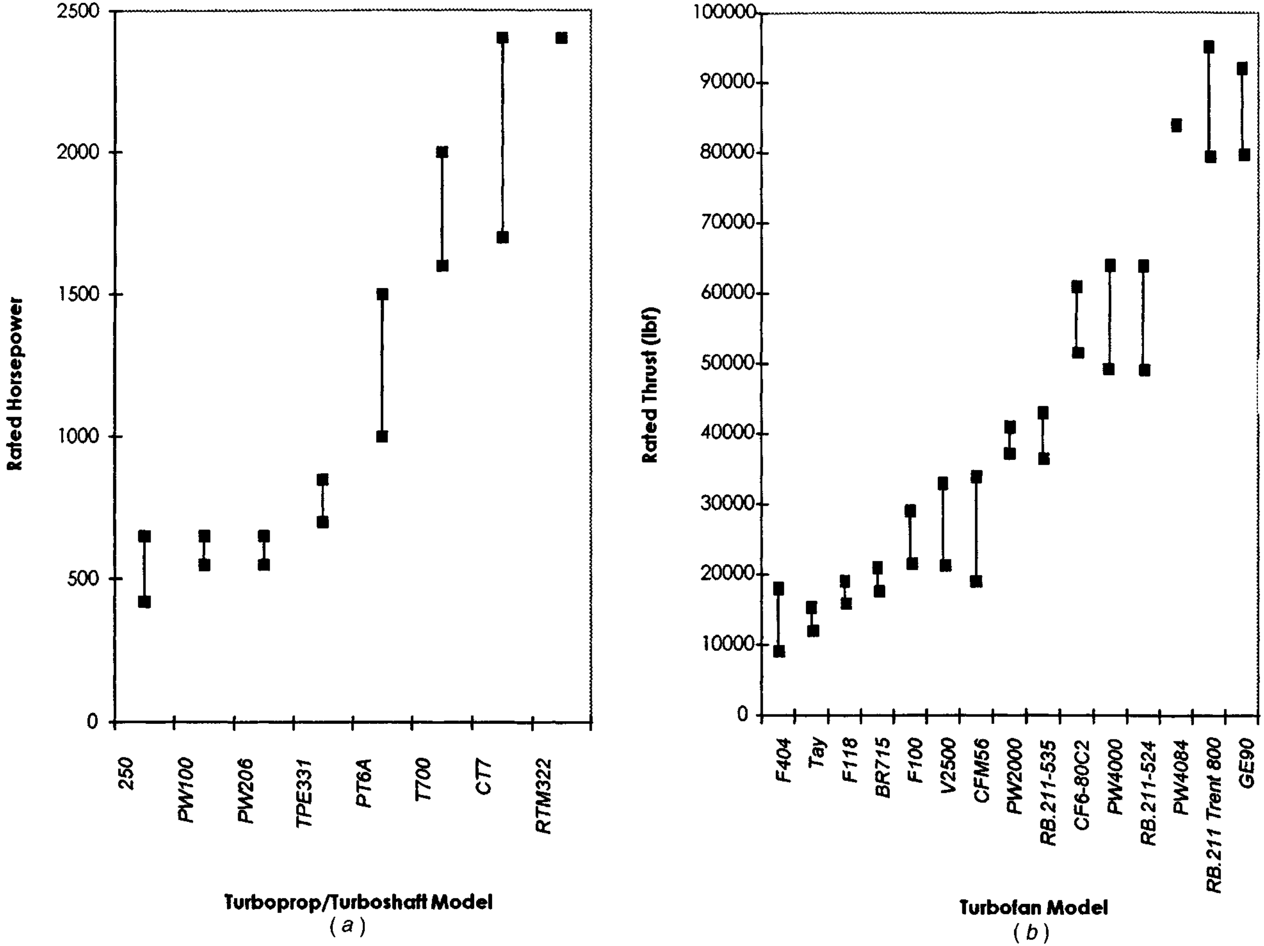

Fig. 57.35 Output performance ratings of some aircraft propulsion gas turbines in recent and near-term applications. Ratings data from industry publications. Note that the shaft engine ratings are in horsepower and the turbofans are rated by pounds of thrust. The conversion between horsepower and thrust varies, since it depends on propeller or rotor efficiency.

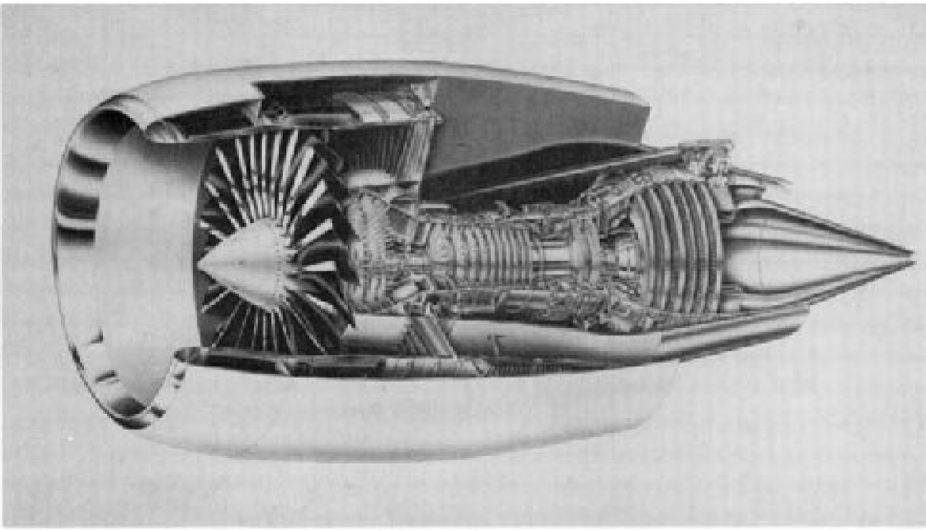

Fig. 57.36 Sectional view of the GE 90, a high-bypass-ratio two-shaft turbofan engine rated at over 80,000 lbf thrust (courtesy of General Electric Company).

Ship propulsion by gas turbine is more commonplace. One recent report summarizing orders and installations over an 18-month period listed 10 orders for a total of 64 gas turbines—775 MW in all. Military applications accounted for 55% of the total. The remaining 45% were applied to fast ferries and similar craft being built in Europe, Australia, and Hong Kong.[15] Gas turbine outputs in the 3–5 MW range, around 10 MW, and in the 20–30 MW range account for all the applications. Small industrial engines were selected in the 3–5 MW range and aero-derivative, free-turbine engines accounted for the remainder.

Successful application of gas turbines aboard ship requires protection from the effects of salt water and, in the case of military vessels, maneuver and sudden seismic loads. In addition to the common problems with salt water-induced corrosion of unprotected metal, airborne sodium, in combination with sulfur usually found in the fuel or air, presents a problem for buckets, nozzles, and combustors. Hot corrosion—also called *sulfidation*—has led to the development of alloys that combine the creep strength of typical aircraft engine bucket and nozzle alloys with superior corrosion resistance. Inconel 738 was the first such alloy. This set of alloys is used in marine propulsion engines. Special corrosion-resistant coatings are applied to further improve the corrosion resistance of nickel-based superalloy components. The level of sodium ingested by the engine can also be controlled with proper inlet design and filtration.[14]

Although there was a period when gas turbines were being applied as prime movers on railroad locomotives, the above report contained only one small railroad application.

57.4 EVALUATION AND SELECTION

57.4.1 Maintenance Intervals, Availability, and Reliability

Service requirements of aircraft and industrial gas turbines differ from other power plants principally in the fact that several key components, because they operate at very high temperatures, have limited lives and must be repaired or replaced periodically to avoid failures during operation. These components include combustion chambers, buckets, and nozzles. Other components, such as wheels or casings, may occasionally require inspection or retirement.

Wear-out mechanisms in hot gas path components include creep, low cycle fatigue, corrosion, and oxidation. All combustors, buckets, and nozzles, if operated for significantly longer than their design life, will eventually fail in one of these modes. Repair or replacement is required to avoid failure. Most of the failure mechanisms give some warning prior to loss of component integrity. Corrosion and oxidation are observable by visual inspection. The creep deflection of nozzles can be detected by measuring changes in clearances. Low-cycle fatigue cracks can occur in nozzles, buckets, and combustors without causing immediate failure of these components. These can be detected visually or by more sophisticated nondestructive inspection techniques. Tolerance of cracks depends on the particular component design, service conditions, and other forces or temperatures superimposed on the component at the location of the crack. Inspection intervals are set by manufacturers, based on laboratory data and field experience, so that components with some degree of distress can be removed from service or repaired prior to component failure.

Bucket creep often gives no advance warning, due to several factors. First, the ability of bucket alloys to withstand alternating stress and the rate of creep progression are both affected by the existence of creep void formation. Local creep void formation is difficult to observe even in individual buckets exposed to radiographic and other nondestructive inspections. Destructive inspection of samples taken from a turbine are not useful in predicting the conditions in the particular single bucket in a stage that will exhibit the most advanced creep conditions. This is due to the statistical distribution of creep conditions in a sample set. Such a large number of samples would be required to accurately predict the condition of the worst part in a set that the cost of such an inspection would be higher than the set of replacement components. Because of this, creep failure is avoided by the retirement of sets of buckets as the risk of the failure of the worst bucket in the set increases above a preselected level.

Some of the wear-out mechanisms are time-related, while others are start-related. Thus, the actual service profile is significant to determining when to inspect or retire gas path components. Manufacturers differ in the philosophy applied. Aircraft engine maintenance recommendations are based on a particular number of mission hours of operation. Each mission contains a number of hours at takeoff conditions, a number at cruise, a number of rapid accelerations, thrust-reversals, and so on. Component lives are calculated and expressed in terms of a number of mission cycles. Thus, the life of any component can be expressed in hours, even if the mechanism of failure expected is low-cycle fatigue related to the number of thermal excursions to which the component is exposed. Inspection and component retirement intervals, based on mission-hours, can be set to detect distress and remove or repair components before the actual failure is likely to occur.

Industrial gas turbine manufacturers have historically designed individual products to be suitable for both continuous duty and frequent starts and stops. A particular turbine model may be applied to missions ranging from twice-daily starts to continuous operation for over 8000 hours per year and virtually no start cycles. To deal with this, manufacturers of industrial gas turbines have developed two ways of expressing component life and inspection intervals. One is to set two criteria for inspection: one based on hours and one based on starts. The other is to develop a formula for "equivalent hours" that counts each start as a number of additional hours of operation. These two methods are illustrated in Fig. 57.37. The figure is a simplification in that it considers only normal starts and base-load hours. Both criteria evaluate hours of operation at elevated firing temperature, fast starts, and emergency shutdown events as more severe than normal operating hours and starts. Industry practice is to establish maintenance factors that can be used to account for effects that shorten the intervals between inspections. Table 57.3 gives typical values. The hours to inspection or starts to inspection in Fig. 57.37 would be divided by the factor in Table 57.3.

The values shown here are similar to those used by manufacturers but are only approximate, since recommendations are modified and updated periodically. Also, the number, extent, and types of inspections vary across the industry. To compare the frequency of inspection recommended for competing gas turbines, the evaluator must forecast the number of starts and hours expected during the evaluation period and, using the manufacturers' recommendation and other experience, determine the inspection frequency for the particular application.

Reliability and *availability* have specific definitions where applied to power generation equipment.[22]

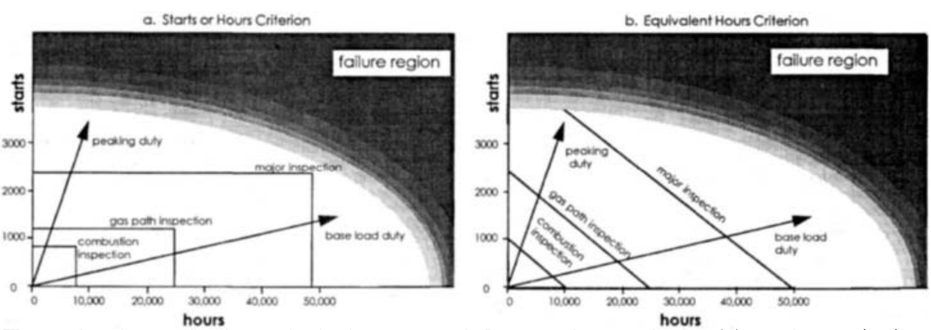

Fig. 57.37 Inspection interval criteria compared. Starts-or-hours criterion (*a*) requires major inspection after 48,000 hours or 2400 starts. Equivalent-hours criterion (*b*) reflects each start being equivalent to 10 hours, with major inspection required after 50,000 equivalent hours.

Table 57.3 Maintenance Factors—Industrial Gas Turbine Nozzles and Buckets

Factors Increasing Maintenance Frequency		
		Hours factors
Fuel	Natural gas	1
	Distillate	1–1.5
	Residual	3–4
Peak load	Elevated firing temp.	5–10
Diluent injection	Water or steam	1–2
		Starts Factors
Trip from full-load		6–10+
Fast load		2–4
Emergency start		10–20

$$\text{Reliability} = 1 - (\text{FOH}/\text{PH}) \text{ (expressed as a percentage)}$$
$$\text{FOH} = \text{total forced outage hours}$$
$$\text{PH} = \text{period hours (8760 hrs/year)}$$

Reliability is used to reflect the percentage of time for which the equipment is not on forced outage. It is the probability of not being forced out of service when the unit is needed, and includes forced outage hours while in service, while on reserve shutdown, and while attempting to start.

$$\text{Availability} = 1 - (\text{UH}/\text{PH}) \text{ (expressed as a percentage)}$$
$$\text{UH} = \text{total unavailable hours (forced outage, failure to start, unscheduled maintenance hours, maintenance hours)}$$
$$\text{PH} = \text{period hours}$$

Availability reflects the probability of being available, independent of whether or not the unit is needed, and includes all unavailable hours normalized by period hours.

There are some minor differences in the definitions across the industry, reflecting the way different databases treat particular types of events, but the equations given above reasonably represent industry norms.

Availability and reliability figures used in power-generation industry literature reflect the performance not only of the turbomachinery, but of the generator, control system, and accessories. Historically, less than half of the unavailability and forced outage hours are due to the turbomachinery. Accessories, controls and driven devices account for the remainder.

Availability is affected by the frequency and duration of inspections as well as the duration of forced outages. Improvements in analytical capability, understanding of material behavior, operating practices, and design sophistication have led to improvements in both availability and reliability over the past decades. The availability of industrial gas turbines has grown from 80% in the early 1970s to better than 95% in the mid-1990s.

57.4.2 Selection of Engine and System

In the transportation field, gas turbines are the engine of choice in large and increasingly in small aircraft where the number of hours per year flown is sufficiently high that the higher speed and lower fuel and service costs attributable to gas turbines justify the higher first-cost. Private automobiles, which operate nominally 400 hours per year and where operating characteristics favor the Otto and Diesel cycles, are not likely to be candidates for gas turbine power. Exhaust-driven superchargers are a more acceptable application of turbomachinery technology to this market. Long-haul trucks, buses, and military applications may be served by gas turbines if the economics that made them commonplace on aircraft can be applied.

Gas turbine technology finds application in mechanical drive and electric power generation. In mechanical drive application, the turbine rotor shaft typically drives a pump, compressor, or drive system. Mechanical drive applications usually employ "two-shaft" gas turbines, in which the output shaft is controllable in speed to match the varying load/speed characteristic of the application. In electric power generation, the shaft drives an electrical generator at a constant synchronous speed. Mechanical drive applications typically find application for gas turbines in the 5–25 MW range. Over the last five years, this market has been approximately 1000 MW per year. Power-generation applications are typically in the larger size ranges, from 25–250 MW and have averaged over 20,000 MW per year.

Gas turbine technology competes with other technologies in both power generation and mechanical drive applications. In both applications, the process for selecting which thermodynamic cycle or engine type to apply is similar. Table 57.4 summarizes the four key choices in electric power generation.

Steam turbine technology utilizes an externally fired boiler to produce steam and drive a Rankine cycle. This technology has been used in power generation for nearly a century. Because the boiler is fired external to the working fluid, steam, any type of fuel may be used, including coal, distillate oil, residual oil, natural gas, refuse, and bio-mass. The thermal efficiencies are typically in the 30% range for small (20–40 MW) industrial and refuse plants to 35% for large (400 MW) power-generation units, to 40% for large, ultra-efficient, ultra-supercritical plants. These plants are largely assembled and erected at the plant site and have relatively high investment cost per kW of output. Local labor costs and labor productivity influence the plant cost. Thus, the investment cost can vary considerably throughout the world.

Diesel technology uses the Diesel cycle in a reciprocating engine. The diesels for power generation are typically medium speed (800 rpm). The diesel engine has efficiencies from 40–45% on distillate oil. If natural gas is the fuel, the ordinary Diesel cycle is not applicable, but a spark ignition system based on the Otto cycle's can be employed. The Otto cycle leads to three percentage points lower efficiency than the diesel. Diesel engines are available in smaller unit sizes than the gas turbines that account for most of the power generated for mechanical drive and power generation (1–10 MW). The investment cost of medium-speed diesels is relatively high per kW of output when compared with large gas turbines, but is lower than that of gas turbines in this size range. Maintenance cost of diesels per kW of output is typically higher than gas turbine technology.

The life-cycle cost of power-generation technology projects is the key factor in their application. The life-cycle cost includes the investment cost charges and the present worth of annual fuel and operating expenses. The investment cost charges are the present worth costs of financing, depreciation, and taxes. The fuel and operating expenses include fuel-consumption cost, maintenance expenses, operational material costs (lubricants, additives, etc.), and plant-operation and maintenance labor costs. For a combined-cycle technology plant, investment charges can contribute 20%, fuel 70%, and operation and maintenance costs 10%. The magnitude and composition of costs is very technology and geographic location dependent.

One way to evaluate the application of technology is to utilize a screening curve, as shown in Fig. 57.38. This chart represents one particular power output and set of economic conditions, and is used here to illustrate a principle, not to make a general statement on the relative merits of various power generation means. The screening curve plots the total $/kW/year annual life-cycle cost of a powerplant versus the number of hours per year of operation. At zero hours of operation (typical of a standby plant used only in the event of loss of power from other sources), the only life-cycle cost component is from investment financing charges and any operating expense associated with providing manpower to be at the site. As the operating hours increase toward 8000 hours per year, the costs of fuel, maintenance, labor, and direct materials are added into the annual life cycle cost.

Table 57.4 Fossil Fuel Technologies for Mechanical Drive and Electric Power Generation

Technology	Power Cycle	Performance Level	Primary Advantages	Primary Disadvantages
Steam Turbine	Rankine cycle	30–40%	Custom size Solid fuels Dirty fuels	Low efficiency Rel. high $/kW Slow load change
Gas Turbine	Brayton cycle	30–40%	Packaged power plant Low $/kW Med. fast starts Fast load delta	Clean fuels Ambient dependence
Combined Cycle	Brayton Topping/ Rankine Bottoming	45–60%	Highest efficiency Med. $/kW Limited fast load delta	Clean fuels Ambient dependence Med. start times
Diesel	Diesel cycle	40–50%	Rel. high efficiency Packaged power plant Fast start Fast construction	High maint. Small size (5 MW)

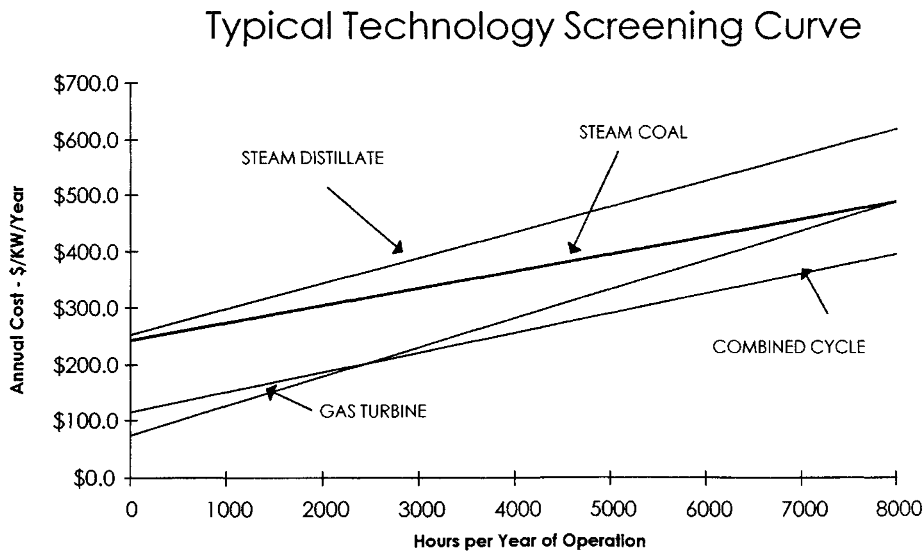

Fig. 57.38 Hypothetical screening curve for selecting power-generation technology from among various thermodynamic cycles and fuel alternatives. This curve would indicate that the most economic choice for few operating hours per year is the simple-cycle gas turbine, and the combined cycle for base load applications.

If the application has only a few hours per year of operation (less than 2000) simple-cycle gas turbine technology has typically the lowest annual life-cycle cost and is therefore chosen. Simple-cycle gas turbine has the lowest annual life-cycle cost in this region in view of its low investment cost. If the application has more than 2000 hours per year of operation, then combined-cycle technology provides the lowest annual life-cycle cost and is selected for application.

Other technology choices are the higher investment cost alternatives of coal-fired steam turbine technology and IGCC technology. In the example of Fig. 57.38, these technologies do not have the lowest annual life-cycle cost in any region and consequently would not find application. However, the screening curve of Fig. 57.38 is based on a specific set of fuel prices and investment costs. In other regions of the world, coal prices may be lower or natural gas prices may be higher. In this case, the coal technologies may have the lowest annual life-cycle cost, in the 6000–8000-hour range. These technologies would then be selected for application.

In summary, there is a range of fuel prices and investment costs for power generation technology. This range influences the applicability of the power-generation technology. In some countries with large, low-priced coal resources, coal steam turbine technology is the most widely used. Where natural gas is available and modestly priced, gas turbine and combined-cycle technology is widely selected.

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