CHAPTER 57

GAS TURBINES

Harold Miller
GE Power Systems
Schenectady, New York

57.1 INTRODUCTION

57.1.1 Basic Operating Principles

Gas turbines are heat engines based on the Brayton thermodynamic cycle. This cycle is one of the four that account for most of the heat engines in use. Other cycles are the Otto, Diesel and Rankine. The Otto and Diesel cycles are cyclic in regard to energy content. Steady-flow, continuous energy transfer cycles are the Brayton (gas turbine) and Rankine (steam turbine) cycles. The Rankine cycle involves condensing and boiling of the working fluid, steam, and utilizes a boiler to transfer heat to the working fluid. The working fluid in the other cycles is generally air, or air plus combustion products. The Otto, Diesel and Brayton cycles are usually internal combustion cycles wherein the fuel is burned in the working fluid. In summary, the Brayton cycle is differentiated from the Otto and Diesel cycle in that it is continuous, and from the Rankine in that it relies on internal combustion, and does not involve a phase change in the working fluid.

In all cycles, the working fluid experiences induction, compression, heating, expansion, and exhaust. In a non-steady cycle, these processes are performed in sequence in the same closed space,

This chapter was written as an update to chapter 72 of the Handbook's previous edition. Much of the structure and significant portions of the text of the previous edition's chapter is retained. The new edition has increased emphasis on the most significant current and future projected gas turbine configurations and applications. Thermodynamic cycle variations are presented here in a consistent format, and the description of current cycles replaces the discussions of some interesting and historical, but less significant, cycles described in the earlier edition. In addition, there is a new discussion of economic and regulatory trends, of supporting technologies, and their interconnection with gas turbine development. The author of the previous version had captured the history of the gas turbine's development, and this history is repeated and supplemented here.

Mechanical Engineers' Handbook, 2nd ed., Edited by Myer Kutz.
ISBN 0-471-13007-9

formed by a piston and cylinder that operate on the working fluid one mass at a time. In contrast, the working fluid flows through a steam turbine power plant or gas turbine engine, without interruption, passing continuously from one single-purpose device to the next.

Gas turbines are used to power aircraft and land vehicles, to drive generators (alternators) to produce electric power, and to drive other devices, such as pumps and compressors. Gas turbines in production range in output from below 50 kW to over 200 MW. Design philosophies and engine configurations vary significantly across the industry. Aircraft engines are optimized for high power-to-weight ratios, while heavy-duty, industrial, and utility gas turbines are heavier, being designed for low cost and long life in severe environments.

The arrangement of a simple gas turbine engine is shown in Fig. 57.1*a*. The rotating compressor acts to raise the pressure of the working fluid and force it into the combustor. The turbine rotation is caused by the work produced by the fluid while expanding from the high pressure at the combustor discharge to ambient air pressure at the turbine exhaust. The resulting mechanical work drives the mechanically connected compressor and output load device.

The nomenclature of the gas turbine is not standardized. In this chapter, the term *blading* refers to all rotating and stationary airfoils in the gas path. Turbine (expander) section rotating blades are *buckets,* a term derived from steam turbine practice. Turbine section stationary blades are *nozzles.* The combustion components in contact with the working fluid are called *combustors;* major combustor components are fuel nozzles and combustion liners. Some combustors (Can-annular and silo-types) have transition pieces that conduct hot gas from the combustion liners to the first-stage nozzles. A stage of the compressor consists of a row of rotor blades, all at one axial position in the gas turbine, and the stationary blade row downstream of it. A turbine stage consists of a set of nozzles occupying one axial location and the set of buckets immediately downstream. Rotating blading is attached either to a monolithic rotor structure or to individual discs or wheels designed to support the blading against centrifugal force and the aerodynamic loads of the working fluid. The terms *discs* and *wheels* are used interchangeably.

Gas turbine performance is established by three basic parameters: mass flow, pressure ratio, and firing temperature. Compressor, combustor, and turbine efficiency have significant, but secondary, effects on performance, as do inlet and exhaust systems, turbine gas path and rotor cooling, and heat loss through turbine and combustor casings.

In gas turbine catalogues and other descriptive literature, mass flow is usually quoted as compressor inlet flow, although turbine exit flow is sometimes quoted. Output is proportional to mass flow.

Pressure ratio is quoted as the compressor pressure ratio. Aircraft engine practice is to define the ratio as the total pressure at the exit of the compressor blading divided by the total pressure at the inlet of the compressor blading. Industrial/utility turbine manufacturers generally refer to the static pressure in the plenum downstream of the compressor discharge diffuser (upstream of the combustor) divided by the total pressure downstream of the inlet filter and upstream of the inlet of the gas turbine. Similarly, there are various possibilities for defining turbine pressure ratio. All definitions yield values within 1 or 2% of one another. Pressure ratio is the primary determinant of simple cycle gas turbine efficiency. High pressure results in high simple cycle efficiency.

Firing temperature is defined differently by each manufacturer, and the differences are significant. Heavy-duty gas turbine manufacturers use three definitions. There is an ISO definition of firing temperature, which is a calculated temperature. The compressor discharge temperature is increased by a calculated enthalpy rise based on the compressor inlet air flow and the fuel flow. This definition is valuable in that it can be used to compare gas turbines or monitor changes in performance through calculations made on the basis of field measurements. To determine ISO firing temperature, one does not require knowledge of the secondary flows within the gas turbine. A widely used definition of

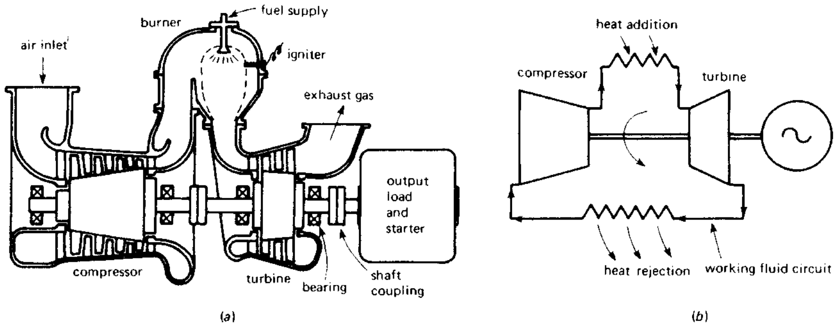


Fig. 57.1 Simple engine type: (*a*) open cycle; (*b*) closed cycle (diagrammatic).[1]

firing temperature is the average total temperature in the exit plane of the first stage nozzle. This definition is used by General Electric for its industrial engines. Westinghouse refers to "turbine inlet temperature," the temperature of the gas entering the first stage nozzle. Turbine inlet temperature is approximately 100°C above nozzle exit firing temperature, which is in turn approximately 100°C above ISO firing temperature. Since firing temperature is commonly used to compare the technology level of competing gas turbines, it is important to compare on one definition of this parameter.

Aircraft engines and aircraft-derivative industrial gas turbines have other definitions. One nomenclature establishes numerical stations—here, station 3.9 is combustor exit and station 4.0 is first-stage nozzle exit. Thus, $T_{3.9}$ is very close to "turbine inlet temperature" and $T_{4.0}$ is approximately equal to GE's "firing temperature." There are some subtle differences relating to the treatment of the leakage flows near the first-stage nozzle. This nomenclature is based on SAE ARP 755A, a recommended practice for turbine engine notation.

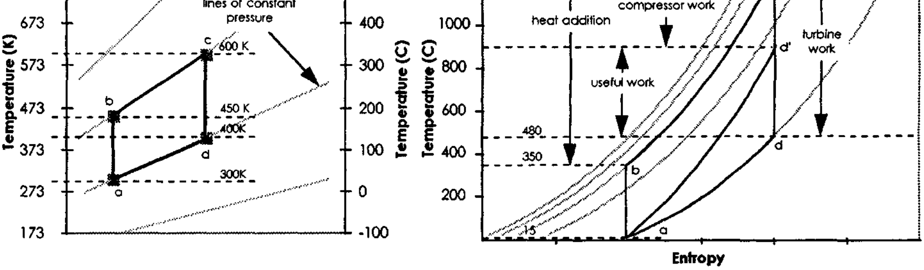
Firing temperature is a primary determiner of power density (specific work) and combined cycle (Brayton–Rankine) efficiency. High firing temperature increases the power produced by a gas turbine of a given physical size and mass flow. The pursuit of higher firing temperatures by all manufacturers of large, heavy-duty gas turbines used for electrical power generation is driven by the economics of high combined cycle efficiency.

Pressures and temperatures used in the following descriptions of gas turbine performance will be total pressures and temperatures. Absolute, stagnation, or total values are those measured by instruments that face into the approaching flow to give an indication of the energy in the fluid at any point. The work done in compression or expansion is proportional to the change of stagnation temperature in the working fluid, in the form of heating during a compression process or cooling during an expansion process. The temperature ratio, between the temperatures before and after the process, is related to the pressure ratio across the process by the expression $T_b/T_a = (P_b/P_a)^{(\gamma-1)/\gamma}$, where γ is the ratio of working fluid specific heats at constant pressure and volume. The temperature and pressure are stagnation values. It is the interaction between the temperature change and ratio, at different starting temperature levels, that permits the engine to generate a useful work output.

This relationship between temperature and pressure can be demonstrated by a simple numerical example using the Kelvin scale for temperature. For a starting temperature of 300°K (27°C), a temperature ratio of 1.5 yields a final temperature of 450°K and a change of 150°C. Starting instead at 400°K, the same ratio would yield a change of 200°C and a final temperature of 600°K. The equivalent pressure ratio would ideally be 4.13, as calculated from solving the preceding equation for P_b/P_a; $P_b/P_a = T_b/T_a^{\gamma/\gamma-1} = 1.5^{1.4/0.4} = 4.13$. These numbers show that, working over the same temperature ratio, the temperature change and, therefore, the work involved in the process vary in proportion to the starting temperature level.[2]

This conclusion can be depicted graphically. If the temperature changes are drawn as vertical lines *a–b* and *c–d*, and are separated horizontally to avoid overlap, the resultant is Fig. 57.2*a*. Assuming the starting and finishing pressures to be the same for the two processes, the thin lines through *a–d* and *b–c* depict two of a family of lines of constant pressure, which diverge as shown. In this ideal case, expansion processes could be represented by the same diagram, simply by proceeding down the lines *b–a* and *c–d*. Alternatively, if *a–b* is taken as a compression process, *b–c* as heat addition, *c–d* as an expansion process, and *d–a* as a heat rejection process, then the figure *a–b–c–d–a* represents the ideal cycle to which the working fluid of the engine is subjected.

Over the small temperature range of this example, the assumption of constant gas properties is justified. In practice, the 327°C (600°K) level at point *d* is too low a temperature from which to start



(*a*) (*b*)

Fig. 57.2 Temperature changes and temperature–entropy diagram for ideal simple gas turbine cycles.

the expansion. Figure 57.2*b* is more realistic. Here, the lines of constant pressure have been constructed for ideal gas–air properties that are dependent upon temperature. Expansion begins from a temperature of 1250°C. With a pressure ratio of 16 : 1, the end point of the expansion is approximately 480°C. Now *a–b* represents the work input required by the compressor. Of the expansion work capacity *c–d*, only the fraction *c–d'* is required to drive the compressor. An optical illusion makes it appear otherwise, but line *a–d'* is displaced vertically from line *b–c* by the same distance everywhere. The remaining 435°C, line *d'–d*, is energy that can be used to perform useful external work, by further expansion through the turbine or by blowing through a nozzle to provide jet thrust.

Now consider line *b–c*. The length of its vertical projection is proportional to the heat added. The ability of the engine to generate a useful output arises from its use of the energy in the input fuel flow, but not all of the fuel energy can be recovered usefully. In this example, the heat input proportional to 1250–350 = 900°C compares with the excess output proportional to 435°C (line *d'–d*) to represent an efficiency of (435/900), or 48%. If more fuel could be used, raising the maximum temperature level at the same pressure, then more useful work could be obtained at nearly the same efficiency.

The line *d–a* represents heat rejection. This could involve passing the exhaust gas through a cooler before returning it to the compressor, and this would be a closed cycle. But, almost universally, *d–a* reflects discharge to the ambient conditions and intake of ambient air (Fig. 57.1*b*). Figure 57.1*a* shows an open-cycle engine, which takes air from the atmosphere and exhausts back to the atmosphere. In this case, line *d–a* still represents heat rejection, but the path from *d* to *a* involves the whole atmosphere and very little of the gas finds its way immediately from *e* to *a*. It is fundamental to this cycle that the remaining 465°C, the vertical projection of line *d–a*, is wasted heat because point *d* is at atmospheric pressure. The gas is therefore unable to expand further and so can do no more work.

Designers of simple cycle gas turbines—including aircraft engines—have pursued a course of reducing exhaust temperature through increasing cycle pressure ratio, which improves the overall efficiency. Figure 57.3 is identical to Fig. 57.2*b* except for the pressure ratio, which has been increased from 16 : 1 to 24 : 1. The efficiency is calculated in the same manner. The total turbine work is proportional to the temperature difference across the turbine, 1250–410 = 840°C. The compressor work, proportional to 430–15 = 415°C, is subtracted from the turbine temperature drop 840–415 = 425°C. The heat added to the cycle is proportional to 1250–430 = 820°C. The ratio of the net work to the heat added is 425/820 = 52%. The approximately 8% improvement in efficiency is accompanied by a 70°C drop in exhaust temperature. When no use is made of the exhaust heat, the 8% efficiency may justify the mechanical complexity associated with higher pressure ratios. Where there is value to the exhaust heat, as there is in combined Brayton–Rankine cycle power plants, the lower pressure ratio may be superior. Manufacturers forecast their customer requirements and understand

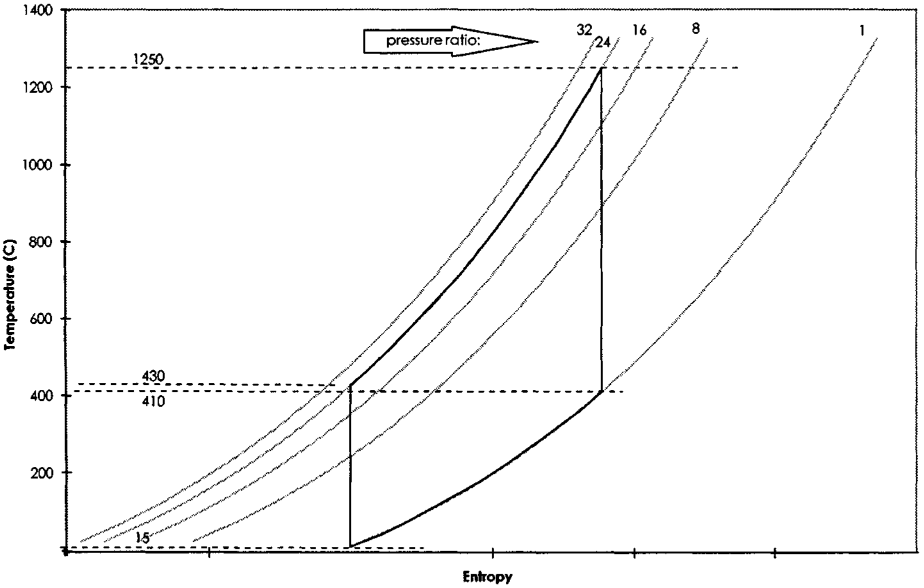


Fig. 57.3 Simple cycle gas turbine temperature–entropy diagram for high (24 : 1) pressure ratio and 1250°C firing temperature.

the costs associated with cycle changes and endeavor to produce gas turbines featuring the most economical thermodynamic designs.

The efficiency levels calculated in the preceding example are very high because many factors have been ignored for the sake of simplicity. Inefficiency of the compressor increases the compressor work demand, while turbine inefficiency reduces turbine work output, thereby reducing the useful work output and efficiency. The effect of inefficiency is that, for a given temperature change, the compressor generates less than the ideal pressure level while the turbine expands to a higher temperature for the same pressure ratio. There are also pressure losses in the heat addition and heat rejection processes. There may be variations in the fluid mass flow rate and its specific heat (energy input divided by consequent temperature rise) around the cycle. These factors can easily combine to reduce the overall efficiency.

57.1.2 A Brief History of Gas Turbine Development and Use

The use of a turbine driven by the rising flue gases above a fire dates back to Hero of Alexandria in 150 BC. It was not until AD 1791 that John Barber patented the forerunner of the gas turbine, proposing the use of a reciprocating compressor, a combustion system, and an impulse turbine. Even then, he foresaw the need to cool the turbine blades, for which he proposed water injection.

The year 1808 saw the introduction of the first explosion type of gas turbine, which in later forms used valves at entry and exit from the combustion chamber to provide intermittent combustion in a closed space. The pressure thus generated blew the gas through a nozzle to drive an impulse turbine. These operated successfully but inefficiently for Karavodine and Holzwarth from 1906 onward, and the type died out after a Brown, Boveri model was designed in 1939.[3]

Developments of the continuous flow machine suffered from lack of knowledge, as different configurations were tried. Stolze in 1872 designed an engine with a seven-stage axial flow compressor, heat addition through a heat exchanger by external combustion, and a 10-stage reaction turbine. It was tested from 1900 to 1904 but did not work because of its very inefficient compressor. Parsons was equally unsuccessful in 1884, when he tried to run a reaction turbine in reverse as a compressor. These failures resulted from the lack of understanding of aerodynamics prior to the advent of aircraft. As a comparison, in typical modern practice, a single-stage turbine drives about six or seven stages of axial compressor with the same mass flow.

The first successful dynamic compressor was Rateau's centrifugal type in 1905. Three assemblies of these, with a total of 25 impellers in series giving an overall pressure ratio of 4, were made by Brown, Boveri and used in the first working gas turbine engine, built by Armengaud and Lemale in the same year. The exhaust gas heated a boiler behind the turbine to generate low-pressure steam, which was directed through turbines to cool the blades and augment the power. Low component efficiencies and flame temperature (828°K) resulted in low work output and an overall efficiency of 3%. By 1939, the use of industrial gas turbines had become well established: experience with the Velox boiler led Brown, Boveri into diverging applications; a Hungarian engine (Jendrassik) with axial flow compressor and turbine used regeneration to achieve an efficiency of 0.21; and the Sun Oil Co. in the United States was using a gas turbine engine to improve a chemical process.[2]

The history of gas turbine engines for aircraft propulsion dates from 1930, when Frank Whittle saw that its exhaust gas conditions ideally matched the requirements for jet propulsion and took out a patent.[4] His first model was built by British Thomson-Houston and ran as the Power Jets Type U in 1937, with a double-sided centrifugal compressor, a long combustion chamber that was curled round the outside of the turbine and an exhaust nozzle just behind the turbine. Problems of low compressor and turbine efficiency were matched by hardware problems and the struggle to control the combustion in a very small space. Reverse-flow, can-annular combustors were introduced in 1938, the aim still being to keep the compressor and turbine as close together as possible to avoid shaft whirl problems (Fig. 57.4). Whittle's first flying engine was the W1, with 850 lb thrust, in 1941. It was made by Rover, whose gas turbine establishment was taken over by Rolls-Royce in 1943. A General Electric version of the W1 flew in 1941. A parallel effort at General Electric led to the development of a successful axial-flow compressor. This was incorporated in the first turboprop engine, the TG100, later designated the T31. This engine, first tested in May of 1943, produced 1200 horsepower from an engine weighing under 400 kg. Flight testing followed in 1949. An axial-compressor turbojet version was also constructed, designated the J35. It flew in 1946. The compressor of this engine evolved to the compressor of the GE MS3002 industrial engine, which was introduced in 1950 and is still in production.[5]

A Heinkel experimental engine flew in Germany in 1939. Several jet engines were operational by the end of the Second World War, but the first commercial engine did not enter service until 1953, the Rolls-Royce Dart turboprop in the Viscount, followed by the turbojet de Havilland Ghost in the Comet of 1954. The subsequent growth of the use of jet engines has been visible to most of the world, and has forced the growth of design and manufacturing technology.[6] By 1970, a range of standard configurations for different tasks had become established, and some aircraft engines were established in industrial applications and in ships.

Gas turbines entered the surface transportation fields also during their early stages of development. The first railway locomotive application was in Switzerland in 1941, with a 2200-hp Brown, Boveri

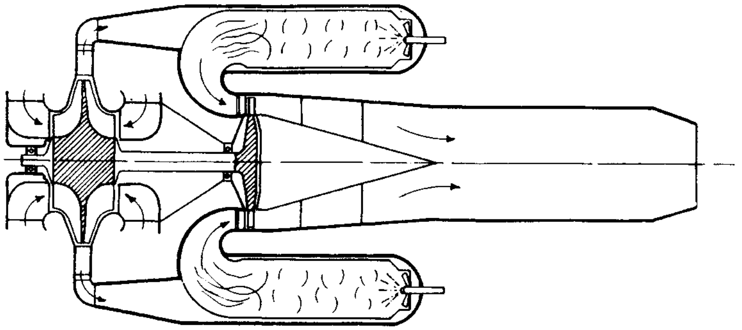


Fig. 57.4 Simplified arrangement of an early Whittle jet engine, with double-sided compressor and reverse-flow combustion chambers. (Redrawn from Ref. 4 by permission of the Council of the Institution of Mechanical Engineers.)

engine driving an electric generator and electric motors driving the wheels. The engine efficiency approached 19%, using regeneration. The next decade saw several similar applications of gas turbines by some 43 different manufacturers. A successful application of gas turbines to transportation was the 4500 draw-bar horsepower engine, based on the J35 compressor. Twenty-five locomotives so equipped were delivered to the Union Pacific railroad between 1952 and 1954. The most powerful locomotive gas turbine was the 8500-hp unit offered by General Electric to the Union Pacific railroad for long-distance freight service.[7] This became the basis of the MS5001 gas turbine, which is the most common heavy-duty gas turbine in use today. Railroad applications continue today, but relying on a significantly different system. Japan Railway uses large stationary gas turbines to generate power transmitted by overhead lines to their locomotives.

Automobile and road vehicle use started with a Rover car of 1950, followed by Chrysler and other companies, but commercial use has been limited to trucks, particularly by Ford. Automotive gas turbine development has been largely independent of other types, and has forced the pace of development of regenerators.

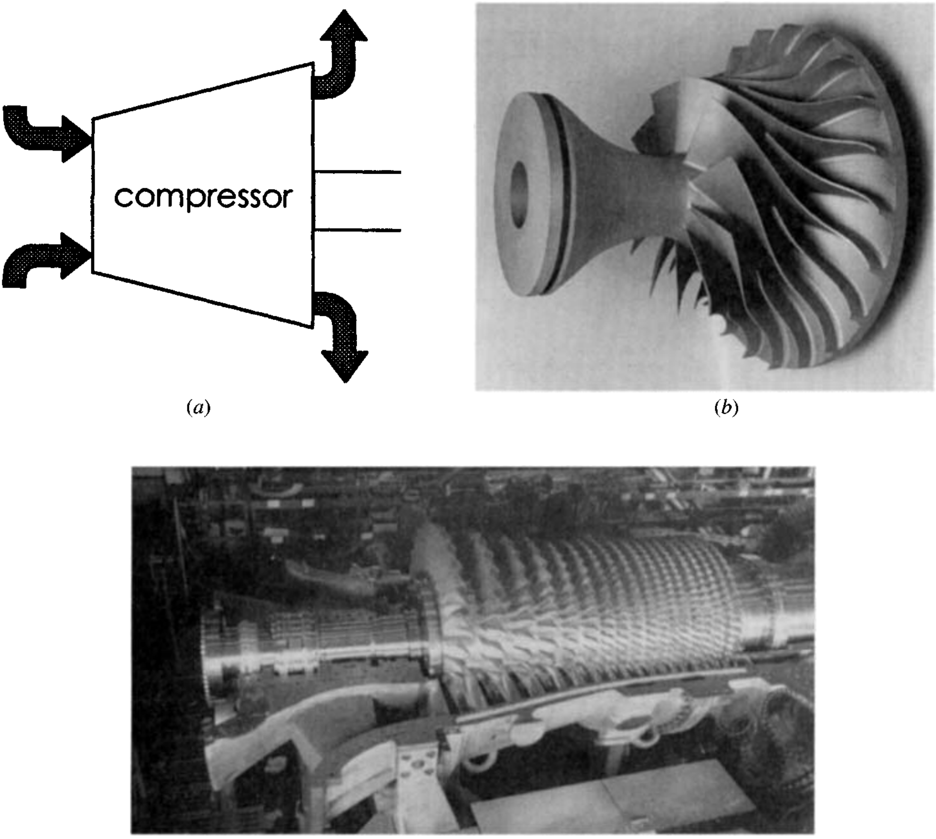
57.1.3 Component Characteristics and Capabilities

Compressors

Compressors used in gas turbines are of the dynamic type, wherein air is continuously ingested and raised to the required pressure level—usually, but not necessarily, between 8 and 40 atmospheres. Larger gas turbines use axial types; smaller ones use radial outflow centrifugal compressors. Some smaller gas turbines use both—an axial flow compressor upstream of a centrifugal stage.

Axial compressors feature an annular flowpath, larger in cross-section area at the inlet than at the discharge. Multiple stages of blades alternately accelerate the flow of air and allow it to expand, recovering the dynamic component and increasing pressure. Both rotating and stationary stages consist of cascades of airfoils, as can be seen in Fig. 57.5. Physical characteristics of the compressor determine many aspects of the gas turbine's performance. Inlet annulus area establishes the mass flow of the gas turbine. Rotor speed and mean blade diameter are interrelated, since optimum blade velocities exist. A wide range of pressure ratios can be provided, but today's machines feature compressions from 8:1 to as high as 40:1. The higher pressure ratios are achieved using two compressors operating in series at different rotational speeds. The number of stages required is partially dependent on the pressure ratio required, but also on the sophistication of the blade aerodynamic design that is applied. Generally, the length of the compressor is a function of pressure ratio, regardless of the number of stages. Older designs have stage pressure ratios of 1.15:1. Low-aspect ratio blading designed with three-dimensional analytical techniques have stage pressure ratios of 1.3:1. There is a trend toward fewer stages of blades of more complicated configuration. Modern manufacturing techniques make more complicated forms more practical to produce, and minimizing parts count usually reduces cost.

Centrifugal compressors are usually chosen for machines of below 2 or 3 MW in output, where their inherent simplicity and ruggedness can largely offset their lower compression efficiency. Such compressors feature a monolithic rotor with a shaped passage leading from the inlet circle or annulus to a volute at the outer radius, where the compressed air is collected and directed to the combustor. The stator contains no blades or passages and simply provides a boundary to the flow path, three sides of which are machined or cast into the rotor. Two or more rotors can be used in series to achieve the desired pressure ratio within the mechanical factors that limit rotor diameter at a given rotational speed.[8]



(a) (b) (c)

Fig. 57.5 Diagram, and photos of centrifugal compressor rotor (courtesy of Nuovo Pignone Company) and axial compressor during assembly (courtesy of General Electric Company).

Two efficiency definitions are used to describe compressor performance. Polytropic efficiency characterizes the aerodynamic efficiency of low-pressure-ratio individual stages of the compressor. Isentropic, or adiabatic, efficiency describes the efficiency of the first step of the thermodynamic process shown in Fig. 57.6 (the path from *a* to *b*). The isentropic efficiency can be calculated from the temperatures shown for the compression process on this figure. The isentropic temperature rise is for the line *a–b*: 335°C. The actual rise is shown by line *a–b*′, and this rise is 372°C. The compressor efficiency η_c is the ratio 335/372 = 90%.

Successful compressor designs achieve high component efficiency while avoiding compressor surge or stall—the same phenomenon experienced when airplane wings are forced to operate at too high an angle of attack at too low a velocity. Furthermore, blade and rotor structures must be designed to avoid vibration problems. These problems occur when natural frequencies of components and assemblies are coincident with mechanical and aerodynamic stimuli, such as those encountered as blades pass through wakes of upstream blades. The stall phenomenon may occur locally in the compressor or even generally, whereupon normal flow through the machine is disrupted. A compressor must have good stall characteristics in order to operate at all ambient pressures and temperatures and to operate through the start, acceleration, load, load-change, unload, and shutdown phases of turbine operation. Compressors are designed with features and mechanisms for avoiding stall. These include air bleed at various points, variable-angle stator (as opposed to rotor) blades, and multiple spools.

Recent developments in the field of computational fluid dynamics (CFD) provide analytical tools that allow designers to substantially reduce aerodynamic losses due to shock waves in the supersonic flow regions. Using this technique, stages that have high tip Mach numbers can attain efficiencies comparable to those of completely subsonic designs. With these tools, compressors can be designed with higher tip diameters, hence higher flows. The same tools permit the design of low

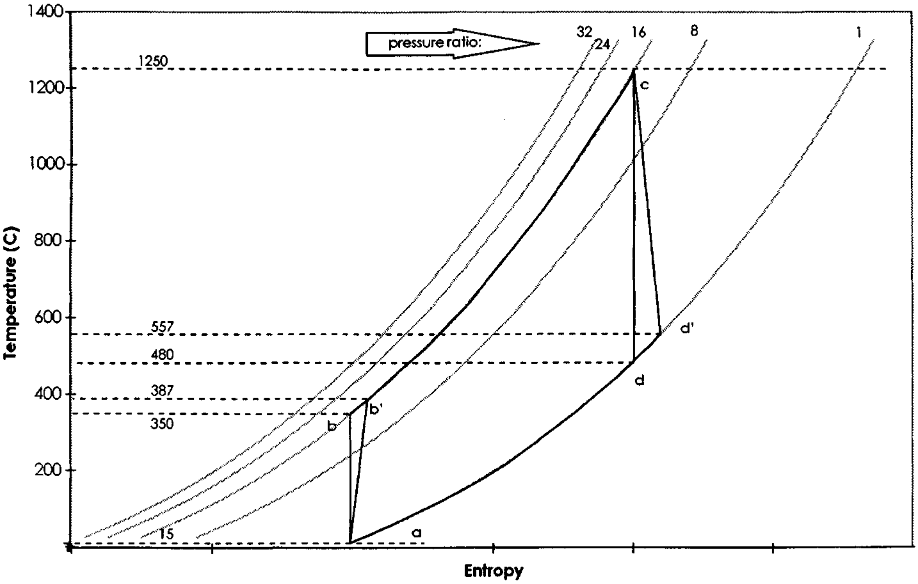


Fig. 57.6 Temperature–entropy diagram showing the effect of compressor and turbine efficiency.

aspect ratio, high stage pressure ratio blades for reducing the number of blade rows. Both capabilities contribute to lower cost gas turbine designs with no sacrifice in performance.

Gas Turbine Combustion System

The gas turbine combustor is a device for mixing large quantities of fuel and air and burning the resulting mixture. A flame burns best when there is just enough fuel to react with the available oxygen. This is called a *stoichiometric condition,* and combustion here produces the fastest chemical reaction and the highest flame temperatures, compared with excess air (fuel-lean) and excess fuel (fuel-rich) conditions, where reaction rates and temperatures are lower. The term *equivalence ratio* is used to describe the ratio of fuel to air relative to the stoichiometric condition. An equivalence ratio of 1.0 corresponds to the stoichiometric condition. Under fuel-lean conditions, the ratio is less than 1, and under fuel-rich conditions it is greater than 1. The European practice is to use the reciprocal, which is the Lambda value (λ).

In a gas turbine, since air is extracted from the compressor for cooling the combustor, buckets, nozzles, and other components and to dilute the flame—as well as support combustion—the overall equivalence ratio is far less than the value in the flame zone, ranging from 0.4–0.5 ($\lambda = 2.5$ to 2).[9]

Historically, the design of combustors required providing for the near-stoichiometric mixture of fuel and air locally. The combustion in this near-stoichiometric situation results in a diffusion flame of high temperature. Near-stoichiometric conditions produce a stable combustion front without requiring designers to provide significant flame-stabilizing features. Since the temperatures generated by the burning of a stoichiometric mixture greatly exceed those at which materials are structurally sound, combustors have to be cooled, and also the gas heated by the diffusion flame must be cooled by dilution before it becomes the working fluid of the turbine.

Gas turbine operation involves a startup cycle that features ignition of fuel at 20% of rated operating speed where air flow is proportionally lower. Loading, unloading, and part-load operation, however, require low fuel flow at full compressor speed, which means full air flow. Thermodynamic cycles are such that the lowest fuel flow per unit mass flow of air through the turbine exists at full speed and no-load. The fuel flow here is about 1/6 of the full-load fuel flow. Hence, the combustion system must be designed to operate over a 6:1 range of fuel flows with full rated air flow.

Manufacturers have differed on gas turbine combustor construction in significant ways. Three basic configurations have been used: annular, can-annular, and "silo" combustors. All have been used successfully in machines with firing temperatures up to 1100°C. Annular and can-annular combustors feature a combustion zone uniformly arranged about the centerline of the engine. All aircraft engines and most industrial gas turbine feature this type of design. A significant number of units equipped with silo combustors have been built as well. Here, one or two large combustion vessels are constructed on top of or beside the gas turbine. All manufacturers of large machines have now abandoned silo combustors in their state-of-the-art products. The can-annular, multiple combustion chamber assembly consists of an arrangement of cylindrical combustors, each with a fuel injection system, and a transition piece that provides a flow path for the hot gas from the combustor to the inlet of the turbine. Annular combustors have fuel nozzles at their upstream end and an inner and outer liner surface extending from the fuel nozzles to the entrance of the first-stage stationary blading. No transition piece is needed.

The current challenge to combustion designers is providing the cycle with a sufficiently high firing temperature while simultaneously limiting the production of oxides of nitrogen, NO_x, which refers to NO and NO_2. Very low levels of NO_x have been achieved in special low-emission combustors. NO_x is formed from the nitrogen and oxygen in the air when it is heated. The nitrogen and oxygen combine at a significant rate at temperatures above 1500°C, and the formation rate increases exponentially as temperature increases. Even with the high gas velocities in gas turbines, NO_x emissions will reach 200 parts per million by volume, dry (ppmvd), in gas turbines with conventional combustors and no NO_x abatement features. Emissions standards throughout the world vary, but many parts of the world require gas turbines to be equipped to control NO_x to below 25 parts per million by volume, dry (ppmvd) at base load.

Emissions

Combustion of common fuels necessarily results in the emission of water vapor and carbon dioxide. Combustion of near-stoichiometric mixtures results in very high temperatures. Oxides of nitrogen are formed as the oxygen and nitrogen in the air combine, and this happens at gas turbine combustion temperatures. Carbon monoxide forms when the combustion process is incomplete. Unburned hydrocarbons (UHC) are discharged as well when combustion is incomplete. Other pollutants are attributed to fuel; principal among these is sulfur. Gas turbines neither add nor remove sulfur; hence, what sulfur enters the gas turbine in the fuel exits as SO_2 in the exhaust.

Much of the gas turbine combustion research and development of the 1980s and 1990s focused on lowering NO_x production in mechanically reliable combustors while maintaining low CO and UHC emissions. Early methods of reducing NO_x emissions included removing it from the exhaust by selective catalytic reduction (SCR) and by diluent injection, that is, the injection of water or steam into the combustor. These methods continue to be employed. The lean-premix combustors now in general use are products of ongoing research.

Thermal NO_x is generally regarded as being generated by a chemical reaction sequence called the Zeldovich mechanism,[10] and the rate of NC_x formation is proportional to temperature, as shown in Fig. 57.7. In practical terms, a conventional gas turbine emits approximately 200 ppmvd when its combustors are not designed to control NO_x. This is because a significant portion of the combustion zone has stoichiometric or near-stoichiometric conditions, and temperatures are high. Additional oxygen, and of course nitrogen on the boundary of the flame, is heated to sufficiently high temperatures, and held at these temperatures for sufficient time, to produce NO_x.

Water- and steam-injected combustors achieve low flame temperatures by placing diluent in the neighborhood of the reacting fuel and air. Among low NO_x combustion systems operating today, water and steam injection is the most common means of flame temperature reduction. Several hundred large industrial turbines operating with steam or water injection have accumulated over 2-1/2 million hours of service. Water is not the only diluent used for NO_x control. In the case of integrated gasification combined cycle plants, nitrogen and CO_2 are available and can be introduced into the combustion region. The NO_x emissions measured at the Cool Water IGCC plant in the United States rival those of the cleanest natural gas plants in the world.[11]

Water or steam injection can achieve levels that satisfy all current standards, but water consumption is sometimes not acceptable to the operator because of cost, availability, or the impact on efficiency. Steam injection sufficient to reduce NO_x emissions to 25 ppmvd can increase fuel consumption in combined cycle power plants by over 3%. Water injection increases fuel use by over 4% for the same emissions level. In base-load power plants, fuel cost is so significant that it has caused the development of systems that do not require water.[12]

In all combustion processes, when a molecule of methane combines with two molecules of oxygen, a known and fixed amount of heat is released. When only these three molecules are present, a minimum amount of mass is present to absorb the energy not radiated and the maximum temperature is realized. Add to the neighborhood of the reaction the nitrogen as found in air (four times the volume of oxygen involved in the reaction) and the equilibrium temperature is lower. When even more air is added to the combustion region, more mass is available to absorb the energy and the resulting observable temperature is lower still. The same can be achieved through the use of excess fuel. Thus, moving away from the stoichiometric mixture means that observable flame temperature is lowered and the production of NO_x is also reduced. On a microscopic level, lean-burning low-NO_x combustors are designed to force the chemical reaction to take place in such a way that the energy released is in the neighborhood of as much mass not taking part in the reaction as possible. By transferring heat to neighboring material immediately, the time-at-temperature is reduced. On a larger

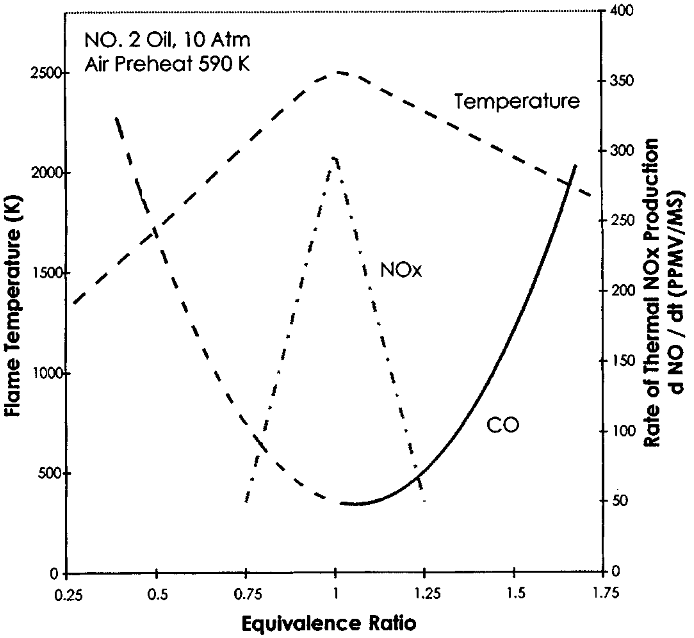


Fig. 57.7 NO_x formation rate driven by temperature (drawn from figure in Ref. 9; courtesy of General Electric Company).

scale, a high measurable temperature will never be reached in a well-mixed lean system and thus NO_x generation is minimized. Both rich-mixture and lean-mixture systems have led to low NO_x schemes. Although those featuring rich flames followed by lean burning zones are sometimes suggested for situations where there is nitrogen in the fuel, most of today's systems are based on lean burning.

Early lean premix dry low-NO_x combustors were operated in GE gas turbines at the Houston Light and Power Wharton Station in 1980 in the United States, in Mitsubishi units in Japan in 1983, and were introduced in Europe in 1986 by Siemens KWU. These combustors control the formation of NO_x by premixing fuel with air prior to its ignition while conventional combustors mix essentially at the instant of ignition. Dry low-NO_x combustors, as the name implies, achieve NO_x control without consuming water and without imposing efficiency penalties on combined-cycle plants.

Figures 57.8 and 57.9 show dry low-NO_x combustors developed for large gas turbines. In the GE system, several premixing chambers are located at the head end of the combustor. A fuel nozzle assembly is located in the center of each chamber. By the manipulation of valves external to the gas turbine, fuel can be directed to several combinations of chambers and to various parts of the fuel nozzles. This is to permit the initial ignition of the fuel and to maintain a relatively constant local fuel–air ratio at all load levels. There is one flame zone, immediately downstream of the premixing chambers. The Westinghouse combustor illustrated in Fig. 57.9 has three concentric premixing chambers. The two nearest the centerline of the combustor are designed to swirl the air passing through them in opposite directions and discharge into the primary combustion zone. The third, which has a longer passage, is directed to the secondary zone. Modulating fuel flow to the various mixing passages and combustion zones ensures low NO_x production over a wide range of operating temperatures. Both the combustors shown are designed for state-of-the-art, high-firing-temperature gas turbines.

Low-NO_x combustors feature multiple premixing features and a more complex control system than more conventional combustors, to achieve stable operation over the required range of operating conditions. The reason for this complexity is explained with the aid of Fig. 57.10. Conventional combustors operate with stability over a wide range of fuel–air mixtures—between the rich and lean flammability limits. A sufficiently wide range of fuel flows could be burned in a combustor with a fixed air flow, to match the range of load requirements from no-load to full-load. In a low-NO_x combustor, the fuel–air mixture feeding the flame must be regulated between the point of flame loss and the point where the NO_x limit is exceeded. When low gas turbine output is required, the air premixed with the fuel must be reduced to match the fuel flow corresponding to the low power output. The two combustors shown above hold nearly constant fuel–air ratios over the load range by

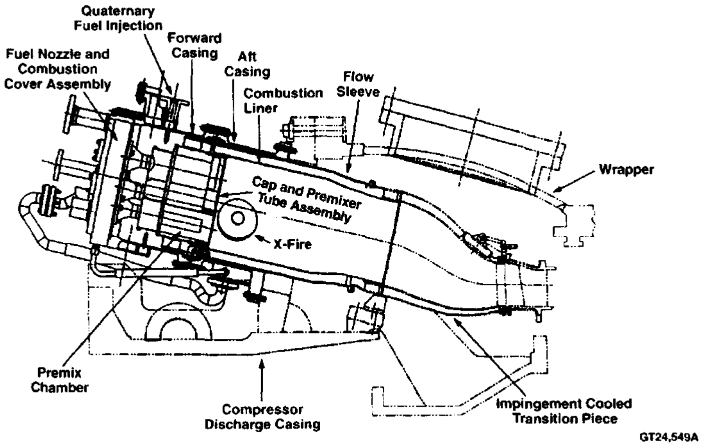


Fig. 57.8 GE DLN-2 lean-premix combustor designed for low emissions at high firing temperatures (courtesy of General Electric Company).

having multiple premixing chambers, each one flowing a constant fraction of the compressor discharge flow. By directing fuel to only some of these passages at low load, the design achieves both part load and optimum local fuel–air ratio. Three, four, or more sets of fuel passages are not uncommon, and premixed combustion is maintained to approximately 50% of the rated load of the machine.[9,13]

Catalytic combustion systems are under investigation for gas turbines. These systems have demonstrated stable combustion at lower fuel–air ratios than those using chamber, or nozzle, shapes to

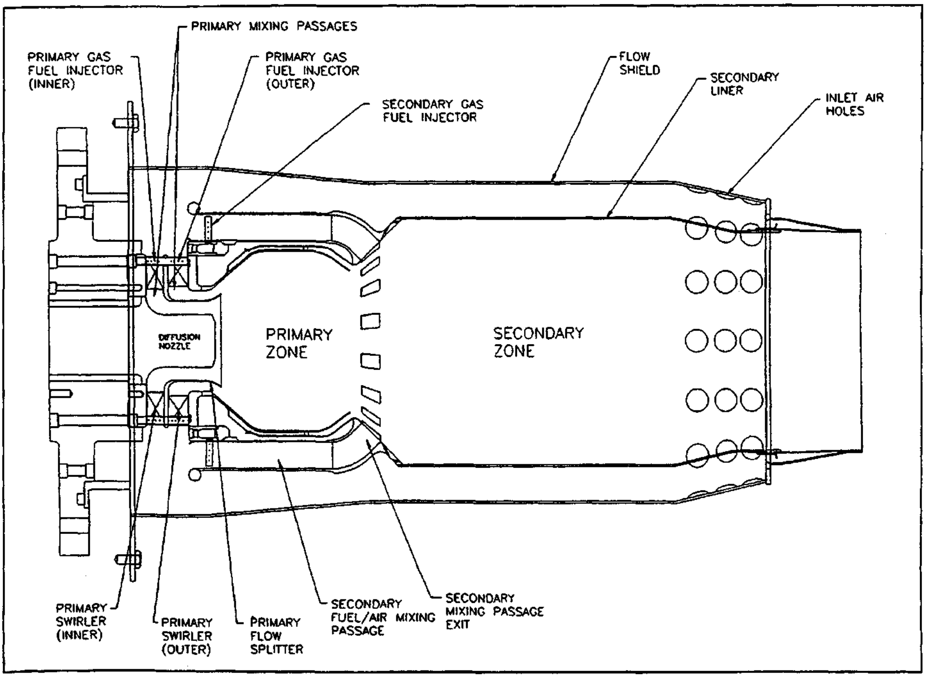


Fig. 57.9 Westinghouse dry low-NO_x combustor for advanced gas turbines (courtesy of Westinghouse Corporation).

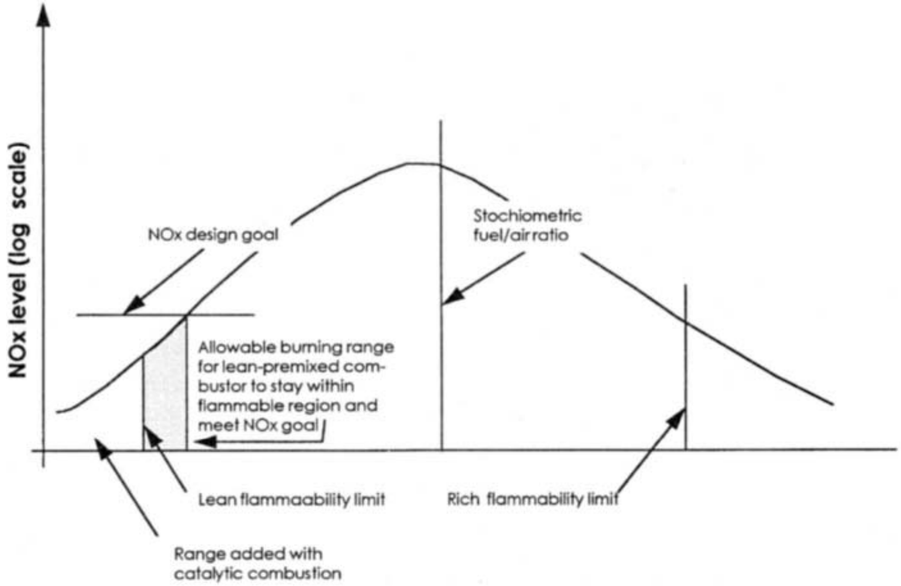


Fig. 57.10 Fuel–air mixture ranges for conventional and premixed combustors (courtesy of Westinghouse Corporation).

stabilize flames. They offer the promise of simpler fuel regulation and greater turn-down capability than low-NO_x combustors now in use. In catalytic combustors, the fuel and air react in the presence of a catalytic material that is deposited on a structure having multiple parallel passages or mesh. Extremely low NO_x levels have been observed in laboratories with catalytic combustion systems.

Turbine

Figure 57.11 shows an axial flow turbine. Radial in-flow turbines similar in appearance to centrifugal compressors are also produced for some smaller gas turbines.

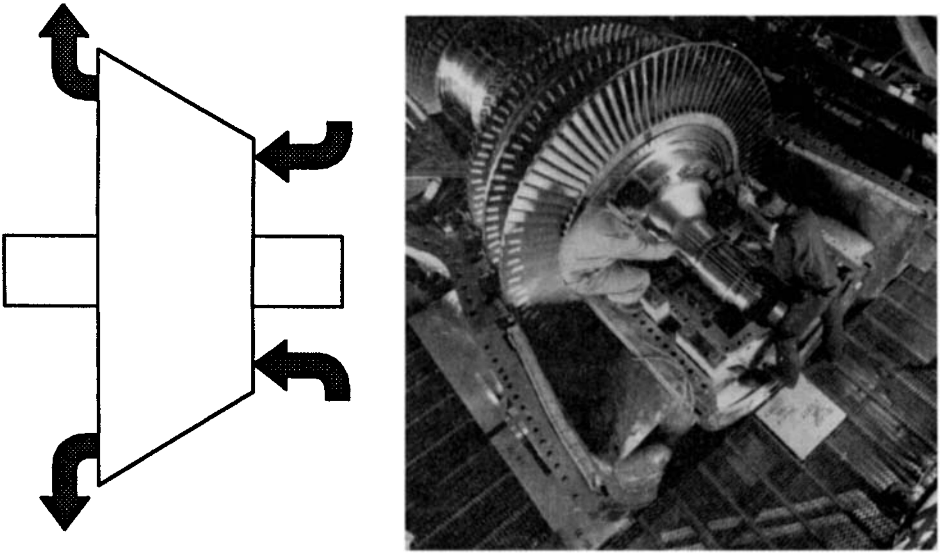


Fig. 57.11 Turbine diagram, and photo of an axial flow turbine during assembly (courtesy of General Electric Company).

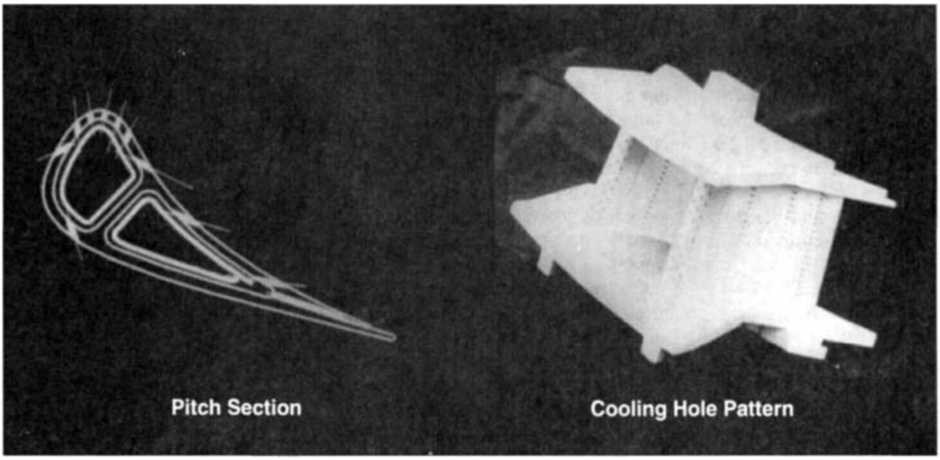


Fig. 57.12 Gas turbine first stage nozzle. Sketch shows cooling system of one airfoil (courtesy of General Electric Company).

By the time the extremely hot gas leaves the combustor and enters the turbine, it has been mixed with compressor discharge air to cool it to temperatures that can be tolerated by the first-stage blading in the turbine: temperatures ranging from 950°C in first-generation gas turbines to over 1500°C in turbines currently being developed and in state-of-the-art aircraft engines. Less dilution flow is required as firing temperatures approach 1500°C.

The first-stage stationary blades, or nozzles, are located at the discharge of the combustor. Their function is to accelerate the hot working fluid and turn it so as to enter the following rotor stage at the proper angle. These first-stage nozzles are subjected to the highest gas velocity in the engine. The gas entering the first-stage nozzle can regularly be above the melting temperature of the structural metal. These conditions produce high heat transfer to the nozzles, so that cooling is necessary.

Nozzles (Fig. 57.12) are subjected to stresses imposed by aerodynamic flow of the working fluid, pressure loading of the cooling air, and thermal stresses caused by uneven temperatures over the nozzle structure. First-stage nozzles can be supported at both ends, by the inner and outer sidewalls. But later-stage nozzles, because of their location in the engine, can be supported only at the outer end, intensifying the effect of aerodynamic loading.

The rotating blades of the turbine, or buckets (Fig. 57.13), convert the kinetic energy of the hot gas exiting the nozzles to shaft power used to drive the compressor and load devices. The blade consists of an airfoil section in the gas path, a dovetail or other type of joint connecting the blade to the turbine disc, and often a shank between the airfoil and dovetail allowing the dovetail to run at lower temperature than the root of the airfoil. Some bucket designs employ tip shrouds to limit

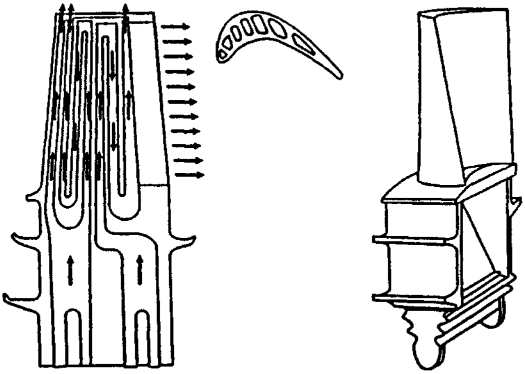


Fig. 57.13 Gas turbine first-stage air-cooled bucket. Cut-away view exposes serpentine cooling passages (courtesy of General Electric Company).

deflection at the outer ends of the buckets, raise natural vibratory frequencies, and provide aerodynamic benefits. Exceptions from this configuration are radial inflow turbines like those common to automotive turbochargers and axial turbines, wherein the buckets and wheels are made of one piece of metal or ceramic.

The total temperature of the gas relative to the bucket is lower than that relative to the preceding nozzles. This is because the tangential velocity of the rotor-mounted airfoil is in a direction away from the gas stream and thus reduces the dynamic component of total temperature. Also, the gas temperature is reduced by the cooling air provided to the upstream nozzle and the various upstream leakages.

Buckets and the discs on which they are mounted are subject to centrifugal stresses. The centrifugal force acting on a unit mass at the blades' midspan is 10,000 to 100,000 times that of gravity. Midspan airfoil centrifugal stresses range from 7 kg/mm^2 (10,000 psi) to over 28 kg/mm^2 (40,000 psi) at the airfoil root in the last stage (longest buckets).

Turbine efficiency is calculated similarly to compressor efficiency. Figure 57.6 also shows the effect of turbine efficiency. Line c–d represents the isentropic expansion process and c–d' the actual. Turbine efficiency η_t is the ratio of the vertical projections of the lines. Thus, (1250–557)/(1250–480) = 90%. It is possible at this point to compute the effect of a 90% efficient compressor and a 90% efficient turbine upon the simple cycle efficiency of the gas turbine represented in the figure. The turbine work is proportional to 693°C and the compressor work to 372°C. The heat added by combustion is proportional to 887°C, the temperature rise from b' to c. The ratio of the useful work to the heat addition is thus 36.2%. It was shown previously that the efficiency with ideal components is approximately 48.3%.

The needs of gas turbine blading have been responsible for the rapid development of a special class of alloys. To tolerate higher metal temperatures without decrease in component life, materials scientists and engineers have developed, and continue to advance, families of temperature-resistant alloys, processes, and coatings. The "superalloys" were invented and continue to be developed primarily in response to turbine needs. These are usually based on Group VIIIA elements: cobalt, iron, and nickel. Bucket alloys are austenitic with gamma/gamma-prime, face-centered cubic structure (Ni_3Al). The elements titanium and columbium are present and partially take the place of aluminum, with beneficial hot corrosion effect. Carbides are present for grain boundary strength, along with some chromium to further enhance corrosion resistance. The turbine industry has also developed processes to produce single-crystal and directionally solidified components that have even better high-temperature performance. Coatings are now in universal use that enhance the corrosion and erosion performance of hot gas path components.[14]

Cooling

Metal temperature control is addressed primarily through airfoil cooling, with cooling air being extracted from the gas turbine flow ahead of the combustor. Since this air is not heated by the combustion process, and may even bypass some turbine stages, the cycle is less efficient than it would be without cooling. Further, as coolant re-enters the gas path, it produces quenching and mixing losses. Hence, for efficiency, the use of cooling air should be minimized. Turbine designers must make tradeoffs among cycle efficiency (firing temperature), parts lives (metal temperature), and component efficiency (cooling flow).

In early, first-generation gas turbines, buckets were solid metal, operating at the temperature of the combustion gases. In second-generation machines, cooling air was conducted through simple, radial passages to keep metal temperatures below those of the surrounding gas. In today's advanced-technology gas turbines, most manufacturers utilize serpentine air passages within the first-stage buckets, with cooling air flowing out the tip, leading, and trailing edges. Leading edge flow is used to provide a cooling film over the outer bucket surface. Nozzles are often fitted with perforated metal inserts attached to the inside of hollow airfoils. The cooling air is introduced inside of the inserts. It then flows through the perforations, impinging on the inner surface of the hollow airfoil. The cooling thus provided is called *impingement cooling*. The cooling air then turns and flows within the passage between the insert and the inner surface of the airfoil, cooling it by convection until it exits the airfoil in either leading edge film holes or trailing edge bleed holes.

The effectiveness of cooling η is defined as the ratio of the difference between gas and metal temperatures to the difference between the gas temperature and the coolant temperature:

$$\eta = (Tg-Tm)/(Tg-Tc)$$

Figure 57.14 portrays the relationship between this parameter and a function of the cooling air flow. It can be seen that, while increased cooling flows have improved cooling effectiveness, there are diminishing returns with increased cooling air flow.

Cooling can be improved by precooling the air extracted from the compressor. This is done by passing the extracted air through a heat exchanger prior to using it for bucket or nozzle cooling. This does increase cooling, but presents several challenges, such as increasing temperature gradients and

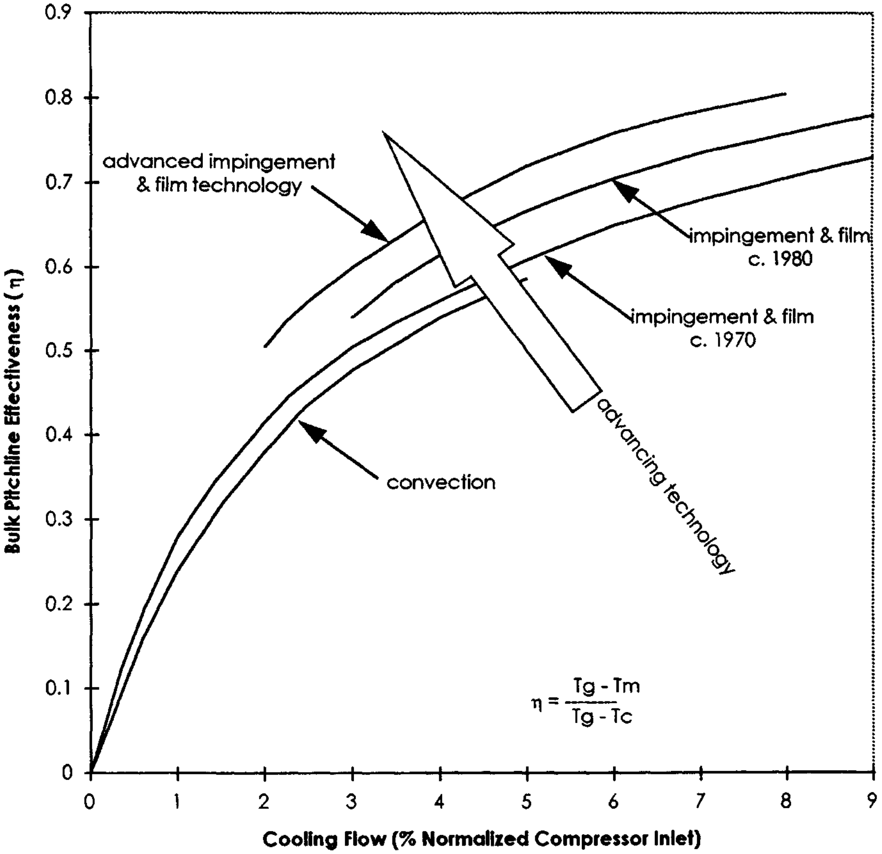


Fig. 57.14 Evolution of turbine airfoil cooling technology.

the cost and reliability of the cooling equipment. Recent advanced gas turbine products have been designed with both cooled and uncooled cooling air.

Other cooling media have been investigated. In the late 1970s, the U.S. Department of Energy sponsored the study and preliminary design of high-temperature turbines cooled by water and steam. Nozzles of the water-cooled turbine were cooled by water contained in closed passages and kept in the liquid state by pressurization; no water from the nozzle circuits entered the gas path. Buckets were cooled by two-phase flow; heat was absorbed as the coolant was vaporized and heated. Actual nozzles were successfully rig tested. Simulated buckets were tested in heated rotating rigs. Recent advanced land-based gas turbines have been configured with both buckets and nozzles cooled with a closed steam circuit. Steam, being a more effective cooling medium than air, permits high firing temperatures and, since it does not enter the gas path, eliminates the losses associated with cooling air mixing with the working fluid. The coolant, after being heated in the buckets and nozzles, returns to the steam cycle of a combined cycle plant. The heat carried away by the steam is recovered in a steam turbine.

57.1.4 Controls and Accessories

Controls

The control system is the interface between the operator and the gas turbine. More correctly, the control system of modern industrial and utility gas turbines interfaces between the operator and the entire power plant, including the gas turbine, generator, and all related accessories. In combined cycle power plants where a steam turbine, heat recovery steam generator, condensing system, and all related accessories are also present, the control system interfaces with these as well.

Functions provided are described below in Section 57.1.5 plus protection of the turbine from faults such as overspeed, overheating, combustion anomalies, cooling system failures, and high vibrations. Also, controls facilitate condition monitoring, problem identification and diagnosis, and monitoring of thermodynamic and emissions performance. Sensors placed on the gas turbine include speed pickups, thermocouples at the inlet, exhaust, compressor exit, wheelspaces, bearings, oil supplies and drains. Vibration monitors are placed on each bearing. Pressure is also monitored at the compressor exit. Multiple thermocouples in the exhaust can detect combustor malfunction by noting abnormal differences in exhaust temperature from one location to another. Multiple sensors elsewhere allow the more sophisticated control systems to self-diagnose, to determine if a problem reading is an indication of a dangerous condition or the result of a sensor malfunction.

Control system development over the past two decades has contributed greatly to the improved reliability of power-generation gas turbines. The control systems are now all computer based. Operator input is via keyboard and cursor movement. Information is displayed to the operator via color graphic displays and tabular and text data on color monitors.

Inlet Systems

Inlet systems filter and direct incoming air, and provide attenuation of compressor noise. They also can include heating and cooling devices to modify the temperature of the air drawn into the gas turbine. Since fixed-wing aircraft engines operate most of the time at high altitudes, where air is devoid of heavier and more damaging particles, these engines are not fitted with inlet air treatment systems performing more than an aerodynamic function. The premium placed on engine weight makes this so. Inertial separators have been applied to helicopter engines to reduce their ingestion of particulates.

Air near the surface of the earth contains dust and dirt of various chemical compositions. Because of the high volume of air taken into a gas turbine, this dirt can cause erosion of compressor blades, corrosion of both turbine and compressor blades, and plugging of passages in the gas path as well as cooling circuits. The roughening of compressor blade surfaces can be due to particles sticking to airfoil surfaces, erosion, or corrosion caused by their chemical composition. This fouling of the compressor can, over time, reduce mass flow and lower compressor efficiency. Both effects will reduce the output and efficiency of the gas turbine. "Self-cleaning" filters collect airborne dirt. When the pressure drop increases to a preset value, a pulse of air is used to reverse the flow briefly across the filter medium, cleaning the filter. More conventional, multistage filters also find application.

Under low-ambient-temperature, high-humidity conditions, it is possible to form frost or ice in the gas turbine inlet. Filters can be used to remove humidity by causing frost to form on the filter element. The frost is removed by the self-cleaning feature. Otherwise, a heating element can be installed in the inlet compartment. These elements use higher-temperature air extracted from the compressor. This air is mixed with the ambient air, raising its temperature. Compressors of most robust gas turbines are designed so that these systems are required only at part load or under unusual operating conditions.

Inlet chillers have been applied on gas turbines installed in high-ambient-temperature, low-humidity regions of the world. The incoming air is cooled by the evaporation of water. Cooling the inlet air increases its density and increases the output of the gas turbine.

Exhaust Systems

The exhaust systems of industrial gas turbines perform three basic functions. Personnel must be protected from the high-temperature gas and from the ducts that carry it. The exhaust gas must be conducted to an exhaust stack or to where the remaining heat from the gas turbine cycle can be effectively used. The exhaust system also contains baffles and other features employed to reduce the noise generated by the gas turbine.

Enclosures and Lagging

Gas turbines are enclosed for four reasons: noise, heat, fire protection, and visual aesthetics. Gas turbines are sometimes provided for outdoor installation, where the supplier includes a sheet metal enclosure that may be part of the factory-shipped package. Other times, gas turbines are installed in a building. Even in a building, the gas turbine is enclosed for the benefit of maintenance crews or other occupants. Some gas turbines are designed to accommodate an insulating wrapping that attaches to the casings of the gas turbine. This prevents maintenance crews from coming into contact with the hot casings when the turbine is operating and reduces some of the noise generated by the gas turbine. Proponents cite the benefit of lowering the heat transferred from the gas turbine to the environment. Theoretically, more heat is carried to the exhaust which can be used for other energy needs. Others contend that the larger internal clearances resulting from hotter casings would offset this gain by lower component efficiencies.

Where insulation is not attached to the casings, and sometimes when it is, a small building-like structure is provided. This structure is either attached to the turbine base or to the concrete foundation. Such a structure provides crew protection and noise control, and assists in fire protection. If a fire is detected on the turbine, within the enclosure, its relative small volume makes it possible to quickly flood the area with CO_2 or other firefighting chemical. The fire is thereby contained in a small volume and more quickly extinguished. Even in a building, the noise control provided by an enclosure is beneficial, especially in buildings containing additional gas turbines or other equipment. By lowering the noise 1 m from the enclosure to below 85 or 90 dba, it is possible to safely perform maintenance on this other equipment, yet continue to operate the gas turbine. Where no turbine enclosure is provided within a building, the building becomes part of the fire-protection and acoustic system.

Fuel Systems

The minimum functions required of a gas turbine fuel system are to deliver fuel from a tank or pipeline to the gas turbine combustor fuel nozzles at the required pressure and flow rate. The pressure required is somewhat above the compressor discharge pressure, and the flow rate is that called for by the controls. On annular and can-annular combustors, the same fuel flow must be distributed to each nozzle to ensure minimum variation in the temperature to which gas path components are exposed. Other fuel system requirements are related to the required chemistry and quality of the fuel.

Aircraft engine fuel quality and chemistry are closely regulated, so extensive on-board fuel conditioning systems are not required. Such is not the case in many industrial applications. Even the better grades of distillate oil may be delivered by oceangoing tanker and run the risk of sodium contamination from the salt water sometimes used for ballast. Natural gas now contains more of the heavier, LP gases. Gas turbines are also fueled with crude oil, heavy oils, and various blends. Some applications require the use of non-lubricating fuels such as naphtha. Most fuels today require some degree of on-site treatment.

Complete liquid fuel treatment includes washing to remove soluble trace metals, such as sodium, potassium, and certain calcium compounds. Filtering the fuel removes solid oxides and silicates. Inhibiting the vanadium in the fuel with magnesium compounds in a ratio of three parts of magnesium (by weight) to one part of vanadium limits the corrosive action of vanadium on the alloys used in high-temperature gas path parts.

Gas fuel is primarily methane, but it contains varying levels of propane, butane, and other heavier hydrocarbons. When levels of these heavier gases increase, the position of the flame in the combustor may change, resulting in local hot spots that could damage first-stage turbine stator blades. Also, sudden increases could cause problems for dry low-NO_x premixed combustors. These combustors depend on being able to mix fuel and air in a combustible mixture before the mixture is ignited. Under some conditions, heavier hydrocarbons can self-ignite in these mixtures at compressor exit temperatures, thus causing flame to exist in the premixing portion of the combustor. The flame in the premixing area would have to be extinguished and reestablished in the proper location. This process interferes with normal operation of the machine.

Lubricating Systems

Oil must be provided to the bearings of the gas turbine and its driven equipment. The lubricating system must maintain the oil at sufficiently low temperature to prevent deterioration of its properties. Contaminants must be filtered out. Sufficient volume of oil must be in the system so that any foam has time to settle out. Also, vapors must be dealt with; they are preferably recovered and the oil returned to the plenum. The oil tank for large industrial turbines is generally the base of the lubricating system package. Large utility machines are provided with tanks that hold over 12,000 liters of oil. The oil is generally replaced after approximately 20,000 hours of operation. More oil is required in applications where the load device is connected to the gas turbine by a gearbox.

The lubrication system package also contains filters and coolers. The turbine is fitted with mist-elimination devices connected to the bearing air vents. Bearings may be vented to the turbine exhaust, but this practice is disappearing for environmental reasons.

Cooling Water and Cooling Air Systems

Several industrial gas turbine applications require the cooling of some accessories. The accessories requiring cooling include the starting means, lubrication system, atomizing air, load equipment (generator/alternator), and turbine support structure. Water is circulated in the component requiring cooling, then conducted to where the heat can be removed from the coolant. The cooling system can be integrated into the industrial or powerplant hosting the gas turbine, or can be dedicated to the gas turbine. In this case, the system usually contains a water-to-air heat exchanger with fans to provide the flow of air past finned water tubes.

Water-Wash Systems

Compressor fouling related to deposition of particles that are not removed by the air filter can be dealt with by water-washing the compressor. A significant benefit in gas turbine efficiency over time can be realized by periodic cleaning of the compressor blades. This cleaning is most conveniently done when the gas turbine is fitted with an automatic water-wash system. Washing is initiated by the operator. The water is preheated and detergent is added. The gas turbine rotor is rotated at a low speed and the water is sprayed into the compressor. Drains are provided to remove waste water.

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