CHAPTER 7

COGENERATION SYSTEMS AND ENGINE AND TURBINE DRIVES

COGENERATION is the simultaneous production of electrical or mechanical energy (power) and useful thermal energy from a single energy stream, such as oil, coal, natural or liquefied gas, biomass, or solar. By capturing and applying heat from an effluent energy stream that would otherwise be rejected to the environment, cogeneration systems can operate at efficiencies greater than those achieved when heat and power are produced in separate or distinct processes. Recovering this thermal energy for a useful purpose from reciprocating engines or steam or combustion turbines can take the following forms:

- Direct heating (e.g., a drying process)
- Indirect heating to generate a flow of steam or hot water for remote heating devices or to generate shaft power in a steam turbine
- Extracting the latent heat of condensation from a recovered flow of steam when the load served permits condensation (e.g., a steam-to-water exchanger) instead of rejecting the latent heat to a cooling tower (e.g., a full condensing turbine with a cooling tower)

Cogeneration can be a topping, bottoming, or combined cycle. In a **topping cycle**, the fuel generates power first, and the resulting thermal energy is recovered and productively used. In a **bottoming cycle**, the power is generated last from the thermal energy left over after the higher level thermal energy has been used to satisfy thermal loads. A typical topping cycle recovers heat from the operation of a prime mover and uses this thermal energy for the process (cooling or heating). A bottoming cycle recovers heat from the process to generate power. A **combined cycle** uses the thermal output from a prime mover to generate additional shaft power (e.g., combustion turbine exhaust generates steam for a steam turbine generator).

When a prime mover generates power, and the thermal output is used productively, it is a cogeneration cycle; if the thermal energy is wasted, it is not. The discussion in this chapter about the prime mover and its driven device applies to both cases, the only difference being that one uses the available thermal energy (cogeneration) while the other does not.

Isolated cogeneration systems, the electrical output of which is used on site to satisfy all site power requirements, are referred to as **total energy systems**. A cogeneration system that is actively tied (paralleled) to the utility grid can, on a contractual basis or on a tariff basis, exchange power with the public utility. This lessens the need for redundant on-site generating capacity and allows operation at maximum thermal efficiency when satisfying the facility's thermal load requires more electric power than the facility needs.

System feasibility and design depend on the magnitude, duration, and coincidence of electrical and thermal loads, as well as on the selection of the prime mover and waste heat recovery system. Integrating the design of the project's electrical and thermal requirements with the cogeneration plant is required for optimum economic benefit. The basic components of the cogeneration plant are (1) prime mover and its fuel supply system, (2) generator, (3) waste heat recovery system, (4) control system, (5) electrical and thermal transmission and distribution systems, and (6) connections to building mechanical and electrical services.

The prime mover converts fuel or thermal energy to shaft energy. The conversion devices normally used are reciprocating internal combustion engines, combustion turbines, expansion turbines, and steam boiler-turbine combinations.

This chapter describes prime movers for a variety of uses, including generators and compressors for refrigerants and other gases. Heat recovery, electrical power and control systems, design concepts, plant auxiliaries, installation, feasibility, and utilization systems are also discussed. Thermal distribution systems are covered in Chapter 11 and Chapter 12.

RECIPROCATING ENGINES

Reciprocating engines are the most common prime mover used in smaller (i.e., under 15 MW) cogeneration plants. These engines are available in sizes up to 27,000 brake horsepower (bhp) and use all types of liquid and gaseous fuels, including methane from landfills or sewage treatment plant digesters. Internal combustion engines that use the **diesel (compression ignition) cycle** can be fueled by a wide range of petroleum products (up to No. 6 oil), although No. 2 diesel oil is the most commonly used. Diesel cycle engines can also be fired with gaseous fuel in combination with liquid fuel. In such **dual-fuel engines**, the liquid acts as the ignition agent and is called pilot oil.

Spark ignition Otto cycle engines are produced in sizes up to 18,000 bhp and use natural gas, liquefied petroleum gas (LPG), and other gaseous and volatile liquid fuels. Engines usually operate in the range of 360 to 1200 rpm, with some up to 1800 rpm. The specific operating speed selected depends on the size and brand of machine, the generator, and the desired length of time between complete engine overhauls. Engines are usually selected to provide a minimum of 15,000 to 30,000 operating hours between minor overhauls, and up to 50,000 operating hours between major overhauls; the higher operating hours correspond to the larger, lower speed engines.

The engine components include the starter system; fuel input systems; fuel-air mixing cycle; ignition system; combustion chamber; exhaust gas collection and removal system; lubrication systems; and power transmission gear with pistons, connecting rods, crankshafts, and flywheel. Some engines also increase power output using turbochargers (engine exhaust, turbine-driven compressors) that increase the mass flow of air delivered to the combustion chamber. The engines are generally 20 to 40% efficient in converting fuel to shaft energy.

The preparation of this chapter is assigned to TC 9.5, Cogeneration Systems.

The **four-stroke** diesel cycle engine, also known as a compression ignition (CI) engine, involves the following four piston strokes for each power stroke:

Intake stroke—Piston travels from top dead center to bottom dead center and takes fresh air into the cylinder. The intake valve is open during this stroke.

Compression stroke—Piston travels from the cylinder bottom to the top with all valves closed. As the air is compressed, its temperature increases. Shortly before the end of the stroke, a measured quantity of diesel fuel is injected into the cylinder. Combustion of the fuel begins just before the piston reaches top dead center.

Power stroke—Burning gases exert pressure on the piston, pushing it to bottom dead center. All valves are closed until shortly before the end of the stroke, when the exhaust valves are opened.

Exhaust stroke—Piston returns to top dead center, venting products of combustion from the cylinder through the exhaust valves.

A four-stroke Otto cycle engine, also known as a spark ignition (SI) engine, operates through the same cycle with two variations. First, a fuel-air mixture rather than pure air flows into the cylinder during the intake stroke, and second, an externally supplied spark is used to initiate combustion at the end of the compression stroke.

The **two-stroke cycle** (Figure 1) requires only two piston strokes for each power stroke. The intake and compression functions of the four-stroke cycle are combined into a single stroke, as are the power and exhaust functions. The intake/compression stroke starts as the piston nears bottom dead center, uncovering intake ports located in the cylinder walls. When the intake port is uncovered in a CI engine, a "scavenging" blower forces air into the cylinder. That intake air forces the combustion products out of the cylinder through the exhaust valves, which were opened at the end of the downward stroke. When the upward moving piston passes the intake ports, sealing them, the exhaust valves are closed and compression occurs. When the piston approaches top dead center and the air temperature is greater than the fuel's ignition temperature, the fuel is injected into the cylinder, where it initiates combustion. During the second or power/exhaust stroke, the piston is forced toward the cylinder bottom, and the intake/exhaust ports are opened to allow the intake/exhaust function.

Two-stroke engines cost less than four-stroke engines of the same capacity. However, two-stroke engines are less efficient, so they reject more heat. As a result, the heat recovery system tends to be larger and more costly than it would be for the four-stroke engine. Moreover, the effective compression ratio of the two-stroke engine is lower than that of a four-stroke engine of the same size. The lower compression ratio is a result of the placement of the air intake ports in the cylinder walls, which reduces the effective volume of the cylinder. Additionally, the scavenging blower is a parasitic load on the engine, and the opening of the intake port and exhaust valve prior to the end of the power stroke causes a loss of power. Two-stroke engines also require 40% more combustion air than do four-stroke engines. In general, two-stroke engines are used in standby or peak shaving duty where low efficiency is not as critical to project economics.

Pistons are usually arranged in an in-line or a V configuration. However, a particularly interesting two-stroke design is the **opposed-piston engine** (Figure 2). These engines contain two crankshafts, one on top of the engine and the other at the bottom. Each cylinder contains two pistons, and there are no cylinder heads. Opposed-piston engines are especially compact and for this reason are widely used for marine propulsion, especially submarines, where space is limited. For marine and other stationary applications, they are typically operated on diesel oil, but for cogeneration applications, dual-fuel CI or SI operation is usually the most cost-effective.

The cycle begins with the pistons at the outer end of their stroke and a fresh charge of air in the cylinder. As the pistons move inward, the air charge rapidly reaches a high temperature as it is compressed. At approximately 9° before the lower piston reaches inner dead center, injection of fuel oil begins. The high temperature of the air charge causes the fuel oil to ignite, and combustion takes place as the pistons pass through their inner centers. The pressure from combustion forces the pistons apart, thereby delivering power to the crankshafts.

The gases expand until nearly the end of the power stroke as the lower piston begins to uncover the exhaust ports, allowing the burned gases to escape to the atmosphere through the exhaust system. At about the time the pressure in the cylinder has dropped to almost atmospheric, the upper piston starts uncovering the inlet ports. Scavenging air in the air receiver rushes into the cylinder under pressure supplied by the blower.

The cylinder is swept clean of the remaining exhaust gases and refilled with fresh air for the next compression stroke. This cycle is

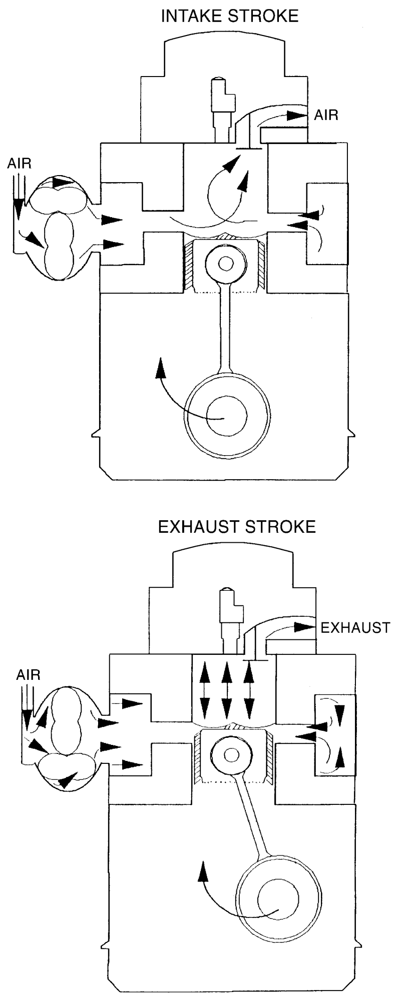

Fig. 1 Two-Stroke Diesel Cycle

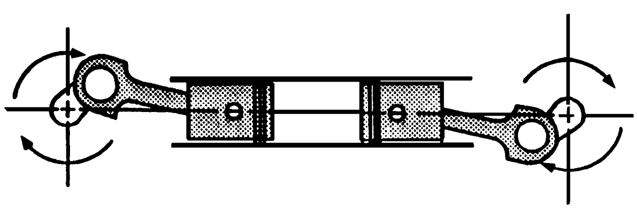

Fig. 2 Opposed-Piston Engine

the same for both diesel and dual-fuel cycles except during fuel admission and combustion. When gas is the primary fuel, shortly after the lower piston covers the exhaust ports, gas is admitted and continues to be admitted for approximately one-quarter revolution of the crankshaft. After the combined air and gas charge is compressed, a small amount of fuel oil (pilot oil) is injected into the cylinder. Ignition of the pilot oil in turn ignites the air-gas charge, and combustion occurs.

PERFORMANCE CHARACTERISTICS

Some of the more important performance characteristics of an engine are its power rating, fuel consumption, and thermal output. Manufacturers base their engine ratings on the engine duty: prime power, standby operations, and peak shaving. Because a cogeneration system is most cost-effective when operating at its baseload, the rating at prime power is usually of greatest interest. Prime power implies that the engine is the primary source of power. This rating is based on providing extended operating life with minimum maintenance. When used for standby, the engine produces continuously for 24 h per day for the length of the primary source outage. Peak power implies an operation level for only a few hours per day to meet peak demand in excess of the prime power capability.

Many manufacturers rate engine capacities according to ISO *Standard* 3046-1. This standard specifies that continuous net brake power under standard reference conditions (total barometric pressure 14.5 psi, corresponding to approximately 330 ft above sea level, air temperature 77°F, and relative humidity 30%) can be exceeded by 10% for 1 h, with or without interruptions, within a period of 12 h of operation. ISO *Standard* 3046-1 defines prime power as power available for continuous operation under varying load factors and 10% overload as previously described. The standard defines standby power as power available for operation under normal varying load factors, not overloadable (for applications normally designed to require a maximum of 300 h of service per year).

However, the basis of the manufacturer's ratings (ambient temperature, altitude, and atmospheric pressure of the test conditions) must be known to determine the engine rating at site conditions. Various derating factors are used. Naturally aspirated engine output typically decreases 3% for each 1000 ft increase in altitude, while turbocharged engines lose 2% per 1000 ft. Output decreases 1% per 10°F increase in ambient temperature, so it is important to avoid the use of heated air for combustion. In addition, an engine must be derated for those fuels with a heating value that is significantly greater than the base specified by the manufacturer. For cogeneration applications, natural gas is usually the fuel of choice.

Power rating is determined by a number of engine design characteristics, the most important of which is displacement; but it also depends on rotational speed, method of ignition, compression ratio, aspiration, cooling system, jacket water temperature, and intercooler temperature. Most engine designs are offered in a range of displacements achieved by different bore and stroke, but with the same number of cylinders in each case. Many larger engine designs retain the same basic configuration, and displacement increases are achieved by simply lengthening the block and adding more cylinders.

Figure 3 illustrates the capacity of some typical SI natural gas engine/generator sets operating at the prime power rating and at 1200 and 900 rpm. The lower value at each speed represents a naturally aspirated design; the upper value represents either an intercooled, turbocharged design or a naturally aspirated design with a higher compression ratio.

Fuel consumption is the greatest contributor to operating cost and should be carefully considered in the planning and design phases of a cogeneration system. It is influenced by combustion cycle, speed, compression ratio, and type of aspiration. It is often expressed in terms of power (Btu/h) for natural gas engines, but for purposes of comparison, it may be expressed as a ratio such as Btu/h per brake horsepower or Btu/kWh. The latter is known as the **heat rate** and equals 3412/efficiency.

The heat rates of several SI engines are shown in Figure 4. Heat rate for an engine of a given size is affected by design and operating factors other than displacement. The most efficient (lowest heat rate) of these engines is naturally aspirated and achieves its increased performance due to the slightly higher compression ratio.

The **thermal-to-electric ratio** is a measure of the useful thermal output for the electrical power being generated. For most reciprocating engines, the recoverable thermal energy is that of the exhaust and jacket. Figure 5 shows the thermal-to-electric ratio of the SI engines shown in Figure 3 and Figure 4. In these example engines, all jacket heat was recovered, and the exhaust gases were cooled to 325°F.

Figure 6 and Figure 7 show the capacities of a large four-stroke and an opposed-piston engine. Capacity varies by the number of cylinders, so performance characteristics relative to output are constant. When the four-stroke engine operates at a heat rate of 10,690 Btu/kWh, the thermal-to-electric ratio is 3880 Btu/kWh. When the two-stroke, opposed-piston engine operates at 720 rpm and a heat rate of 11,030 Btu/kWh, the thermal-to-electric ratio is 2360 Btu/kWh. The large amount of air used to scavenge the exhaust gases from two-stroke engines reduces their exhaust temperatures, thus reducing their usefulness for cogeneration, in which exhaust gas heat recovery is important.

Ideally, a cogeneration plant should operate at full output to achieve maximum cost-effectiveness. In plants that must operate at part load some of the time, part-load fuel consumption and thermal output are important factors that must be considered in the overall economics of the plant. Figure 8 shows the part-load heat rate and

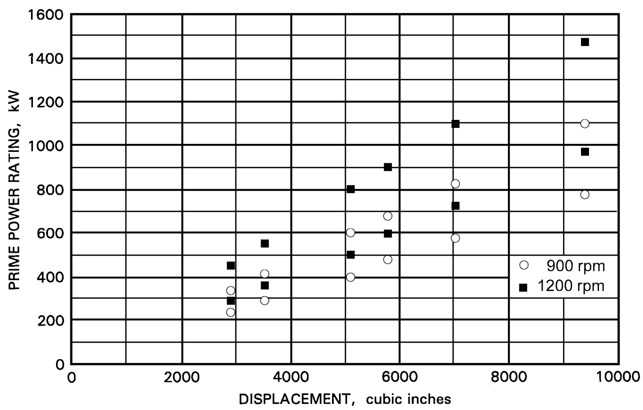

Fig. 3 Capacity of Spark Ignition Engine

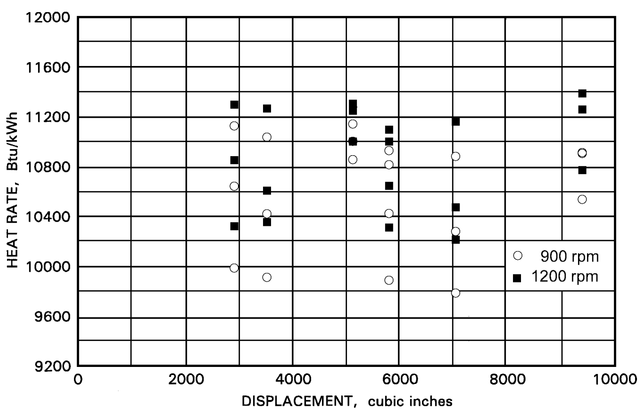

Fig. 4 Heat Rate of Spark Ignition Engines

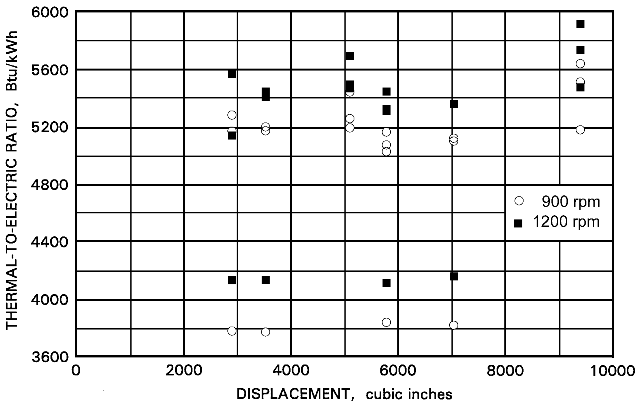

Fig. 5 Thermal-to-Electric Ratio of Spark Ignition Engines

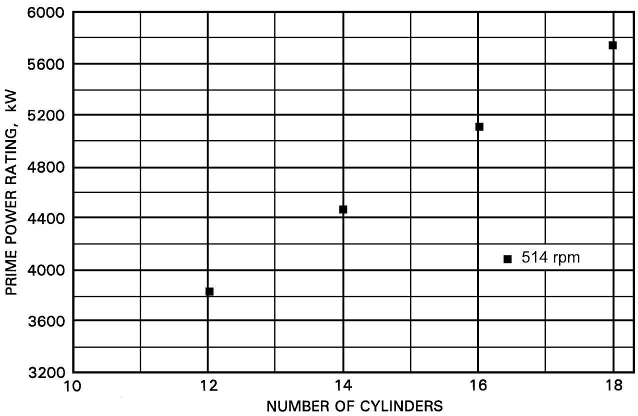

Fig. 6 Capacity of Large Four-Stroke Dual-Fuel Engine

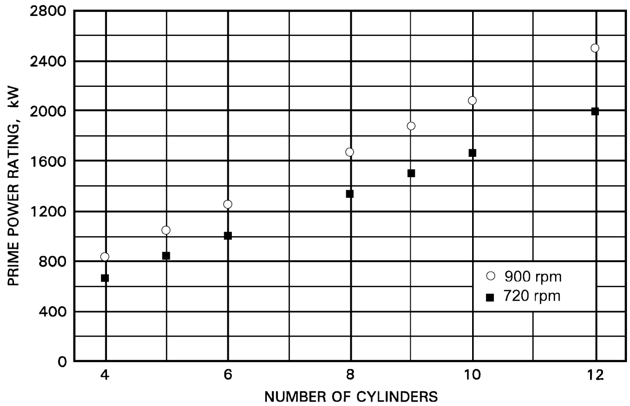

Fig. 7 Capacity of Opposed-Piston, Dual-Fuel Engine

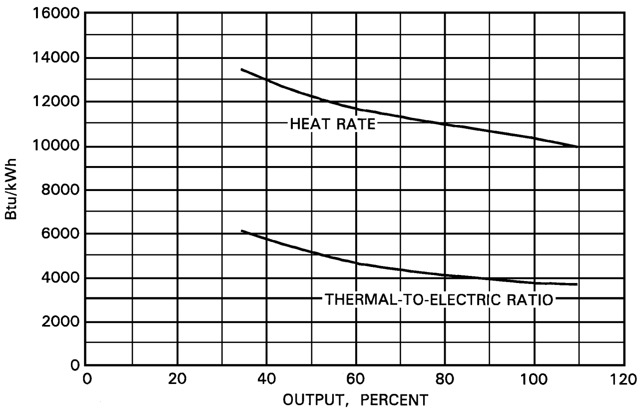

Fig. 8 Part-Load Performance of 725 kW Gas Engine

thermal-to-electric ratio as a function of load for a 725 kW naturally aspirated gas engine, assuming a final exhaust temperature of 375°F.

FUELS

Fuel Selection

Fuel specifications, grade, and characteristics have a marked effect on engine performance. Fuel standards for internal combustion engines are designated by the American Society for Testing and Materials (ASTM) and are substantially different from those for heating use.

Engines may be fueled with gasoline, natural gas, propane, sludge gas, or diesel and heavier oils. Multifuel engines using diesel oil as one of the fuels are available in sizes larger than 220 hp. A small amount of diesel oil is used as the compression ignition agent in gas-fueled engines.

Gasoline engines are generally not used because of fuel storage hazards, fuel cost, and the higher maintenance required due to deposits of combustion products on internal parts.

Methane-rich gas obtained from sewage treatment processes can be used as a fuel for both engines and other heating services. The fuel must be dried and cleaned prior to its injection into the engines. Because of the lower heat content of methane-rich gas (approximately 600 to 700 Btu/ft^3), it is sometimes mixed with natural gas, and the engine must be fitted with a larger carburetor.

Dual carburetors are often installed, with sewage gas as the primary fuel and natural gas as backup. The large amount of hydrogen sulfide in the fuel requires the use of special materials, such as aluminum in the bearings and bushings and low-friction plastic in the O rings and gaskets.

The final choice of fuel should be based on fuel availability, cost, storage requirements, emissions requirements, and fuel rate. Except for gasoline engines, maintenance costs tend to be similar for all engines.

Fuel Heating Value

Fuel consumption data may be reported in terms of either high heating value (HHV) or low heating value (LHV). HHV is used by the gas utility industry and is the basis for evaluating most gaseous fuel. Most natural gases have an LHV/HHV factor of 0.9 to 0.95. This factor ranges from 0.87 for hydrogen to 1.0 for carbon monoxide. For fuel oils, LHV/HHV ranges from 0.96 (heavy oils) to 0.93 (light oils). The HHV is customarily used for oil (including pilot oil in dual-fuel engines).

Many manufacturers base engine power ratings on an LHV of approximately 900 Btu/ft^3 for natural gas. The LHV deducts the energy necessary to keep the water produced during combustion in the vapor state. This latent heat does no work on the piston. Manufacturers sometimes suggest derating engine output about 2% per 10% decrease in fuel heating value below the base specified by the manufacturer. Additional power required to drive auxiliary equipment like compressors, pumps, or generators should not be counted.

FUEL SYSTEMS

Fuel Oil Systems

Storage, handling, and cleaning of liquid fuels are covered in Chapter 26. Most oil-fueled cogeneration is with No. 2 diesel fuel, which is much simpler to handle than the heavier grades. Residual oil is used only for large industrial projects with low-speed engines.

The fuel injection system is the heart of the diesel cycle. Performance functions are as follows:

- Meter a constant quantity to each cylinder at any load during each combustion cycle.
- Inject fuel (1) with a precise and rapid beginning and ending at the correct timing point in each cycle and (2) at a rate needed for controlled combustion and pressure rise.
- Atomize the fuel and distribute it evenly through the air in the combustion chamber.

The two methods for atomization are high-pressure air injection and mechanical injection. Older systems used air injection until satisfactory mechanical injection systems were developed that avoid the high initial cost and parasitic operating cost of the air compressor.

Earlier mechanical injection pressurized a header to provide a common fuel pressure near 5000 psi. A camshaft opened a spray nozzle in each cylinder, with the length of spray time proportioned to the load through a governor or throttle control. With this system, a leaky valve allowed a steady drip into the cylinder throughout the cycle, which caused poor fuel economy and smoking.

Currently, three injection designs are in use:

- Individual plunger pumps for each cylinder, with controlled bypass, controlled suction, variable-suction orifice, variable stroke, or port-and-helix metering
- Common high-pressure metering pump with a separate distribution line to each cylinder that delivers fuel to each cylinder in firing order sequence
- Common low-pressure metering pump and distributor with a mechanically operated, high-pressure pump and nozzle at each cylinder

Spark Ignition Gas Systems

Fuels vary widely in composition and cleanliness, from pipeline natural gas requiring only a meter, a pressure regulating valve, and safety devices to those from sewage or biomass, which may require scrubbers and holding tanks in addition. The following are gas system accessories.

Line-Type Gas Pressure Regulators. Turbocharged (and aftercooled) engines, as well as many naturally aspirated units, are equipped with line regulators designed to control the gas pressure to the engine regulator, as shown in Table 1. The same regulators (both line and engine) used on naturally aspirated gas engines may be used on turbocharged equipment.

Line-type gas pressure regulators are commonly called service[b] regulators (and field regulators). They are usually located just upstream of the engine regulator to ensure that the required pressure range exists at the inlet to the engine regulator. A remote location is sometimes specified; authorities having jurisdiction should be consulted. Although this intermediate regulation does not constitute a safety device, it does permit initial regulation (by the gas utility at the meter inlet) at a higher outlet pressure, thus allowing an extra cushion of gas between the line regulator and the meter for both full gas flow at engine start-ups and delivery to any future branches from the same supply line. The engine manufacturer specifies the size, type, orifice size, and other regulator characteristics based on the anticipated gas-pressure range.

Table 1 Line Regulator Pressures

Line Regulator	Turbocharged Engine[a]	Naturally Aspirated Engine[a]
Inlet	14 to 20 psig	2 to 30 psig
Outlet[b]	12 to 15 psig[c]	7 to 10 in. of water

[a]Overall ranges—not the variation for individual installations.
[b]Also inlet to engine regulator.
[c]Turbocharger boost plus 7 to 10 in. of water.

Engine-Type Gas Pressure Regulators. This engine-mounted pressure regulator, also called a carburetor regulator (and sometimes a secondary regulator or a B regulator), controls the fuel pressure to the carburetor. Regulator construction may vary with the fuel used. The unit is similar to a zero governor.

Air-Fuel Control. The flow of air-fuel mixtures must be controlled in definite ratios under all load and speed conditions required of engines.

Air-Fuel Ratios. High-rated, naturally aspirated, spark ignition engines require closely controlled air-fuel ratios. Excessively lean mixtures cause excessive lubricating oil consumption and engine overheating. Engines using pilot oil ignition can run at rates above 0.18 ppm/hp without misfiring. Air rates may vary with changes in compression ratio, valve timing, and ambient conditions.

Carburetors. In these venturi devices, the airflow mixture is controlled by a governor-actuated butterfly valve. This air-fuel control has no moving parts other than the butterfly valve. The motivating force in naturally aspirated engines is the vacuum created by the intake strokes of the pistons. Turbocharged engines, on the other hand, supply the additional energy as pressurized air and pressurized fuel.

Ignition. An electrical system or pilot oil ignition may be used. Electrical systems are either low-tension (make-and-break) or high-tension (jump spark). Systems with breakerless ignition distribution are also in use.

Dual-Fuel and Multifuel Systems

Engines using gas and pilot oil for diesel ignition are commonly classified as dual-fuel engines, while those using natural gas (NG) with LPG standby are called multifuel engines. Dual-fuel engines operate either on full oil or on gas and pilot oil, with automatic online switchover when appropriate, while NG/LPG multifuel engines operate safely on one fuel or the other. In sewage gas systems, a blend with natural gas may be used to maintain a minimum LHV or to satisfy fuel demand when sewage gas production is short.

JACKET WATER SYSTEM

Various jacket water circuits to control the engine's operating temperature while recovering its conducted thermal waste are illustrated and described in the section on Engine Jacket Heat Recovery on page 7.20. Ultimately, this heat is (1) recovered and productively used, (2) carried away by an air-cooled radiator, (3) dissipated in a cooling tower or raw water stream (e.g., groundwater), or (4) ejected into the engine room.

LUBRICATING SYSTEMS

All engines use the lubricating system to remove some heat from the machine. Some configurations cool only the piston skirt with oil; other designs remove more of the engine heat with the lubricating system. The operating temperature of the engine may be significant in determining the proportion of engine heat removed by the lubricant. Between 5 and 10% of the total fuel input is converted to heat that must be extracted from the lubricating oil; this may warrant using oil coolant at temperatures high enough to permit economic use in a process such as domestic water heating.

Radiator-cooled units generally use the same fluid to cool the engine water jacket and the lubricant; thus, the temperature difference between the oil and the jacket coolant is not significant. If the oil temperature rises in one area (such as around the piston skirts), the heat may be transferred to other engine oil passages and then removed by the jacket coolant. When the engine jacket temperatures are much higher than the lubricant temperatures, the reverse process occurs, and the oil removes heat from the engine oil passages.

Determining the lubricant cooling effect is necessary in the design of heat exchangers and coolant systems. Heat is dissipated to the lubricant in a four-cycle engine with a high-temperature (225 to

250°F) jacket water coolant at a rate of about 7 or 8 Btu/min·bhp; oil heat is rejected in the same engine at 3 to 4 Btu/min·bhp. However, this engine uses more moderate (180°F) coolant temperatures for both lubricating oil and engine jacket.

The characteristics of each lubricant, engine, and application are different, and only periodic laboratory analysis of oil samples can establish optimum lubricant service periods. The following factors should be considered in selecting an engine:

- High-quality lubricating oils are generally required for operation at temperatures between 160 and 200°F, with longer oil life expected at lower temperatures. Moisture may condense in the crankcase if the oil is too cool, which reduces the useful life of the oil.
- Copper piping should be avoided in oil-side surfaces in oil coolers and heat exchangers to reduce the possibility of oil breakdown caused by contact with copper.
- A full-flow filter provides better security against oil contamination than one that filters only a portion of circulated lubricating oil and bypasses the rest.

COMPRESSED AIR SYSTEMS

Larger engines are frequently started with compressed air, either by direct cylinder injection or by air-driven motors. In large plants, one of the smallest of multiple compressors is usually engine driven for a "black start" (dead plant start-up). The same procedure is true for fuel oil systems when the main storage tank cannot gravity feed the day tank. However, the storage tanks must have the capacity for several starting procedures on any one engine, in case of repeated failure to start.

Another start-up concept eliminates all auxiliary engine drives and powers the motor-driven auxiliaries directly through a segregated circuit served by a portable, engine-driven emergency generator. This circuit can be sized for the black start power and control requirements as well as for emergency lighting and receptacles for power tools and welding devices. For a black start, or after any major damage causing a plant failure, this circuit can be used for required repairs at the plant and at other buildings in a given complex.

EXHAUST GAS SYSTEMS

The exhaust stream from the engine can (1) exit directly to the atmosphere through a silencer; (2) pass through a jacket water-cooled exhaust manifold; (3) drive a jacket water-cooled turbocharger; or (4) flow through a heat recovery/silencer device. The section on Reciprocating Engines under Heat Recovery on page 7.20 discusses these various methods of exhaust gas heat recovery.

SUPERCHARGERS

A supercharged engine is generally less costly than a larger, naturally aspirated engine for a given engine output. The turbocharger must match the engine to provide the required pressure ratio and mass flow under all conditions of engine operation, while staying out of the field of instability of any centrifugal blower.

Centrifugal blowers operate at 10,000 to 50,000 rpm to attain pressure ratios up to 3:1 in a single stage. Gear trains for these speeds are much more elaborate than for rotary blowers; therefore, such blowers are normally driven by small exhaust gas turbines called turbochargers.

Rotary blowers are positive displacement blowers using rotary lobes that mesh together with small clearances and have no direct contact with one another. They operate at 2000 to 6000 rpm and can be directly driven by the engine or by a separate motor drive.

Turbochargers may be air or water cooled, and they have an aftercooler on the discharge air side to avoid feeding hot, less dense air to the engine. Larger ones may have their own lubricating systems and oil cooler, while smaller ones operate with the main engine oil.

In two-cycle naturally aspirated engines, the turbocharger increases the output by about 50%, while in four-cycle engines the increase can be from 30% to 150% or more, depending on the pressure ratio. Aftercooling following compression can raise performance approximately another 17%.

The turbocharger extends the optimum fuel consumption curve because the usual limitation on larger gas-fueled engines is the volume of combustion air that can be inserted in the available combustion chamber. Many stationary reciprocating engines were developed for use with diesel fuel. The energy ratings with the liquid fuel are higher than with naturally aspirated gaseous fuel. Turbocharging on diesel service applies more air pressure in the cylinder so that larger quantities of the separately injected fuel can be burned efficiently.

In gaseous fuel systems, the fuel must always have a pressure high enough to enter the carburetor and mix with the combustion air, which is at a boosted pressure. Because the gas and air mixture ignite at a specific temperature-pressure relationship, the lower the inlet air temperature is, the higher the compression ratio can be before spontaneous combustion (preignition) occurs.

COMBUSTION AIR SYSTEMS

All internal combustion engines require clean, cool air for optimum performance. Highly humid air does not hurt performance, and it may even help by slowing combustion and reducing cylinder pressure and temperature. Provisions must be made to silence air noise, provide an adequate amount of air for combustion, and, in the case of two-cycle engines, provide enough air to scavenge the combusted fuel.

Smaller engines generally use engine-mounted impingement filters, often designed for some silencing, while larger engines commonly use a cyclone filter or various oil-bath filters. Selection considerations are (1) efficiency or dirt removal capacity; (2) airflow resistance (high intake pressure drop affects performance); (3) ease, frequency, and cost of cleaning or replacement; and (4) first cost. Many of the filter types and media commonly found in HVAC systems are used, but they are designed specifically for engine use. Air piping is designed for low pressure drop to maintain high engine performance. A low pressure drop is more important for naturally aspirated than for supercharged engines. For engine intakes, conventional velocities range from 3000 to 7200 fpm, governed by the engine manufacturer's recommended pressure drop of approximately 5.5 in. of water.

Evaporative coolers are sometimes used to cool the air before it enters the engine intake, while a recirculated water coolant recovery system is used for aftercooling.

INSTRUMENTS AND CONTROLS

Starting Systems

The start-stop control may include manual or automatic activation of the engine fuel supply, the engine cranking cycle, and establishment of the engine heat removal circuits. Stop circuits always shut off the fuel supply, and, for spark ignition engines, the ignition system is generally grounded as a precaution against incomplete fuel valve closing.

Alarm and Shutdown Controls

The prime mover is protected from malfunction by alarms that warn of unusual conditions and by safety shutdown under unsafe conditions. The control system must protect against failure of (1) speed control (underspeed or overspeed); (2) lubrication (low oil pressure, high oil temperature); (3) heat removal (high coolant temperature or lack of coolant flow); (4) combustion process (fuel, ignition); (5) lubricating oil level; and (6) water level.

Controls for alarms preceding shutdown are provided as needed. Monitored alarms without shutdown include lubricating oil and fuel filter, lubricating oil temperature, manifold temperature, jacket water temperature, etc. Automatic start-up of the standby engine when an alarm/shutdown sequence is triggered is often provided.

Both a low-lubrication pressure switch and a high jacket water temperature cutout are standard for most gas engines. Other safety controls used include (1) an engine speed governor, (2) ignition current failure shutdown (battery-type ignition only), and (3) the safety devices associated with a driven machine. These devices shut down the engine to protect it against mechanical damage. They do not necessarily shut off the gas fuel supply unless they are specifically set to do so.

Governors

A governor senses speed (and sometimes load), either directly or indirectly, and acts by means of linkages to control the flow of gas and air through engine carburetors or other fuel-metering devices to maintain a desired speed. Speed control with electronic, hydraulic, or pneumatic governors extends engine life by minimizing forces on engine parts, permits automatic throttle response without operator attention, and prevents destructive overspeeding. A separate overspeed device, sometimes called an overspeed trip, prevents runaway in the event of a failure that renders the governor inoperative. Both constant and variable engine speed controls are available. For constant speed, the governor is set at a fixed position, which can be reset manually.

Gas Leakage Prevention

The first method of avoiding gas leakage due to engine regulator failure is to install a solenoid shutdown valve with a positive cutoff either upstream or downstream of the engine regulator. The second method is a sealed combustion system that carries any leakage gas directly to the outdoors (i.e., all combustion air is ducted to the engine directly from the outdoors).

NOISE AND VIBRATION

Because engine exhaust must be muffled to reduce ambient noise levels, most recovery units also act as silencers. Figure 9 illustrates a typical noise curve. Figure 10 shows typical attenuation curves for various silencers. Table 34 in Chapter 47 of the *ASHRAE Handbook—Applications* lists acceptable noise level criteria for various applications. The section on Noise and Vibration Control on page 7.39 has further information.

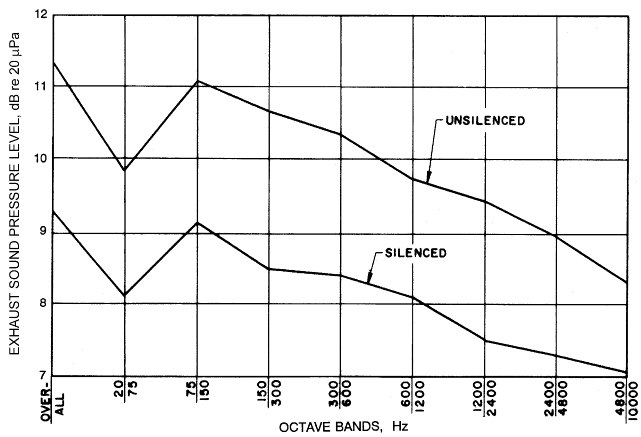

Fig. 9 Typical Reciprocating Engine Exhaust Noise Curves

MAINTENANCE

Engines require periodic servicing and replacement of parts, depending on usage and the type of engine. Transmission drives require periodic gearbox oil changes and the operation and care of external lubricating pumps. Log records should be kept of all servicing; checklists should be used for this purpose.

Table 2 shows ranges of typical maintenance routines for both diesel and gas-fired engines, based on the number of hours run. The actual intervals vary according to the cleanliness of the combustion

Table 2 Recommended Engine Maintenance

	Hours Between Procedures	
Procedure	Diesel Engine	Gas Engine
1. Take lubricating oil sample	Once per month plus once at each oil change	Once per month plus once at each oil change
2. Change lubricating oil filters	350 to 750	500 to 1000
3. Clean air cleaners, fuel	350 to 750	350 to 750
4. Clean fuel filters	500 to 750	n.a.
5. Change lubricating oil	500 to 1000	1000 to 2000
6. Clean crankcase breather	350 to 700	350 to 750
7. Adjust valves	1000 to 2000	1000 to 2000
8. Lubricate tachometer, fuel priming pump, and auxiliary drive bearings	1000 to 2000	1000 to 2000 (fuel pump n.a.)
9. Service ignition system; adjust breaker gap, timing, spark plug gap, and magneto	n.a.	1000 to 2000
10. Check transistorized magneto	n.a.	6000 to 8000
11. Flush lubrication oil piping system	3000 to 5000	3000 to 5000
12. Change air cleaner	2000 to 3000	2000 to 3000
13. Replace turbocharger seals and bearings	4000 to 8000	4000 to 8000
14. Replace piston rings, cylinder liners (if applicable), connecting rod bearings, and cylinder heads; recondition or replace turbochargers; replace gaskets and seals	8000 to 12,000	8000 to 12,000
15. Same as item 14, plus recondition or replace crankshaft; replace all bearings	24,000 to 36,000	24,000 to 36,000

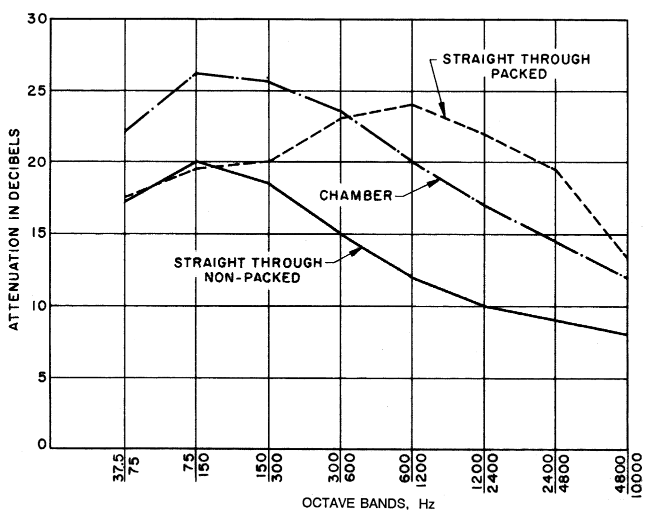

Fig. 10 Typical Attenuation Curves for Engine Silencers

air, the cleanliness of the engine room, the engine manufacturer's recommendations, the number of engine starts and stops, and lubricating conditions indicated by the oil analysis. With some engines and some operating conditions, the intervals between procedures listed in Table 2 may be extended.

A preventive maintenance program should include inspections for

- Leaks (a visual inspection, which is facilitated by a clean engine)
- Abnormal sounds and odors
- Unaccountable speed changes
- Condition of fuel and lubricating oil filters

Daily logs should be kept on all pertinent operating parameters, such as

- Water and lubricating oil temperatures
- Individual cylinder compression pressures, which are useful in indicating blowby
- Changes in valve tappet clearance, which indicate the extent of wear in the valve system

Lubricating oil analysis is a low-cost method of determining the physical condition of the engine and a guide to maintenance procedures. Commercial laboratories providing this service are widely available. The analysis should measure the concentration of various elements found in the lubricating oil, such as bearing metals, silicates, and calcium. It should also measure the dilution of the oil, suspended and nonsuspended solids, water, and oil viscosity. The laboratory can often assist in interpreting the readings and alert the user to impending problems.

Lubricating oil manufacturers' recommendations should be followed. Both the crankcase oil and oil filter elements should be changed at least once every six months.

COMBUSTION TURBINES

TYPES

Combustion gas turbines, although originally used for aircraft propulsion, have been developed for stationary use as prime movers. Turbines are available in sizes from 32 to 173,000 bhp and can burn a wide range of liquid and gaseous fuels. Turbines, and some dual-fuel engines, are capable of shifting from one fuel to another without loss of service.

Combustion turbines consist of an air compressor section to boost combustion air pressure, a combination fuel-air mixing and combustion chamber (combustor), and an expansion power turbine section that extracts energy from the combustion gases.

In addition to these components, some turbines use regenerators and recuperators as heat exchangers to preheat the combustion air entering the combustion chamber with heat from the turbine discharge gas, thereby increasing machine efficiency. Most turbines are the single-shaft type, (i.e., air compressor and turbine on a common shaft). However, split-shaft machines that use one turbine stage on the same shaft as the compressor and a separate power turbine driving the output shaft are available.

Turbines rotate at speeds varying from 3600 to 60,000 rpm and often need speed reduction gearboxes to obtain shaft speeds suitable for generators or other machinery. Turbine motion is completely rotary and relatively vibration-free. This feature, coupled with low mass and high power output, provides an advantage over reciprocating engines with regard to space, foundation requirements, and ease of start-up.

However, in smaller sizes, the combustion gas turbine has a heat rate efficiency of 12 to 30%. Larger turbines approach a heat rate efficiency above 35%, with advanced cycles above 50%.

Combustion turbines have the following advantages over other internal combustion engine drivers:

- Small size, high power-to-weight ratio
- Ability to burn a variety of fuels, though more limited than reciprocating engines
- Ability to meet stringent pollution standards
- High reliability
- Available in self-contained packages
- Instant power—no warm-up required
- No cooling water required
- Vibration-free operation
- Easy maintenance
- Low installation cost
- Clean, dry exhaust
- Lubricating oil not contaminated by combustion oil

Gas Turbine Cycle

The basic gas turbine cycle (Figure 11) is the **Brayton cycle (open cycle)**, which consists of adiabatic compression, constant pressure heating, and adiabatic expansion. Figure 11 shows that the thermal efficiency of a gas turbine falls below the ideal value because of inefficiencies in the compressor and turbine and because of duct losses. Increases in entropy occur during the compression and expansion processes, and the area enclosed by points 1, 2, 3, and 4 is reduced. This loss of area is a direct measure of the loss in efficiency of the cycle.

Nearly all turbine manufacturers present gas turbine engine performance in terms of power and specific fuel consumption. A comparison of fuel consumption in specific terms is the quickest way to compare overall thermal efficiencies of gas turbines.

Components

Figure 12 shows the major components of the gas turbine unit, which include the air compressor, the combustor, and the power turbine. Atmospheric air is compressed by the air compressor. Fuel is then injected into the airstream and ignited in the combustor, with the leaving gases attaining temperatures between 1800 and 2300°F. These high-pressure hot gases are then expanded through a turbine, which provides not only the power required by the air compressor, but also power to drive the load.

Gas turbines are available in two major classifications—single-shaft (Figure 12) and split-shaft (Figure 13). The **single-shaft turbine** has the air compressor, the gas-producer turbine, and the power turbine on the same shaft. The **split-shaft or dual-shaft turbine** has the section required for air compression on one shaft and the section producing output power on a separate shaft. For a split-shaft turbine, the portion that includes the compressor, combustion

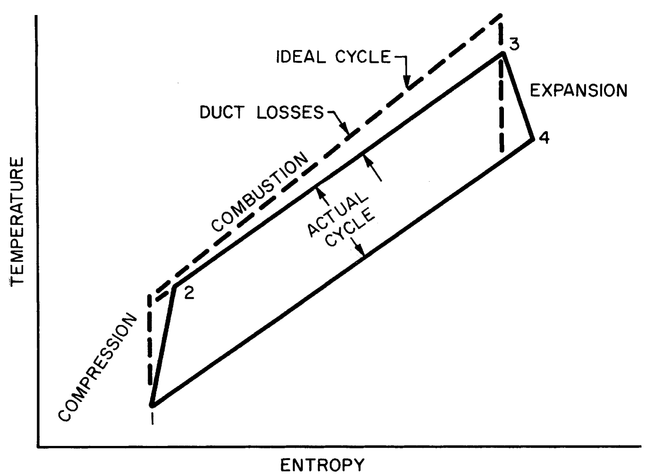

Fig. 11 Temperature-Entropy Diagram for Brayton Cycle

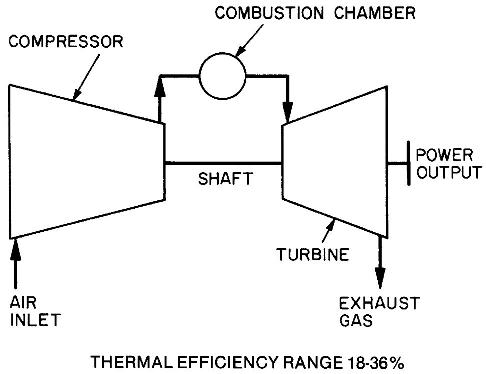

Fig. 12 Single-Shaft Turbine

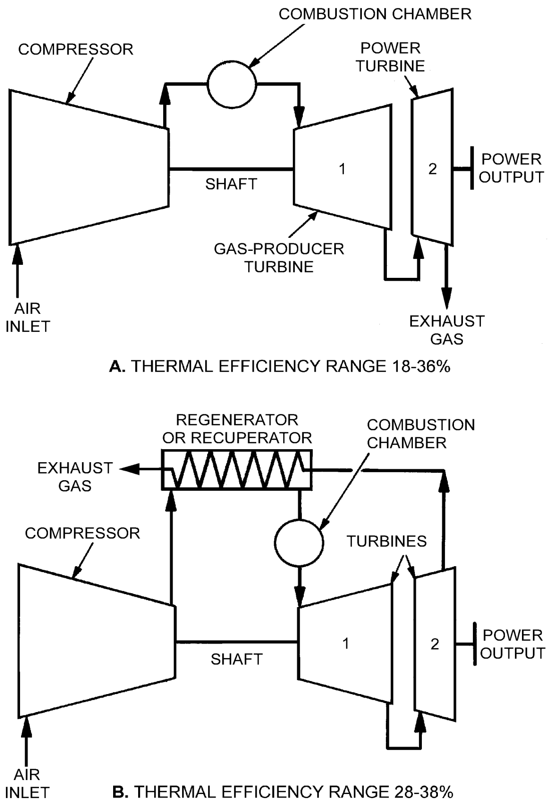

Fig. 13 Split-Shaft Turbines

chamber, and first turbine section is the hot gas generator or gas producer. The second turbine section is the power turbine.

The turbine used depends on job requirements. Single-shaft engines are usually selected when a constant-speed drive is required, as in generator drives, and when starting torque requirements are low. A single-shaft engine can be used to drive centrifugal compressors, but the starting system and the compressor match point must be considered. Split-shaft engines allow for variable speed at full load. Additional advantages of the split-shaft engine are that it can easily be started with a high torque load connected to the power output shaft, and the power turbine can be more optimally configured to match load requirements.

FUELS AND FUEL SYSTEMS

The ability to burn almost any combustible fluid is a key advantage of the gas turbine. Natural gas is the fuel of choice over other gaseous fuels because it is readily available, has good combustion characteristics, and is relatively easy to handle. A typical fuel gas control system is a two-stage system that uses pressure control in combination with flow control to achieve a turndown ratio of about 100:1. Other fuel gases include liquefied petroleum gases, which are considered "wet" gases because they can form condensables at normal gas turbine operating conditions; and a wide range of refinery waste and coal-derived gases, which have a relatively high fraction of hydrogen. Both of these features lead to problems in fuel handling and preparation, as well as in gas turbine operation. Heat tracing to heat the piping and jacketing of valves is required to prevent condensation at start-up. Piping runs should be designed to eliminate pockets where condensate might drop out and collect.

Distillate oil is the most common liquid fuel, and except for a few installations where natural gas is not available, it is primarily used as a backup and alternate start-up fuel. Crude oils are common as primary fuels in many oil-producing countries because of their abundance. Both crude and residual oils require treatment for sodium salts and vanadium contamination. The most common multiple-fuel combination is natural gas and distillate. The combustion turbine may be started on either fuel, and transfers from one fuel to the other may be initiated by the operator at any time after completion of the starting sequence without interrupting operation.

Steam and demineralized water injection are currently used in gas turbines for NO_x abatement in quantities up to 2% of compressor inlet airflow. An additional 3% steam may be injected independently at the compressor discharge for power augmentation. The required steam conditions are 300 to 350 psig and a temperature that is no more than 150°F above compressor discharge temperature, but not less than 50°F of superheat. Steam contaminants should be guarded against, and the steam supply system should be designed to supply dry steam under all operating conditions.

LUBRICATING OIL SYSTEMS

Lubricating systems provide filtered and cooled oil to the gas turbine, the driven equipment, and the gear reducer. Lubricating systems include a motor-driven, start-up/coast-down oil pump, a primary oil pump mounted on and driven by the gear reducer, filters, an oil reservoir, an oil cooler, and automatic controls. Along with lubricating the gas turbine bearings, gear reducer, and driven equipment bearings, the lubrication system provides hydraulic oil to the gas turbine control system.

The start-up/coast-down oil pump circulates oil until the gas turbine reaches a speed at which the primary pump can take over. Where emergency alternating current (ac) power is not available in case of lost outside power, an emergency direct current (dc) motor-driven pump may be required to provide lubricant during start-up and coast-down.

Oil filters serve the full flow of the pumps. Two filters are provided so that one can be changed while the other remains in operation.

The oil reservoir is mounted in the base of the gas turbine's supporting structure. Heater systems in the reservoir maintain the oil temperature above a minimum level. A combination mechanical/coalescer filter in the reservoir's vent removes oil from the vent. The oil cooler can be either water cooled or air cooled, depending on the availability of cooling water.

STARTING SYSTEMS

Starting systems can use pneumatic, hydraulic, or electric motor starters. A pneumatic starting system uses either compressed air or fuel gas to power a pneumatic starter motor or a starting subsystem integrated directly with the rotating components. A hydraulic system uses a hydraulic motor for starting. Hydraulic fluid is provided to the hydraulic motor by either an ac motor-driven pump or a diesel engine-driven pump. An electric motor system couples an electric

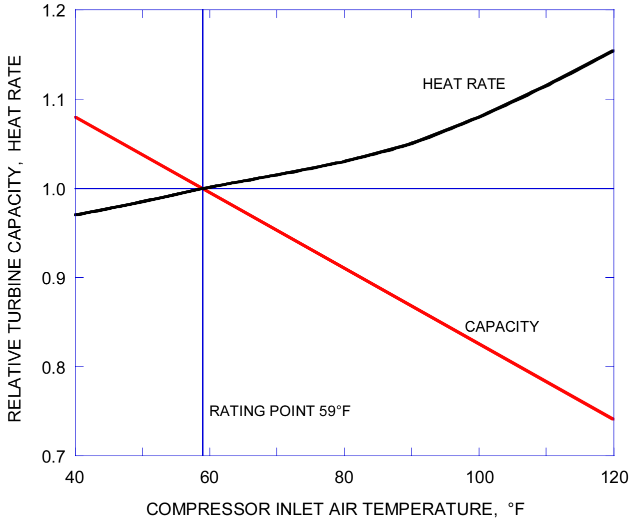

Fig. 14 Relative Turbine Power Output and Heat Rate Versus Inlet Air Temperature

motor directly to the gas engine for starting. All systems use a one-way clutch to couple the starter motor to the gas engine so that as the engine accelerates above the start speed, the starter can shut down. Black starts can be accomplished if a pneumatic or diesel/hydraulic starting system is used.

EXHAUST GAS SYSTEMS

Exhaust systems of gas turbines used in cogeneration systems consist of gas ducts, expansion joints, an exhaust silencer, a dump (or bypass) stack, and a diverter valve (or damper). The exhaust silencer is installed in a dump stack. The diverter valve is used to modulate the flow of exhaust gas into the heat recovery equipment or divert 100% of the exhaust gas to the dump stack when heat recovery is not required.

COMBUSTION TURBINE INLET AIR COOLING

Combustion turbine inlet air cooling (CTIAC) systems increase the capacity of turbine-generators by increasing the density of the combustion air. Since the volumetric flow to most turbines is constant, increasing the air density increases the mass flow rate. As the inlet air temperature increases, as on hot summer days, the capacity decreases (MacCracken 1994). Cooling inlet air increases the power and typically decreases the heat rate, after all parasitic cooling power usage is considered (Figure 14). Factors that affect CTIAC installation and operation include the turbine type, climate, hours of operation, ratio of airflow rate to power generated, ratio of generation increase with increased airflow, and monetary value of power generated.

Cooling Methods

Some CTIAC designs are for turbines that operate only a few hours per year, to demonstrate power reserve or provide peak demand power. **Peaking** combustion turbines are operated when utilities experience the greatest demand. Both inlet evaporative cooling systems and thermal energy storage (TES) systems allow CTIAC during turbine operation with no coincident parasitic power usage except for pumps. TES systems allow the use of small-capacity refrigeration systems, operated only during off-peak hours. Utilities and independent power producers may operate turbines at **baseload** and experience a need for continuous cooling for a significant number of hours per year. For turbines operating continuously or for several hours per day, fuel cost and availability are important factors that favor on-line cooling systems such as direct refrigeration without thermal storage.

The most prevalent CTIAC system is **evaporative cooling** using wetted media, due to low installation and operating costs. Ideal evaporative cooling occurs at a constant wet-bulb temperature, cooling the air to near 100% relative humidity. The typical evaporative cooling system allows the air-water vapor mixture to reach 85 to 95% of the difference between the dry-bulb air temperature and the wet-bulb temperature. Evaporative cooling can be used before or after indirect (secondary fluid) cooling. If a combination of sensible cooling (cooling coils) and evaporative cooling is used, sensible cooling should be used first, and then evaporative cooling, to reach the minimum temperature without latent cooling by the cooling coils.

Chilled water or direct refrigeration can also be used. These processes decrease the enthalpy (and temperature) of the air-water vapor mixture. The water vapor content (humidity ratio) remains constant as the mixture cools to near the dew-point temperature. Continued cooling follows the cooling coil performance curve, lowering the humidity ratio by forcing part of the water vapor to condense out from the mixture, while holding the mixture's relative humidity near 100%.

Chilled water systems can be used in conjunction with either ice or chilled water TES (Ebeling 1994). From a cost standpoint, the TES system is usually justified only for turbines that operate a few hours per week or to increase reserve power. The TES system typically has a higher capital cost and uses more energy than a direct refrigeration cooling system because of the secondary fluid loop, the pumping required, and the increased size of cooling coils. The chilled water system, however, requires less refrigerant piping and inventory and is therefore less susceptible to refrigerant leakage. Thermal storage allows the reduction of refrigeration equipment size and on-peak parasitic energy usage, which can decrease costs. Parasitic loads for a TES system that operates only a few hours per week usually do not severely affect the economic value of a CTIAC system.

A **direct** refrigerant cooling system consists of either a vapor compression system or an absorption system where the liquid refrigerant is used directly in air-cooling coils. The cooling process is identical to that of a chilled water system. A direct system can provide cooling during all hours of turbine operation but must be sized to meet the peak cooling required; therefore, it is larger than a TES system (Stewart 1997).

Advantages of CTIAC

Capacity Enhancement. Use of CTIAC for newer turbines, with lower airflow rates per unit of power generated, is even more economical than for older turbines. The lower flow rates require less cooling capacity to lower inlet air temperatures and therefore smaller evaporative coolers or refrigeration equipment, including TES systems.

Heat Rate Improvement. Fuel mass flow rates increase with inlet airflow and turbine output, but typically at a lower rate. CTIAC systems may be used primarily for decreased heat rate and corresponding fuel cost savings.

Turbine Life Extension. Turbines operating at lower inlet air temperatures have extended life and reduced maintenance. Lower and constant turbine inlet air temperatures reduce the wear on turbines and turbine components.

Increased Combined-Cycle Efficiency. Lower inlet air temperatures result in lower exhaust gas temperatures, potentially decreasing the capacity of the heat recovery steam generator to provide heat to steam turbines and absorption equipment. However, the greater airflow rate of a CTIAC system usually produces an overall increase in capacity because the effect of increased exhaust mass flow rate exceeds the effect of decreased temperature.

Delayed Capacity Addition. With the increased generation capacity provided by a CTIAC system, the addition of actual or reserve generation capacity can be delayed.

Baseload Efficiency Improvements. An ice or chilled water TES system can help level the baseload of a power generation facility by storing energy using electric chiller equipment during off-peak periods; this tends to increase the efficiency of power production. Electric chillers operated at cooler nighttime temperatures are more efficient and operate at reduced condenser temperatures, which can also use less source energy.

When maximum power is desired every hour of the year, a continuous CTIAC system is justified in warm climates to maximize turbine output and minimize heat rate.

Other Benefits. Other advantages include the following:

- Evaporative media filter the inlet air.
- CTIAC systems that reduce the air temperature below saturation can produce a significant amount of condensed water, a potentially valuable resource that can also provide makeup water for cooling towers or evaporative condensers.
- CTIAC systems are simple, energized only when required.
- Emissions can decrease due to increased overall efficiency.
- A CTIAC system can match the inlet air temperature to the required turbine generating capacity, allowing 100% open inlet guide vanes, which eliminate inlet guide vane pressure loss penalties.

Disadvantages

- CTIAC systems require additional space and increase maintenance.
- Evaporative media or cooling coils pose a constant inlet air pressure loss.

INSTRUMENTS AND CONTROLS

Control systems are typically microprocessor based. The control system sequences all systems during starting and running, monitors performance, and protects the equipment. The operator interface is a monitor and keyboard with analog gages provided for redundancy.

Where operating the gas turbine engine at the unit's maximum rating is desirable, the load is controlled based on the temperature of the combustion gases in the turbine section and on the ambient air temperature. When the engine combustion gas temperature reaches a set value, the control system begins to control the engine so that the load (and therefore the temperature) does not increase further. With changes in the ambient air temperature, the control system adjusts the load to maintain the set temperature value in the gas engine's turbine section. Where maintaining a constant load level is desirable, the control system allows the operator to dial in any load, and the system controls the engine accordingly.

PERFORMANCE CHARACTERISTICS

The rating of a gas turbine is greatly affected by altitude, ambient temperature, inlet pressure to the air compressor, and exhaust pressure from the turbine. In most applications, filters and silencers must be installed in the air inlet. Silencers, waste heat boilers, or both are used on the exhaust. The pressure drop of these accessories and piping losses must be considered when determining the power output of the unit.

Gas turbine ratings are usually given at standard conditions defined by the International Organization for Standardization (ISO): 59°F, 60% rh, and sea level pressure at the inlet flange of the air compressor and the exhaust flange of the turbine. Corrections for other conditions must be obtained from the manufacturer, as they vary with each model depending primarily on gas turbine efficiency. Inlet air cooling has been used to increase capacity. The following approximations may be used for design:

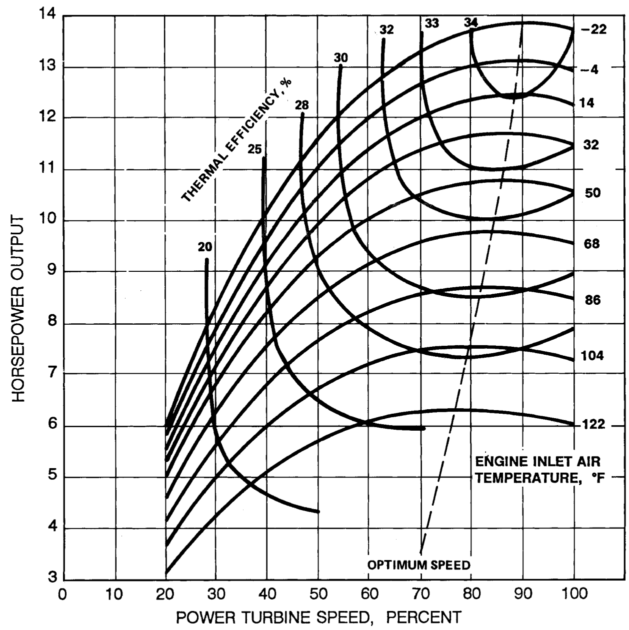

Fig. 15 Turbine Engine Performance Characteristics

- Each 18°F rise in inlet temperature decreases power output by 9%.
- An increase of 1000 ft in altitude decreases power output by approximately 3.5%.
- Inlet pressure loss in filter, silencer, and ducting decreases power output by approximately 0.5% for each inch of water pressure loss.
- Discharge pressure loss in boiler, silencer, and ducting decreases power output by approximately 0.3% for each inch of water pressure loss.

Gas turbines operate with a wide range of fuels. For refrigeration service, a natural gas system is usually provided with an option of a standby No. 1 or 2 grade fuel oil system.

Figure 15 shows a typical performance curve for a 10,000 hp turbine engine. For example, at an air inlet temperature of 86°F, the engine develops its maximum power at about 82% of maximum speed. The shaft thermal efficiency of the prime mover is 18 to 36% with exhaust gases from the turbine ranging from 806 to 986°F. If the exhaust heat can be used, overall thermal efficiency can increase.

Figure 13B shows a regenerator that uses the heat of the exhaust gases to heat the air from the compressor prior to combustion. Overall shaft efficiency can be increased to between 28 and 38% by using a regenerator or recuperator.

If process heat is required, the exhaust can satisfy a portion of that heat, and the combined system is a cogeneration system. The exhaust can be used (1) directly as a source of hot air, (2) in a large boiler or furnace as a source of preheated combustion air (the exhaust contains about 16% oxygen), or (3) to heat a process or working fluid such as the steam system shown in Figure 16. Overall thermal efficiency is [(shaft energy + heat energy) × 100] / (fuel energy). Thermal efficiencies of these systems vary from 50% to greater than 90%. The exhaust of a gas turbine has about 4000 to 8000 Btu/h of available heat per horsepower.

Additionally, because of the high oxygen content, the exhaust stream can support the combustion of an additional 30,000 Btu/h of fuel per horsepower. This additional heat can then be used in general manufacturing processes.

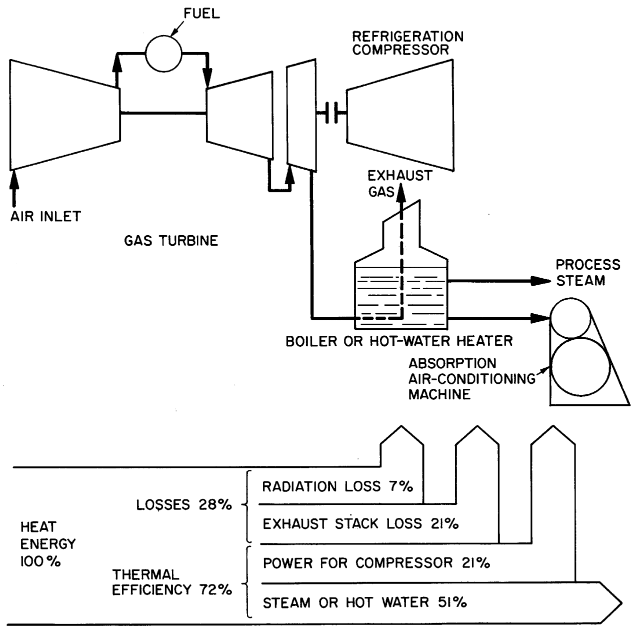

Fig. 16 Gas Turbine Refrigeration System Using Exhaust Heat

EMISSIONS AND NOISE

Emissions. Gas turbine power plants emit relatively low levels of CO_x and NO_x compared to other internal combustion engines; however, for each application, the gas turbine manufacturer should be consulted to ensure that applicable codes are met. Special care should be taken if high-sulfur fuel is being used because gas turbine exhaust stacks are typically not high, and dilution is not possible.

Noise. Gas turbine manufacturers have developed sound-attenuated enclosures that cover the turbine and gear package. Turbine drivers, when properly installed with a sound-attenuated enclosure, an inlet silencer, and an exhaust silencer, meet the strictest noise standards. The turbine manufacturer should be consulted for detailed noise level data and recommendations on the least expensive method of attenuation for a particular installation.

MAINTENANCE

Industrial gas turbines are designed to operate for 12,000 to 30,000 h between overhauls, with normal maintenance. Normal maintenance includes checking filters and oil level, inspecting for leaks, and so forth, all of which can be done by the operator with ordinary mechanics' tools. However, factory-trained service personnel are required to inspect engine components such as combustors and nozzles. These inspections, depending on the manufacturer's recommendations, are required as frequently as every 4000 h of operation.

Most gas turbines are maintained by condition monitoring and inspection rather than by specific overhaul intervals (called predictive maintenance). Gas turbines specifically designed for industrial applications may have an indefinite life for the major housings, shafts, and low-temperature components. Hot-section repair intervals for combustor and turbine components can vary from 10,000 to 100,000 h. The total cost of maintaining a gas turbine includes (1) cost of operator time, (2) normal parts replacement, (3) lubricating oil, (4) filter changes (combustion inlet air, fuel, and lubricating oil), (5) overhauls, and (6) factory service time (to conduct engine inspections). The cost of all these items can be estimated by the manufacturer and must be taken into account to determine the total operating cost.

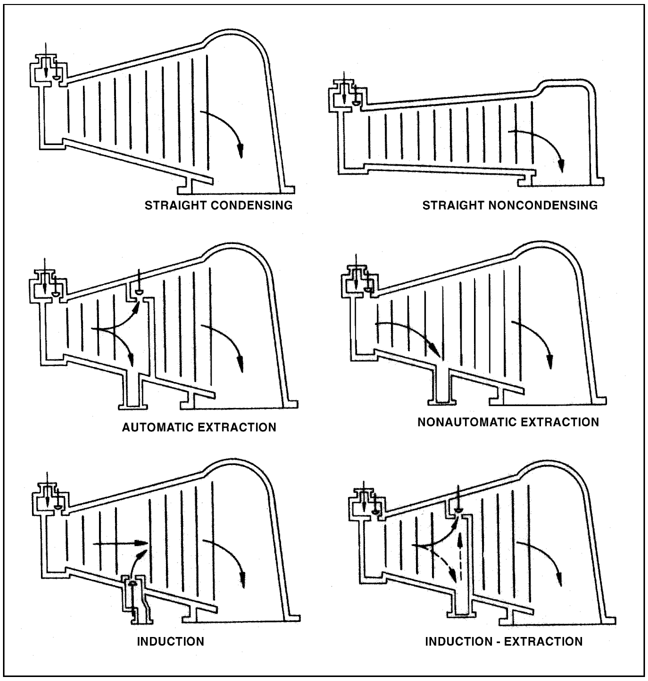

Fig. 17 Basic Types of Axial Flow Turbines

STEAM TURBINES

TYPES

Axial Flow Turbines

Conventional axial flow steam turbines direct steam axially through the peripheral blades of one or more staged turbine wheels (much like a pinwheel) one after another on the same shaft. Figure 17 shows basic types of axial turbines. NEMA *Standard* SM 24 defines these and further subdivisions of their basic families as follows:

Noncondensing (back pressure) turbine. A steam turbine designed to operate with an exhaust steam pressure at any level that may be required by a downstream process, where all condensing takes place.

Condensing turbine. A steam turbine with an exhaust steam pressure below atmospheric pressure, such that steam is directly and completely condensed.

Automatic extraction turbine. A steam turbine that has opening(s) in the turbine casing for the extraction of steam and means for directly regulating the extraction steam pressure.

Nonautomatic extraction turbine. A steam turbine that has opening(s) in the turbine casing for the extraction of steam without a means for controlling its pressure.

Induction (mixed pressure) turbine. A steam turbine with separate inlets for steam at two pressures, with an automatic device for controlling the pressure of the secondary steam induced into the turbine and means for directly regulating the flow of steam to the turbine stages below the induction opening.

Induction-extraction turbine. A steam turbine with the capability of either exhausting or admitting a supplemental flow of steam through an intermediate port in the casing, thereby maintaining a process heat balance. Turbines of the extraction and induction-extraction type may have several casing openings, each passing steam at a different pressure.

The necessary rotative force for shaft power in a turbine may be imposed on the turbine through the velocity of the steam, the pressure energy of the steam, or both. If velocity energy is used, the movable wheels are usually fitted with crescent-shaped blades. A row of fixed nozzles in the steam chest increases the steam velocity into the blades with little or no steam pressure drop across them and causes wheel rotation. Such combinations of nozzles and velocity-powered wheels are characteristic of an **impulse turbine**.

A **reaction turbine** uses alternate rows of fixed and moving blades generally of an airfoil shape. Steam velocity increases in the fixed nozzles and drops in the movable ones, while the steam pressure drops through both.

The power capability of a reaction turbine is maximum when the moving blades travel at about the velocity of the steam passing through them; in the impulse turbine, maximum power is produced with a blade velocity of about 50% of steam velocity. Steam velocity is related directly to pressure drop. To achieve the desired relationship between steam velocity and blade velocity without resorting to large wheel diameters or high rotative speeds, most turbines include a series of impulse or reaction stages or both, thus dividing the total steam pressure drop into manageable increments. A typical commercial turbine may have two initial rows of rotating impulse blading with an intervening stationary row (called a Curtis stage), followed by several alternating rows of fixed and movable blading of either the impulse or the reaction type. Most multistage turbines use some degree of reaction.

Construction. Turbine manufacturers' standards prescribe casing materials for various limits of steam pressure and temperature. The choice between built-up or solid rotors depends on turbine speed or inlet steam temperature. Water must drain from pockets within the turbine casing to prevent damage caused by condensate accumulation. Carbon rings or closely fitted labyrinths prevent leakage of steam between pressure stages of the turbine, outward steam leakage, and inward air leakage at the turbine glands. The erosive and corrosive effect of moisture entering with the supply steam must be considered. Heat loss is controlled by installation (often at the manufacturer's plant) of thermal insulation and protective metal jacketing on the hotter portions of the turbine casing.

Radial Inflow Turbines

Radial inflow turbines have a radically different configuration from axial flow machines. Steam enters through the center or eye of the impeller and exits from the periphery, much like the path of fluid through a compressor or pump; but in this case the steam actuates the wheel, instead of the wheel actuating the air or the water.

Radial, multistage arrangements comprise separate, single-stage wheels connected with integral reduction gearing in a factory-assembled package. Induction, extraction, and moisture elimination are accomplished in the piping between stages, giving the radial turbine a greater tolerance of condensate.

LUBRICATION SYSTEMS

Small turbines often have only simple oil rings to handle bearing lubrication, but most turbines for cogeneration service have a complete pressure lubrication system. Basic components include a shaft-driven oil pump, an oil filter, an oil cooler, a means of regulating oil pressure, a reservoir, and interconnecting piping. Turbines having a hydraulic governor may use oil from the lubrication circuit or, with certain types of governors, use a self-contained governor oil system. To ensure an adequate supply of oil to bearings during acceleration and deceleration periods, many turbines include an auxiliary motor or turbine-driven oil pump. Oil pressure-sensing devices act in two ways: (1) to stop the auxiliary pump once the shaft-driven pump has attained proper flow and pressure or (2) to start the auxiliary pump if the shaft-driven pump fails or loses pressure when decelerating. In some industrial applications, the lubrication systems of the turbine and the driven compressor are integrated. Proper oil pressure, temperature, and compatibility of the lubricant qualities must be maintained.

INSTRUMENTS AND CONTROLS

Starting Systems

Unlike reciprocating engines and combustion turbines, steam turbines do not require auxiliary starting systems. Steam turbines are started through controlled opening of the main steam valve, which is in turn controlled by the turbine governing system. Larger turbines with multiple stages and/or split shafting arrangements are started in a gradual manner to allow for controlled expansion and thermal stressing. Many of these turbines are provided with electrically powered turning gears that slowly rotate the shaft(s) during the initial stages of start-up.

Governing Systems

The wide variety of available governing systems permits the selection of a governor ideally matched to the characteristics of the driven machine and the load profiles. The principal and most common function of a fixed-speed steam turbine governing system is to maintain constant turbine speed despite load fluctuations or minor variations in supply steam pressure. This arrangement assumes that close control of the output of the driven component, such as a generator in a power plant, is primary to plant operation and that the generator can adjust its capacity to varying loads.

Often it is desirable to vary the turbine speed in response to an external signal. In centrifugal water chilling systems, for example, reduced speed generally reduces steam rate at partial load. An electric, electronic, or pneumatic device responds to the system load or the temperature of the fluid leaving the water chilling heat exchanger (evaporator). To avoid compressor surge and to optimize the steam rate, the speed is controlled initially down to some part load, then controlled in conjunction with the compressor's built-in capacity control (e.g., inlet vanes).

Process applications frequently require placing an external signal on the turbine governing system to reset the speed control point. Such external signals may be necessary to maintain a fixed compressor discharge pressure, regardless of load or condenser water temperature variations. Plants relying on a closely maintained heat balance may control turbine speed to maintain an optimum pressure level of steam entering, being extracted from, or exhausting from the turbine. One example is the combination turbine absorption plant, where control of pressure of the steam exhausting from the turbine (and feeding the absorption unit) is an integral part of the plant control system.

Components. The steam turbine governing system consists of (1) a speed governor (mechanical, hydraulic, electrical, or electronic); (2) a speed control mechanism (relays, servomotors, pressure- or power-amplifying devices, levers, and linkages); (3) governor-controlled valve(s); (4) a speed changer; and (5) external control devices, as required.

The **speed governor** responds directly to turbine speed and initiates action of the other parts of the governing system. The simplest speed governor is the direct-acting flyball, which depends on changes in centrifugal force for proper action. Capable of adjusting speeds through an approximate 20% range, it is widely used on single-stage, mechanical-drive steam turbines with speeds of up to 5000 rpm and steam pressure of up to 600 psig.

The speed governor used most frequently on centrifugal water chilling system turbines is the oil pump type. In its direct-acting form, oil pressure, produced by a pump either directly mounted on the turbine shaft or in some form responsive to turbine speed, actuates the inlet steam valve.

The oil relay hydraulic governor, as shown in Figure 18, has greater sensitivity and effective force. Here, the speed-induced oil pressure changes are amplified in a **servomotor** or **pilot-valve relay** to produce the motive effort required to reposition the steam inlet valve or valves.

The least expensive turbine has a single governor-controlled steam admission **throttle valve**, perhaps augmented by one or more small auxiliary valves (usually manually operated), which close off nozzles supplying the turbine steam chest for better part-load efficiency. Figure 19 shows the effect of auxiliary valves on part-load turbine performance.

For more precise speed governing and maximum efficiency without manual valve adjustment, multiple automatic nozzle control is used (Figure 20). Its principal application is in larger turbines where a single governor-controlled steam admission valve would be too large to permit sensitive control. The greater power required to actuate the multiple-valve mechanism dictates the use of hydraulic servomotors. **Speed changers** adjust the setting of the governing systems while the turbine is in operation. Usually, they comprise either a means of changing spring tension or a means of regulating oil flow by a needle valve. The upper limit of a speed changer's capability should not exceed the rated turbine speed. Such speed changers, while usually mounted on the turbine, may sometimes be remotely located at a central control point.

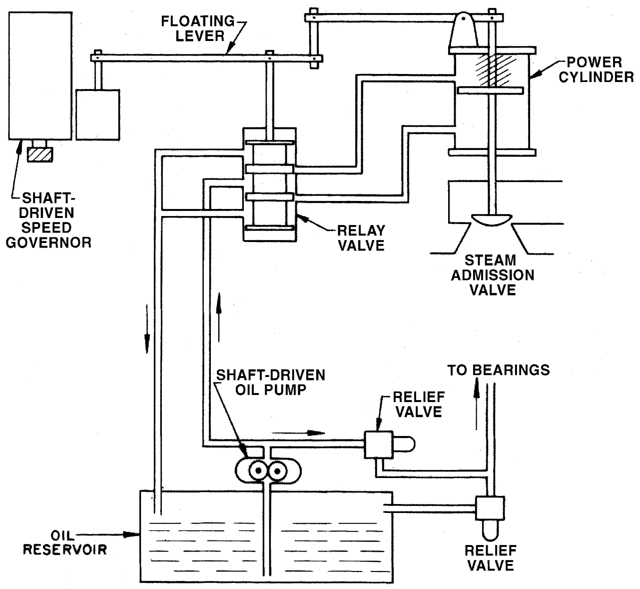

Fig. 18 Oil Relay Governor

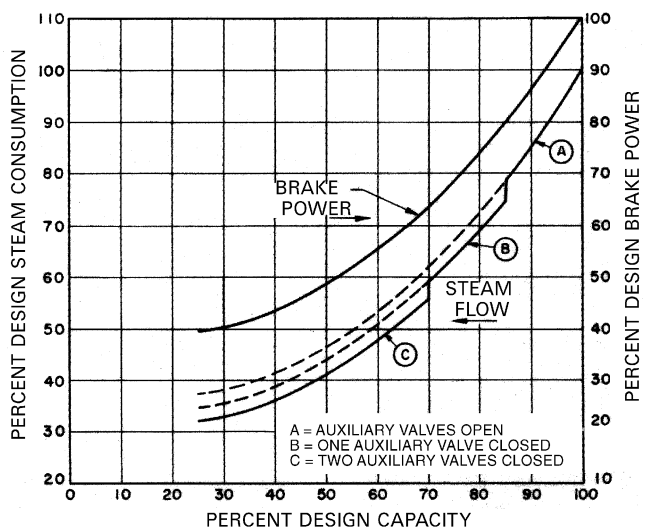

Fig. 19 Part-Load Turbine Performance Showing Effect of Auxiliary Valves

As stated previously, **external control devices** are often used when some function other than turbine speed is controlled. In such cases, a signal overrides the turbine speed governor's action, and the latter assumes a speed-limiting function. The external signal controls steam admission either by direct inlet valve positioning or by adjustment of the speed governor setting. The valve-positioning method either exerts mechanical force on the valve-positioning mechanism or, if power has to be amplified, regulates the pilot valve in a hydraulic servomotor system.

Where more precise control is required, the speed governor adjusting method is preferred. Although the external signal continually resets the governor as required, the speed governor always provides ideal turbine speed control. Thus, it maintains the particular set speed, regardless of load or steam pressure variations.

Classification. The National Electrical Manufacturers Association (NEMA) classifies steam turbine governors as shown in Table 3. **Range of speed changer adjustment**, expressed as a percentage of rated speed, is the range through which the turbine speed may be adjusted downward from rated speed by the speed changer, with the turbine operating under the control of the speed governor and

Table 3 NEMA Classification of Speed Governors

Class of Governor	Range of Speed Changer Adjustment, %	Maximum Steady-State Speed Regulation, %	Maximum Speed Variation, % Plus or Minus	Maximum Speed Rise, %	Trip Speed, % Above Rated Speed
A	10 to 65	10	0.75	13	15
B	10 to 80	6	0.50	7	10
C	10 to 80	4	0.25	7	10
D	10 to 90	0.50	0.25	7	10

Source: NEMA *Standard* SM 24.

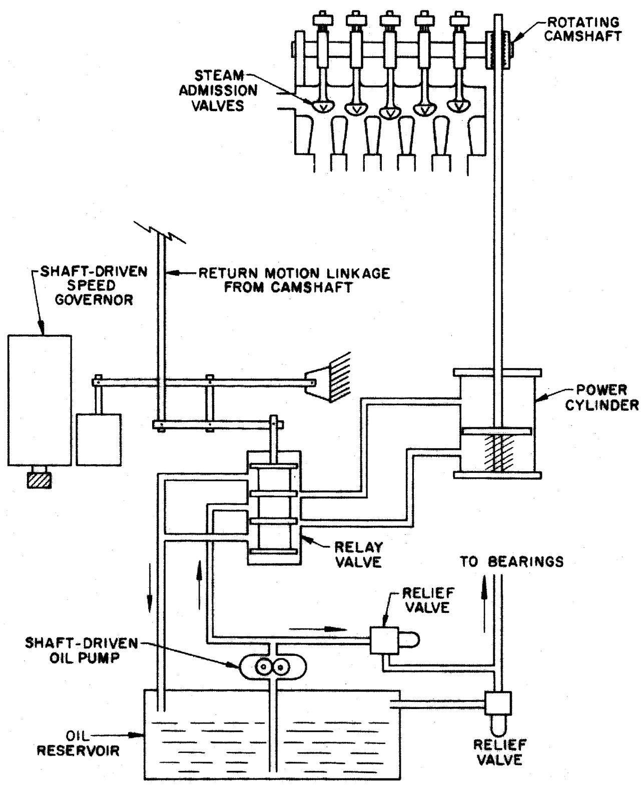

Fig. 20 Multivalve Oil Relay Governor

passing a steam flow equal to the flow at rated power, output, and speed. The range of the speed changer adjustment, expressed as a percentage of rated speed, is derived from the following equation:

$$\text{Range (\%)} = \frac{\begin{pmatrix}\text{Rated}\\\text{speed}\end{pmatrix} - \begin{pmatrix}\text{Minimum}\\\text{speed setting}\end{pmatrix}}{\text{Rated speed}} \times 100$$

Steady-state speed regulation, expressed as a percentage of rated speed, is the change in sustained speed when the power output of the turbine is gradually changed from rated power output to zero power output under the following conditions:

- Steam conditions (initial pressure, initial temperature, and exhaust pressure) are set at rated values and held constant.
- Speed changer is adjusted to give rated speed with rated power output.
- Any external control device is rendered inoperative and blocked in the open position to allow the free flow of steam to the governor-controlled valve(s).

The steady-state speed regulation, expressed as a percentage of rated speed, is derived from the following equation:

$$\text{Regulation (\%)} = \frac{\begin{pmatrix}\text{Speed at zero}\\\text{power output}\end{pmatrix} - \begin{pmatrix}\text{Speed at rated}\\\text{power output}\end{pmatrix}}{\text{Speed at rated power output}} \times 100$$

Steady-state speed regulation of automatic extraction or mixed pressure turbines is derived with zero extraction or induction flow and with the pressure-regulating system(s) inoperative and blocked in the position corresponding to rated extraction or induction pressure(s) at rated power output.

Speed variation, expressed as a percentage of rated speed, is the total magnitude of speed change or fluctuations from the speed setting. It is defined as the difference in speed variation between the governing system in operation and the governing system blocked to be inoperative, with all other conditions constant. This characteristic includes dead band and sustained oscillations. Expressed as a percentage of rated speed, the speed variation is derived from the following equation:

$$\begin{matrix}\text{\% Speed}\\\text{Variation}\end{matrix} = \frac{\begin{pmatrix}\text{Speed change}\\\text{above set speed}\end{pmatrix} - \begin{pmatrix}\text{Speed change}\\\text{below set speed}\end{pmatrix}}{\text{Rated speed}} \times 100$$

Dead band, also called wander, is a characteristic of the speed-governing system. It is the insensitivity of the speed-governing system and the total speed change during which the governing valve(s) do not change position to compensate for the speed change.

Stability is a measure of the ability of the speed-governing system to position the governor-controlled valve(s); thus, sustained oscillations of speed are not produced during a sustained load demand or following a change to a new load demand. Speed oscillations, also called hunt, are characteristics of the speed-governing system. The ability of a governing system to keep sustained oscillations to a minimum is measured by its stability.

Maximum speed rise, expressed as a percentage of rated speed, is the maximum momentary increase in speed obtained when the turbine is developing rated power output at rated speed and the load is suddenly and completely reduced to zero. The maximum speed rise, expressed as a percentage of rated speed, is derived from the following equation:

$$\text{Speed rise (\%)} = \frac{\begin{pmatrix}\text{Maximum speed at}\\\text{zero power output}\end{pmatrix} - \begin{pmatrix}\text{Rated}\\\text{speed}\end{pmatrix}}{\text{Rated speed}} \times 100$$

Protective Devices

In addition to speed-governing controls, certain safety devices are required on steam turbines. These include an overspeed mechanism, which acts through a quick-tripping valve independent of the main governor valve to shut off the steam supply to the turbine, and a pressure relief valve in the turbine casing. Overspeed trip devices may act directly, through linkages to close the steam valve, or hydraulically, by relieving oil pressure to allow the valve to close. Also, the turbine must shut down if other safety devices, such as oil pressure failure controls or any of the driven system's protective controls, so dictate. These devices usually act through an electrical interconnection to close the turbine trip valve mechanically or hydraulically. To shorten the coast-down time of a tripped condensing turbine, a vacuum breaker in the turbine exhaust opens to admit air on receiving the trip signal.

PERFORMANCE CHARACTERISTICS

The topping cogeneration cycle steam turbine is typically either a back pressure or an extraction condensing type that makes downstream, low-pressure thermal energy available for process use. Bottoming cycles commonly use condensing turbines because these yield more power, having lower grade throttle energy to begin with.

The highest steam plant efficiency is obtainable with a back pressure turbine when 100% of its exhaust steam is used for thermal process. The only inefficiencies are the gear drive, alternator, and inherent steam-generating losses. A large steam system topping cycle using an efficient water-tube boiler, economizer, and preheater can easily achieve an overall efficiency (fuel to end use) of more than 90%.

Full condensing turbine heat rates (Btu/hp·h) are the highest in the various steam cycles because the turbine's exhaust condenses, rejecting the latent heat of condensation (1036 Btu/lb at a condensing pressure of 1 psia and 101°F) to a waste heat sink (e.g., a cooling tower or river).

Conversely, the incremental heat that must be added to a low-pressure [e.g., 15 psig (30 psia)] steam flow to produce high-pressure, superheated steam for a topping cycle is only a small percentage of its latent heat of vaporization. For example, to produce 250°F, 30 psia saturated steam for a single-stage absorption chiller in a low-pressure boiler requires 1164 Btu/lb, of which 945 Btu/lb is the heat of vaporization. To boost this to 600°F, 320 psia requires an additional 146 Btu/lb, which is only 15% of the latent heat at 30 psia, for an enthalpy of 1310 Btu/lb.

A low-pressure boiler generating 30 psia steam to the absorber, directly, has a 75% fuel-to-steam efficiency, which is 15% lower than the 90% efficiency of a high-pressure boiler used in the cogeneration cycle. Therefore, from the standpoint of fuel cost, the power generated by the back pressure turbine is virtually free when its 30 psia exhaust is discharged into the absorption chiller.

The potential power-generating capacity and size of the required turbine are determined by its efficiency and steam rate (or water rate). This capacity is, in turn, the system's maximum steam load, if the turbine is sized to satisfy this demand. Efficiencies range from 55 to 80% and are the ratio of actual to theoretical steam rate, or actual to theoretical enthalpy drop from throttle to exhaust conditions.

NEMA *Standard* SM 24 defines theoretical steam rate as the quantity of steam per unit of power required by an ideal Rankine cycle, which is an isentropic or reversible adiabatic process of expansion. This can best be seen graphically on an enthalpy-entropy (Mollier) chart. Expressed algebraically, the steam rate is

$$w_t = \frac{2546}{h_i - h_e} \tag{1}$$

where

w_t = theoretical steam rate, lb/hp·h
h_e = enthalpy of steam at exhaust pressure and inlet entropy, Btu/lb
h_i = enthalpy of steam at throttle inlet pressure and temperature, Btu/lb
2546 = Btu/hp·h

This isentropic expansion through the turbine represents 100% conversion efficiency of heat energy to power. An example is shown on the Mollier chart in Figure 21 as the vertical line from 320 psia, 600°F, 1310 Btu/lb to 30 psia, 250°F, 93% quality, 1100 Btu/lb.

On the other hand, zero efficiency is a throttling, adiabatic, nonreversible horizontal line terminating at 30 psia, 552°F, 1310 Btu/lb. An actual turbine process would lie between 0 and 100% efficiency, such as the one shown at h_a actual exhaust condition of saturated steam at 30 psia, 250°F, 1164 Btu/lb; the actual turbine efficiency is

$$E_a = \frac{h_i - h_a}{h_i - h_e} \tag{2}$$

and the actual steam rate is

$$w_a = \frac{3412}{h_i - h_a} \tag{3}$$

where

w_a = actual steam rate, lb/kWh
h_a = enthalpy of steam at actual exhaust conditions, Btu/lb
3412 = Btu/kWh

For the case described,

$$E_a = (1310 - 1164)/(1310 - 1100) = 0.69 \text{ or } 69\%$$

As a cogeneration cycle, if the previously described absorption chiller has a capacity of 2500 tons, which requires 45,000 lb/h of 30 psia saturated steam, it can be provided by the 69% efficient turbine at an actual steam rate of 3412/146 = 23.4 lb/kWh; the potential power generation is

$$45{,}000/23.4 = 1923 \text{ kWh}$$

The incremental turbine heat rate to generate this power is only

$$(146 \times 45{,}000)/1923 = 3416 \text{ Btu/kWh}$$

instead of a typical 9000 Btu/kWh (thermal efficiency of 38%) for an efficient steam power plant with full condensing turbines and cooling towers.

Turbine performance tests should be conducted in accordance with the appropriate American Society of Mechanical Engineers (ASME) *Performance Test Code*: PTC 6, PTC 6S Report, or PTC 6A. The steam rate of a turbine is reduced with higher turbine speeds, a greater number of stages, larger turbine size, and a higher difference in heat content between entering and leaving steam

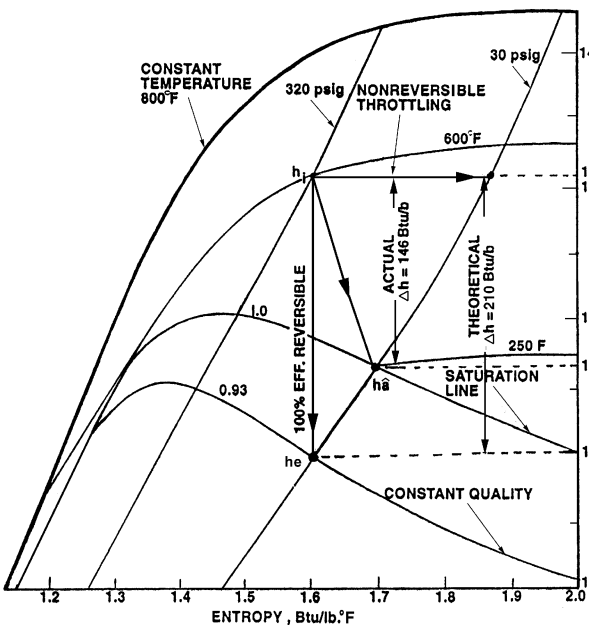

Fig. 21 Isentropic Versus Actual Turbine Process

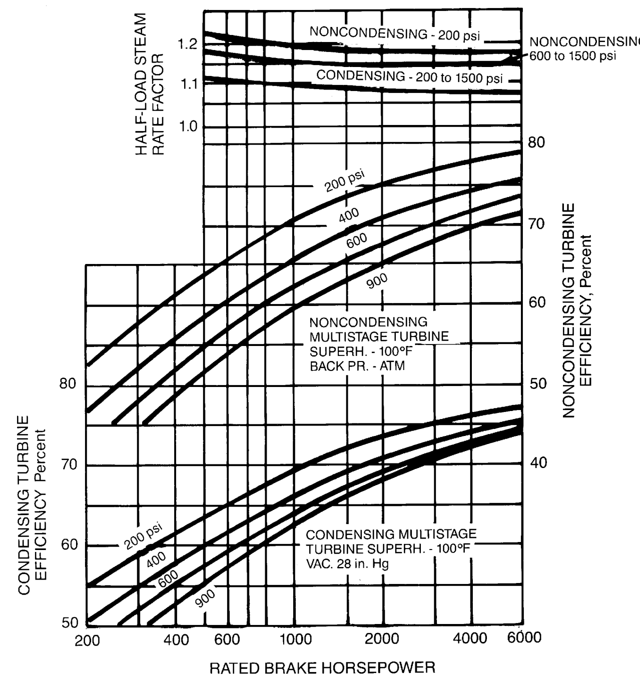

Fig. 22 Efficiency of Typical Multistage Turbines

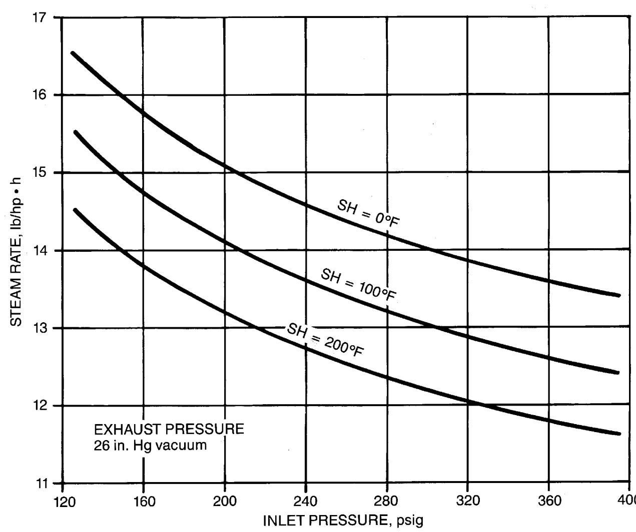

Fig. 23 Effect of Inlet Pressure and Superheat on Condensing Turbine

conditions. Often, one or more of these factors can be improved with only a nominal increase in initial capital cost. Cogeneration applications range, with equal flow turbines, from approximately 100 to 10,000 hp and from 3000 to 10,000 rpm, with the higher speeds generally associated with lower power outputs, and lower speeds with higher power outputs. (Some typical characteristics of turbines driving centrifugal water chillers are shown in Figure 15 and Figure 22 through Figure 26.)

While initial steam pressures for small turbines commonly fall in the 100 to 250 psig range, wide variations are possible. Turbines in the range of 2000 hp and above commonly have throttle pressures of 400 psig or greater.

Back pressure associated with noncondensing turbines generally ranges from 50 psig to atmospheric, depending on the use for the exhaust steam. Raising the initial steam temperature by superheating improves steam rates.

NEMA *Standards* SM 23 and SM 24 govern allowable deviations from design steam pressures and temperatures. Because of possible unpredictable variations in steam conditions and load requirements, turbines are selected for a power capability of 105 to 110% of design shaft output and speed capabilities of 105% of design rpm.

Because no rigid standards prevail for the turbine inlet steam pressure and temperature, fixed design conditions proposed by ASME/IEEE should be used to size the steam system initially. These values are 400 psig at 750°F, 600 psig at 825°F, 850 psig at 900°F, and 1250 psig at 950 or 1000°F.

Table 4 lists theoretical steam rates for steam turbines at common conditions. If project conditions dictate different throttle/exhaust conditions from the steam tables, theoretical steam rate tables or graphical Mollier chart analysis may be used.

Steam rates for multistage turbines depend on many variables and require extensive computation. Manufacturers provide simple tables and graphs to estimate performance, and these data are good guides for preliminary sizing of turbines and associated auxiliaries for the complete system.

If the entire exhaust steam flow from a base-loaded back pressure turbine is fully used, the maximum efficiency of a steam turbine cogeneration cycle is achieved. However, if the facility's thermal/electrical load cannot absorb the fully loaded output of the turbine, whichever profile is lower can be tracked, and the reduced power output or steam flow has to be accepted unless the output remaining is exported. The annual efficiency can still be high if the machine operates at significant combined loads for substantial periods. Straight steam condensing turbines offer no opportunity for topping cycles but are not unusual in bottoming cycles because the waste steam from the process can be most efficiently used by full-condensing turbines when there is no other use for low-pressure steam. Either back pressure or extraction condensing turbines may be used as extraction turbines.

The steam in an extraction turbine expands part of the way through the turbine until the pressure and temperature required by the external thermal load are attained. The remaining steam continues through the low-pressure turbine stages; however, it is easier to adjust for noncoincident electrical and thermal loads.

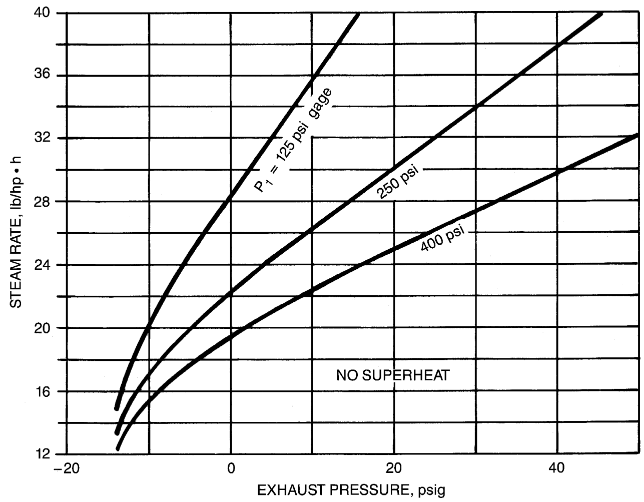

Fig. 24 Effect of Exhaust Pressure on Noncondensing Turbine

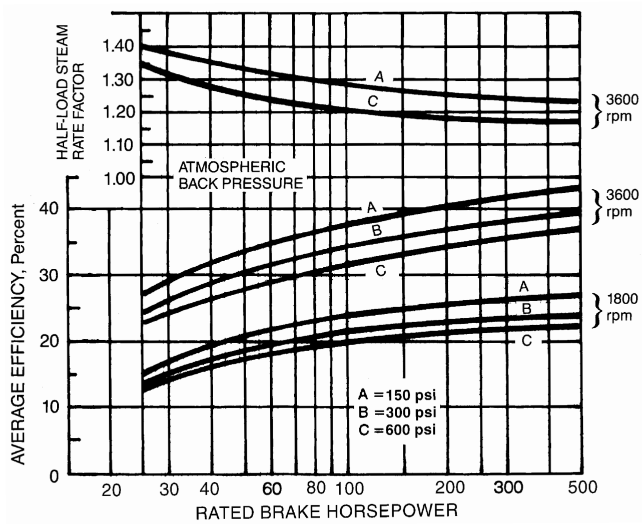

Fig. 25 Single-Stage Noncondensing Turbine Efficiency

Table 4 Theoretical Steam Rates for Steam Turbines at Common Conditions, lb/kWh

	Throttle Steam Conditions							
Exhaust Pressure	150 psig, 366°F, Saturated	200 psig, 388°F, Saturated	250 psig, 500°F, 94°F Superheat	400 psig, 750°F, 302°F Superheat	600 psig, 750°F, 261°F Superheat	600 psig, 825°F, 336°F Superheat	850 psig, 825°F, 298°F Superheat	850 psig, 900°F, 373°F Superheat
2 in. Hg (abs.)	10.52	10.01	9.07	7.37	7.09	6.77	6.58	6.28
4 in. Hg (abs.)	11.76	11.12	10.00	7.99	7.65	7.28	7.06	6.73
0 psig	19.37	17.51	15.16	11.20	10.40	9.82	9.31	8.81
10 psig	23.96	21.09	17.90	12.72	11.64	10.96	10.29	9.71
30 psig	33.6	28.05	22.94	15.23	13.62	12.75	11.80	11.07
50 psig	46.0	36.0	28.20	17.57	15.36	14.31	13.07	12.21
60 psig	53.9	40.4	31.10	18.75	16.19	15.05	13.66	12.74
70 psig	63.5	45.6	34.1	19.96	17.00	15.79	14.22	13.25
75 psig	69.3	48.5	35.8	20.59	17.40	16.17	14.50	13.51

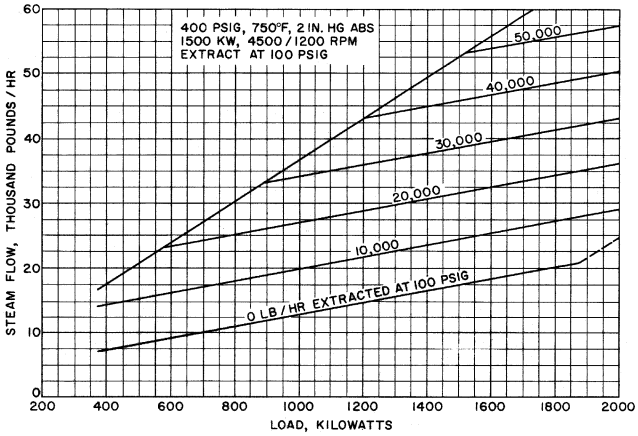

Fig. 26 Effect of Extraction Rate on Condensing Turbine

Because steam cycles operate at pressures exceeding those allowed by ASME and local codes for unattended operation, their use in cogeneration plants is limited to large systems where attendants are required for other reasons or the labor burden of operating personnel does not seriously affect overall economics.

Figure 26 shows the performance of a 1500 kW extraction condensing turbine, indicating the effect of various extraction rates on total steam requirements as follows: at zero extraction and 1500 kW, 17,500 lb/h or a water rate of 11.67 lb/kWh is required. When 45,000 lb/h is extracted at 100 psig, only 4000 lb/h more (49,000 – 45,000) is required at the throttle condition of 400 psig to develop the same 1500 kW, chargeable to the generation of electric power. The portion of input energy chargeable to the power is represented by the sum of the enthalpy of this 4000 lb/h at throttle conditions and the difference in enthalpy between the throttle and extraction conditions of the extracted portion of steam.

In effect, as the extraction rate increases, the overall efficiency increases. However at "full" extraction, a significant flow of "cooling" steam must still pass through the final turbine stages for condensing. At this condition, a simple back pressure turbine would be more efficient, if all of the exhaust steam could be used.

Full-condensing steam turbines have a maximum plant shaft efficiency (power output as a percentage of input fuel to the boiler) ranging from 20 to 36%, but they have no useful thermal output. Therefore, with overall plant efficiencies no better than their shaft efficiencies, they are unsuitable for topping cogeneration cycles.

At maximum extraction, the heat/power ratio of extraction turbines is relatively high. This high ratio makes it difficult to match facility loads, except those with very high base thermal loads, if reasonable annual efficiencies are to be achieved. As extraction rates decrease, plant efficiency approaches that of a condensing turbine, but can never reach it. Thus, the 17,500 lb/h (11.67 lb/kWh) illustrated for Figure 26 at 1500 kW and zero extraction represents a steam-to-electric efficiency of 28.6%, but a fuel-to-electric efficiency of 25%, with a boiler plant efficiency of 85%, developed as follows:

Isentropic Δh from 400 psig steam to 2 in. Hg (absolute) condensing pressure = 1022 Btu/lb output.

$$\text{Actual } \Delta h = \frac{3413 \text{ Btu/kWh}}{\text{Actual steam rate (lb/kWh)}}$$

Actual Δh = 3413/11.67 = 292 Btu/lb
Plant shaft efficiency = (100 × 292 × 0.85)/1022
= 24.3% (fuel input to electric output)

Radial inflow turbines are more efficient than single-stage axial flow turbines of the same output. They are available up to 15,000 hp from several manufacturers, with throttle steam up to 2100 psig, wheel speeds up to 60,000 rpm, and output shaft speeds as low as 3600 or 1800 rpm. It is these high wheel speeds that yield efficiencies of 70 to 80%, compared with single-stage axial turbines spinning at only 10,000 rpm with efficiencies of up to 40%.

MAINTENANCE

Maintenance requirements for steam turbines vary greatly with complexity of design, throttle pressure rating, duty cycle, and steam quality (both physical and chemical). Typically, several common factors can be attributed to operational problems with steam turbines. First are problems resulting from solid particle erosion. Turbines that undergo frequent cycling may see significant deterioration of initial pressure stages. Solid particles from steam lines, superheaters, or reheats can become dislodged and enter the turbine. These solids, over time, erode the nozzle and blade materials and initiate cracks and weakening of the rotating blades. Turbines subject to cyclical operation should be examined carefully every 18 to 36 months. Usually, nondestructive testing is used to establish material loss trends and predictable maintenance requirements for sustained planned outages.

Cycling duty as well as sustained operation at low loads can create problems in the lower pressure section of the turbine. Chemicals can concentrate in these sections and corrode the blades, especially where steam becomes saturated. Both stress and corrosion fatigue are common in lower pressure stages.

The best way to minimize both corrosion and erosion in steam turbines is to maintain proper feedwater/steam chemistry. The complexity of feedwater chemistry increases with steam pressures. To protect steam turbines from unnecessary damage, both mechanical and vapor carryover from the steam generator must be minimized.

Turbine seals, glands, and bearings are also common areas of deterioration and maintenance. Bearings require frequent examination, especially in systems that experience cyclical duty. Oil samples from the lubrication system should be taken regularly to determine concentration of solid particle contamination and changes in viscous properties. Filters and oil should be recycled according to manufacturers recommendations and the operational history of the turbine system.

Large multistage steam turbines usually contain instrumentation that monitors vibration within the casing. As deposits build on the blades, blade material erodes or corrodes; or as mechanical tolerance of bearing surfaces increases, nonuniform rotation creates increasing turbine vibration. Vibration instrumentation, consequently, is used to determine maintenance intervals for turbines, especially those subject to extensive base-load operations where visual examination is not feasible.

ECONOMICS

The thermal efficiency associated with the steam turbine prime mover (specifically the Rankine cycle) is relatively low. The cycle depends mostly on steam throttle conditions (temperature/pressure), regeneration, and condensing condition. Consequently, the capital cost per unit of output is relatively high when compared to combustion turbines or reciprocating engines. Furthermore, plants that operate with the elevated steam pressures required by turbines require attended operation, which adds to overall operational costs. Consequently, the steam turbine prime movers are not economically appealing in small cogeneration applications (i.e., less than 15 to 20 MW). Exceptions may include the following:

- Applications where the fuel source is inexpensive, such as municipal waste, process gas, or waste streams in which incineration with waste heat recovery can be applied
- District heating/cooling plants that have high process loads, thermal loads, or both

THERMAL OUTPUT AND RECOVERY

THERMAL OUTPUT CHARACTERISTICS

Cogeneration provides an opportunity to use the fuel energy that the prime mover does not convert into shaft energy. If the heat cannot be used effectively, the plant efficiency is limited to that of the prime mover. However, if the site heat energy requirements can be met effectively by the normally wasted heat at the level it is available from the prime mover, this salvaged heat will reduce the normal fuel requirements of the site and increase overall plant efficiency. The prime mover furnishes (1) mechanical energy from the shaft and (2) unused heat energy that remains after the fuel or steam has acted on the shaft. Shaft loads (generators, centrifugal chillers, compressors, and process equipment) require a given amount of rotating mechanical energy. Once the prime mover is selected to provide the required shaft output, it has a fixed relationship to heat availability and system efficiency, depending on the prime mover fuels versus heat balance curves. The ability to use the prime mover waste heat determines overall system efficiency and is one of the critical factors in economic feasibility.

Reciprocating Engines

In all reciprocating internal combustion engines except small air-cooled units, heat can be reclaimed from the jacket cooling system, lubricating system, turbochargers, exhaust, and aftercoolers. These engines require extensive cooling to remove excess heat not conducted into the power train during combustion and the heat resulting from friction. Coolant fluids and lubricating oil are circulated to remove this engine heat. Some engines permit the coolant to reach 250°F at above atmospheric pressure, which allows the coolant to flash into low-pressure steam (15 psig) after it has left the engine jacket (ebullient cooling).

Waste heat in the form of hot water or low-pressure steam is recovered from the engine jacket manifolds and exhaust, and additional heat can be recovered from the lubrication system (see Figure 30 through Figure 34).

Provisions similar to those used with gas turbines are necessary if supplemental heat is required, except an engine exhaust is rarely fired with a booster because it contains insufficient oxygen. If electrical supplemental heat is used, the additional electrical load is reflected back to the prime mover, which reacts accordingly by producing additional waste heat. This action creates a feedback effect, which can stabilize system operation under certain conditions. The approximate distribution of input fuel energy under selective control of the thermal demand for an engine operating at rated load is as follows:

Shaft power	33%
Convection and radiation	7%
Rejected in jacket water	30%
Rejected in exhaust	30%

These amounts vary with engine load and design. Four-cycle engine heat balance for naturally aspirated (Figure 27) and turbocharged gas engines (Figure 28) show typical heat distribution. The exhaust gas temperature for these engines is about 1200°F at full load and 1000°F at 60% load.

Two-cycle lower speed (900 rpm and below) engines operate at lower exhaust gas temperatures, particularly at light loads, because the scavenger air volumes remain high through the entire range of

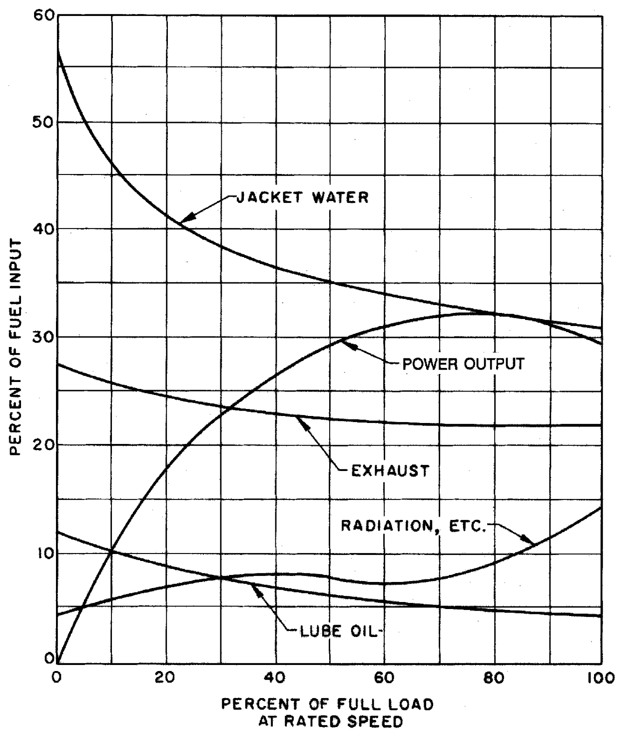

Fig. 27 Heat Balance for Naturally Aspirated Engine

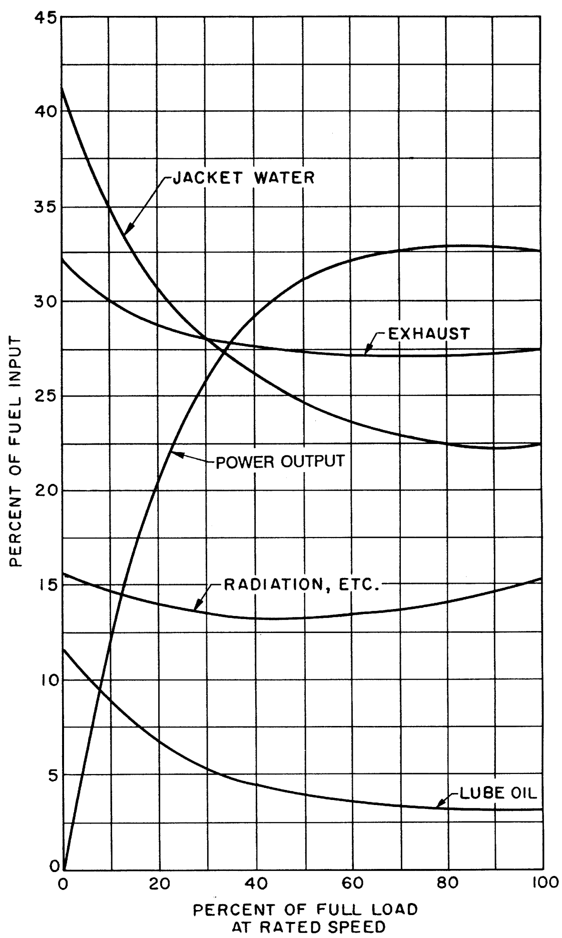

Fig. 28 Heat Balance for Turbocharged Engine

capacity. High-volume, lower temperature exhaust gas offers less efficient exhaust gas heat recovery possibilities. The exhaust gas temperature is approximately 700°F at full load and drops below 500°F at low loads. Low-speed engines generally cannot be ebulliently cooled because their cooling systems often operate at 170°F and below.

Combustion Turbines

In the gas turbine cycle, the average fuel to electrical shaft efficiency ranges from approximately 12% to above 35%, with the remainder of the fuel energy discharged in the exhaust and through radiation or internal coolants in large turbines. A minimum stack exhaust temperature of about 300°F is required to prevent condensation. Because the heat rate efficiency is lower, the quantity of heat recoverable per unit of power is greater for a gas turbine than for a reciprocating engine. This heat is generally available at the higher temperature of the exhaust gas. The net result is an overall thermal efficiency of 60% and higher. Because gas turbine exhaust contains a large percentage of excess air, afterburners or boost burners may be installed in the exhaust to create a supplementary boiler system. This system can provide additional steam or level the steam production during reduced turbine loads. Boost burning can increase cycle efficiency to a maximum of 93%.

Absorption chillers capable of operating directly off turbine exhausts with coefficients of performance of 1.14 or more are also available. The conventional method of controlling steam or hot water production in a heat recovery system at part load is to by-pass a portion of the exhaust gases around the boiler tubes and out the exhaust stack through a gas bypass valve assembly (Figure 29).

Steam Turbines

Steam turbine exhaust or extracted steam, reduced in both pressure and temperature from throttle conditions as it generates the prime mover shaft power, may be fed to heat exchangers, absorption chillers, steam turbine-driven centrifugal chillers, pumps, or other equipment. Its value as recovered energy is the same as that of the fuel otherwise needed to generate the same steam flow at the condition leaving the prime mover in an independent boiler plant cycle.

Heat/Power Ratio and Quality of Heat

Selection of the prime mover depends on the thermal and power profiles required by the end user and on the contemporaneous relationship of these profiles. Ideally, the recoverable heat is fully used as the prime mover follows the power load, but this ideal never occurs over extended periods.

For maximum equipment use and least energy waste, the following methods are used to produce only the power and thermal energy that is required on site:

- Match the heat/power ratio of the prime mover to that of the user's hourly load profile.
- Store excess power as chilled water or ice when the thermal demand exceeds the coincident power demand.
- Store excess thermal production as heat when the power demand exceeds the heat demand. Either cool or heat storage must be able to productively discharge most of its energy before it is dissipated to the environment.
- Sell excess power or heat on a mutually acceptable contract basis to a user outside of the host facility. Usually the buyer is the local utility, but sometimes it is a nearby facility.

The quality of recovered energy is the second major determinant in selecting the prime mover. If the quantity of high-temperature (above 260°F) recoverable heat available from an engine's exhaust is significantly less than that demanded by end users, a combustion or steam turbine may offer greater opportunity.

Low heat/power ratios of 1 to 3 lb/h steam per horsepower output of the prime mover indicate the need for one with a high shaft efficiency of 30 to 45% (shaft energy/fuel input). This efficiency is a good fit for an engine because its heat output is available as 15 psig steam or 250°F water. Higher temperature/pressures are available, but only from a exhaust gas recovery system, separated in that case from the low-temperature jacket water system. However, for a typical case in which 30% of the fuel energy is in the exhaust, approximately 50% of this energy is recoverable (with 300°F final exhaust gas temperature); less than 50% is recoverable if higher steam pressures are required.

Medium heat/power ratios of 4 to 11 lb steam/hp·h can be provided by combustion turbines, which are inherently low in shaft efficiency. Smaller turbines, for example, are only 20 to 25% efficient, with 75 to 80% of their fuel energy released into the exhaust.

High heat/power ratios of 8 to 40 lb steam/hp·h, provided by various steam turbine configurations make this prime mover highly flexible for higher thermal demands. The designer can vary throttle, exhaust and/or extraction conditions, and turbine efficiency to attain the most desirable ratio for varying heat/power loads in many applications; thus furnishing a wide variety of thermal energy quality levels.

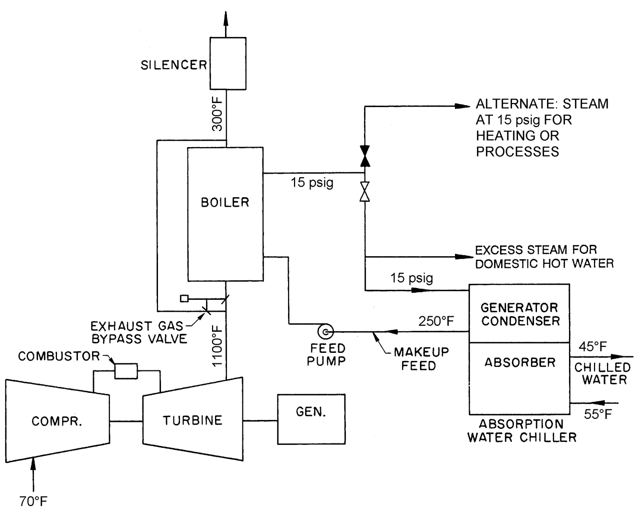

Fig. 29 Typical Heat Recovery Cycle for Gas Turbine

HEAT RECOVERY

Reciprocating Engines

Engine Jacket Heat Recovery. Engine jacket cooling passages for reciprocating engines, including the water cooling circuits in the block, heads, and exhaust manifolds, must remove about 30% of the heat input to the engine. If the machine operates at above 180°F coolant temperature, condensation of combustion products should produce no ill effects. Some engines have modified gaskets and seals to enable satisfactory operation up to 250°F at 30 psia. To avoid thermal stress, the temperature rise through the engine jacket should not exceed 15°F. Flow rates must be kept within the engine manufacturers' design limitations to avoid erosion from excessive flow or inadequate distribution within the engine from low flow rates.

Engine-mounted, water circulating pumps driven from an auxiliary shaft can be modified with proper shaft seals and bearings to give good service life at the elevated temperatures. Configurations that have a circulating pump for each engine increase reliability because the remaining engines can operate if one engine pump assembly fails. An alternative design uses an electric-drive pump battery to circulate water to several engines and has a standby pump

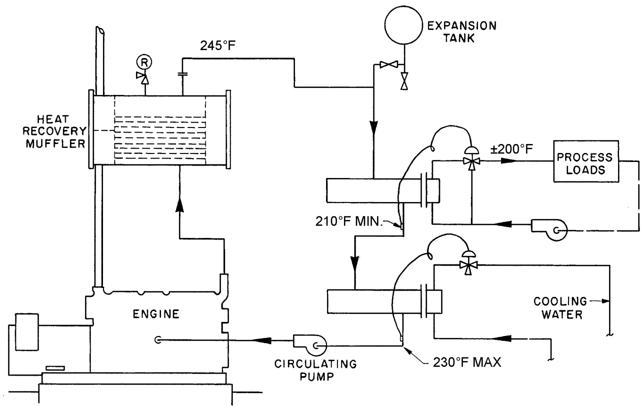

Fig. 30 Hot Water Heat Recovery

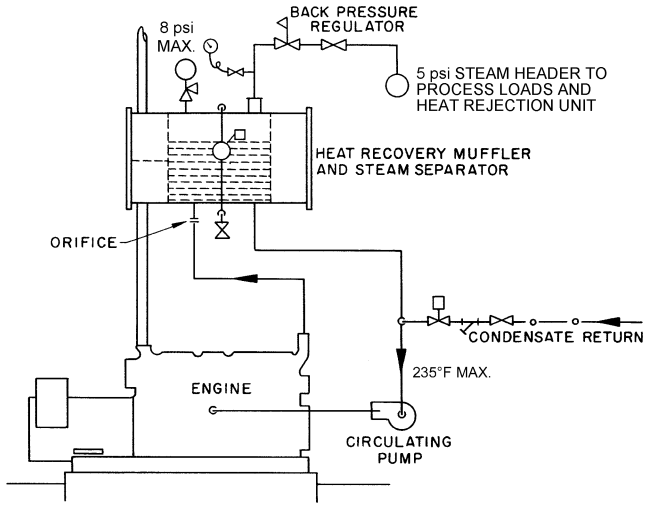

Fig. 31 Hot Water Engine Cooling with Steam Heat Recovery (Forced Recirculation)

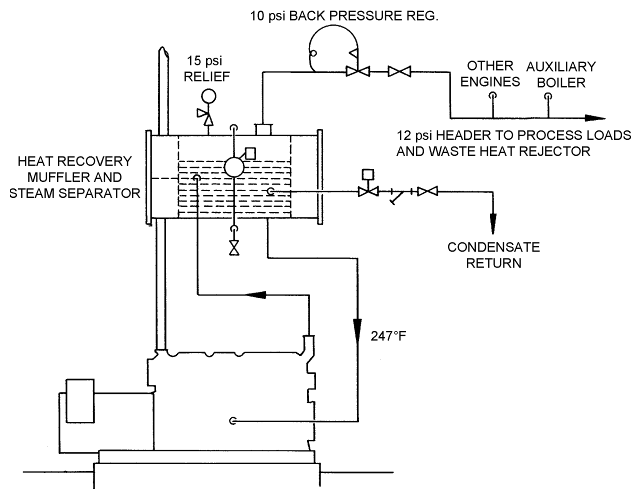

Fig. 32 Engine Cooling with Gravity Circulation and Steam Heat Recovery

assembly in reserve, interconnected so that any engine or pump can be cut off without disabling the jacket water system.

Forced-circulation hot water, at 250°F and 30 psia, can be used for many process loads, including water heating systems for comfort and process loads, absorption refrigeration chillers, and service water heating. Distribution of the engine jacket coolant must be limited to reduce the risk of leaks, contamination, or other failures in downstream equipment that could prevent engine cooling.

One solution is to confine each engine circuit to its individual engine, using a heat exchanger to transfer the salvaged heat to another circuit that serves several engines via an extensive distribution system. An additional heat exchanger is needed in each engine circuit to remove heat whenever the waste heat recovery circuit does not extract all the heat produced (Figure 30). The reliability of this approach is high, but because the salvage circuit must be at a lower temperature than the engine operating level, it requires either larger heat exchangers, piping, and pumps or a sacrifice of system efficiency that might result from these lower temperatures.

Low-temperature limit controls prevent excessive system heat loads from seriously reducing the operating temperature of the engines. Engine castings may crack if they are suddenly cooled because a large demand for heating temporarily overloads the system. A heat storage tank is an excellent buffer for such occasions because it can provide heat at a very high rate for short periods to protect the machinery serving the heat loads. The heat level can be controlled with supplementary heat input such as an auxiliary boiler.

Another method of protecting the engine heat recovery system from instantaneous load is to flash the recovered heat into steam (**ebullient cooling** system). These pressurized, reciprocating engine cooling, heat recovery systems are also called high-temperature systems because they operate with a few degrees of temperature differential in the range of 212 to 250°F.

Engine builders have approved various conventionally cooled models for ebullient cooling. Azeotropic antifreeze solutions should be used to ensure constant coolant composition in boiling. However, the engine power outputs are generally derated by the manufacturer when the jackets are cooled ebulliently.

Ebullient cooling or hot water cooling equipment usually replaces the radiator, the belt-driven fan, and the gear-driven water pump. Many heat recovery systems use the jacket water in water-to-liquid exchangers when the thermal loads are at temperature levels below 180°F, particularly when their load profiles can absorb most or all of the jacket heat. Such systems are easier to design and have historically developed fewer problems than those with ebullient cooling.

Ebullient systems use the steam as the distribution fluid (Figure 31). By using a back pressure regulator, the steam pressure on the engine can be kept uniform, and the auxiliary steam boilers can supplement the distribution steam header. The same engine coolant system using forced-circulation water at 235°F entering and 250°F leaving the engine can produce steam at 235°F, 8 psig. The engine cooling circuit is the same as in other high-temperature water systems in that an adequate static pressure is maintained above the circulating pump inlet to avoid cavitation at the pump, and flow is restricted at the entrance to the steam flash chamber. The flow restriction prevents flash steam, which can severely damage the engine.

A limited distribution system that distributes steam through nearby heat exchangers can salvage heat without contaminating the main engine cooling system. The salvage heat temperature is kept high enough for most low-pressure steam loads. Returning condensate must be treated to prevent engine oxidation. The flash tank can accumulate the sediment from treatment chemicals.

Some engines can use natural-convection ebullient cooling, in which water circulated by gravity flows to the bottom of the engine. There, it is heated by the engine to form steam bubbles (Figure 32). This heat lowers the density of the fluid, which then rises to a separating chamber, or flash tank, where the steam is released and the water is recirculated to the engine.

Maintaining the flash tank at a static pressure slightly above that of the highest part of the engine causes a rapid flow of coolant through the engine, so only a small temperature difference is maintained between inlet and outlet. A constant back pressure must be maintained at the steam outlet of the flash chamber to prevent sudden lowering of the operating pressure. If the operating pressure changes rapidly, steam bubbles in the engine could quickly expand, interfering with heat transfer and causing overheating at critical points in the engine. This method of engine cooling is suitable for operation at up to 250°F, 15 psig steam, using components built to meet Section VIII of the ASME *Boiler and Pressure Vessel Code*.

The engine coolant passages must be designed for gravity circulation and must eliminate the formation of freely bubbling steam. It is important to consider how coolant flows from the engine heads and exhaust manifolds. Each coolant passage must vent upward to encourage gravity flow and the free release of steam toward the steam separating chamber. Engine heat removal is most satisfactory when these circuits are used because fluid temperatures are uniform throughout the machine. The free convection cooling system depends less on mechanical accessories and is more readily arranged for completely independent assembly of each engine with its own coolant system than other engine cooling systems.

Any system that generates steam from an engine has inherent design hazards. It is preferable, when the downstream facility thermal profiles and system arrangements permit, to use conventional hot water heat recovery such as shown in Figure 30.

Lubricant Heat Recovery. Lubricant heat exchangers should maintain oil temperatures at 190°F, with the highest coolant temperature consistent with the economical use of the salvaged heat. Engine manufacturers usually size their oil cooler heat transfer surface on the basis of 130°F entering coolant water and without provision for additional lubricant heat gains that occur with high engine operating temperatures. The cost of obtaining a reliable supply of lower temperature cooling water must be compared with the cost of increasing the size of the oil cooling heat exchanger and operating at a cooling water temperature of 165°F. In applications where engine jacket coolant temperatures are above 220°F and where there is use for heat at 155 to 165°F, the heat from the lubricant can be salvaged profitably.

The coolant should not foul the oil cooling heat exchanger. Untreated water should not be used unless it is free of silt, calcium carbonates, sand, and other contaminants. A good solution is a closed-circuit, treated water system using an air-cooled heat transfer coil with freeze-protected coolant. A domestic water heater can be installed on the closed circuit to act as a reserve heat exchanger and to salvage some useful heat when needed. Inlet air temperatures are not as critical with diesel engines as with turbocharged natural gas engines, and the aftercooler water on diesel engines can be run in series with the oil cooler (Figure 33). A double-tube heat exchanger can also be used to prevent contamination from a leaking tube.

Turbocharger Heat Recovery. Turbochargers on natural gas engines require medium fuel gas pressure (12 to 20 psi) and rather low aftercooler water temperatures (90°F or less) for high compression ratios and best fuel economy. Aftercooler water at 90°F is a premium coolant in many applications because the usual sources are raw domestic water and evaporative cooling systems such as cooling towers. Aftercooler water at temperatures as high as 135°F can be used, although the engine is somewhat derated. Using a domestic water solution may be expensive because (1) the coolant is continuously needed while the engine is running, and (2) the available heat exchanger designs require a large amount of water even though the load is less than 200 Btu/hp·h. A cooling tower can be used, but unless required for other reasons, it can increase initial costs. Also, it requires winter freeze protection and water quality control. If a cooling tower is used, the lubricant cooling load must be included in the tower design load for periods when there is no use for salvage lubricant heat.

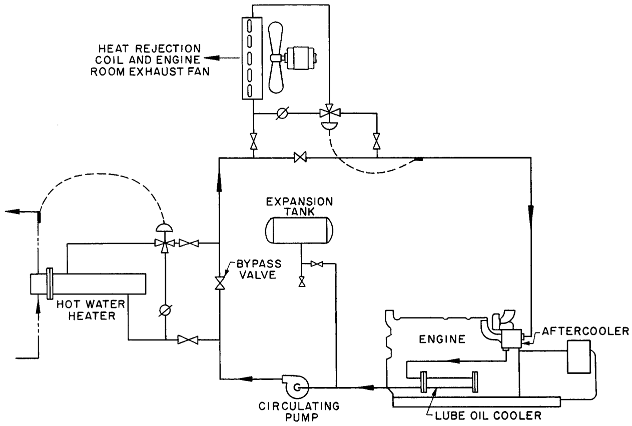

Fig. 33 Lubricant and Aftercooler System

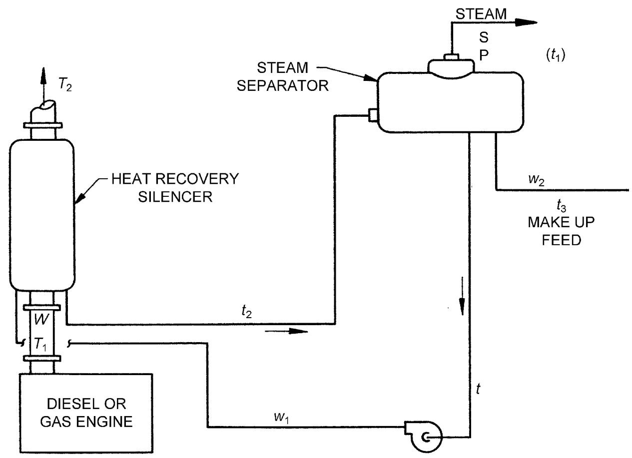

Fig. 34 Exhaust Heat Recovery with Steam Separator

Exhaust Gas Heat Recovery. Almost all heat transferred to the engine jacket cooling system can be reclaimed in a standard jacket cooling process or in combination with exhaust gas heat recovery. However, only part of the exhaust heat can be salvaged due to the limitations of heat transfer equipment and the need to prevent flue gas condensation (Figure 34). Energy balances are often based on standard air at 60°F; however, exhaust temperature cannot easily be reduced to this level.

A minimum exhaust temperature of 250°F was established by the Diesel Engine Manufacturers Association (DEMA). Many heat recovery boiler designs are based on a minimum exhaust temperature of 300°F to avoid condensation and acid formation in the exhaust piping. Final exhaust temperature at part load is important on generator sets that operate at part load most of the time. Depending on the initial exhaust temperature, approximately 50 to 60% of the available exhaust heat can be recovered.

A complete heat recovery system, which includes jacket water, lubricant, turbocharger, and exhaust, can increase the overall thermal efficiency from 30% for the engine generator alone to approximately 75%. Exhaust heat recovery equipment is available in the same categories as standard fire-tube and water-tube boilers.

Other heat recovery silencers and boilers include coil-type water heaters with integral silencers, water-tube boilers with steam separators for gas turbine, and engine exhausts and steam separators for high-temperature cooling of engine jackets. Recovery boiler design should facilitate inspection and cleaning of the exhaust gas and water sides of the heat transfer surface. Diesel engine units should

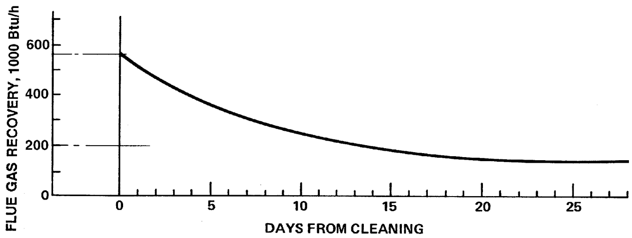

Fig. 35 Effect of Soot on Energy Recovery from Flue Gas Recovery Unit on Diesel Engine

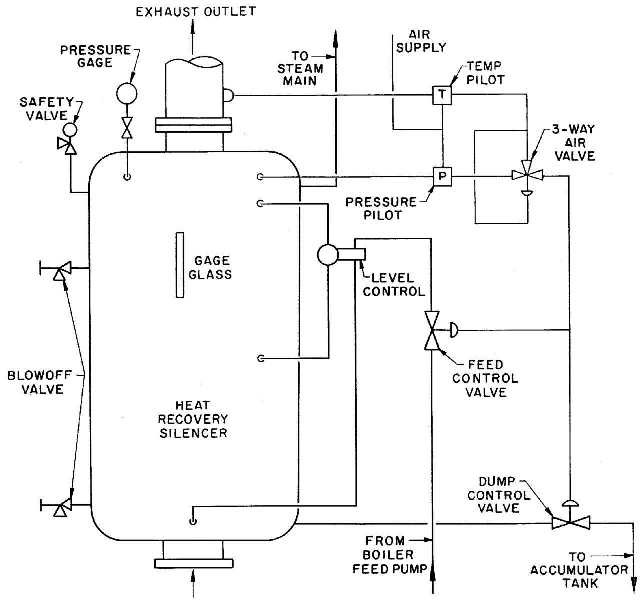

Fig. 36 Automatic Boiler System with Overriding Exhaust Temperature Control

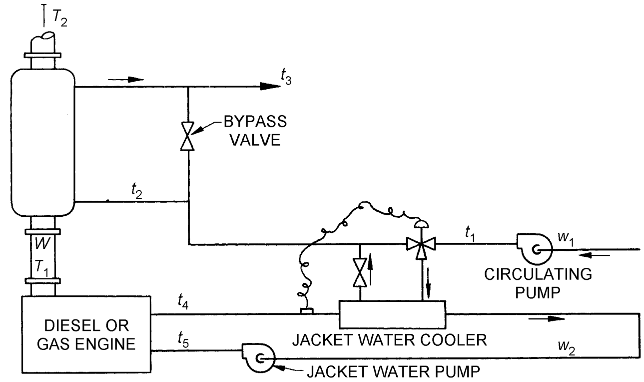

Fig. 37 Combined Exhaust and Jacket Water Heat Recovery System

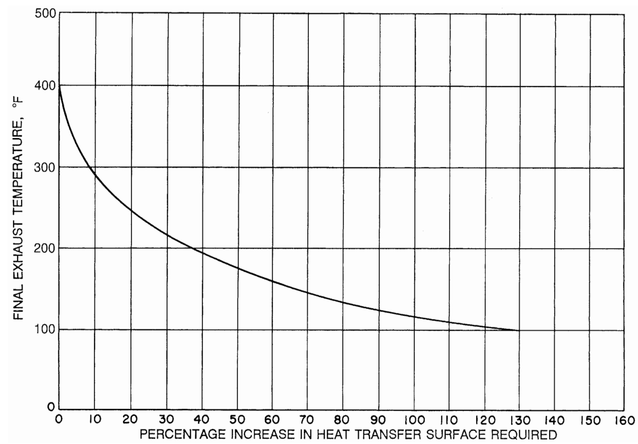

Fig. 38 Effect of Lowering Exhaust Temperature below 300°F

have a means of soot removal because soot deposits can quickly reduce the heat exchanger effectiveness (Figure 35). These recovery boilers can also serve other requirements of the heat recovery system, such as surge tanks, steam separators, and fluid level regulators.

In many applications for heat recovery equipment, the demand for heat requires some method of **automatic control**. In vertical recovery boilers, control can be achieved by varying the water level in the boiler. Figure 36 shows a control system using an air-operated pressure controller with diaphragm or bellows control valves. When steam production begins to exceed demand, the feed control valve begins to close, throttling the feedwater supply. Concurrently, the dump valve begins to open, and the valves reach an equilibrium position that maintains a level in the boiler to match the steam demand.

This system can be fitted with an overriding exhaust temperature controller that regulates the boiler output to maintain a preset minimum exhaust temperature at the outlet. This type of automatic control is limited to vertical boilers because the ASME *Boiler and Pressure Vessel Code* does not permit horizontal boilers to be controlled by varying the water level. Instead, a control condenser, radiator, or thermal storage can be used to absorb excess steam production.

In hot water units, a temperature-controlled bypass valve can divert the water or the exhaust gas to achieve automatic modulation with heat load demand (Figure 37). If the water is diverted, precautions must be taken to limit the temperature rise of the lower flow of water in the recovery device, which could otherwise cause steaming. The heat recovery equipment should not adversely affect the primary function of the engine to produce work. Therefore, the design of waste heat recovery boilers should begin by determining the back pressure imposed on the engine exhaust gas flow. Limiting back pressures vary widely with the make of engine, but the typical value is 6 in. of water gage. The next step is to calculate the heat transfer area that gives the most economical heat recovery without reducing the final exhaust temperature below 300°F.

Heat recovery silencers are designed to adapt to all engines; efficient heat recovery depends on the **initial exhaust temperature**. Most designs can be modified by adding or deleting heat transfer surface to suit the initial exhaust conditions and to maximize heat recovery down to a minimum temperature of 300°F.

Figure 38 illustrates the effect of lowering the exhaust temperature below 300°F. This curve is based on a specific heat recovery silencer design with an initial exhaust temperature of 1000°F. Lowering the final temperature from 300°F to 200°F increases heat recovery 14% but requires a 38% surface increase. Similarly, a reduction from 300°F to 100°F increases heat recovery 29% but requires a 130% surface increase. Therefore, the cost of heat transfer surface is a factor that must be considered when determining the final temperature.

Another factor to consider is the problem of water vapor condensation and acid formation if the exhaust gas temperature falls below the dew point. This point varies with fuel and atmospheric conditions and is usually in the range of 125 to 150°F. This gives an adequate margin of safety for the 250°F minimum temperature recommended by DEMA. Also, it allows for other conditions that could cause condensation, such as an uninsulated boiler shell or other cold surface in the exhaust system, or part loads on an engine.

The quantity of water vapor varies with the type of fuel and the intake air humidity. Methane fuel, under ideal conditions and with only the correct amount of air for complete combustion, produces 2.25 lb of water vapor for every pound of methane burned. Similarly, diesel fuel produces 1.38 lb of water vapor per pound of fuel. In the gas turbine cycle, these relationships would not hold true because of the large quantities of excess air. The condensates formed at low exhaust temperatures can be highly acidic. Sulfuric acid from diesel fuels and carbonic acid from natural gas fuels can cause severe corrosion in the exhaust stack as well as in colder sections of the recovery device.

If engine exhaust flow and temperature data are available, and maximum recovery to 300°F final exhaust temperature is desired, the basic **exhaust recovery equation** is

$$q = \dot{m}_e (c_p)_e (t_e - t_f) \tag{4}$$

where

q = heat recovered, Btu/h
$\dot{m}_e$ = mass flow rate of exhaust, lb/h
t_e = exhaust temperature, °F
t_f = final exhaust temperature, °F
$(c_p)_e$ = specific heat of exhaust gas = 0.25 Btu/lb·°F

Equation (4) applies to both steam and hot water units. To estimate the quantity of steam obtainable, the total heat recovered q is divided by the latent heat of steam at the desired pressure. The latent heat value should include an allowance for the temperature of the feedwater return to the boiler. The basic equation is

$$q = \dot{m}_s (h_s - h_f) \tag{5}$$

where

$\dot{m}_s$ = mass flow rate of steam, lb/h
h_s = enthalpy of steam, Btu/lb
h_f = enthalpy of feedwater, Btu/lb

Similarly, the quantity of hot water can be determined by

$$q = \dot{m}_w (c_p)_w (t_o - t_i) \tag{6}$$

where

$\dot{m}_w$ = mass flow rate of water, lb/h
$(c_p)_w$ = specific heat of water = 1.0 Btu/lb·°F
t_o = temperature of water out, °F
t_i = temperature of input water, °F

If the shaft power is known but engine data are not, the heat available from the exhaust is about 1000 Btu/h per horsepower output or 1 lb/h steam per horsepower output. The exhaust recovery equations also apply to gas turbines, although the rate of flow will be much greater. The estimate for gas turbine boilers is 8 to 10 lb/h of steam per horsepower output. These values are reasonably accurate for steam pressures in the range of 15 to 150 psig.

The normal procedure is to design and fabricate heat recovery boilers to the ASME *Boiler and Pressure Vessel Code* (Section VIII) for the working pressure required. Because the temperatures in most exhaust systems are not excessive, it is common to use flange or firebox quality steels for the pressure parts and low-carbon steels for the nonpressure components. Wrought iron or copper can be used for extended-fin surfaces to improve heat transfer capacities.

In special applications such as sewage gas engines, where exhaust products are highly corrosive, wrought iron or special steels are used to improve corrosion resistance. Exhaust heat may be used to make steam, or it may be used directly for drying or other processes. The steam provides space heating, hot water, and absorption refrigeration, which may supply air conditioning and process refrigeration. Heat

Table 5 Temperatures Normally Required for Various Heating Applications

Application	Temperature, °F
Absorption refrigeration machines	190 to 245
Space heating	120 to 250
Water heating (domestic)	120 to 200
Process heating	150 to 250
Evaporation (water)	190 to 250
Residual fuel heating	212 to 330
Auxiliary power producers, with steam turbines or binary expanders	190 to 350

Table 6 Full-Load Exhaust Mass Flows and Temperatures for Various Engines

Type of Engine	Mass Flow, lb/bhp·h	Temperature, °F
Two-cycle		
Blower-charged gas	16	700
Turbocharged gas	14	800
Blower-charged diesel	18	600
Turbocharged diesel	16	650
Four-cycle		
Naturally aspirated gas	9	1200
Turbocharged gas	10	1200
Naturally aspirated diesel	12	750
Turbocharged diesel	13	850
Gas turbine, nonregenerative	18 to 48[a]	800 to 1050[a]

[a]Lower mass flows correspond to more efficient gas turbines.

recovery systems generally involve equipment specifically tailored for the job, although conventional fire-tube boilers are sometimes used. Exhaust heat may be recovered from reciprocating engines by a muffler-type exhaust heat recovery unit. Table 5 gives the temperatures normally required for various heat recovery applications.

In some engines, exhaust heat rejection exceeds jacket water rejection. Generally, gas engine exhaust temperatures run from 700 to 1200°F, as shown in Table 6. About 50 to 75% of the sensible heat in the exhaust may be considered recoverable. The economics of exhaust heat boiler design often limits the temperature differential between exhaust gas and generated steam to a minimum of 100°F. Therefore, in low-pressure steam boilers, the gas temperature can be reduced to 300 to 350°F; the corresponding final exhaust temperature range in high-pressure steam boilers is 400 to 500°F.

Because higher airflows are required to purge the cylinders of two-cycle engines, they have lower temperatures than four-cycle engines. As a result, they are less desirable for heat recovery. Gas turbines have even larger flow rates, but at high enough temperatures to make heat recovery worthwhile when the recovered heat can be efficiently used.

Combustion Turbines

The information in the preceding section on Exhaust Gas Heat Recovery applies to combustion turbines as well.

Steam Turbines

Noncondensing Turbines. The back pressure or noncondensing turbine is the simplest turbine. It consists of a turbine in which the steam is exhausted at atmospheric pressure or higher. These turbines are generally used when there is a process need for high-pressure steam, and all steam condensation takes place downstream of the turbine cycle and in the process. Figure 39 illustrates the steam path for a noncondensing turbine. The back pressure steam turbine operates on the enthalpy difference between steam inlet and exhaust conditions. The noncondensing turbine's Carnot cycle efficiency

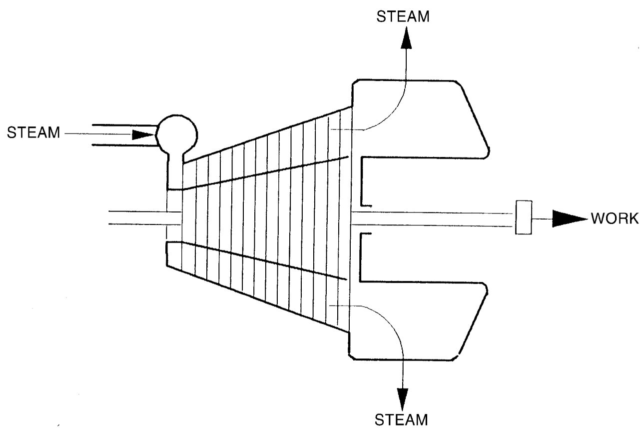

Fig. 39 Back Pressure Turbine

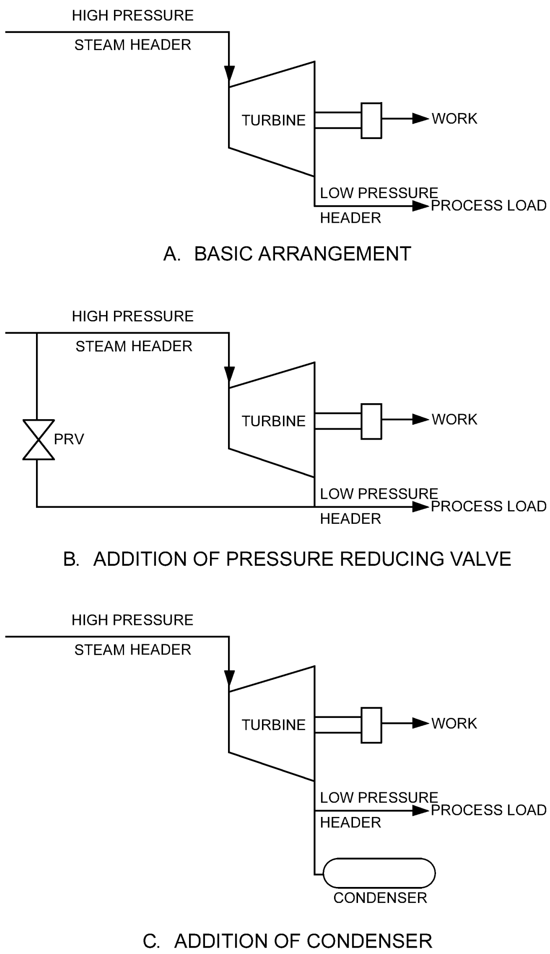

Fig. 40 Integration of Back Pressure Turbine with Facility

tends to be lower than is possible with other turbines because the difference between the turbine inlet and exhaust temperatures tends to be lower. Because much of the steam's heat, including the latent heat of vaporization, is exhausted and then used in a process, the back pressure cogeneration system process efficiency or total energy efficiency can be very high. One potential application for back pressure turbines is as a substitute for pressure reducing valves; they provide the same function (pressure regulation), but also produce a useful product—power.

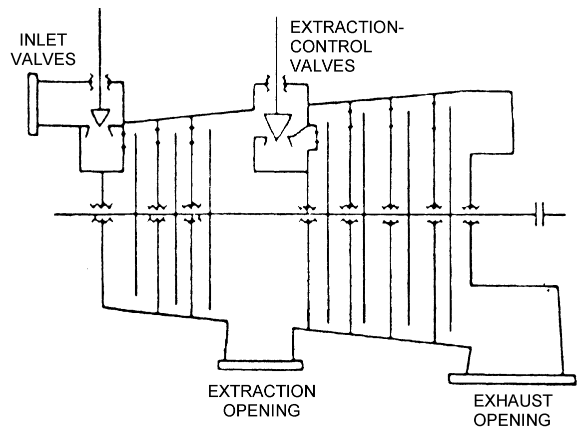

Fig. 41 Condensing Automatic Extraction Turbine

The back pressure turbine has one major disadvantage in cogeneration applications. Because the process load is the heat sink for the steam, the amount of steam passed through the turbine depends on the heat load at the site. Thus, the back pressure turbine provides little flexibility in directly matching electrical output to electrical requirements; electrical output is controlled by the thermal load. The direct linkage of the site steam requirements and electrical output can result in electric utility charges for standby service or increased supplemental service unless some measures are taken to increase system flexibility.

Figure 40 illustrates several ways to achieve some flexibility in performance when the electrical and thermal loads do not match the capability of the back pressure turbine. Figure 40A shows the basic arrangement of a noncondensing steam turbine and its relationship to the facility. Figure 40B illustrates the addition of a pressure reducing valve (PRV) to bypass some or all of the steam around the turbine. Thus, if the process steam demand exceeds the capability of the turbine, the additional steam can be provided through the PRV. Figure 40C illustrates the use of a load condenser to permit the generation of electricity, even when there is no process steam demand. The use of either of these techniques to match thermal and electrical loads is very inefficient, and operating time at these off-design conditions must be minimized by careful analysis of the coincident, time-varying process steam and electrical demands.

Process heat recovered from the noncondensing turbine can be easily estimated using steam tables combined with knowledge of the steam flow, steam inlet conditions, steam exit pressure, and turbine isentropic efficiency.

Extraction Turbines. Figure 41 illustrates the internal arrangement of an automatic extraction turbine where steam is exhausted from the turbine at one or more stations along the steam flow path. Conceptually, the extraction turbine is a hybrid combination of condensing and noncondensing turbines. The advantage of this turbine is that it allows extraction of the quantity of steam required at each temperature or pressure needed by the industrial process. Multiple extraction ports allow great flexibility in matching the cogeneration cycle to the thermal requirements at the site. Extracted steam can also be used in the power cycle for feedwater heating or powerhouse pumps. Depending on cycle constraints and process requirements, the extraction turbine's final exit conditions can be either back pressure or condensing.

A diaphragm in the automatic extraction turbine separates the high-pressure section from the low-pressure section. All the steam passes through the high-pressure section just as it does in a single back pressure turbine. A throttle controls steam flow into the low-pressure section. This throttle is controlled by the pressure in the process steam line. If the pressure drops, the throttle closes, allowing more steam to the process. If the pressure rises, the throttle

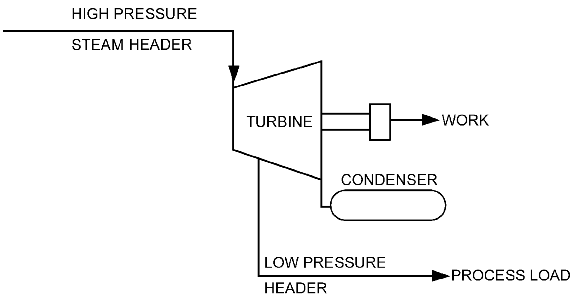

Fig. 42 Automatic Extraction Turbine Cogeneration System

opens to allow steam to flow through the low-pressure section, where additional power is generated.

An automatic extraction turbine (Figure 42) is uniquely designed to meet the specific power and heat capability of a given site; therefore, no simple relationship generally applies. For preliminary design analyses, the procedures presented by Newman (1945) can be used to estimate performance. The product of such an analysis is a performance map similar to that shown in Figure 43 for a 5000 kW generator. The performance map provides the steam flow to the turbine as a function of generator output with extraction flow as a parameter.

HEAT-ACTIVATED CHILLERS

Waste heat may be converted and used to produce chilled water by several methods. The conventional method is to use hot water (>200°F) or low-pressure steam (<15 psig) in single-stage absorption chillers. These single-stage absorbers have a COP of 0.6 or less; 18,000 Btu/h of recovered heat can produce about 1 ton of cooling (12,000 Btu/h).

If a direct exhaust, two-stage absorption chiller is used, the equation to estimate cooling produced from recoverable heat is

$$q = \frac{\dot{m}_e c_p (t_1 - t_2)(1.14 \times 0.97)}{12{,}000} \tag{7}$$

where

q = cooling produced, tons
$\dot{m}_e$ = mass flow of exhaust gas, lb/h
c_p = specific heat of gas = 0.268 Btu/lb·°F
t_1 = exhaust temperature in, °F
t_2 = exhaust temperature out = 375°F
1.14 = coefficient of performance
0.97 = connecting duct system efficiency
12,000 = Btu/ton·h

If it is available, the manufacturer's COP rating should be used to replace the assumed value.

For internal combustion engines, jacket water heat at 180 to 210°F may be added to the recovered heat of the engine exhaust to produce chilled water in a single-stage absorption machine. The equation to estimate the cooling produced from the heat recovered from the water is

$$q = \frac{0.6\dot{m}_w (c_p)_w (t_1 - t_2)}{12{,}000} \tag{8}$$

where

$\dot{m}_w$ = mass flow of water, lb/h
$(c_p)_w$ = specific heat of water, 1.0 Btu/lb·°F
t_1 = water temperature out of engine, °F
t_2 = water temperature returned to engine, °F [Typically $(t_1 - t_2)$ = 15°F.]
0.6 = COP

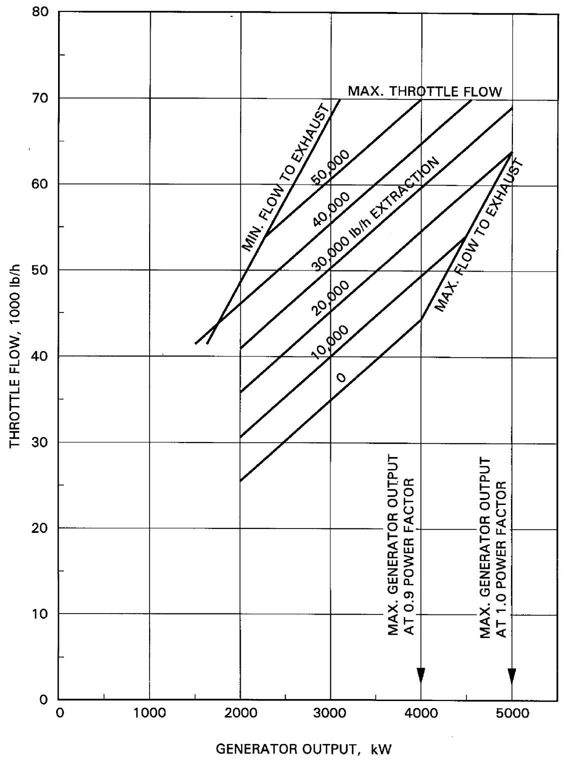

Fig. 43 Performance Map of Automatic Extraction Turbine

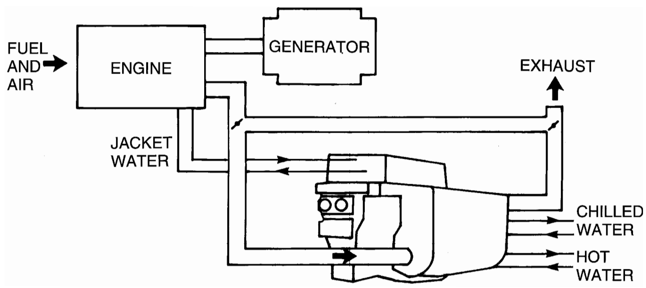

Fig. 44 Exhaust Gas Chiller-Heater

Heat may be recovered from engines and gas turbines as high-pressure steam, depending on exhaust temperature. Steam pressures from 15 to 200 psig are common. When steam is produced at pressures over 43 psig, two-stage steam absorption chillers can also be considered. The COP of a two-stage absorption chiller is 1.14 or greater. The steam input required is 9.3 to 9.7 lb/ton·h. This compares to 18 lb/ton·h for the single-stage absorption machine using 15 psig steam. Two-stage absorption chillers can take advantage of the dual temperatures available from the engine exhaust and jacket water.

The use of the engine exhaust heat to provide cooling is described in the section on Exhaust Gas Heat Recovery, and some absorption machines have been designed specifically for heat recovery in cogeneration applications. These units use both the jacket water and the exhaust gas directly. Ebullient cooling of the engine is not required (Figure 44). These chillers range in capacity from 50 to 200 tons.

Another type of absorption chiller uses gas engine or turbine exhaust directly in a waste heat absorption chiller. These units are available in sizes ranging from 100 to 1500 tons. Using oil/diesel exhaust for this purpose has not been successful due to fouling and corrosion problems in such direct-fired or waste exhaust-fired chillers.

ELECTRICAL SYSTEMS

UTILITY INTERFACING

All electric utility interfacing requires safety on the electric grid and the ability to meet the operating problems of the electric grid and its generating system. Additional control functions depend on the desired operating method during loss of interconnection. For example, the throttle setting on a single generator operating in parallel with the utility is determined by either the heat recovery requirements or the power requirements, whichever governs, and its exciter current, which is set by the reactive power flow through the interconnection. When the interconnection with the utility is lost, the generator control system must detect that loss, assume voltage and frequency control, and immediately disconnect the intertie to prevent an unsynchronized reconnection.

With the throttle control now determined solely by the electrical load, the heat produced may not match the requirements for supplemental or discharge heat from the system. When the utility source is reestablished, the system must be manually or automatically synchronized, and the control functions restored to normal operation.

Loss of the utility source may be sensed through the following factors: overfrequency, underfrequency, overcurrent, overvoltage, undervoltage, or any combination of these. The most severe condition occurs when the generator is delivering all electrical requirements of the system up to the point of disconnection, whether it is on the electric utility system or at the plant switchgear. Under such conditions, the generator tends to operate until the load changes.

At this time, it either speeds up or slows down, allowing the over- or underfrequency device to sense loss of source and to reprogram the generator controls to isolated system operation. The interconnection is normally disconnected during such a change and automatically prevented from reclosing to the electric system until the electric source is reestablished and stabilized and the generator is brought back to synchronous speed. Additional utility interfacing aspects are covered in following sections.

GENERATORS

Criteria influencing the selection of alternating current (ac) generators for cogeneration systems are (1) machine efficiency in converting mechanical input into electrical output at various loads; (2) electrical load requirements, including frequency, power factor, voltage, and harmonic distortion; (3) phase balance capabilities; (4) equipment cost; and (5) motor-starting current requirements.

Generator speed is a direct function of the number of poles and the output frequency. For 60 Hz output, the speed can range from 3600 rpm for a two-pole machine to 900 rpm for an eight-pole machine. A wide latitude exists in matching generator speed to prime mover speed without reducing the efficiency of either unit. This range in speed and resultant frequency suggests that electrical equipment with improved operation at a special frequency might be accommodated.

Induction generators are similar to induction motors in construction and control requirements. The generator draws excitation current from the utility's electrical system and produces power when driven above its synchronous speed. In the typical induction generator, full output occurs at 5% above synchronous speed.

In order to prevent large transient overvoltage in the induction generator circuit, special precautions are required to prevent the generator from being isolated from the electrical system while connected to power factor correction capacitors. Also, an induction generator cannot operate without excitation current from the utility; only the **synchronous generator** has its own excitation.

Generator efficiency is a nonlinear function of the load and is usually at its maximum at or near the rated load (see Figure 45). The rated load estimate should include a safety factor to cover such

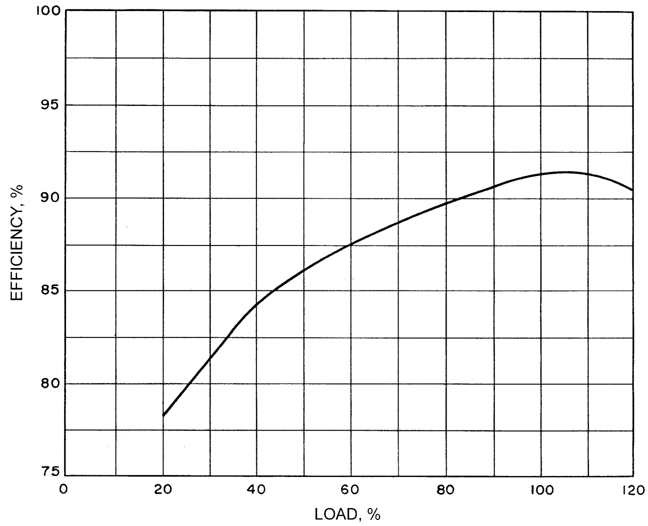

Fig. 45 Typical Generator Efficiency

transient conditions as short-term peaking and equipment start-up. Industrial generators are designed to handle a steady-state overload of 20 to 25% for several hours of continuous operation. If sustained overloads are possible, the generator ventilation system must be able to relieve the temperature rise of the windings, and the prime mover must be able to accommodate the overload.

Proper phase balance is extremely important. Driving three-phase motors from the three-phase generator presents the best phase balance, assuming that power factor requirements are met. Driving single-phase motors and building lighting or distribution systems may cause an unbalanced distribution of the single-phase loads that leads to harmonic distortion, overheating, and electrical imbalance of the generator. In practice, maximum phase imbalance can be held within 5 to 10% by proper distribution system loading.

Voltage is regulated by using static converters or rotating dc generators to excite the alternator. Voltage regulation should be within 0.5% from full load to no load during static conditions. Good electronic three-phase voltage sensing is necessary to control the system response to load changes and the excitation of paralleled alternators to ensure reactive load division.

The system power factor is reflected to the generator and should be no less than 0.8 for reasonable generator efficiency. To fall within this limit, the planned electrical load may have to be adjusted so the combined leading power factor substantially offsets the combined lagging power factor. Although more expensive, individual power factor correction at each load with properly sized capacitors is preferred to total power factor correction on the bus. On-site generators can correct some power factor problems, and for those cogeneration systems interconnected to the grid, they improve the power factor seen by the grid.

Generators operating in **parallel** with the utility system grid have different control requirements than those that operate **isolated** from the utility grid. A system that operates in parallel and provides emergency standby power if a utility system source is lost must also be able to operate in the same control mode as the system that normally operates isolated from the electric utility grid.

Control requirements for systems that provide electricity and heat for equipment and electronic processors differ depending on the number of energy sources and the type of operation relative to the electric utility grid. Isolated systems generally use more than one prime mover during normal operation to allow for load following and redundancy.

Table 7 shows the control functions required for systems operating isolated from the utility grid and systems operating in parallel

Table 7 Generator Control Functions

	Isolated		Parallel	
Control Functions	**One Engine**	**Two or More Engines**	**One Engine**	**Two or More Engines**
Frequency	Yes	Yes	No	No
Voltage	Yes	Yes	No	No
Power	No (Load following)	Yes (Division of load)	No	Yes (Division of load)
Reactive kVAR	No (Load following)	Yes (Division of kVAR)	Yes	Yes
Heat t_1	Supplement only	Supplement only	Load following	Load following
Heat $t_2 - t_x$	Reduce from t_1 or supplement	Reduce from t_1 or supplement	Reduce from t_1 and load following	Reduce from t_1 and load following
Cooling	Remove excess heat (tower, fan, etc.)	Remove excess heat (tower, fan, etc.)	Normally no (Emergency yes)	Normally no (Emergency yes)
Synchronizing	No	Yes	Yes	Yes
Black start	Yes	Yes (one engine)	Emergency use	Emergency use (one engine)

with the grid, both with single and with multiple prime movers. Frequency and voltage are directly controlled in a single-engine isolated system. The power is determined by the load characteristics and is met by automatic adjustment of the throttle. Reactive power is also determined by the load and is automatically met by the exciter in conjunction with voltage control.

In parallel operation, both frequency and voltage are determined by the utility service. The power output is determined by the throttle setting, which responds to the system heat requirements if thermal tracking governs, or to system power load if that governs. Only the reactive power flow is independently controlled by the generator controls.

When additional generators are added to the system, there must be a means for controlling the power division between multiple prime movers and for continuing to divide and control the reactive power flow. All units require synchronizing equipment.

The generator system must be protected from overload, overheating, short-circuit faults, and reverse power. The minimum protection is a properly sized circuit breaker with a shunt-trip coil for immediate automatic disconnect in the event of low voltage, overload, or reverse power. The voltage regulation control must prevent overvoltage. Circuit breakers for low voltage (below 600 V) may be air-type, and circuit breakers for medium voltage (up to 12,000 V) should be vacuum-type and should operate within 5 cycles.

Voltage must be held to close tolerances by the voltage regulator from no load to full load. A tolerance of 0.5% is realistic for steady-state conditions from no load to full load. The voltage regulator must allow the system to respond to load changes with minimum transient voltage variations. During parallel operation, the reactive load must be divided through the voltage regulator to maintain equal excitation of the alternators connected to the bus. True reactive load sensing is of prime importance to good reactive load division. An electronic voltage control responds rapidly and, if all three phases are sensed, better voltage regulation is obtained even if the loads are unbalanced on the phases. The alternator construction of a well-designed voltage regulator dictates the transient voltage variation.

Engine sizing can be influenced by the control system's accuracy in dividing real load. If one engine lags another in carrying its share of the load, the capacity that it lags is never used. Therefore, if the load sharing tolerance is small, the engines can be sized more closely to the power requirements. A load-sharing tolerance of less than 5% of unit rating is necessary to use the engine capacity to good advantage.

A load-sharing tolerance of 5% is also true for reactive load sharing and alternator sizing. If reactive load sharing is not close, a circulating current results between the alternators. The circulating current uses up alternator capacity, which is determined by the heat generated by the alternator current. The heat generated, and thus the alternator capacity, is proportional to the square of the current. Therefore, a precise control system should be installed; the added cost is justified by the possible installation of smaller engines and alternators.

POWER QUALITY

Electrical energy can be delivered to the utility grid, directly to the user, or to both. Generators for on-site power plants can deliver electrical energy equal in quality to that provided by the electric utility in terms of voltage regulation, frequency control, harmonic content, reliability, and phase balance. They can be more capable than the utility of satisfying stringent requirements imposed by user computer applications, medical equipment, high-frequency equipment, and emergency power supplies because other end users on the utility grid can create quality problems. The generator's electrical interface should be designed according to user or utility electrical characteristics.

COGENERATION REGULATORY ISSUES

REGULATORY AUTHORITIES

In the interest of energy conservation, a number of laws were developed in the United States to encourage the use of cogeneration and renewable energy sources by removing or limiting barriers to such plants.

United States Laws and Regulations

Public Utilities and Regulatory Policies Act (PURPA) mandates the following benefits for a qualifying cogenerator:

- Electric utilities must purchase power from cogenerators and small power producers (SPPs) at the price that the utility avoids by not producing it (avoided cost); and a utility must allow cogenerators to be paralleled (interconnected) with its grid if the cogenerator or SPP pays for it and complies with the utility's safety and protective requirements.
- The utility must provide supplemental power for facilities that do not cogenerate 100% of their power, standby power for emergency power during a cogenerator's outage, and maintenance power at nondiscriminatory rates during a planned outage for maintenance.
- Cogenerators and SPPs selling power to a utility are exempt from the "rate of return" (allowable return on investment) of utility regulations of the Federal Energy Regulatory Commission (FERC) and the Securities and Exchange Commission (SEC).
- Utilities may wheel (transmit) the cogenerated power over its grid to a location remote from the cogeneration plant and, under certain conditions, can be ordered by FERC to wheel it.

Qualifying Facility. FERC extends these benefits if the cogenerator becomes a qualifying facility (QF) by meeting the following qualifications:

- No more than 50% of a facility may be owned by an electric utility.
- At least 5% of its annual useful output must be useful thermal energy.
- Its efficiency must be at least 45% for topping cycles, and if more than 15% of its output is thermal, the minimum efficiency is 42.5%. There is no efficiency standard for bottoming cycles or for renewable resource plants. FERC efficiency is defined as

$$(\text{Power Output} + \text{Thermal Output}/2)\,/\,\text{LHV of Fuel Input}$$

- A cogenerator's retail power and steam rates may be regulated by state bodies under a broad authority and subject to local sales or gross receipts taxes.
- An SPP plant must be no greater than 80 MW unless powered by waste products or renewable energy.

State Laws and Regulations

PURPA provides FERC with the obligation to encourage cogeneration through rule making. FERC further charges each state with the responsibility of implementing PURPA rules. While these rules require electric utilities to buy from QFs that are interconnected or that wheel the power to the utility, FERC rules do not require purchase of QF power if the purchase will result in the utility's bearing a cost greater than if it had generated the power itself.

During the design of a facility, the purchasing utility should be consulted regarding its operation. For example, the facility may need to be designed to reduce the generation output at times when the purchasing utility is at reduced load.

While PURPA provides for rules that require all electric utilities to purchase from QFs, there are local cooperatives and municipalities in which the utility may ignore the purchase requirements due to lack of jurisdiction.

PURPA regulations provide for the sale of the power at an avoided cost. The methodologies and options for pricing vary from state to state. The local utility or the state Public Service Commission or Utility Commission can also provide some assistance.

In order to obtain QF certification, an application must be submitted to FERC. This process is typically undertaken at the early stages of the facility design. FERC may reject the certification if it finds that the representations in the application for certification were not obtained. For this reason, it is not uncommon to amend the application with FERC when the design of the facility changes.

Emission Regulations

Number 6 fuel oil, particularly in developing countries, is sometimes used in internal combustion engines and combustion turbines; but it can cause serious environmental pollution hazards, especially because it has a high sulfur content. Air pollution control authorities may require engine exhaust gases to be conditioned or may require special engines or fuels. Turbines may need water injection systems or selective catalyst reduction (SCR) systems. Requirements are based on the best available control technology (BACT). Typically the requirements become more stringent as the technology becomes available. The permitting agency can provide the most current requirements.

Reciprocating engines may be required to have catalytic converters or SCR. Boilers for steam turbine cogeneration normally require low NO_x burner design even if they are operating on high-quality natural gas. In most cases, the cost involved with the monitoring and control of emissions from larger systems is substantial. In addition, many cogenerators sustain substantial costs in permitting related to the emissions of the plant. Emissions trading allows one company that has an excess of emissions to purchase emission credits from another that is below its allocated emission level.

DISTRIBUTION SYSTEMS

Cogeneration system designers have a greater number of systems to choose from when the three major subsystems—production, distribution, and utilization (by the end user)—are all new. Significant limits are imposed on design choices when any one of these three elements already exists, has a substantial remaining life expectancy, and can carry out its assigned task(s) in a fairly efficient manner. Only when replacement is cost-effective, or necessary for a reason such as significant load increase, can options be broadened.

DESIGN CONSIDERATIONS

The geographical density of new or existing loads and the variety and type of existing systems to be connected to the cogeneration system have a profound effect on economic feasibility and choice of distribution.

In a totally new multifunction or multibuilding complex for a single owner or tenant who has total decision control, all energy systems can be designed for best compatibility with basic cogeneration concepts and criteria. However, the design is more difficult for an existing or new complex with multiple owners or tenants and diverse energy systems. In that situation, the cogeneration developer must (1) trail the various needs to cost-effectively pick up all possible services, (2) negotiate agreements with some of the occupants to make their own provisions, or (3) establish criteria to be followed by new facility participants.

In an industrial park or shopping center, for example, the developer might be required to serve one buyer with cooling and power only while another may need heating and cooling but no power. Or the developer may mandate that all buyers take all services and use compatible building energy systems. An example might be that all chilled water air handlers contain coils that meet the users' requirements with a Δt of 18°F (from 45 to 63°F). In any event, the developer's choices of plant design and distribution means will be limited by the extent or the variety of individual production, distribution, and terminal criteria desired by each buyer.

Optimized cogeneration mandates that maximum use be made of all forms of output energy from the production plant and from any existing mechanical/electrical equipment and systems. The prime mover outputs must be either in a form and quality of energy compatible with systems to be serviced by its main output or in some form suitable for conversion, such as from steam to absorption chilling.

OUTPUT ENERGY DISTRIBUTION MEDIA

Interconnects may have to be made to electrical systems of one or more voltages; to low-, medium-, or high-pressure steam systems; to chilled water or secondary coolant systems; to low-, medium-, or high-temperature hot water or thermal fluid systems; or to thermal energy storage systems. Each variation should be addressed in the planning stages.

Electrical

Electrical energy can produce work, heating, or cooling; it is the most transmittable form of energy. As a cogeneration output, it can be used to refrigerate or to supplement the prime mover's thermal output during periods of high thermal, low cooling demand. Mechanical aspects of a cogeneration system must be coordinated with electrical system designers who are familiar with power plant switchgear and utility and building interface requirements. See the section on Electrical Systems on page 7.27 for more information.

Steam

Engines and combustion and steam turbines can provide a range of pressure/temperature characteristics encountered in almost all steam systems. Their selection is basically a matter of choosing the prime mover and heat recovery steam generator combination that best suits the economic and physical goals.

Steam can also provide work, heating, or cooling, but with somewhat less range and flexibility than electric power. Distribution to remote users is more expensive with steam than with electricity and is less adaptable for remote production of work.

Economics limit the pressure and/or temperature (P/T) of steam available from gas turbine exhaust because the incremental cost/benefit ratio of increasing the heat recovery generator surface to yield a higher P/T is limited by a relatively fixed exhaust gas temperature, unless the turbine is refired with an auxiliary duct burner. However, steam turbines are not similarly limited, except by throttle conditions, because extraction can be accomplished from any point in the P/T reduction process of the turbine. Chapter 10 and Chapter 11 have further information on steam systems.

Chilled Water

The entire output of any prime mover can be converted to refrigeration and then chilled water, serving the wide variety of terminal units in conventional systems. In widely spread service distribution systems, choices must be made whether to serve outlying facilities with electric, steam, or hot water. All three can be used directly, for building or process heating and/or cooling, or indirectly, through heat exchangers and mechanical or thermally activated chillers located at remote facilities.

Central chilled water production and distribution to existing individual or multibuilding complexes is most practical if a chilled water network already exists and all that is required is an interconnect at or near the cogeneration plant. If the user building(s) already have one or more types of chillers in good condition, cogeneration and chilled water distribution may have diminished economic prospects unless applied on a small scale to individual buildings.

If chilled water distribution is feasible, the central cogeneration concept is easier to justify, and several techniques can be used to improve the viability of a cogenerated chilled water system by significantly reducing the owning and operating costs of piping and pumping systems and their associated components (e.g., valves, insulation, etc.). Such systems have been widely discussed, successfully developed, modified, and specified by many firms (Avery et al. 1990; Mannion 1988; Becker 1975).

The following are cost reduction concepts that are embodied in variable-flow water systems:

- Let the main pump(s) and the primary distribution system flow rate match the instantaneous sum of the demand flows from all cooling coils served by the primary loop. The chilled water should not be pumped off the primary loop in such a way as to circulate more chilled water through the secondary pump of the outlying buildings than the flow that it draws from the primary loop.
- Use two-way throttling control valves on all coils. Avoid three-way control valves for coil control or for bypassing chilled water supply into the chilled water return (e.g., end-of-line bypass to maintain a constant pump flow or system pressure differential). Valves must have suitable control characteristics for the system and full shutoff capability at the maximum pressure differential encountered.
- Select and circuit cooling coils for a large chilled water temperature difference (Δt as much as 24°F) while maintaining coil tube velocities of 5 to 10 fps and the required supply air conditions off the coil. Such coils may require 8 to 10 rows, but the additional pressure drop and cost are offset by the lower cost of the pump and distribution piping of long distribution systems. A system with Δt of 24°F requires only one-third the flow rate required by one with a Δt of 8°F.
- Care must be exercised by the designer for successful implementation of these concepts. The *Air-Conditioning Systems Design Manual* (ASHRAE 1993) has further design information.

Hot Water

Cogeneration thermal output is well adapted to low-, medium-, or high-temperature water (LTW, MTW, HTW) distribution systems. The major difference between chilled water and hot water systems is that even LTW systems (up to 250°F) can be designed with a Δt as high as 100°F with low flow by using different series-parallel terminal circuiting, as described in Chapter 12. This way, even equipment that is limited to Δt = 20°F (e.g., radiators, convectors) can be adapted to large system temperature differences. For example, besides those circuits given in Chapter 12, unit heaters can be piped in series and parallel on a single hot water building loop without a conventional supply and return line. Parallel runs of five heaters each can drop in 20°F increments. The first group drops the temperature from 250 to 230°F, the last from 170 to 150°F, and all are sized at the 170 to 150°F range. Fan cycling off each local thermostat maintains control despite the different temperatures.

Similarly, larger heaters with conventional small-row coils (not metal cores) can be fitted with three-way modulating bypass valves, sized for a 10 to 40°F drop, with the through-flow and bypass flow from the first flowing to the second, and so forth, using only one primary loop.

Medium- (250 to 350°F) and high- (350°F and higher) temperature water systems are designed with an even higher Δt, but they are not customarily connected directly to the primary loop. These systems can be connected to steam generators in outlying buildings that have steam distribution and steam terminal devices.

When a choice can be made, a prime mover's thermal output should be used according to the following priorities. Apply the output first for useful work, second for an efficient form of thermal conversion, and third for productive thermal use. For example, if a combustion turbine's exhaust can cost-effectively produce shaft power or, if not, heat some process, it should be considered for these functions rather than for a low-pressure absorption function, which is the least efficient application.

For both hot water and chilled water distribution systems, a common approach is to lower the hot water supply temperature as the ambient temperature rises and to raise chilled water supply temperature as loads are reduced. Both techniques reduce pipe transmission and fuel or electrical costs for heating or cooling, and stabilize valve control; but the impact of increased pumping costs is often overlooked.

Below some part-load condition in both hot water and chilled water systems, the cost of reducing flow by varying pump speed and air volume may be more than the saving in energy. This is especially true for chilled water systems, when the cascading effect of VAV fan power reduction from lower supply air temperatures, together with pumping savings, becomes more significant than the low-load chiller savings from raising the chilled water system temperature. Also, humidity control for the space limits how much the chilled water temperature can rise.

These phenomena need to be examined in determining the part-load condition at which hot water and chilled water system scheduling might be advantageously modified.

COGENERATION UTILIZATION SYSTEMS

Good cogeneration planning responds to the requirements of the end user and strives to use to the maximum the equipment and the energy it produces. In order for cogeneration to be economically feasible, the energy recovered must match the site requirements well and avoid as much waste as possible. Depending on the design and operating decisions, users may tie into the electric utility grid for some or all of their electric energy needs.

The selection of prime movers is based partially on user thermal requirements. For maximum heat recovery, the thermal load must remove sufficient energy from the heat recovery medium to lower its temperature to that required to cool the prime mover effectively. A supplementary means for rejecting heat from the prime mover may be required if the thermal load does not provide adequate cooling during all modes of operation or as a backup to thermal load loss.

Internal combustion reciprocating engines have the lowest heat/power ratio, yielding most of the heat at a maximum temperature of 200 to 250°F. This jacket water heat can be used by applications requiring low-temperature heat or to preheat inlet air. Higher temperature levels from the exhaust are limited to approximately one-third the engine's recoverable thermal energy.

Gas turbines can provide a larger quantity and a better quality of heat per unit of power, while extraction steam turbines can provide even greater flexibility in both the quantity and the quality (temperature and pressure) of heat delivered. Externally fired prime mover cycles, such as the Stirling cycle, gas turbines with steam injection (Cheng cycle), and steam-operated absorption chillers for inlet air cooling are also flexible in the quantity and quality of thermal energy they can provide.

If a gas turbine plant is designed to serve a variety of loads (e.g., direct drying, steam generation for thermal heating or cooling, and shaft power), it is even more flexible than one that serves only one or two such loads. Of course, such diverse equipment service must be economically justified.

AIR SYSTEMS

Large, central air handlers with deep and/or suitably circuited coils that operate with a large cooling and heating Δt are available to reduce distribution piping and pumping costs. These units serve a multiplicity of control zones or large single-zone spaces with air distribution ductwork. Smaller units such as perimeter fan-coil units directly condition spaces with small lengths of duct or no ductwork. Thus, they are totally decentralized from the air side of the system. The maximum Δt through the coil is only 12 to 14°F.

Both central and decentralized air handlers can be coupled with cogeneration in mildly cold climates in a two-pipe changeover configuration with a small, intermediate season electric heating coil. This arrangement can heat or cool different zones simultaneously during the intermediate seasons. Chilled water is available to cool any zone. Zones needing heat can cut off the coil and turn on the electric heat. A four-pipe system is unnecessary in these conditions.

When the building's balance point is reached (i.e., when all zones need heat), the pumping system is changed over to hot water. The concept applies best (and mostly) to perimeter zone layouts and to large air handlers with economizer cycles that do not need chilled water below the ambient changeover point (i.e., when economizer cooling can satisfy their loads). However, the application must have a high enough thermal demand for process or other non-space-heating loads to absorb the extra thermal energy produced by the engine generator for this additional electric heating load.

If the predominant thermal load during this period is for space heating and cooling (both being at a low demand level), it makes no sense to exacerbate the already low heat/power ratio by designing for more electrical load with no use for the heat generated. Hospitals and apartment houses with high process heating demands are examples of suitable applications, but single-function office buildings are not.

The significance of this cogeneration design is that the prime mover's electrical output can be swung from a motor-driven refrigeration load, which is less during intermediate seasons, to the electric heating function as long as the additional thermal energy can be absorbed. This can work well with engine-generators and electric refrigeration.

An absorption chiller might be a better match where the site's heat/power ratio is low, such as for an office building. But a mechanical chiller without cogeneration may offer an even better return than an absorption system.

HYDRONIC SYSTEMS

Hydronics, particularly in buildings with no need for process or high-pressure steam, is a much more widely used transport medium than steam. Information on the various types of terminals and systems may be found in Chapter 3. Loosely defined, hydronics covers all liquid transport systems, including (1) chilled water, hot water, and thermal fluids that convey energy to locations where space and process heating or cooling occur; (2) domestic or service hot water; (3) coolant for refrigeration or a process; (4) fresh or raw water for potable or process purposes; and (5) wastewater.

From the standpoint of cogeneration design, all these applications are relevant, but some are not HVAC applications. Each application may offer an opportunity to improve the cogeneration system. For example, a four-pipe, two-coil system and a two-pipe, common coil system offer similar options in plant design, with the four-pipe system offering superior flexibility for individual control of space conditions. All-electric, packaged terminal air conditioners offer little opportunity for a sizable plant, unless a substantial thermal demand (e.g., for process heat) exists in addition to the normal comfort space-conditioning needs. Without the thermal demand, the only option is to install a plant that generates a fraction of the total electrical demand while satisfying service water heating requirements, for example, and to buy the bulk of the electricity required. If the site's heat/power ratio is so low that it cannot support the lowest ratio prime mover for a large portion of the electrical demand, then a smaller plant that can operate close to a base-loaded electric generating condition might be considered.

The temperature required by the site loads also influences the feasibility of cogeneration. If a temperature of 110 to 120°F can satisfy most of the site's heating requirements (with air handlers, fan-coil units, or multitiered finned radiators), a central motor-driven heat pump might offer a more cost-effective alternative to a prime mover in a cogeneration plant that produces more heat than required.

Even if refrigeration from the heat pump is not used, the heat pump takes only 42% of the source fuel from the electric utility's boiler to produce the same heat energy as an 80% efficient on-site boiler. The section on Economic Coefficient of Performance (ECOP) on page 7.33 has further information. (ECOP is a dimensionless performance index representing the energy output of a system per unit energy input of the source fuel measured in the ratio of costs of different energy streams per unit of energy.)

The heat pump is even more effective if there is a simultaneous demand for both refrigeration and heating. To the extent that refrigeration is in excess of demand, it can be used instead of air handler economizer cooling or for fan-coil units not equipped with economizers. If a cogeneration plant incorporates a heat pump, it may produce too much heat for the site to absorb, thereby reducing the heat pump utilization factor. More information on heat pumps may be found in Chapter 8.

UNITARY AND PACKAGED HVAC SYSTEMS

When a building has a predominance of self-contained HVAC units for cooling and/or heat pump applications, as discussed in the previous section on Hydronic Systems, cogeneration is not feasible unless the site has another, substantial demand for heat.

SERVICE HOT WATER SYSTEMS

Service hot water systems can be a major and preferred user of thermal energy from prime movers and can often constitute a fairly level year-round load, when averaged over a 24 to 48 h period. Service hot water use in hospitals, domiciliary facilities, etc., is usually quite variable in a 24 h weekday or 48 h weekend profile; heat storage permits expanded use of the thermal output and justification for larger prime movers.

The service hot water demand often provides a strong case for consuming the entire thermal output with packages sized for the 24 h demand, instead of for space cooling or heating. For the larger prime movers with ebullient cooling, and particularly on sites with wide climate variations, the incremental benefits of using the thermal output for space heating as well as for service hot water may not be great, and the annual utilization factor of the incremental thermal output may be relatively low.

DISTRICT HEATING/COOLING SYSTEMS

The high cost of a central plant and distribution system generally mandates that the economic returns develop soon after the plant and distribution systems are complete. Because of the high risk, the developer must have satisfactory assurance that there are enough buyers for the product. Furthermore, the developer must know what distribution media and quality are best for connection to existing buyers and must install a system that is flexible enough for future buyers.

Generally, the load on a district system tends to level out because of the great diversity factor of the many loads and noncoincident peaking. This variety also makes plant sizing and consumption estimate aspects difficult. Statistical data from case studies and broad assumptions may be the only source of information. Chapter 11 has further information on district systems.

HYBRID, UNCONVENTIONAL PROCESS SYSTEMS

Commercial and Industrial HVAC

Residential and domiciliary facilities without a high ratio of occupants to ground floor area (e.g., single-family residences) are less desirable for large cogeneration projects than those with a high geographical load density or those that can use a substantial portion of the plant output for comfort or process heating and cooling.

Most applications do not have heat/power ratios that match a prime mover sized to meet the electrical peak demand. Without a suitable market for excess electricity or heat, the only rational option is to downsize the generator so that all the heat generated is totally consumed by highly efficient equipment. Any electrical demand not generated by the plant must be purchased, and any shortage in thermal capacity must be purchased or generated separately.

Desiccant Systems

Both solid and liquid desiccants are used for dehumidification and process drying. All of them produce heat in the process, and some or all of this heat must be removed. All reusable desiccants must be reactivated with heat. Some or all of this cooling or heating requires an external energy source, which can be incorporated in the cogeneration process. Chapter 22 has further information on desiccants.

Impact of Other Conservation Measures

A feasibility study for cogeneration should be based on the projected baseload of the site. If actual consumption is substantially lower as a result of other conservation measures, the savings will not be as great as projected unless the excess energy can be sold to others. On the other hand, a partial cogeneration installation is not oversized for reduced demands, but it might not be economically feasible because of the unfavorable economy of scale.

COGENERATION SYSTEM CONCEPTS

PACKAGED SYSTEMS

Packaged cogeneration systems are available from 6 to over 1000 kW. Both direct-drive engine-generators and engine-chillers with heat recovery exchangers are available. Standard designs are available, while some units are designed for specific applications.

Engine-Generators. The major advantage of these packages is the relatively simple hookup to building services; but proper planning for economic matching of the output of the package and the requirements of the facility still requires engineering skills. A standard package tends to eliminate the problems of interactions between components.

Engine-Driven Compressors and Chillers. Coupling a reciprocating engine to a reciprocating or rotating compressor has inherent torsional vibration effects that can be corrected. Such defects can be more easily eliminated in the factory design and assembly than on site. These defects do not occur when gas turbine prime movers are used.

CUSTOM-ENGINEERED SYSTEMS

The high cost of engineering and the risk of technical, economic, and environmental failure, as well as regulatory restraints, mandate extraordinary care and skill to successfully design and build custom-engineered cogeneration systems.

LOAD PROFILING AND PRIME MOVER SELECTION

The selection of a prime mover is determined by the facility's thermal or electrical load profile. The choice depends on (1) the ability of the heat/power ratio of the prime mover to match the facility loads; (2) the decision whether to parallel with the public utility or be totally independent; (3) the decision whether to sell excess power to the utility; or (4) the desire to size to the thermal baseload.

No matter which basis is used to choose the prime mover, the degree of use of the available heat determines the overall system efficiency; this is the critical factor in economic feasibility. Therefore, each prime mover's heat/power characteristics must be analyzed over its range of operating loads to make the best choice. Maximizing efficiency may not be as important as maximizing the total economic value of the cogeneration system output at all loads.

Cogeneration systems paralleled with the utility grid can operate at peak efficiency (1) if the electric generator can be sized to meet the valley of the thermal load profile, operate at a base electrical load (100% full load) at all times, and purchase the balance of the site's electric needs from the utility; or (2) if the electric generators are sized for 100% of the site's electrical demand and the heat recovered can be fully used at that condition, with thermal demands in excess of the recovered heat provided by supplementary means and excess power sold to the utility.

The heat output to the primary process is determined by the engine load. It must be balanced with actual requirements by being supplemented or by having excess heat rejected through peripheral devices such as cooling towers. Similarly, if more than one level of heat is required, controls are needed to (1) reduce heat from a higher level when it is required at the primary level, (2) supplement heat if it is not available, or (3) reject heat when availability exceeds the requirements.

In an isolated plant with more than one prime mover, controls must be added to balance the power output of the prime movers and to balance the reactive power flow between the generators.

Generally, an isolated system requires that the prime movers supply the needed electrical output, with the heat availability controlled by the electrical output requirements. Any imbalance in heat requirements results in burning supplemental fuels or wasting surplus recoverable heat through the heat rejection system.

Supplemental firing and heat loss can be minimized during parallel operation of the generators and the electric utility system grid by adjusting the prime mover throttle for the required amount of heat. This procedure is called thermal load following. The amount of electrical energy generated then depends on heat requirements; imbalances between the thermal load and the electrical load are carried by the electric utility either through absorption of excess generation or through the delivery of supplemental electrical energy to the electrical system.

Similarly, electrical load tracking controls the electric output of the generator(s) to follow the site's electrical load, while using, selling, storing, or discarding (or any combination of these methods) the thermal energy output. To minimize waste of thermal energy, the plant can be sized to track the electrical load profile up to the generator capacity, which is selected for a thermal output that matches the valley of the thermal profile. Supplemental electric power is purchased and/or thermal energy generated by other means when the thermal load exceeds the generator's profile valley.

These tracking scenarios require either a fairly accurate set of coincident electric and thermal profiles that are typical for a variety of repetitious operating modes or a set of typical daily, weekend, and holiday accumulated thermal consumption requirements for the sizing of an appropriate thermal energy storage plant.

PEAK SHAVING

High electrical demand charges in many areas, ratchet rates (minimum demand charge for 11 months = x% of the highest annual peak, leading to an actual payment that in many months is more than the metered charge), and utility capacity shortages have led to demand side management (DSM) by utilities and their consumers.

COGENERATION SYSTEM PERFORMANCE

Power Plant Incremental Heat Rate

Typically, cogeneration power plants are rated against incremental heat rates (and thus incremental thermal efficiency) by comparing the incremental fuel requirements with the base case energy needs of a particular site. For example, if a gas engine generator with a design heat rate of 10,000 Btu/kWh (34% thermal efficiency) provided steam or hot water through waste heat recovery to a particular system that would save 4000 Btu/h energy input, the incremental heat rate of the cogeneration power plant would be only 6000 Btu/kWh, which translates to a thermal efficiency of 57%. If the same system is applied to another site where only 2000 Btu/h of the recovered heat can be used (against the availability of 4000 Btu/h), the incremental heat rate for the same power plant rises to 8000 Btu/kWh, with thermal efficiency dropping to 43%.

Thus, the cogeneration power plant performance really depends on the required heat/power ratio for a particular application, and it is only according to this ratio that the type of cogeneration configuration should be chosen.

A system that requires 1000 kWh electrical energy and 7000 lb low-pressure steam at 30 psig (heat/power ratio of about 0.5, or 7 lb steam per kilowatt-hour) can be used to further illustrate the above requirement for measuring cogeneration system performance. In this example, a 1000 kW gas engine-generator cogeneration system provides a maximum of 1500 lb/h steam, and the balance of 5500 lb/h would have to be met by a conventional boiler with 75% thermal efficiency. Thus, the total power requirement for this example cogeneration system is about 10×10^6 Btu/h (34% efficient) for the gas engine and 7.4×10^6 Btu/h for the boiler input for a total of 17.4×10^6 Btu/h.

If instead a gas turbine cogeneration system with the same power and heat requirements is used, the gas turbine-generator with a heat rate of 13,650 Btu/h per kilowatt (thermal efficiency of 25%) would supply both 1000 kW of power and 7000 lb/h steam with only 13.65×10^6 Btu/h fuel input. Thus, the gas engine-generator, although having a very high cogeneration thermal efficiency, is not suitable for the combination system because it would use 3.75×10^6 Btu/h (nearly 28%) more power than the gas turbine for the same total output.

Economic Coefficient of Performance (ECOP)

The normal procedure for evaluating COP does not provide a workable method for comparing efficiencies of dissimilar energy streams. Instead, each energy stream should be valued on the same energy basis and at prevailing rates, with 1 kWh of electrical energy taken as 3412 Btu and fuel input as the low heating value (LHV) in Btu. Then a direct comparison by means of an economic coefficient of performance (ECOP) can be made. Rates with step charges based on the load factor must be carefully evaluated to be sure the appropriate incremental cost is used.

For example, with energy from the utility at \$0.08/kWh and natural gas supply at \$2.75 per thousand cubic feet (LHV of 900 Btu/ft^3), 1000 Btu of electrical energy costs \$0.023 (0.08/3.412), and 1000 Btu of natural gas costs \$0.003 (2.75/900). Thus, the ECOP can be defined as all output energy in desired output forms, converted in terms of economic costs, divided by all energy input, again converted in terms of economic costs of each energy stream. For this example, the electrical energy costs (0.023/0.003) = 7.67 times more than the equivalent energy from natural gas. This ratio is used to calculate the ECOP in the following examples.

Example 1. Calculate the ECOP of a low-pressure steam absorption chiller with motor auxiliaries totaling 25 hp per 1000 tons. The on-site boiler generates 19 lb/ton·h steam at 15 psig (1164 Btu/lb enthalpy) at 80% efficiency from feedwater at 0 psig, 212°F (180 Btu/lb enthalpy).

Solution:

On-site fuel input = 19(1164 – 180)/0.8
= 23,400 Btu/ton·h

The electrical input generated off site supplies 25 hp/1000 tons at a motor efficiency of 0.87.

Off-site electrical input = (0.746 kW/hp × 0.025 hp/ton)/0.87
= 0.0214 kW input/ton
= 3412 × 0.0214 = 73 Btu/ton·h

The equivalent total input per unit of output (cooling only) is

23,400 + (73 × 7.67) = 23,960 Btu/ton·h (equivalent energy)

Thus, the ECOP for 12,000 Btu/ton·h output and 23,960 Btu/ton·h equivalent input (at the preceding power costs) is

12,000/23,960 = 0.501

Example 2. Calculate and compare the ECOP for the same cooling output, using an engine-driven, vapor-compression chiller and piggyback absorption chiller. The engine has an 8600 Btu/hp·h heat rate (30% shaft thermal efficiency) and 3470 Btu/hp·h of saturated steam at 15 psig heat recovery (40% heat recovery rate).

Solution: The total cooling output is

From engine-chiller at 1 hp/ton cooling	12,000 Btu
From the absorption chiller at 19(1164 – 180) = 18,700 Btu/ton·h and for 3470 Btu/ton·h (cooling), 12,000 × 3470/18,700 =	2,227 Btu
Total cooling	14,227 Btu

Off-site electrical input for absorption chiller auxiliaries at 25 hp per 1000 tons, as detailed in Example 1, is 73 Btu/ton·h. The equivalent total input energy is

$$8600 + (73 \times 7.67) = 9160 \text{ Btu}$$

Thus, the ECOP for above is 14,227/9,160 = 1.553, which is more than three times that of the conventional system covered in Example 1.

A similar approach can produce ECOPs for different configurations and with different electrical and fuel (gas or oil) costs. But an ECOP should be considered an indicator only and should be followed with a life-cycle cost analysis to make a final decision.

THERMAL ENERGY STORAGE TECHNOLOGIES

Thermal energy storage (TES) can decouple power generation from the production of process heat, allowing the production of dispatchable power while fully using the thermal energy available from the prime mover. Thermal energy from the prime mover exhaust can be stored as sensible or latent heat and used during peak demand periods to produce electric power or process steam/hot water. However, the additional material and equipment necessary for a TES system add to the capital costs. As a result, the economic benefits of adding TES to a conventional cogeneration system must outweigh the increased cost of the combined system.

The selection of a specific storage system depends on the quality and quantity of recoverable thermal energy and on the nature of the thermal load to be supplied from the storage system. Chapter 34 of the *ASHRAE Handbook—Applications* has more information on thermal storage. The TES systems and technologies that are being considered for power generation applications can be categorized by storage temperature.

High-temperature storage can be used to store thermal energy from sources like the gas turbine exhaust stream at high temperatures (**heat storage**). High-temperature TES options such as oil/rock storage, molten nitrate salt storage, and combined molten salt and oil/rock storage are well developed and commercially available.

Low-temperature TES technologies store thermal energy at temperatures below ambient temperature (**cool storage**) and can be used for cooling the air entering gas turbines. Examples of cool storage include commercially available options such as diurnal ice storage, as well as more advanced schemes represented by complex, compound chemisorption TES systems.

A number of emerging issues may limit the number of useful applications of cogeneration. One of these is a mismatch between the demand for electricity and the demand for thermal energy on a daily basis. Increasingly, utilities are requiring cogenerators to provide dispatchable power, while most industrial thermal loads are relatively constant during the day. **Diurnal** TES can decouple the generation of electricity from the production of thermal energy, allowing the cogeneration facility to supply dispatchable power. Diurnal TES stores thermal energy recovered from the exhaust of the prime mover (gas turbine) to meet daily variations in the demand for electric power and in thermal loads.

COMBINED CYCLES

Combined-Cycle Power Plants

Combined-cycle power plants are based on combination gas turbine and steam turbine generators to achieve high efficiency in power generation. High-temperature exhaust from the gas turbine produces high-temperature, high-pressure steam that operates the steam turbine generator. Normally, a combined-cycle station consists of the following major components (Figure 46):

- Gas turbine generator set
- Steam turbine generator set
- Waste heat recovery boiler
- Steam and feedwater makeup system
- Condenser
- Electrical equipment
- Control, safety, and instrumentation systems

Gas turbines, with normal efficiencies in the range of 28 to 38%, can provide approximately 25 to 30% more energy in the form of high-quality steam from the exhaust stream. This energy can be recovered in waste heat boilers, and for large sizes, such boilers have separate drums for high-pressure and low-pressure steam. For smaller systems, only a high-pressure steam circuit may be provided to lower costs. The low-pressure steam heat exchanger in the waste heat boiler can be used for feedwater heating at the required temperatures. The superheated steam is normally fed to condensing steam turbines. For higher overall

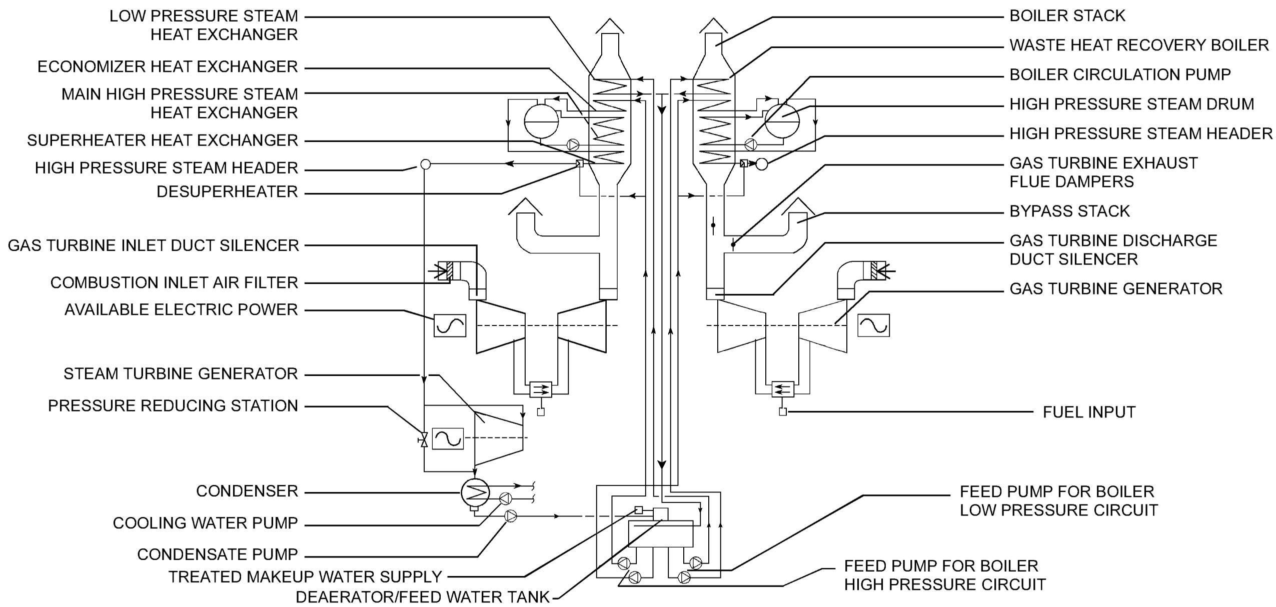

Fig. 46 Typical Combined-Cycle Power Plant Schematic

thermal efficiency, the steam turbine can be of the extraction condensing type when there is a considerable, continuous low-pressure steam demand.

Combined-cycle power plants offer the following advantages compared to conventional steam thermal power plants:

- Higher thermal efficiencies, presently up to 57% on natural gas fuel, compared to peak efficiencies of only 38 to 39% for large steam turbine power stations
- Shorter starting time, especially for gas turbines that can be loaded to nearly 70% of station capacity in only 15 to 20 min
- Better part-load performance with multiple gas turbine installations, allowing flexibility in meeting load demands
- Shorter completion time normally, with 70% power available within 1 year with simple gas turbine cycle operation, and the addition of waste heat recovery boiler and steam turbine cycle taking 1 more year
- Lower capital costs
- Lower water consumption/cooling requirements because the condensing steam cycle is for only one-third of the final power station capacity
- Lower environmental pollution

UNCONVENTIONAL SYSTEMS

Fuel Cells

Fuel cells convert the chemical energy of a hydrogen-based fuel directly into electricity without combustion. In the cell, a hydrogen-rich fuel passes over the anode, while an oxygen-rich gas (air) passes over the cathode. The catalysts help split the hydrogen into hydrogen ions and electrons. The hydrogen ions move through an external circuit, thus providing a direct current at a fixed voltage potential. A typical packaged fuel cell power plant consists of a fuel processor, which chemically combines the supply fuel with steam to form a hydrogen-rich gas; a fuel cell section, which consists of many individual cells; and a power conditioner, which transforms the dc power into ac power.

An advanced phosphoric acid fuel cell, the molten carbonate fuel cell, the solid oxide fuel cell, and the alkaline fuel cell are continuing to be developed. Fuel cells are also being developed small enough to power hybrid city buses. The heat generated by and efficiency of several types of fuel cells are as shown in Table 8. Emissions from fuel cells are very low; NO_x emissions are less than 20 ppm. Large phosphoric acid fuel cells are commercially available. For example, one packaged fuel cell power plant has the following characteristics:

Electrical output	200 kW
Thermal energy available	760,000 Btu/h @ 165°F
	or 350,000 Btu/h @ 250°F
	or 350,000 Btu/h @ 140°F
Electrical efficiency	40%
Overall efficiency	85%
Size	10 ft × 10 ft × 18 ft
Weight	40,000 lb

Table 8 Heating Value and Efficiency of Several Fuel Cells

Fuel Cell Type	Lower Heating Value, Btu/kWh	Efficiency, $kW_{out}/kW_{fuel\ in}$
1st generation phosphoric acid	8500	40%
Advanced phosphoric acid	7600	45%
Molten carbonate	6600	52%

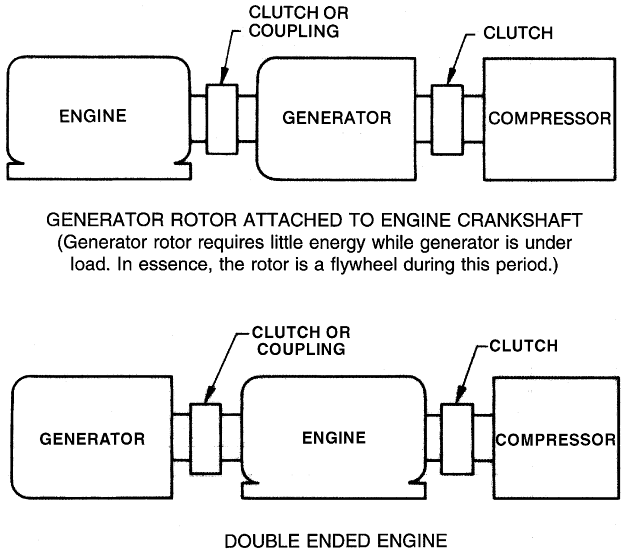

Fig. 47 Dual-Service Applications

Continuous-Duty Standby

An engine that drives a refrigeration compressor can be switched over automatically to drive an electrical generator in the event of a power failure (Figure 47). This plan assumes that loss of the compressor service can be tolerated in an emergency.

If the engine capacity of such a dual-service systems equals 150 or 200% of the compressor load, the power available from the generator can be delivered to the utility grid during normal operation. While induction generators may be used for this application, a synchronous generator is required for emergency operation if there is a grid outage.

Dual-service arrangements have the following advantages:

- In comparison to two engines in single service, lower capital investment and reduced space and maintenance are required, even after allowing for the additional controls needed with dual-service installations.
- Because they operate continuously or on a regular basis, dual-service engines are more reliable than emergency (reserved) single-service engines.
- Engines that are in service and running can be switched over to emergency power generation with minimal loss of continuity. Hospital authorities recognize this and prefer dual-service to the cold start of an emergency generator.

Expansion Engines/Turbines

Expansion turbines are turbines that use compressed gases available from gas-producing processes to produce shaft power. One application compresses natural gas from high-pressure pipelines in much the same way that a steam turbine replaces a pressure reducing valve. These engines are used mainly, however, for cryogenic applications to about –320°F (e.g., oxygen for steel mills, low-temperature chemical processes, and the space program). Relatively high-pressure air or gas expanded in an engine drives a piston and is cooled in the process. At the shaft, about 42 Btu/hp is removed. Available units, developing as much as 600 hp, handle flows ranging from 100 to 10,000 cfm. Throughput at a given pressure is controlled by varying the cutoff point, the engine speed, or both. The conversion efficiency of heat energy to shaft work ranges from 65 to 85%. A 5 to 1 pressure ratio and an inlet pressure of 3000 psig are recommended. Outlet temperatures as low as –450°F have been handled satisfactorily.

Such a process consumes no fuel, but it does provide shaft energy, as well as expansion refrigeration, which fits the definition of cogeneration. Similarly, a back pressure steam turbine, used instead of a pressure reducing valve, generates productive shaft power and permits the use of the residual thermal energy.

Natural gas from a high-pressure pipeline can be run through a turbine to produce shaft energy and then burned downstream. See the section on Combustion Turbines for more information.

DESIGN AND INSTALLATION

ENGINES

Additional information on engines may be found at the beginning of this chapter in the section on Reciprocating Engines. Specific information on the application of engines to chillers, heat pumps, and refrigeration equipment may be found in Chapter 47.

Jacket Water Systems

The circulating water and oil systems must be kept clean because the internal coolant passages of the engine are not readily accessible for service. The installation of piping, heat exchangers, valves, and accessories must include provisions for internal cleaning of these circuits before they are placed in service and, when possible, for maintenance access afterwards.

Coolant fluids must be noncorrosive and free from salts, minerals, or chemical additives that can deposit on hot engine surfaces or form sludge in relatively inactive fluid passages. Generally, engines cannot be drained and flushed effectively without major disassembly, making any chemical treatment of the coolant fluid that can produce sediment or sludge undesirable.

An initial step toward maintaining clean coolant surfaces is to limit fresh water makeup. The coolant system should be tight and leak-free. Softened water or mineral-free water is effective for initial fill and makeup. Forced-circulation hot water systems may require only minor corrosion-inhibiting additives to ensure long, trouble-free service. This feature is one of the major assets of hot water heat recovery systems.

Water-Cooled Engines

Heat in the engine coolant should be removed by heat exchange to a separate water system. Recirculated water can then be cooled in open-circuit cooling towers, where water is added to make up for evaporation. Closed-circuit coolant of all types (e.g., for closed-circuit evaporative coolers, radiators, or engine-side circuits of shell and tube heat exchangers) should be treated with a rust inhibitor and/or antifreeze to protect the engine jacket. Because engine coolant is best kept in a protected closed loop, it is usually circulated on the shell side of an exchanger. A minimum fouling factor of 0.002 should be assigned to the tube side.

Jacket water outlet and inlet temperature ranges of 175 to 190°F and 165 to 175°F, respectively, are generally recommended, except when the engines are used with a heat recovery system. These temperatures are maintained by one or more thermostats that bypass water as required. A 10 to 15°F temperature rise is usually accompanied by a circulating water rate of about 0.5 to 0.7 gpm per engine horsepower.

Installation involves (1) sizing water piping according to the engine manufacturer's recommendations, (2) avoiding restrictions in the water pump inlet line, (3) never connecting piping rigidly to the engine, and (4) providing shutoffs to facilitate maintenance.

If the cost of pumping the water at conventional jacket water temperature can be absorbed in the external system, a hot water system is preferred. However, the cooling tower must be sized for the full jacket heat rejection if there are periods when no load is available to absorb it (see Chapter 36 for information on cooling tower design). Most engines are designed for forced circulation of the coolant.

Where several engines are used in one process, independent coolant systems for each machine can be used to avoid complete plant shutdown from a common coolant system component failure. Such independence can be a disadvantage in that the unused engines are not maintained at operating temperature, as they are when all units are in a common circulating system. If the idle machine temperature drops below the dew point of the combustion products, corrosive condensate may form in the exhaust gas passages each time the idle machine is started.

When substantial water volume and machinery mass must be heated up to operating temperature, the condensate volume is quite significant and must be drained. Some contaminants will get into the lubricant and reduce the service life. If the machinery gets very cold, it may be difficult to start. Units that are started and stopped frequently require an off-cycle heating sequence to lessen the exposure to corrosion.

Forced lubrication by an auxiliary, externally powered lubricant pump is beneficial to the engine when the unit is off. Continuously bathing the engine parts in oil and maintaining engine temperatures near their normal operating temperature improves engine life and lowers the engine maintenance costs. This practice should be evaluated against the pumping losses and the radiation heat losses to the engine room environment.

Still another concept for avoiding a total plant shutdown, and even avoiding the probability of any single engine shutdown, requires the following arrangement:

- A common, interconnected jacket water piping system for all the engines.
- An extra standby device for all auxiliaries such as pumps and heat exchangers. Valves must be installed to isolate any auxiliary that fails or is out for preventive maintenance while permitting continued operation.
- Header isolation valves to permit continued plant operation while any section of the common piping is serviced or repaired.

This common piping arrangement permits continuous full-load plant operation if any one auxiliary suffers an outage or needs maintenance; no more than one engine in the battery can have a forced outage if the headers suffer a problem. On the other hand, independent, dedicated auxiliaries for each engine can force an engine outage whenever an auxiliary is down, unless each such auxiliary is provided with a standby, which is not a practical option. Furthermore, the common header concept avoids the possibility of a second engine outage when any of the second engine's support components fails while the first is out for major repair. It also permits a warm start of any engine by the circulation of a moderate flow of the hot jacket water through any idle engine.

Ebullient Cooling

Ebullient cooling systems of any size are not favored by many designers and manufacturers of large industrial engines, but they are commonly used. This method presents some special hazards for water treatment. Evaporation, which should not take place in the engine, sometimes does, and the concentration of dissolved minerals increases at this critical location. This problem can be minimized by using gravity or forced circulation to encourage high flow rates and by reducing minerals added to the system to a negligible amount.

Elevating the flash tank or using pumps to maintain a high static pressure also reduces the tendency for vaporization in the engine, which can cause engine failure or fouling. The water treatment must be adequate to prevent corrosion from free oxygen brought in with returned condensate and makeup water in the flash tank. Additional treatment should control corrosion in the heat recovery muffler, the steam separator, and the condensate system.

The water level of separate steam-producing engine cooling systems must be controlled to prevent backflow through the steam nozzle of idle recovery apparatus. A back pressure regulating valve, a steam check valve, or an equalizing line can prevent this problem.

Exhaust Systems

Engine exhaust must be safely conveyed from the engine through piping and any auxiliary equipment to the atmosphere at an allowable pressure drop and noise level. Allowable back pressures, which vary with engine design, run from 2 to 25 in. of water. For low-speed engines, this limit is typically 6 in. of water; for high-speed engines, it is typically 12 in. Adverse effects of excessive pressure drops include power loss, poor fuel economy, and excessive valve temperatures, all of which result in shortened service life and jacket water overheating.

General installation recommendations include the following:

- Install a high-temperature, flexible connection between the engine and the exhaust piping. Exhaust gas temperature does not normally exceed 1200°F, but 1400°F may be reached for short periods. An appropriate stainless steel connector may be used.
- Adequately support the exhaust system downstream from the connector. At the maximum operating temperature, no weight should be exerted on the engine or its exhaust outlet.
- Minimize the distance between the silencer and the engine.
- Use a 30 to 45° tail pipe angle to reduce turbulence.
- Specify tail pipe length (in the absence of other criteria) in odd multiples of $12.5(T_e^{0.5}/P)$, where T_e is the temperature of the exhaust gas (Rankine), and P is exhaust frequency (pulses per second). The value of P is calculated as follows:

  P = rpm/120 = (rev/s)/2 for four-stroke engines

  P = rpm/60 = rev/s for two-stroke engines

  Note that for V-engines with two exhaust manifolds, rpm or rev/s equals engine speed.
- A second, but less desirable, exhaust arrangement is a Y-connection with branches entering the single pipe at about a 60° angle; never use a T-connection, as the pulses of one branch will interfere with the pulses from the other.
- Use an engine-to-silencer pipe length that is 25% of the tail pipe length.
- Install a separate exhaust for each engine to reduce the possibility of condensation in the engine that is not running.
- Install individual silencers to reduce the condensation that results from an idle engine.
- Limit heat radiation from exhaust piping with a ventilated sleeve around the pipe or with high-temperature insulation.
- Use large enough fittings to minimize pressure drop.
- Allow for thermal expansion in exhaust piping, which is about 0.09 in. per foot of length.
- Specify muffler pressure drops to be within the back pressure limits of the engine.
- Do not connect the engine exhaust pipe to a chimney that serves natural-draft gas appliances.
- Slope the exhaust away from the engine to prevent condensate backflow. Drain plugs in silencers and drip legs in long, vertical exhaust runs may also be required. Raincaps may prevent the entrance of moisture but might add back pressure and prevent adequate upward ejection velocity.

Proper effluent discharge and weather protection can be maintained in continuously operated systems by maintaining sufficient discharge velocity (in excess of 2500 fpm) through a straight stack; in intermittently operated systems, protection can be maintained by installing drain-type stacks. Drain-type stacks effectively eliminate rainfall entry into a vertical stack terminal without destroying the upward ejection velocity, as a raincap does.

Drain-Type Stack. This design places a stack head, rather than a stack cap, over the discharge stack. The height of the upper section is important for adequate rain protection, just as the height of the stack is important for adequate dispersal of effluent. Stack height should be great enough to discharge above the building eddy zone. Bolts for inner stack fastening should be soldered, welded, or brazed, depending on the tack material.

Table 9 Exhaust Pipe Diameter[a]

Output Power, hp	Minimum Pipe Diameter, in. — Equivalent Length of Exhaust Pipe: 25 ft	50 ft	75 ft	100 ft
25	2	2	3	3
50	2.5	3	3	3.5
75	3	3.5	3.5	4
100	3.5	4	4	5
200	4	5	5	6
400	6	6	8	8
600	6	8	8	8
800	8	8	10	10

[a]Minimum exhaust pipe diameter to limit engine exhaust back pressure to 8 in. of water.

Powerhouse Stack. In this design, a fan discharge intersects the stack at a 45° angle. A drain lip and drain are added in the fume discharge version.

Offset Design. This design is recommended for round ductwork and can be used with sheet metal or glass fiber reinforced polyester ductwork.

The exhaust pipe may be routed between an interior engine installation and a roof-mounted muffler through (1) an existing unused flue or one serving power-vented gas appliances only (this should not be used if exhaust gases may be returned to the interior); (2) an exterior fireproof wall with provision for condensate drip to the vertical run; or (3) the roof, provided that a galvanized thimble with flanges having an annular clearance of 4 to 5 in. is used. Sufficient clearance is required between the flue terminal and the rain cap on the pipe to permit the venting of the flue. A clearance of 30 in. between the muffler and roof is common. Vent passages and chimneys should be checked for resonance.

When interior mufflers must be used, minimize the distance between the muffler and the engine, and insulate inside the muffler portion of the flue. Flue runs exceeding 25 ft may require power venting, but vertical flues help to overcome the pressure drop (natural draft).

The following design and installation features should be used for flexible connections:

- Material—Convoluted steel (Grade 321 stainless steel) is favored for interior installation.
- Location—Principal imposed motion (vibration) should be at right angles to the connector axis.
- Assembly—The connector (not an expansion joint) should not be stretched or compressed; it should be secured without bends, offsets, or twisting (the use of float flanges is recommended).
- Anchor—The exhaust pipe should be rigidly secured immediately downstream of the connector in line with the downstream pipe.
- Exhaust piping—Some alloys and standard steel alloy or steel pipe may be joined by fittings of malleable cast iron. Table 9 shows exhaust pipe sizes. The exhaust pipe should be at least as large as the engine exhaust connection. Stainless steel double-wall liners may be used.

Combustion Air Systems

The following factors apply to combustion air requirements:

- Supply 2 to 5 cfm per brake horsepower, depending on type, design, and size. Two-cycle units consume about 40% more air than four-cycle units.

- Avoid heated air because power output varies by $(T_r/T_a)^{0.5}$, where T_r is the temperature at which the engine is rated and T_a is the engine air intake temperature, both in Rankine.
- Locate the intake away from contaminated air sources.
- Install properly sized air cleaners that can be readily inspected and maintained (pressure drop indicators are available). Air cleaners minimize cylinder wear and piston ring fouling. About 90% of valve, piston ring, and cylinder wall wear is the result of dust. Both dry and wet cleaners are used. If wet cleaners are undersized, oil carryover may reduce filter life. Filters may also serve as flame arresters.
- Engine room air-handling systems may include supply and exhaust fans, louvers, shutters, bird screens, and air filters. The total static pressure opposing the fan should be 0.35 in. of water maximum.\

The following sections on Heating and Ventilating Systems, Off-Engine Instruments and Controls, Noise and Vibration Control, Maintenance, and Engine Applications for internal combustion engines also apply, with slight modifications, to combustion and steam turbines.

Heating and Ventilating Systems

Methods of dissipating heat from the jacket water system, exhaust system, lubrication and piston cooling oil, turbocharger, and air intercooler have been covered in previous sections. In addition, radiation and convection losses from the surfaces of the engine components and accessories and piping must be dissipated by ventilation. If the radiated heat is more than 8 to 10% of the fuel input, an air cooler may be required. In some cases, this rejected heat can be productively applied as tempered makeup air in an adjacent space, with consideration given to life/fire safety requirements, but in most instances it is simply vented.

This heat must be removed to maintain acceptable working conditions and to avoid overloading the electrical systems from high ambient conditions. Heat can be removed by outdoor air ventilation systems that include dampers and fans and thermostatic controls regulated to prevent overheating or excessively low temperatures in extreme weather. The manner and amount of heat rejection varies with the type, size, and make of engine and the extent of engine loading.

An **air-cooled engine** installation includes the following:

- An outside air entrance at least as large as the radiator face and 25 to 50% larger if protective louvers impede airflow.
- Auxiliary means (e.g., a hydraulic, pneumatic, or electric actuator) to open the louvers blocking the heated air exit, rather than a gravity-operated actuator.
- Control of jacket water temperature by radiator louvers in lieu of a bypass for freeze protection.
- Thermostatically controlled shutters that regulate airflow to maintain the desired temperature range. In cold climates, louvers should be closed when the engine is shut down to help maintain engine ambient temperature at a safe level. A crankcase heater can be installed on backup systems located in unheated spaces.
- Positioning of the engine so that the face of the radiator is in a direct line with an air exit leeward of the prevailing wind.
- An easily removable shroud so that exhaust air cannot reenter the radiator.
- Separated units in a multiple installation to avoid air short-circuiting among them.
- Low-temperature protection against snow and ice formation. Propeller fans cannot be attached to long ducts because they can only achieve low static pressure. Radiator cooling air directed over the engine promotes good circulation around the engine; thus, the engine runs cooler than for airflow in the opposite direction.
- Adequate sizing to dissipate the other areas of heat emissions to the engine room.

Table 10 Ventilation Air for Engine Equipment Rooms

Room Air Temperature Rise,[a] °F	Airflow, cfm/hp		
	Muffler and Exhaust Pipe[b]	Muffler and Exhaust Pipe[c]	Air- or Radiator-Cooled Engine[d]
10	140	280	550
20	70	140	280
30	50	90	180

[a]Exhaust minus inlet.
[b]Insulated or enclosed in ventilated duct.
[c]Not insulated.
[d]Heat discharged in engine room.

Sufficient ventilation must also be provided to protect against minor fuel supply leaks (not rupture of the supply line). Table 10 is sometimes used for minimum ventilation air requirements. Ventilation may be provided by a fan that induces the draft through a sleeve surrounding the exhaust pipe. A slight positive pressure should be maintained in the engine room.

Ventilation efficiency for operator comfort and equipment reliability is improved by (1) taking advantage of a full wiping effect across sensitive components (e.g., electrical controls and switchgear) with the coolest air; (2) taking cool air in as low as possible and forcing it to travel at occupancy level; (3) letting cooling air pass subsequently over the hottest components; (4) exhausting from the upper, hotter strata; (5) avoiding any short-circuiting of the cool air directly to the exhaust while bypassing equipment; and (6) arranging equipment locations, when possible, to permit such an airflow path.

Larger engines with off-engine filters should accomplish the following:

- Temper cold outside combustion air when its temperature is low enough to delay ignition timing and inhibit good combustion, which leads to a smoky exhaust.
- Permit the silencer and/or recovery device's hot surfaces to warm this cold air to an automatically controlled temperature. As this air enters the machine room it also provides some cooling to a hot machine room or heating to a cold room.
- Manipulate the dampers with a thermostat at the inlet to the machine room, which is reset by room temperature.
- Cool hot combustion air with a cooling device downstream of the air filter to increase engine performance. This practice is particularly helpful with large, slow-speed, naturally aspirated engines. The Diesel Engine Manufacturers Association (DEMA) recommended that engines rated at 90°F and 1500 ft above sea level be derated in accordance with the particular manufacturer's ratings. In a naturally aspirated engine, a rating of 100 hp at 1500 ft drops to 50 hp at 16,000 ft. Also, a rating can drop from 100 hp at 90°F to 88 hp at 138°F.

Off-Engine Instruments and Controls

Controls for cogeneration systems are required for (1) system output, (2) safety, (3) prime mover automation, and (4) waste heat recovery and disposal. Building requirements determine the level of automation. Every system must have controls to regulate the output energy and to protect the equipment. Independent power plants require constantly available control energy to actuate cranking motors, fuel valves, circuit breakers, alarms, and emergency lighting.

Smaller engines and combustion turbines generally use battery systems for these functions because the stored energy is available if the generating system malfunctions. Automatic equalizer battery chargers maintain the energy level with minimum battery maintenance.

Larger plants can use either batteries or an emergency generator technique as described in the section on Compressed Air Systems on page 7.6. Generally, a simple control system is adequate where labor is available to make minor adjustments and to oversee system operation. Fully automatic, completely unattended systems have the same advantages as automatic temperature control systems and are gaining acceptance.

Fully automated generator controls should be considered for office buildings, hotels, motels, apartments, large shopping centers, some schools, and manufacturing plants where close control of frequency and voltage is needed. The control system must be highly reliable, regardless of load change or malfunction, and it must protect the system equipment from electrical transients and malfunctions. To provide this reliability, each generator must be controlled.

Generators that operate in **parallel** require interconnecting controls. The complete system must be integrated to the building use, give properly sequenced operation, and provide overall protection. In addition to those controls required for a single prime mover installation, the following further controls are required for multiple-generator installations:

- Simultaneous regulation of fuel flow to each prime mover to maintain required shaft output
- Load division and frequency regulation of generators by a signal to the fuel controls
- System voltage regulation and reactive load division by maintaining the generator output at the required level
- Automatic starting and stopping of each unit for protection in the event of a malfunction
- Load demand and unit sequencing by determining when a unit should be added or taken off the bus as a function of total load
- Automatic paralleling of the oncoming unit to the bus after it has been started and reaches synchronous speed
- Safety protection for prime movers, generators, and waste heat recovery equipment in the event of overload or abnormal operation, including a means of load dumping (automatic removal of building electrical loads) in case the prime mover overloads or fails

When the system is operated in **isolation** of the utility grid, the engine speed control must maintain frequency within close tolerances, both at steady-state and transient conditions. Generators operating in parallel require a speed control to determine frequency and to balance real load between operating units. To obtain precise control, an electronic governing system is generally used to throttle the engine; the engine responds quickly and is capable of responding to control functions as follows:

- Sensing engine speed and controlling to operating frequency (60 Hz). See the section on Generators on page 7.27 for other generator control information.
- Synchronizing the oncoming generating unit to the bus so it can be paralleled.
- Sensing true real load, rather than current, to divide real load between parallel units. (This must be done through the speed control to the throttle.)
- Proper throttle operation during start-up to ensure engine starting and prevention of overspeed as the engine approaches rated speed.
- Time reference control is desirable to maintain clock accuracy to within 60 s per month.

When starting a unit, the following actions must be performed in sequence: (1) initiate engine crank; (2) open fuel valve and throttle; (3) when engine starts, terminate cranking; (4) when desired speed is reached, eliminate overshoot; (5) if the engine refuses to start in a given period, register a malfunction and start the next unit in sequence; and (6) reaching the desired speed, synchronize the started engine-alternator to the bus to be paralleled without causing severe mechanical and electrical transients. Some time should be allowed for synchronization; if it does not occur during the allotted time, a malfunction should register, and the next unit in sequence should start.

Once a unit is online, real and reactive load division should be effective immediately. The throttle of the unit coming online must be advanced so that it accepts its share of the load. The unit or units that were carrying all the load have their throttle(s) retarded so that they give up load to the oncoming unit until all units share equally or, if they are sized differently, in proportion to unit size. During the load-sharing process, control frequency must be maintained, and once paralleled, the load-sharing control must correct continuously to maintain the load balance. Reactive load sharing is similar but is done by the voltage regulator varying the excitation of the alternator exciter field.

Problems that increase installation cost can arise during installation. These can be minimized by carefully checking out the cogeneration system, particularly the engine-alternator units. Typical problems arising during installation include (1) engine actuators placed too near exhaust pipes, causing excessive wear of actuator bearings and loose parts in the actuator; (2) improper wiring; (3) sloppy linkage between engine actuator and carburetor; and (4) difficulties due to a failure to coordinate electrical design with the utility.

Sensors used to send a malfunction signal to the control system can also cause trouble when not properly installed. It is appropriate to use two sensors in sequence—one for alarm indication of an abnormal condition and the other for shutdown when a malfunction grows worse (e.g., lubricant reaches a shutdown temperature).

Noise and Vibration Control

Engine-driven machines installed indoors, even where the background noise level is high, usually require noise attenuation and isolation from adjoining areas. Air-cooled radiators, noise radiated from surroundings, and exhaust heat recovery boilers may also require silencing. Boilers that operate dry do not require separate silencers. Installations in more sensitive areas may be isolated, receive sound treatment, or both.

Basic attenuation includes (1) turning air intake and exhaust openings away (usually up) from the potential listener; (2) limiting blade-tip speed (if forced-draft air cooling is used) to 12,000 fpm for industrial applications, 10,000 fpm for commercial applications, and 8000 fpm for critical locations; (3) acoustically treating the fan shroud and plenum between blades and coils; (4) isolating (or covering) moving parts, including the unit, from their shelter (where used); (5) properly selecting the gas meter and regulator(s) to prevent singing; and (6) adding sound traps or silencers on ventilation air intake, exhaust, or both.

Further attenuation means include (1) lining the intake and exhaust manifolds with sound-absorbing materials; (2) mounting the unit, particularly a smaller engine, on vibration isolators, thereby reducing foundation vibration; (3) installing a barrier between the prime mover and the listener (often a concrete block enclosure); (4) enclosing the unit with a cover of absorbing material; and (5) locating the unit in a building constructed of massive materials, paying particular attention to the acoustics of the ventilating system and the doors.

Noise levels must meet legal requirements (see Chapter 47 of the *ASHRAE Handbook—Applications* for details).

Foundations. Multicylinder, medium-speed engines may not require massive concrete foundations, although concrete offers advantages in cost and in maintaining alignment for some driven equipment. Fabricated steel bases are satisfactory for direct-coupled, self-contained units, such as electric sets. Steel bases mounted on steel spring or equal-vibration isolators are adequate and need no special foundation other than a floor designed to accommodate the weight. Concrete bases are also satisfactory for such units, provided the bases are equally well isolated from the supporting floor or subfloor.

Glass fiber blocks are effective as isolation material for concrete bases, which should be thick enough to prevent deflection. Excessively thick bases only increase subfloor or soil loading, and they still should be supported by a concrete subfloor. In addition, some acceptable isolation material should be placed between the base and the floor. To avoid the transmission of vibration, an engine base or foundation should never rest directly on natural rock formations. Under some conditions, such as shifting soil on which an outlying cogeneration plant might be built, a single, very thick concrete pad for all equipment and auxiliaries may be required to avoid a catastrophic shift between one device and another.

Alignment and Couplings. Proper alignment of the prime mover to the driven device is necessary to prevent undue stresses to the shaft, coupling, and seals of the assembly. Installation instructions usually suggest that the alignment of the assembly be performed and measured at maximum load condition and maximum heat input to the turbine or engine.

Torsional vibrations can be a major problem when matching these components, particularly when matching a reciprocating engine to any higher speed centrifugal device. The stiffness of a coupling and its dynamic response to small vibrations from an angular misalignment at critical speed(s) affect the natural frequencies that exist in the assembly. Encotech (1992) describes how changing the mass, stiffness, or damping of the coupling can alter the natural frequency during start-up, synchronization, or load change or when a natural frequency exists within the assembly's operating envelope. Proper grouting is needed to preserve the alignment.

Flexible Connections. Greater care is required in the design of piping connections to turbines and engines than for other HVAC equipment because the larger temperature spread causes greater expansion. (See Chapter 41 for further information on piping.)

Maintenance

Preventive Maintenance. One of the most important provisions for healthy and continuous plant operation is implementation of a comprehensive preventive maintenance program. This should include written schedules of daily wipedown and observation of equipment, weekly and periodic inspection for replacement of degradable components, engine oil analysis, and maintenance of proper water treatment. Immediate access to repair services may be furnished by subcontract or by in-house plant personnel. An inventory of critical parts should be maintained on site. (See Chapters 35 through 42 of the *ASHRAE Handbook—Applications* for further information on building operation and maintenance.)

Predictive Maintenance. Given the tremendous advancement and availability of both fixed and portable instrumentation for monitoring sound, vibration, temperatures, pressures, flow, and other online characteristics, many key aspects of equipment and system performance can be logged manually or by computer to observe trends. Such factors as fuel rate, heat exchanger approach, and cylinder operating condition can be compared against new and/or optimized baseline conditions to indicate when maintenance may be required. This monitoring permits periods between procedures to be longer, catches incipient problems before they create outages or major repairs, and avoids unnecessary maintenance.

Equipment Rotation. When more than one engine, pump, or other component is serving a given distribution system, it is undesirable to operate the units with the equal life approach mode, which puts each unit in the same state of wear and component deterioration. If only one standby unit (or none) is available to a given battery of equipment, and one unit suffers a major failure or shutdown, all units now needed to carry the full load would be prone to additional failure while the first failed unit is undergoing repair.

The preferred operating procedure is to keep one unit in continuous reserve, with the shortest possible running hours between overhauls or major repair, and to schedule operation of all others for unequal running hours. Thus any two units would have a minimum statistical chance of a simultaneous failure. All units, however, should be exercised for several hours in any week.

Engine Applications

Engine Sizing. In sizing an engine, proper evaluation of the various electrical loads, building heat gains and losses, and their time of occurrence avoids overestimating the actual simultaneous load. For existing buildings, an energy audit should be performed to access correct load.

Consumption rates must be known to determine operating costs. Specific data can be obtained from the manufacturer; however, a range of consumption rates is given in Table 11. Atmospheric corrections included in the following equation may be used to match prime movers to loads at various ambient conditions.

$$P_u = P_{max} \times (\text{Rating or derating factor}) \tag{9}$$

where

P_u = usable power, kW
P_{max} = power rating under ideal conditions, kW
P_{max} is based on manufacturer's performance data (usually a dynamometer test) and corrected in accordance with the manufacturer's specifications. Examples of specifications are

60°F and 29.92 in. Hg
80°F and 1000 ft elevation (NEMA)
85°F and 500 ft elevation (SAE)
90°F and 1500 ft elevation (DEMA)

$$\text{Rating factor} = \frac{100 - C_a - C_t - C_{hv} - C_r}{100} \tag{10}$$

where

C_a = percent altitude correction
= 3% per 1000 ft above a specific level for naturally aspirated engines
= 2% per 1000 ft for turbocharged engines
C_t = percent temperature correction
= 1% per 10°F rise above a specified base
C_{hv} = percent fuel heating value correction to account for the air intake temperature (see the section on Fuel Heating Value on page 7.4)
C_r = percent reserve, which is an allowance (safety factor) to permit design output under unforeseen operating conditions that would reduce output, such as a dusty environment, poor maintenance, higher ambient temperature, or lowered cooling efficiency. Recommended values are in Table 12.

Table 11 Fuel Consumption Rates

Fuel	Heating Value, Btu/gal	Range of Consumption, Btu/hp·h
Fuel oil	137,000 to 156,000	7,000 to 9,000
Gasoline	130,000	10,000 to 14,000
Type of Gas Engine	**Compression Ratio**	**Typical Gas Consumption, Btu/hp·h**
Turbocharged	10.5:1	8,100
Naturally aspirated	10.5:1	9,200
Naturally aspirated	7.5:1	10,250

Table 12 Percent Minimum Engine Reserves for Air Conditioning and Refrigeration

Altitude, ft	Naturally Aspirated		Turbocharged Aftercooled	
	Air Cond.	Refrigeration	Air Cond.	Refrigeration
Sea level	15	20	20	30
1000	12	17	18	28
2000	10	14	16	26
3000	10	11	14	24
4000	10	10	12	22
5000	10	10	10	20
10,000	10	10	10	10

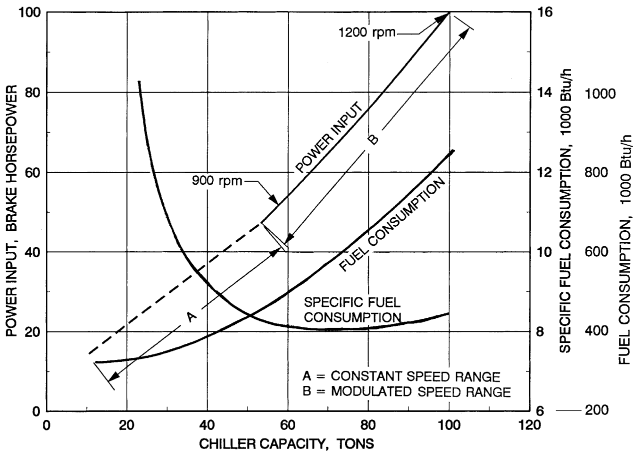

Fig. 48 Performance Curve for Typical 100 Ton, Gas Engine-Driven, Reciprocating Chiller

Table 13 Coefficient of Performance for Engine-Driven Heat Pump

Item	Heat Source: Refrigerant Condenser Only, Btu/ton·h	Refrigerant Condenser and Jacket Water, Btu/ton·h	Refrigerant Condenser, Jacket Water, Exhaust Gas, Btu/ton·h
Total heat input to engine	10,000	10,000	10,000
Cooler heat rejection to condenser (from building load)	12,000	12,000	12,000
Heat of compression	2,545	2,545	2,545
Heat from engine jacket water heater	—	2,500	2,500
Heat from exhaust gas heater	—	—	3,000
Total heat to heating circuit	14,545	17,045	20,045
Economic coefficient of performance	1.45	1.70	2.00

Engine-Driven Reciprocating Compressors. Engine-driven reciprocating compressor water chiller units may be packaged or field assembled from commercially available equipment for comfort service, low-temperature refrigeration, and heat pump applications. Both direct-expansion and flooded chillers are used. Some models achieve a low operating cost and a high degree of flexibility by combining speed variation with cylinder unloading. These units achieve capacity control by reducing engine speed to about 30 to 50% of rated speed; further capacity modulation may be achieved by unloading the compressor in increments. Engine speed should not be reduced below the minimum specified by the manufacturer for adequate lubrication or good fuel economy.

Most engine-driven reciprocating compressors are equipped with a cylinder loading mechanism for idle (unloaded) starting. This arrangement may be required because the starter may not have sufficient torque to crank both the engine and the loaded compressor. With some compressors, not all the cylinders (e.g., four out of 12) unload; in this case, a bypass valve must be installed for a fully unloaded start. The engine first speeds to one-half or two-thirds of full speed. Then, a gradual cylinder load is added, and the engine speed increases over a period of 2 to 3 min. In some applications, such as an engine-driven heat pump, low-speed starting may cause oil accumulation and sludge. As a result, a high-speed start is required.

These systems operate at specific fuel consumptions (SFCs) of approximately 8 to 13 ft^3/h of pipeline quality natural gas (HHV = 1000 Btu/ft^3) per horsepower in sizes down to 25 tons. Comparable heat rates for diesel engines run from 7000 to 9000 Btu/hp·h. Smaller units are also available. Coolant pumps can also be driven by the engine. These direct-connected pumps never circulate tower water through the engine jacket. Figure 48 illustrates the fuel economy effected by varying prime mover speed with reciprocating compressor load until the machine is operating at about half its capacity. Below this level, the load is reduced at essentially constant engine speed by unloading the compressor cylinders.

Frequent operation at low engine idling speed may require an auxiliary oil pump for the compressor. To reduce wear and assist in starts, a tank-type lubricant heater or a crankcase heater and a motor-driven auxiliary oil pump should be installed to lubricate the engine with warm oil when it is not running. Refrigerant piping practices for engine-driven units are the same as for motor-driven units.

Engine-Driven Centrifugal Compressors. Packaged, engine-driven centrifugal chillers that do not require field assembly are available in capacities up to 2100 tons. Automotive derivative engines modified for use on natural gas are typical of these smaller packages because of their compact size and mass. These units may be equipped with either manual or automatic start-stop systems and engine speed controls.

Larger open-drive centrifugal chillers are usually field assembled and normally include a compressor mounted on an individual base and coupled by means of flanged pipes to an evaporator and a condenser. The centrifugal compressor is driven through a speed increaser. Many of these compressors operate at about six times the speed of the engines; compressor speeds of up to 14,000 rpm have been used.

To effect the best compromise between the initial cost of the equipment (engine, couplings, and transmission) and the maintenance cost, engine speeds between 900 and 1200 rpm are generally used. Engine output can be modulated by reducing engine speed. If the operation at 100% of rated speed produces 100% of rated output, approximately 60% of rated output is available at 75% of rated speed. Capacity control of the centrifugal compressor can be achieved by either variable inlet guide vane control with constant compressor speed or a combination of variable-speed control and inlet guide vane control, the latter providing the greatest operating economy.

Engine-Driven Heat Pumps. An additional economic gain can result from operating an engine-driven refrigeration cycle as a heat pump, provided that the facility has a thermal load profile that can adequately absorb its 100 to 120°F low-quality heat. Using the same equipment for both heating and cooling reduces capital investment. A gas engine drive for heat pump operation also makes it possible to operate in a cogeneration mode, which requires a somewhat larger thermal load. Unless a major portion of this larger thermal recovery can be absorbed, the cycle may not be economical.

For larger projects, the economics of engine or turbine generators combined with motor-driven chillers should be compared with the economics of generators plus combustion engine or turbine-driven chillers. Figure 49 shows the total energy available from a typical engine-driven heat pump. Table 13 lists the economic coefficients of performance (ECOPs) for various configurations. The values in the table can be used to illustrate the difference between the COP, the ECOP, and the thermal efficiency of this cycle. Here, the ECOP is based upon the engine fuel input of 10,000 Btu/ton·h, yielding an ECOP of 14,545/10,000 = 1.45; while the COP, based upon the work input (not the fuel input), is 14,545/2,545 = 5.72.

The classic definition of reverse-cycle performance is (heat out)/(work in). The definition does not recognize the fuel input to the engine, just as it ignores the fuel input to generate the electricity for the motor of a motor-driven compressor. No COP is really defined for the cycle that captures the jacket and exhaust heat, but

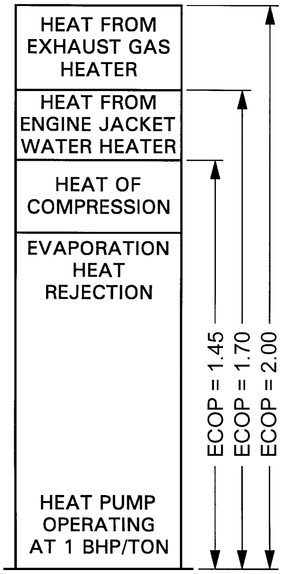

Fig. 49 Heat Balance for Engine-Driven Heat Pump

input) is indeed 17,045/10,000 = 1.70 for jacket recovery and 20,045/10,000 = 2.0 for the jacket and exhaust cycles, as shown in Figure 49.

Engine-Driven Screw Compressors. Chiller packages with these compressors are available for refrigeration applications. Manufacturers offer water chillers that use screw compressors driven directly by natural gas engines. Capacity control is achieved by varying the engine speed and adjusting the slide valve on the compressor. Units have a cooling COP near 5.72 without heat recovery and an ECOP near 1.45 at rated cooling load.

COMBUSTION TURBINE APPLICATIONS

The gas turbine has achieved an increasingly important position as a prime mover for electric power generation up to more than 240 MW and for shaft power drives up to more than 108,000 hp. Figure 50 shows a typical gas turbine refrigeration cycle, with optional combustion air precooling. A gas turbine must be brought up to speed by an auxiliary starter. With a single-shaft turbine, the air compressor, turbine, speed reducer gear, and refrigeration compressor must all be started and accelerated by this starter. The refrigeration compressor must also be unloaded to ease the starting requirement. Sometimes, this may be done by making sure the capacity control vanes close tightly. At other times, it may be necessary to depressurize the refrigeration system to get started.

With a split-shaft design, only the air compressor and the gas producer turbine must be started and accelerated. The rest of the unit starts rotating when enough energy has been supplied to the blades of the power turbine. At this time, the gas producer turbine is up to speed, and the fuel supply is ignited. Electric starters are usually available as standard equipment. Reciprocating engines, steam turbines, and hydraulic or pneumatic motors may also be used. The output shaft of the gas turbine must rotate in the direction required by the refrigeration compressor; in many cases, the manufacturers of split-shaft engines can provide the power turbine with either direction of rotation.

At low loads, both the gas turbine unit and the centrifugal refrigeration machine are affected by surge, a characteristic of all centrifugal and axial flow compressors. At a certain pressure ratio, a minimum flow through the compressor is necessary to maintain stable operation. In the unstable area, a momentary backward flow of gas occurs through the compressor. Stable operation can be maintained, however, by the use of a hot gas bypass valve.

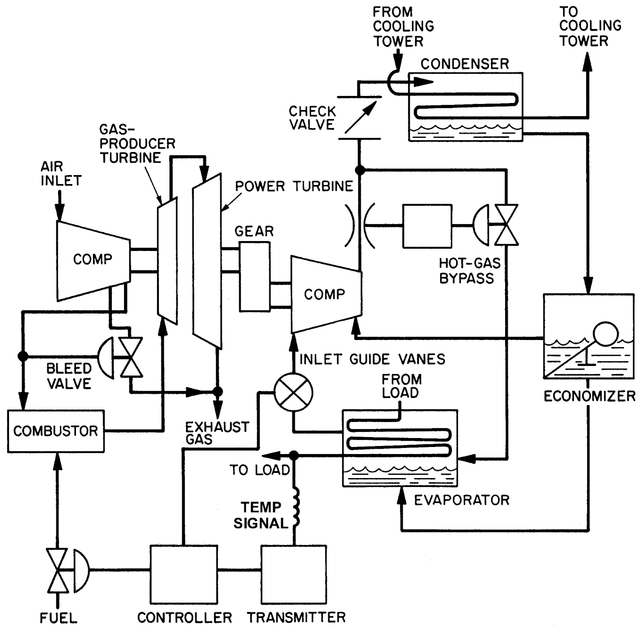

Fig. 50 Typical Gas Turbine Refrigeration Cycle

The turbine manufacturer normally includes automatic surge protection, either as a bleed valve that bypasses a portion of the air directly from the axial compressor into the exhaust duct or by providing for a change in the position of the axial compressor stator vanes. Both methods are used in some cases.

The assembly should be prevented from rotating backward, which may occur if the unit is suddenly stopped by one of the safety controls. The difference in pressure between the refrigeration condenser and cooler can make the compressor suddenly become a turbine and cause it to rotate in the opposite direction. This rotation can force hot turbine gases back through the air compressor, causing considerable damage. Reverse flow through the refrigeration compressor may be prevented in a variety of ways, depending on the system's components.

When there is no refrigerant receiver, quick-closing inlet guide vanes are usually satisfactory because there is very little high-pressure refrigerant to cause reverse rotation. However, when there is a receiver, a substantial amount of energy is available to cause reverse rotation. This can be reduced by opening the hot gas bypass valve on shutdown and installing a discharge check valve on the compressor.

The following safety controls are usually supplied with a gas turbine:

- Overspeed
- Compressor surge
- Overtemperature during operation under load
- Low oil pressure
- Failure to light off during start cycle
- Underspeed during operation under load

A fuel supply regulator can maintain a single-shaft gas turbine at a constant speed. With the split-shaft design, the output shaft of the turbine unit runs at the speed required by the refrigeration compressor. The temperature of the chilled water or brine leaving the cooler of the refrigeration machine controls the fuel. See also the section on Fuels and Fuel Systems on page 7.9.

STEAM TURBINE APPLICATIONS

Steam turbines in the air-conditioning and refrigeration field are principally used to drive centrifugal compressors. Such compressors

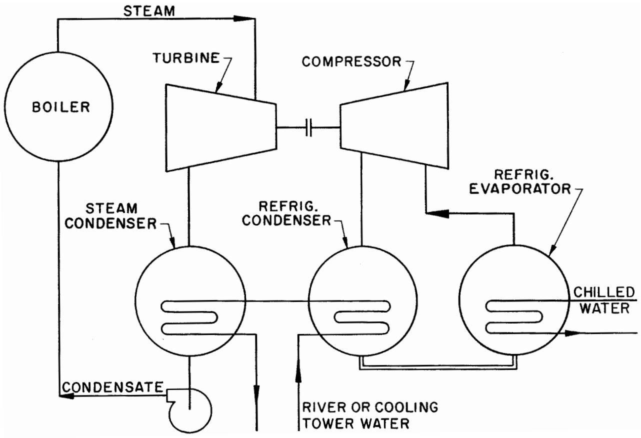

Fig. 51 Condensing Turbine-Driven Centrifugal Compressor

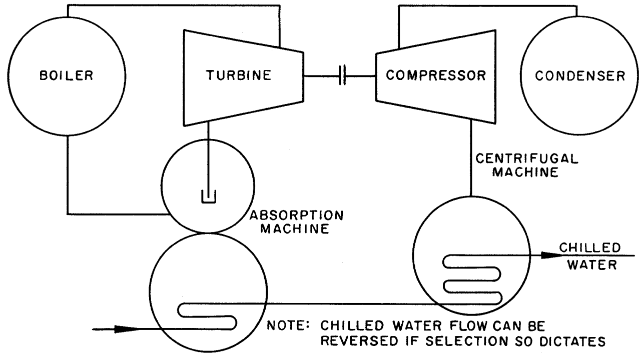

Fig. 52 Combination Centrifugal-Absorption System

are usually part of a water or secondary coolant chilling system using one of the newer or the halogenated hydrocarbon refrigerants. In addition, many industrial processes employ turbine-driven centrifugal compressors with a variety of other refrigerants such as ammonia, propane, and butane, or other process gases.

Related applications of steam turbines include driving chilled water and condenser water circulating pumps and serving as prime movers for electrical generators in cogeneration systems. In industrial applications, the steam turbine may be advantageous, serving either as a work-producing steam pressure reducer or as a scavenger using otherwise wasted low-pressure steam.

Many steam turbines are used in urban areas where commercial buildings are served with steam from a central public utility or municipal source. Institutions where large central plants serve a multitude of buildings with heating and cooling also use steam turbine-driven equipment.

Most steam turbines driving centrifugal compressors for air conditioning are of the multistage condensing type (Figure 51). Such a turbine provides good steam economy at reasonable initial cost. Usually, steam is available at 50 psig or higher, and there is no demand for exhaust steam. However, turbines may work equally well where an abundance of low-pressure steam is available. The wide range of application of this turbine is shown by at least one industrial firm that drives a sizable capacity of water chilling centrifugal compressors with an initial steam pressure of less than 4 psig, thus balancing summer cooling against winter heating with steam from generator-turbine exhausts.

Aside from wide industrial use, the noncondensing (back pressure) turbine is most often used in water chilling plants to drive a centrifugal compressor that shares the cooling load with one or more absorption units (Figure 52). The exhaust steam from the turbine, commonly at about 15 psig, serves as the heat source for the absorption unit's generator (concentrator). This dual use of the heat energy in the steam generally results in a lower energy input per unit of refrigeration output than is attained by either machine operating alone. An important aspect in the design of such combined systems is the need to balance the turbine exhaust steam flow with the absorption input steam requirements over the full range of load.

Extraction and mixed pressure turbines are used mainly in industry or in large central plants. Extracted steam is often used for boiler feedwater heating or other processes where steam with lower heat content is needed. Most motor-driven centrifugal refrigeration compressors are driven at constant speed (some with variable-frequency drives). However, governors on steam turbines can maintain a constant or variable speed without the need for expensive variable-frequency drives.

ECONOMIC FEASIBILITY

TERMINOLOGY

Avoided cost. Incremental cost for the electric utility to generate or purchase electricity that is avoided through the purchase of power from a cogeneration facility.

Backup power. Electric energy available from or to an electric utility during an unscheduled outage to replace energy ordinarily generated by the facility or the utility. Frequently referred to as standby power.

Baseload. Minimum electric or thermal load generated or supplied over one or more periods.

Bottoming cycle. Cogeneration facility in which the energy input to the system is first applied to another thermal energy process; the rejected heat that emerges from the process is then used for power production.

Capability. Maximum load that a generating unit, generating station, or other electrical apparatus can carry under specified conditions for a given period of time without exceeding approved limits of temperature and stress.

Capacity. Load for which a generating unit, generating station, or other electrical apparatus is rated.

Capacity credits. Value included in the utility's rate for purchasing energy, based on the savings accrued through the reduction or postponement of new generation capacity that results from purchasing power from cogenerators.

Capacity factor. Ratio of the actual annual plant electricity output to the rated output.

Central cooling. Same as central heating except that cooling (heat removal) is supplied instead of heating; usually a chilled water distribution system supply and return for air conditioning.

Central heating. Supply of thermal energy from a central plant to multiple points of end use, usually by steam or hot water, for space and/or service water heating; central heating includes large-scale plants serving university campuses, medical centers, or military installations and central building systems serving multiple zones; also district heating plants.

Coefficient of performance (COP). Refrigeration or refrigeration plus thermal output energy divided by the energy input to the refrigeration compressor or absorption device.

Cogeneration. Sequential production of electrical or mechanical energy and useful thermal energy from a single energy stream. To qualify as a cogeneration facility in the United States, a plant must meet certain energy efficiency standards.

Coproduction. Conversion of energy from a fuel (possibly including solid or other wastes) into shaft power (which may be used to generate electricity) and a second or additional useful form of energy. The process generally entails a series topping and bottoming arrangement of conversion to shaft power and either process or space heating. Cogeneration is a form of coproduction.

Demand. Rate at which electric energy is delivered at a given instant or averaged over any designated time, generally over a period of less than 1 h.

Annual demand. Greatest of all demands that occur during a prescribed demand interval billing cycle in a calendar year.

Billing demand. Demand on which customer billing is based, as specified in a rate schedule or contract. It can be based on the contract year, a contract minimum, or a previous maximum and is not necessarily based upon the actual measured demand of the billing period.

Coincident demand. Sum of two or more demands occurring in the same demand interval.

Instantaneous peak demand. Maximum demand at the instant of greatest load.

Demand charge. Specified charge for electrical capacity on the basis of the billing demand.

Demand factor. Average demand over a specific period divided by the maximum demand over the same period (e.g., monthly demand factor, annual demand factor).

Economic coefficient of performance (ECOP). Energy in desired output units converted in terms of economic costs divided by fuel input in units of energy purchased or produced, where energy is measured in equivalent, consistent units again converted in terms of economic costs of each energy stream (e.g., a ton of cooling output is 12,000 Btu/h, while shaft horsepower output is 2545 Btu/h and electrical kilowatt output is 3412 Btu/h.)

Energy charge. That portion of the billed charge for electric service based on the electric energy (kilowatt-hours) supplied, as contrasted with the demand charge.

FERC efficiency. Electrical output plus one-half the thermal heat utilized divided by the energy input; based on the low heating value of the fuel (defined by the Federal Energy Regulatory Commission).

Generating efficiency. Electrical energy output from the engine or turbine generator divided by the energy input to the prime mover, in consistent units.

Grid. System of interconnected transmission lines, substations, and generating plants of one or more utilities.

Grid interconnection. Intertie of a cogeneration plant to an electric utility's distribution network to allow electricity flow in either direction.

Harmonics. Wave forms whose frequencies are multiples of the fundamental (60 Hz or 50 Hz) wave. The combination of harmonics and the fundamental wave causes a nonsinusoidal, periodic wave. Harmonics in power systems are the result of nonlinear effects. Typically, harmonics are associated with rectifiers and inverters, arc furnaces, arc welders, and transformer magnetizing current. Both voltage and current harmonics occur.

Heat rate. Measure of generating station thermal efficiency, generally expressed in Btu per net kilowatt-hour, or lb steam/kWh.

Heating value. Energy content in a fuel that is available as useful heat. The **high heating value** (HHV) includes the energy transmitted to vapor formed during combustion, whereas the **low heating value** (LHV) deducts this energy because it does no work on the piston.

Interruptible power. Electric energy supplied by an electric utility subject to interruption by the electric utility under specified conditions.

Load factor. Ratio of the average load supplied or required during a designated period to the peak or maximum load occurring in that period.

Maintenance power. Electric energy supplied by an electric utility during scheduled outages of the cogenerator.

Off-peak. Time periods when power demands are below average; for electric utilities, generally nights and weekends; for gas utilities, summer months.

Plant efficiency. Net electrical energy output (not including generating plant auxiliaries) plus thermal energy used divided by fuel input to the plant.

Power factor. Ratio of real power (kW) to apparent power (kVA) for any load and time; generally expressed as a decimal.

Selective energy systems. Form of cogeneration in which part, but not all, of the site's electrical needs are met solely with on-site generation, with additional electricity purchased from a utility as needed.

Shaft efficiency. Prime mover's shaft energy output divided by its energy input in equivalent, consistent units. For a steam turbine, input can be the thermal value of the steam or the fuel value to produce the steam. For a fuel-fired prime mover, it is the fuel input.

Standby power. Electric energy supplied by a utility to a cogenerator or vice-versa during either one's unscheduled outage.

Supplemental thermal. Heat required when recovered engine heat is insufficient to meet thermal demands.

Supplementary firing. Injection of fuel into an exhaust gas stream to raise its energy content (heat).

Supplementary power. Electric energy supplied by an electric utility in addition to the energy the facility generates.

Thermal capacity. Maximum amount of instantaneous heat that a system can produce.

Thermal efficiency (First Law efficiency). Electrical or shaft plus thermal output divided by the energy input in consistent units.

Topping cycle. Cogeneration facility in which the energy input to the facility is first used to produce useful power, and the rejected heat from production is used for other purposes.

Total energy system. Form of cogeneration in which all electrical and thermal energy needs are met by on-site systems. A total energy system can be completely isolated or switched over to a normally disconnected electrical utility system for backup.

Voltage flicker. Significant fluctuation of voltage.

Wheeling. Use of the transmission facilities of one system to transmit gas or power for another system.

SIZING AND OPERATING OPTIONS

Selecting a cogeneration system requires an evaluation of a large number of factors. Aside from the selection of the type of prime mover, the most important step in evaluating or planning a cogeneration system involves determining the cogeneration plant capacity and operating options. The following are possible sizing criteria:

- Peak electrical demand
- Minimum (or base-load) electrical demand
- Peak thermal demand
- Minimum (or base-load) thermal demand
- Maximum economic return with available funds
- Marginal cost/benefit ratio

The following are plant operating options:

- Rated output
- Thermally dispatched (tracks facility thermal load)
- Electrically dispatched (tracks facility electrical load)
- Maximum economic return
- Utility dispatched
- Peak shaving

The significance of each of these options is explained in the following discussion.

PRELIMINARY FEASIBILITY

Planning a cogeneration system is considerably more involved than planning an HVAC system. HVAC systems must be sized to meet peak loads, while cogeneration systems need not. Also, HVAC systems do not have to be coordinated and integrated with other energy systems as extensively as do cogeneration systems.

First Estimates

Becker (1988) suggested a quick way to determine whether a study should be undertaken: if the cost of electricity expressed in \$/kWh is more than 0.013 times the cost of fuel expressed in \$/10^6 Btu, a study should be considered. If it is 0.026 times or more, the chances are excellent for a 3 year or less simple payback.

Load Duration Curve Analysis

A much more comprehensive energy analysis, combined with an economic analysis, must be used to select a cogeneration system that maximizes the efficiency and economic return on investment. For better identification and screening of potential candidates, a simplified but accurate performance analysis must be conducted that considers the dynamics of the electrical and thermal loads of the facility, as well as the size and fuel consumption of the prime mover.

The need for an accurate analysis is especially important for commercial and institutional cogeneration applications because of the large time-dependent changes in magnitude of load and the non-coincident nature of the power and thermal loads. A facility containing a generator sized and operated to meet the thermal demand may occasionally have to purchase supplemental power and sometimes may produce power in excess of facility demand.

Even in the early planning stages, a reasonably accurate estimate of the following must be determined:

- Fuel consumed (if it is a topping cycle)
- Amount of supplemental electricity that must be purchased
- Amount of supplemental boiler fuel (if any) that must be purchased
- Amount of excess power available for sale
- Electrical capacity required from the utility for supplemental and standby power.
- Electrical capacity represented by any excess power if the utility offers capacity credits

Obtaining estimates of the these performance values for multiple time-varying loads is difficult and is further complicated by utility rate structures that may be based on time-of-day or time-of-year purchase and sale of power. Data must be collected at intervals short enough to give the desired levels of accuracy, yet taken over a long period and/or a well-selected group of sampling periods.

A basic method for analyzing the performance of HVAC systems is the bin method. The basic tool for sizing and evaluating performance of power systems is the load duration curve. This curve contains the same information as bins, but the load data are arranged in a slightly different manner. The load duration curve is a plot of hourly averaged instantaneous load data over a period; the plot is rearranged to indicate the frequency, or hours per period, that the load is at or below the stated value. The load duration curve is constructed by sorting the hourly averaged load values of the facility into descending order. Large volumes of load data can be easily sorted with desktop computers and electronic spreadsheets or databases. The load duration curve produces a visually intuitive tool for sizing cogeneration systems and for accurately estimating system performance.

Figure 53 shows a hypothetical steam load profile for a plant operating with two shifts each weekday and one shift each weekend day; no steam is consumed during nonworking hours for this example. The data provide little information for thermally sizing a generator, except to indicate that the peak demand for steam is about 46,000 lb/h and the minimum demand is about 13,000 lb/h

Figure 54 is a load duration curve (a descending order sort) of the steam load data in Figure 53. Mathematically, the load duration curve shows the frequency with which load equals or exceeds a given value; the curve is one minus the integral of the frequency distribution function for a random variable. Because the frequency distribution is a continuous representation of a histogram, the load duration curve is simply another arrangement of bin data.

In the frequency domain, or load duration curve form, the baseload and peak load can be readily identified. Note that the practical baseload at the "knee" of the curve is about 21,000 lb/h rather than the 13,000 lb/h absolute minimum identified on the load profile curve.

The cogenerator sized at the baseload achieves the greatest efficiency and best use of capital. However, it may not offer the shortest payback because of the high value of electrical power. An appropriately sized cogeneration plant might be sized somewhat larger than baseload to achieve minimum payback through increased electrical savings. A combination of load analysis and economic analysis must be performed to determine the most economical plant. For this example, the maximum economical plant size is arbitrarily assumed to be 28,000 lb/h. Cogeneration systems sized this way produce high equipment utilization and depend on the utility to serve the peak loads in excess of the plant's 28,000 lb/h capacity.

The load duration curve allows the designer to estimate the total amount of steam generated within the interval. Because the total amount of steam is the area under the curve, the calculation may be performed by using either formulae for rectangles and triangles or an appropriate curve analysis program. For a thermally tracked

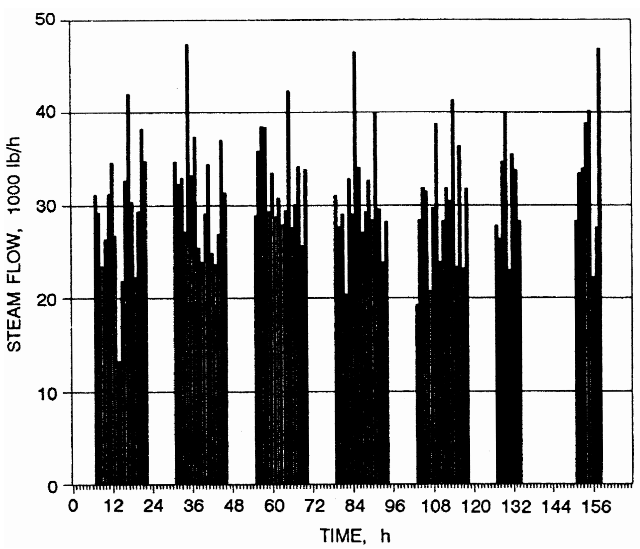

Fig. 53 Hypothetical Steam Load Profile

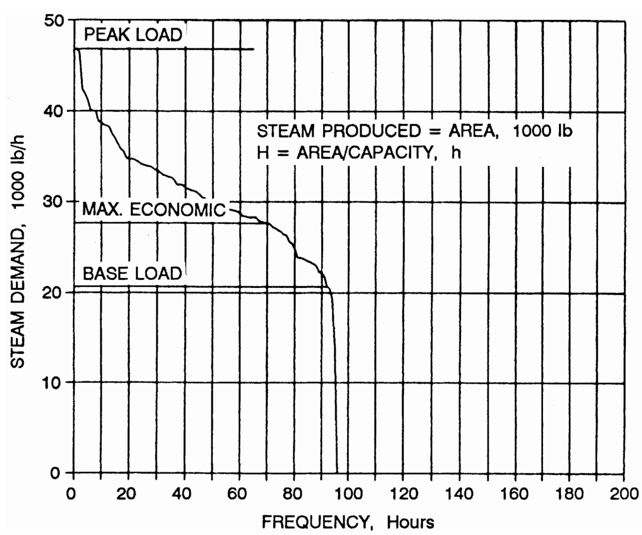

Fig. 54 Load Duration Curve

plant sized at the 21,000 lb/h baseload, the hours of operation are about 93 h/wk. Therefore, based on the area under the rectangle, the total steam produced by the cogeneration plant is

$$\text{Cogenerated steam} = (93 \text{ h/wk})(21 \times 10^3 \text{ lb/h}) = 1.953 \times 10^6 \text{ lb/wk}$$

If the plant is shut down during nonworking hours, no steam is wasted. In addition, if the electrical load profile is always above that needed to produce the 21,000 lb/h steam, the plant can run at full capacity for power and steam, while an electric utility provides peak power and a supplementary boiler provides peak steam. However, if the facility's steam load profile is unable to absorb the steam produced at continuous full power, the electrical output can be reduced accordingly to avoid steam waste.

In cases where it is more cost-effective to generate excess power than to suffer the parasitic cost of condensing the steam, the plant can still be operated at full power. Therefore, it is important to examine whether the electrical load profile matches and/or exceeds the electrical output. If it does, then the load duration curve reveals the quality of boiler-generated steam required. This value, which can be estimated by calculating the size of the triangular area above the baseload, is

$$\text{Boiler steam required} = 93(47{,}000 - 21{,}000)/2 = 1.21 \times 10^6 \text{ lb/wk}$$

$$\text{Cogenerated kW} = (21 \times 10^3 \text{ lb/h}) / (\text{Steam-to-Electric Ratio})$$

where the ratio is that of the prime mover, in lb of steam/kWh. The total weekly electrical production is

$$\text{Cogenerated kWh} = 1.953 \times 10^6/\text{Ratio}$$

Fuel consumption equals cogenerated kilowatt-hours times the full-load heat in Btu, to which is added the boiler fuel consumption.

The **equivalent full-load hours** (EFLH) of both steam and electric cogeneration production in this base-load sizing and operating mode is 93 h/wk, so that

$$\text{Electric production} = \text{EFLH} \times \text{Rated capacity (kW) at full load}$$

If the plant had been sized at the maximum economic return, the cogenerated steam would be the area under the 28×10^3 lb/h level, or

$$\begin{aligned}\text{Cogenerated steam} &= 1.953 \times 10^6 + 70(28{,}000 - 21{,}000) \\ &\quad + (93 - 70)(28{,}000 - 21{,}000)/2 \\ &= 2.52 \times 10^6 \text{ lb/wk}\end{aligned}$$

For this larger size, there is a higher electrical and lower boiler production, but a portion of the steam is condensed without any productive use.

Estimating the fuel consumption and electrical energy output of a system sized for the peak load or sized above the baseload is not as simple as for the base-load design because changes in the performance of the prime mover at part load must be considered. In this case, average value estimates of the performance at part load must be used for preliminary studies.

In some cases, an installation has only one prime mover; however, several smaller units operating in parallel provide increased reliability and performance during part-load operation. Figure 55 illustrates the use of three prime movers rated at 10,000 lb/h each. For this example, generators #1 and #2 operate fully loaded, and #3 operates between full-load capacity and 50% capacity while tracking the facility thermal demand. Because operation at less than 50% load is inefficient, further reduction in total output must be achieved by part-load operation of generators #1 and #2.

Offsetting the advantages of multiple units are their higher specific investment and maintenance costs, the control complexity, and the usually lower efficiency of smaller units.

Facilities such as hospitals often seek to reduce their utility costs by using existing standby generators to share the electrical peak (peak shaving). Such an operation is not strictly cogeneration because heat recovery is rarely justified. Figure 56 illustrates a hypothetical electrical load duration curve with frequency as a percent of total hours in the year (8760). A generator rated at 1000 kW, for example, reduces the peak demand by 500 kW if it is operated between 500 and 1000 kW to avoid extended hours of low-efficiency operation.

Some electrical energy saving as well as a reduction in demand are obtained by operating the generator. However, this saving is idealistic because it can only be obtained if the operators or the control system can anticipate in advance when facility demand will exceed 1400 kW in sufficient time to bring the generator up to operating condition. Nor should the peak shaver be started too early because it would waste fuel.

Also, many utilities include ratchet clauses in their rate schedules. As a result, if the peak shaving generator is inoperative for any reason when the facility monthly peak occurs, the ratchet is set for a year hence, and the demand savings potential of the peaking generator

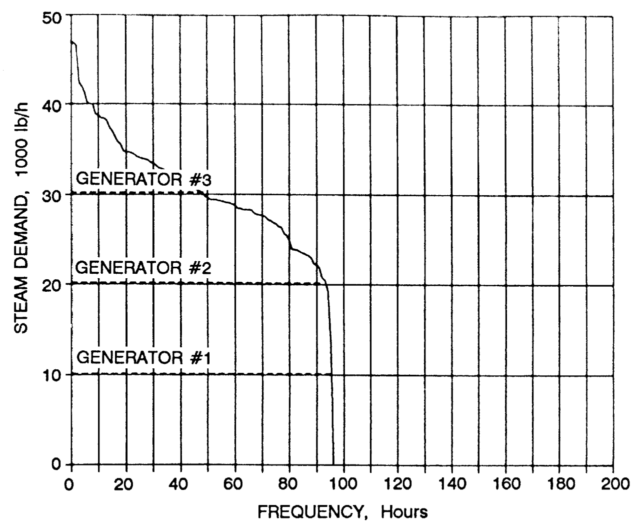

Fig. 55 Load Duration Curve with Multiple Generators

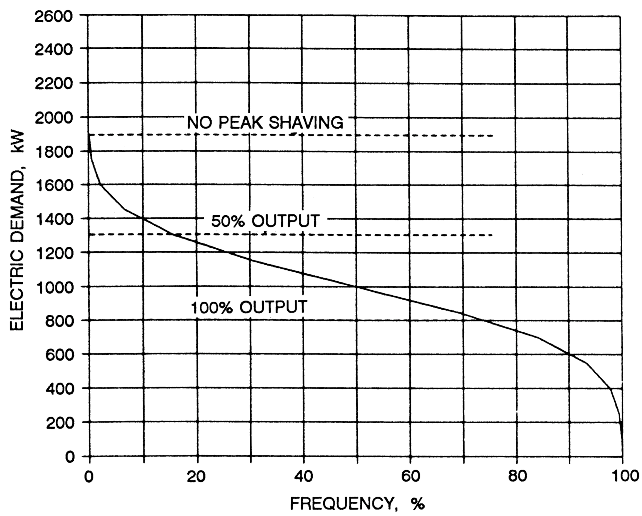

Fig. 56 Hypothetical Peaking Generator

will not be realized until a year later. Even though an existing standby generator may seem to offer "free" peak shaving capacity, careful planning and operation are required to secure its full potential.

Conversely, continuous-duty/standby systems offer the benefits of heat recovery while satisfying the standby requirements of the facility. During emergencies, the generator load is switched from its normal nonessential load to the emergency load.

Two-Dimensional Load Duration Curve

A two-dimensional load duration curve becomes necessary when the designer must consider simultaneous steam and electrical load variations. Such a situation occurs, for example, when the facility electrical demand drops below the output of a steam-tracking generator, and the excess power capacity cannot be exported. During these periods, the generator is throttled to curtail electrical output to that of facility demand. In other words, it is now operating in an electrical tracking mode.

To develop the two-dimensional load duration curve method for simultaneous loads, either the electrical demand or the steam demand must be broken into discrete periods defined by the number of hours per year when the electric or steam demand is within a certain range of values, or bins.

Duration curves for the remaining load are then created for each period as before, using coincident values. Figure 57 illustrates such a representation for three bins. The electrical load values indicated in the figure represent the center value of each bin. In general, a large number of periods gives a more accurate representation of facility loads. Also, the greater the load fluctuation, the greater the number of periods required for accurate representation. Note that the total number of hours for all periods adds up to 8760, the number of hours in a year.

Another situation that requires a two-dimensional load duration curve is when the facility buys or sells power, the price of which depends on the time of day or time of the year. Many electric rate structures contain explicit time periods for the purchase or sale of power. In some cases, there is only a summer-winter distinction. Other rate schedules may have several periods to reflect time-of-use or time-of-sale rates. The two-dimensional analysis is required to consider these rate schedule periods when defining the load bins because the operating schedule with the greatest annual savings is influenced by energy prices as well as energy demands.

Using Figure 57 as an illustration of a summer-peaking utility, the first two bins might coincide with the winter-fall-spring off-peak rate and the third bin with the on-peak rate. Because the analysis becomes burdensome when there are large load swings and several rate periods, the calculations are run by computers.

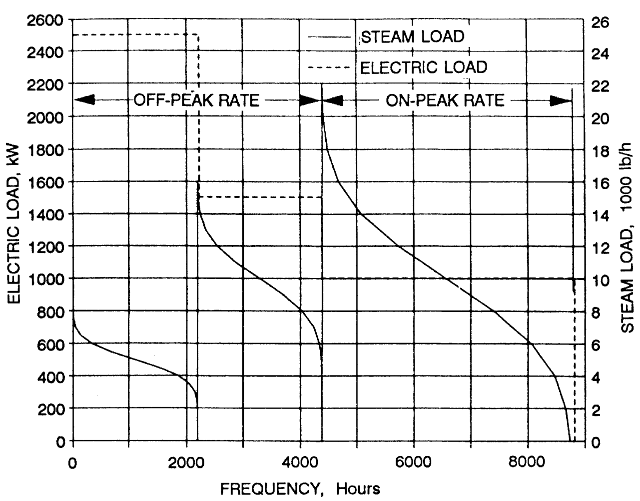

Fig. 57 Example of Two-Dimensional Load Duration Curve

Analysis by Simulations

The load duration curve is a convenient, intuitive graphical tool for preliminary sizing and analysis of a cogeneration system; it lacks the capability, however, for detailed analysis. A commercial or institutional facility, for example, can have as many as four different loads that must be considered simultaneously. These are cooling, noncooling electrical, steam or high-temperature hot water for space heating, and low-temperature service hot water. These loads are never in balance at any instant, which complicates sizing equipment, establishing operating modes, and determining the quantity of heat rejected from the cogenerator that is usefully applied to the facility loads.

Further complicating the evaluation of commercial cogeneration systems is the fact that prime movers rarely operate at full rated load; therefore, part-load operating characteristics such as fuel consumption, exhaust mass flow, exhaust temperature, and heat rejected from the jacket and intercooler of internal combustion engines must be considered. If the prime mover is a combustion gas turbine, then the effects of ambient temperature on full-load capacity and part-load fuel consumption and exhaust characteristics (or on cooling capacity if gas turbine inlet air cooling is used) should be considered.

In addition to the prime movers, a commercial or institutional cogeneration system may include absorption chillers that use jacket heat and exhaust heat to produce chilled water at two different COPs or ECOPs. Some commercial cogeneration systems use thermal energy storage to store hot water, chilled water, or ice to reduce the recovered thermal energy that must be dumped during periods of low demand. Internal combustion engines coupled with steam compressors and steam-injected gas turbines have been produced to allow some variability of the output heat/power ratio from a single package.

Computer programs that analyze cogeneration systems are available. Many of these programs emphasize the financial aspects of cogeneration systems with elaborate rate structures, energy price forecasts, and economic models. However, equipment part-load performance, load schedules, and other technical characteristics that greatly affect system economics are modeled only superficially in many programs. Also, many computer codes contain built-in equipment data and perform system selection and sizing automatically, thereby excluding the designer from important design decisions that could affect system viability. While these programs have their place, they should only be used with care by those who know the program limitations.

The primary consideration in selecting a computer program for analyzing commercial cogeneration technical feasibility is the ability to handle multiple, time-varying loads. The four methods of modeling the thermal and electrical loads are as follows:

- Hourly average values for a complete year
- Monthly average values
- Truncated year consisting of hourly averaged values for one or more typical (usually working and nonworking) days of each month
- Bin methods

Cogeneration simulation using hourly averaged values for load representation provides the greatest accuracy; however, the 8760 values for each type of load can make the management of load data a formidable task. Guinn (1987) describes a public domain simulation that can consider a full year of multiple-load data.

The data needed to perform a monthly averaged load representation can often be obtained from utility billings. This model should only be used as an initial analysis; best accuracy is obtained when thermal and electric profiles are relatively consistent.

The truncated year model is a compromise between the accuracy offered by the full hourly model and the minimal data-handling requirements of the monthly average model. It involves the development of hourly values for each load over a typical average day of each month. Usually, two typical days are considered—a working day and a nonworking day. Thus, instead of 8760 values to represent a load over a year, only 576 values are required. This type of load model is often used in cogeneration computer programs. Pedreyra (1988) and Somasundarum (1986) describe programs that use this method of load modeling.

Bin methods are based on the frequency distribution, or histogram, of load values. The method determines the number of hours per year the load was in different ranges, or bins. This method of representing weather data is widely used to perform building energy analysis. It is a convenient way to condense a large database into a smaller set of values, but it is no more accurate than the time resolution of the original set. Furthermore, bin methods become unacceptably cumbersome for cogeneration analysis if more than two loads must be considered.

CODES AND STANDARDS

In addition to applicable local codes, the following codes and standards should be consulted.

National Electrical Code (NFPA *Standard* 70-98)

Article 440, Air Conditioning and Refrigeration Equipment
Article 445, Generators
Article 700-12(b), Emergency System Set
Articles 701 and 702, Stand-by Power Gas Codes
(These articles cover service at all pressures.)

ASHRAE

Standard 15-1994, Safety Code for Mechanical Refrigeration
(This standard covers the refrigeration section of the system.)

National Fire Protection Association (NFPA)

30-96 Flammable and Combustible Liquids Code
31-97 Installation of Oil-Burning Equipment
37-98 Installation and Use of Stationary Combustion Engines and Gas Turbines
54-99 National Fuel Gas Code
58-98 Liquefied Petroleum Gas Code
59-98 Storage and Handling of Liquefied Petroleum Gases at Utility Gas Plants
59A-96 Production, Storage, and Handling of Liquefied Natural Gas (LNG)
90A-96 Installation of Air Conditioning and Ventilating Systems
211-96 Chimneys, Fireplaces, Vents, and Solid Fuel-Burning Appliances

Other Standards Organizations

National Engine Manufacturers Association (NEMA)
American Society for Testing and Materials (ASTM)
American Refrigeration Institute (ARI)

REFERENCES

ASHRAE. 1993. *Air-conditioning systems design manual*.

ASHRAE. 1996. *Cogeneration design guide*.

ASME. 1982. Appendix A to Test code for steam turbines. ANSI/ASME *Standard* PTC 6A-82 (R 1995). American Society of Mechanical Engineers, New York.

ASME. 1985. Guidance for evaluation of measurement uncertainty in performance tests of steam turbines. ANSI/ASME *Standard* PTC 6 *Report*-85 (R 1997).

ASME. 1988. Procedures for routine performance tests of steam turbines. ANSI/ASME *Standard* PTC 6S *Report*-88 (R 1995).

ASME. 1996. Steam turbines. ANSI/ASME *Standard* PTC 6-96.

ASME. 1998. Rules for the construction of pressure vessels. *Boiler and Pressure Vessel Code*, Section VIII-98.

Avery, G., W.C. Stethem, W.J. Coad, R.A. Hegberg, F.L. Brown, and R. Petitjean. 1990. The pros and cons of balancing a variable flow water system. *ASHRAE Journal* 32(10):32-59.

Becker, H.P. 1975. Energy conservation analysis of pumping systems. *ASHRAE Journal* 17(4):43-51.

Becker, H.P. 1988. Is cogeneration right for you? *Hotel and Resort Industry* (Sept.). New York.

DEMA. 1972. *Standard practices for low and medium stationary diesel and gas engines*, 6th ed. Diesel Engine Manufacturers Association.

Ebeling, J.A. 1994. Combustion turbine inlet air cooling alternatives and case histories. *Proceedings* of the 27th Annual Frontiers of Power Conference, Stillwater, OK (Oct. 24-25).

Encotech. 1992. Coupling effects on the torsional vibration of turbine generator power trains. *Encotech Technical Topics* 7(1). Schenectady, NY.

Guinn, G.R. 1987. Analysis of cogeneration systems using a public domain simulation. *ASHRAE Transactions* 93(2):333-66.

ISO. 1995. Reciprocating internal combustion engines—Performance—Part 1: Standard reference conditions, declarations of power, fuel, and lubricating oil consumptions, and test methods. *Standard* 3046-1-95. International Organization for Standardization, Geneva, Switzerland.

MacCracken, C.D. 1994. Overview of the progress and the potential of thermal storage in off-peak turbine inlet cooling. *ASHRAE Transactions* 100(1):569-71.

Mannion, G.F. 1988. High temperature rise piping design for variable volume pumping systems: Key to chiller energy management. *ASHRAE Transactions* 94(2):1427-43.

National Engine Use Council. 1967. NEUC engine criteria. Chicago.

NEMA. 1991. Land based steam turbine generator sets. *Standard* SM 24-91. National Electrical Manufacturers Association, Rosslyn, VA.

NEMA. 1991. Steam turbines for mechanical drive service. ANSI/NEMA *Standard* SM 23-91.

Newman, L.E. 1945. Modern extraction turbines, Part II—Estimating the performance of a single automatic extraction condensing steam turbine. *Power Plant Engineering* (Feb.).

Pedreyra, D.C. 1988. A microcomputer version of a large mainframe program for use in cogeneration analysis. ASHRAE *Transactions* 94(1): 1617-25.

Somasundarum, S. 1986. An analysis of a cogeneration system at T.W.U. *The Cogeneration Journal* 1(4):16-26.

Stewart, W.E., Jr. 1997. *Design guide for combustion turbine inlet air cooling systems*. ASHRAE.