Related Commercial Resources

CHAPTER 7

COMBINED HEAT AND POWER SYSTEMS

COMBINED heat and power (CHP) is the simultaneous production of electrical or mechanical energy (power) and useful thermal energy from a single energy source. By capturing and using the recovered heat energy from an effluent stream that would otherwise be rejected to the environment, CHP (or *cogeneration*) systems can operate at utilization efficiencies greater than those achieved when heat and power are produced in separate processes, thus contributing to sustainable building solutions.

Recovered thermal energy from fuel used in reciprocating engines, steam or combustion turbines (including microturbines, which are typically less than 500 kW in capacity), Stirling engines, or fuel cells can be used in the following applications:

- Direct heating: exhaust gases or coolant fluids used directly for drying processes, to drive an exhaust-fired absorption chiller, to regenerate desiccant materials in a dehumidifier, or to operate a bottoming cycle
- Indirect heating: exhaust gases or coolant fluids used to heat a secondary fluid (e.g., steam or hot water) for devices, to generate power, or to power various thermally activated technologies
- Latent heat: extracting the latent heat of condensation from a recovered flow of steam when the load served allows 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)

There are many potential applications, including base-load power, peaking power where on-site power generation (distributed generation) is used to reduce the demand or high on-peak energy charges imposed by the electric energy supplier, back-up power, remote power, power quality, and CHP, providing both electricity and thermal needs to the site. Usually, customers own the small-scale, on-site power generators, but third parties may own and operate the equipment. Table 1 provides an overview of typical applications, technologies and uses of **distributed generation (DG)** and CHP systems.

On-site CHP systems are small compared to typical central station power plants. DG systems are inherently modular, which makes distributed power highly flexible and able to provide power where and when it is needed. DG and CHP systems can offer significant benefits, depending on location, rate structures, and application. Typical advantages of an on-site CHP plant include improved power reliability and quality, reduced energy costs, increased predictability of energy costs, lowered financial risk, use of renewable energy sources, reduced emissions, and faster response to new power demands because capacity additions can be made more quickly.

CHP system efficiency is not as simple as adding outputs and dividing by fuel inputs. Nevertheless, using what is normally waste exhaust heat yields overall efficiencies (η_O) of 50 to 70% or more (for a definition of overall efficiency, see the section on Performance Parameters).

The preparation of this chapter is assigned to TC 1.10, Cogeneration Systems.

CHP can operate on a topping, bottoming, or combined cycle. Figure 1 shows an example of topping and bottoming configurations. In a **topping cycle**, energy from the fuel generates shaft or electric power first, and thermal energy from the exiting stream is recovered for other applications such as process heat for cooling or heating systems. In a **bottoming cycle**, shaft or electric power is generated last from thermal energy left over after higher-level thermal energy has been used to satisfy thermal loads. *A typical topping cycle recovers heat from operation of a prime mover and uses this thermal energy for the process (cooling and/or heating). A bottoming cycle recovers heat from the process to generate power.* A **combined cycle** uses thermal output from a prime mover to generate additional shaft power (e.g., combustion turbine exhaust generates steam for a steam turbine generator).

Grid-isolated CHP systems, in which electrical output is used on site to satisfy all site power and thermal requirements, are referred to as **total energy systems**. Grid-parallel CHP systems, which are actively tied to the utility grid, can, on a contractual or tariff basis, exchange power with or reduce load on (thus reducing capacity demand) the public utility. This may eliminate or lessen the need for redundant on-site back-up generating capacity and allows operation at maximum thermal efficiency when satisfying the facility's thermal load; this may produce more electric power than the facility needs.

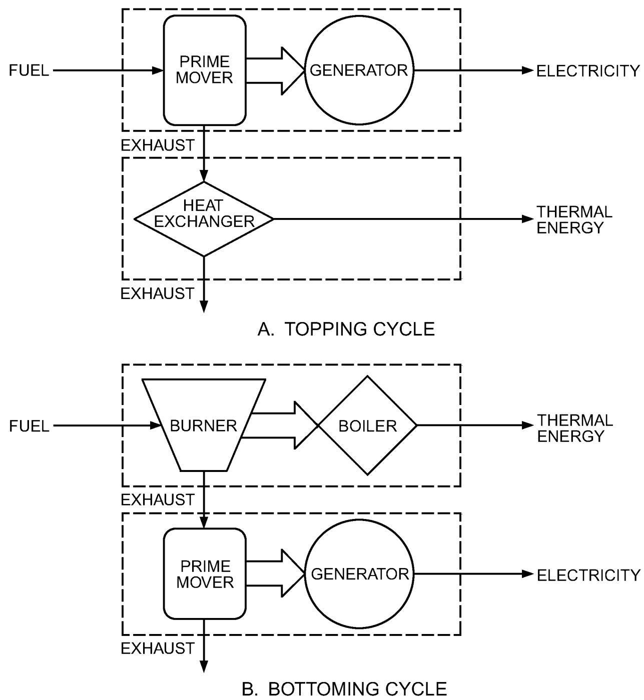

Fig. 1 CHP Cycles

Table 1 Applications and Markets for DG/CHP Systems

DG Technologies	Standby Power	Base-Load Power Only	Demand Response Peaking	Customer Peak Shaving	Premium Power	Utility Grid Support	CHP	Applicable Market Sectors
Reciprocating engines: 50 kW to 16 MW	X	X	X	X	X	X	X	Commercial buildings, institutional, industrial, utility grid (larger units), waste fuels
Gas turbines: 500 kW to 50 MW		X		X	X	X	X	Large commercial, institutional, industrial, utility grid, waste fuels
Steam turbines: 500 kW to 100 MW		X			X		X	Institutional buildings/campuses, industrial, waste fuels
Microturbines: 30 to 500 kW	X	X	X	X	X	X	X	Commercial buildings, light industrial, waste fuels
Fuel cells: 5 kW to 2 MW		X			X	X	X	Residential, commercial, light industrial

Source: Adapted from NREL (2003).

CHP 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, waste heat recovery system, and thermally activated technologies. Integrating design of the project's electrical and thermal requirements with the CHP plant is required for optimum economic performance. Matching the CHP plant's thermal/electric ratio with that of the building load is required for optimum economic benefit. The basic components of the CHP plant are the (1) prime mover and its fuel supply system, (2) generator and accessories, including interconnection and protection systems, (3) waste heat recovery system, (4) thermally activated technologies, (5) control system, (6) electrical and thermal transmission and distribution systems, and (7) connections to mechanical and electrical services.

This chapter describes the increasing role of CHP in sustainable design strategies, presents typical system designs, provides means and methods to understand system performance, and describes prime movers, such as reciprocating and Stirling engines, combustion and steam turbines, and fuel cells, and their characteristics for various uses. It also describes thermally activated technologies (TAT) such as heat recovery, absorption chillers, steam turbine-driven chillers, and desiccant dehumidifiers, as well as organic Rankine cycle (ORC) machines for waste heat recovery. Related issues, such as fuels, lubricants, instruments, noise, vibration, emissions, and maintenance, are discussed for each type of prime mover. Siting, interconnection, installation, and operation issues are also discussed. Thermal distribution systems are presented in Chapters 11 and 12.

TERMINOLOGY

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

Back-up power. Electric energy available from or to an electric utility during an outage to replace energy ordinarily generated by the CHP plant.

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

Black start. A start-up of an off-line, idle, non-spinning generation source without the electric utility.

Bottoming cycle. CHP facility in which energy put into the system is first applied to another thermal energy process; rejected heat from the process is then used for power production.

Capacity. Load for which an apparatus is rated (electrical generator or thermal system) at specific conditions.

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

Capacity factor. Ratio of the actual annual output to the rated output over a specified time period.

Coefficient of performance (COP). Refrigeration or refrigeration plus thermal output energy divided by the energy input to the absorption device, refrigeration compressor, or steam turbine. See also **Combined heat and power (CHP)**.

Combined heat and power (CHP). Simultaneous production of electrical or mechanical energy and useful thermal energy from a single energy stream.

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 of topping or bottoming cycles to generate shaft power and/or useful thermal output. CHP 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 on the actual measured demand of the billing period.

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

Peak demand. Demand at the instant of greatest load.

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

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

Demand-side management (DSM). Process of managing the consumption of energy, generally to optimize available and planned generation resources.

Economic coefficient of performance (ECOP). Output energy in terms of economic costs divided by input fuel in terms of energy purchased or produced, expressed in consistent units (e.g., Btu/h) for each energy stream.

Efficiency. See the section on Performance Parameters.

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

Energy Information Administration (EIA). Independent agency in the U.S. Department of Energy that develops surveys, collects energy data, and analyzes and models energy issues.

Electric tariff. Statement of electrical rate and terms and conditions governing its application.

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

Grid interconnection. System that manages the flow of power and serves as communication, control, and safety gateway between a CHP plant and an electric utility's distribution network.

Harmonics. Wave forms with frequencies that are multiples of the fundamental (60 or 50 Hz) wave. The combination of harmonics and the fundamental wave causes a nonsinusoidal, periodic wave. Harmonics in power systems result from nonlinear effects. Typically 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 **higher heating value (HHV)** includes the energy needed to vaporize water formed during combustion, whereas the **lower heating value (LHV)** deducts this energy because it does not contribute to useful work.

Intermediate load. Range from base electric or thermal load to a point between base load and peak.

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

Isolated plant. A CHP plant not connected to the electric utility grid.

Load factor. Load served by a system over a designated period divided by system capacity over the same period.

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

Peak load. Maximum electric or thermal load in a stated period of time.

Peak shaving. Reduction of peak power demand using on-site power generation, thermally activated technologies, or other load-shifting device.

Point of common coupling. Point where a CHP system is connected to the local grid.

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

Premium power. High reliability power supply and/or high voltage/current power quality.

Reactive power. Reactive power exists in all alternating current (AC) power systems as current leads or lags voltages; inadequate reactive power reserves can contribute to voltage sags or even voltage collapse. CHP and/or on-site power systems can provide reactive power support where they are connected to the grid.

Selective energy systems. Form of CHP 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 required to produce the steam. For a fuel-fired prime mover, it is the fuel input to the prime mover.

Standby power. Electric energy supplied by a cogenerator or other on-site generator during a grid outage.

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

Supplemental firing. Injection and combustion of additional fuel into an exhaust gas stream to raise its energy content (heat).

Transmission and distribution (T&D). System of wires, transformers, and switches that connects the power-generating plant and end user.

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

Total energy system. Form of CHP 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 back-up.

Transmission. Network of high-voltage lines, transformers, and switches used to move electric power from generators to the distribution system.

Voltage flicker. Significant fluctuation of voltage (see Figure 28 in Chapter 55 of the 2007 *ASHRAE Handbook—HVAC Applications*).

Wheeling. Using one system's transmission facilities to transmit gas or power for another system.

See Chapter 55 of the 2007 *ASHRAE Handbook—HVAC Applications* for a more detailed discussion of electricity terminology.

CHP SYSTEM CONCEPTS

CUSTOM-ENGINEERED SYSTEMS

Historically, CHP systems were one-of-a-kind, custom-engineered systems. Because of the high cost of engineering, and technical, economic, environmental, and regulatory complexities, extraordinary care and skill are needed to successfully design and build custom-engineered CHP systems. Custom-engineered systems continued to be the norm among systems greater than 5 MW and where process requirements require significant customization.

PACKAGED AND MODULAR SYSTEMS

Packaged/modular CHP systems are available from 5 to over 5000 kW. Packaged systems are defined as the integration of one or more power component (engines, microturbines, combustion turbines, and fuel cells), powered component (generators, compressors, pumps, etc.), heat recovery device [heat exchangers, heat recovery steam generators (HRSGs)], and/or thermally activated technology (absorption/adsorption chillers, steam turbines, ORCs, desiccant dehumidifiers), prefabricated on a single skid. A modular system is two or more packaged systems designed to be easily interconnected in the field. Modular systems are used largely because of shipping or installation size limitations. Packaged and modular systems often can be functionally tested in the factory before shipment, increasing the ease of commissioning. The simplest packaged and modular CHP systems are found in tightly integrated systems in general categories such as the following:

Reciprocating engine systems:

- Small engine generators (under 500 kW) recover jacket and exhaust heat in the form of hot water. Packaged systems include electronic safety and interconnection equipment.
- Small packaged and split-system engine heat pumps, integrating engines with complete vapor compression heat pumps and engine jacket and exhaust heat recovery, available under 50 rated tons (RT).
- Engine packaged systems (typically 100 to 2000 kW) can drive generators, compressors, or pumps. Engine/generators recover at least jacket heat, and several modular systems integrate jacket water and exhaust systems to directly power single- and two-stage absorption chillers, providing power, heating, and cooling.

Microturbine systems:

- Microturbines integrate exhaust heat recovery (hot water) with electronic safety and interconnection equipment in a single compact package.
- Microturbines are easily integrated electrically and thermally (exhaust), providing ideal CHP systems delivering multiple power offerings, typically up to 500 kW, with hot-water systems or using integrated single- and two-stage absorption chillers.

Combustion turbine systems:

- Modular systems have been developed in the 1 to 6 MW range, combining turbine generators, inlet cooling, exhaust control, heat recovery steam generators and/or absorption chillers.

LOAD PROFILING AND PRIME MOVER SELECTION

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 prime mover's thermal/electric ratio 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, and (4) the desire to size to the thermal baseload. The form and quality of the required thermal energy is very important. If high-pressure steam is required, the reciprocating engine is less attractive as a thermal source than a combustion turbine.

Regardless of how the prime mover is chosen, the degree of use of the available heat determines the overall system efficiency; this is the critical factor in economic feasibility. Therefore, the prime mover's thermal/electric ratio and load must be analyzed as a first step towards making the best choice. Maximizing efficiency is generally not as important as thermal and electric use.

CHP paralleled with the utility grid can operate at peak efficiency if (1) 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) the electric generators are sized for 100% of the site's electrical demand and recovered heat can be fully used at that condition, with additional thermal demands met by supplementary means and excess power sold to a utility or other electric energy supplier or broker.

Heat output to the primary process is determined by the engine type and load. It must be balanced with actual requirements by supplementation or by rejecting excess heat 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, (2) supplement heat if it is not available, or (3) reject heat when availability exceeds requirements.

In plants with more than one prime mover, controls must be added to balance the power output of the prime movers and to balance reactive power flow between the generators. Generally, an isolated system requires that the prime movers supply the needed electrical output, with 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 **thermal load following** (adjusting the prime mover throttle for the required amount of heat). The amount of electrical energy generated then depends on heat requirements; imbalances between the thermal and electrical loads are carried by the electric utility either through absorption of excess generation or by delivering 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. Careful selection of the prime mover type and model is critical in providing the correct thermal/ electric ratio to minimize electric and thermal waste. Supplemental electric power is purchased and/or thermal energy generated by other means when the thermal load exceeds the generator's maximum capacity.

Analysis of these tracking scenarios requires either a fairly accurate set of coincident electric and thermal profiles typical for a variety of repetitious operating modes, or a set of daily, weekend, and holiday accumulated electrical and thermal consumption requirements. Where thermal load profiles are not necessarily coincident with electric load profiles, thermal energy storage may be used to maximize load factor.

PEAK SHAVING

High on-peak energy charges and/or electrical demand charges in many areas, ratchet charges (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 demand charge based on the actual measured usage), and utility capacity shortages have led to **demand-side management (DSM)** by utilities and their consumers. In these cases, a strategy of peak shaving or generating power only during peak cost or peak demand situations may be used.

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 2), if loss of compressor service can be tolerated in an emergency.

If engine capacity of a dual-service system equals 150 or 200% of the compressor load, power available from the generator can be delivered to the utility grid during normal operation. Although 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, required capital investment, space, and maintenance are lower, 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.

POWER PLANT INCREMENTAL HEAT RATE

Typically, CHP 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

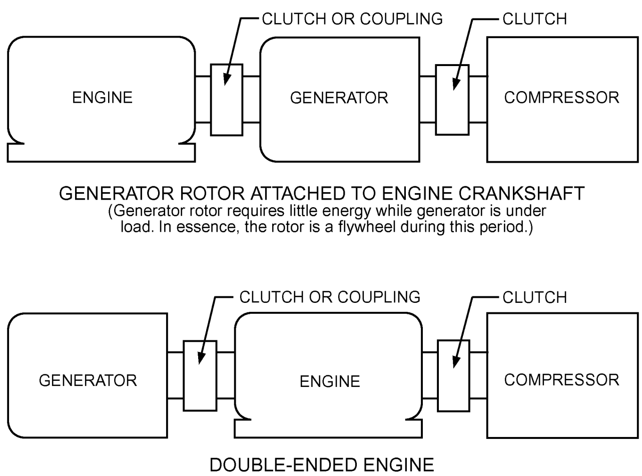

Fig. 2 Dual-Service Applications

particular site. For example, if a gas engine generator with a design heat rate of 10,000 Btu/kWh (34% 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 CHP power plant would be only 6000 Btu/kWh, which translates to an 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 efficiency dropping to 43%. Thus, CHP power plant performance really depends on the required thermal/electric ratio for a particular application, and it is only according to this ratio that the type of CHP configuration should be chosen.

A system that requires 1000 kWh electrical energy and 7000 lb low-pressure steam at 30 psig (thermal/electric ratio of about 2.4, or 7 lb steam per kilowatt-hour) can be used to further illustrate measuring CHP system performance. In this example, a 1000 kW gas engine-generator CHP system provides a maximum of 1500 lb/h steam, with the balance of 5500 lb/h to be met by a conventional boiler with 75% thermal efficiency. Thus, the total power requirement 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 a gas turbine CHP system with the same power and heat requirements is used instead, with a heat rate of 13,650 Btu/h per kilowatt (efficiency of 25%), it 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 overall 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.

PERFORMANCE PARAMETERS

HEATING VALUE

Natural gas is often selected as the fuel for CHP systems, although the same considerations discussed here apply to biofuels and fossil fuels (Peltier 2001). There are two common ways to define the energy content of fuel: higher heating value and lower heating value.

Turbine, microturbine, engine, and fuel cell manufacturers typically rate their equipment using **lower heating value (LHV)**, which accurately measures combustion efficiency; however, LHV neglects the energy in water vapor formed by combustion of hydrogen in the fuel. This water vapor typically represents about 10% of the energy content. LHVs for natural gas are typically 900 to 950 Btu/ft^3.

Higher heating value (HHV) for a fuel includes the full energy content as defined by bringing all products of combustion to 77°F. Natural gas typically is delivered by the local distribution company with values of 1000 to 1050 Btu/ft^3 on this HHV basis. Because the actual value may vary from month to month, some gas companies convert to therms (1 therm = 100,000 Btu). These measures all represent higher heating values.

Consumers purchase natural gas in terms of its HHV; therefore, performances of CHP systems as well as the electric grid for comparison are calculated in HHV.

The **net electric efficiency** η_E of a generator can be defined by the first law of thermodynamics as net electrical output W_E divided by fuel consumed Q_{fuel} in terms of kilowatt-hours of thermal energy content.

$$\eta_E = \frac{W_E}{Q_{fuel}}$$

A CHP system, by definition, produces useful thermal energy (heat) as well as electricity. If the first law is applied, adding the useful thermal energy Q_{TH} to the net electrical output and dividing by the fuel consumed (which is how virtually all CHP system efficiencies are reported), the resulting overall efficiency η_O does not account for the relative value of the two different energy streams:

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}}$$

According to the second law of thermodynamics, the two different energy streams have different relative values; heat and electricity are not interchangeable. The first law describes the quantity of the two energy streams, whereas the second law describes their quality or value (exergy). Electrical energy is generally of higher value because it can do many types of work, and, in theory, 100% of it can be converted into thermal energy. Thermal energy is more limited in use and is converted to work at rates usually much lower than 100% conversion. The theoretical maximum efficiency at which thermal energy can be converted to work is the Carnot efficiency, which is a function of the quality, or temperature, of the thermal energy and is defined as $(T_{high} - T_{low})/T_{high}$.

CHP ELECTRIC EFFECTIVENESS

The current methodology of using net electric efficiency η_E and overall efficiency η_O either separately or in combination does not adequately describe CHP performance because

- η_E gives no value to thermal output
- η_O is an accurate measure of fuel use but does not differentiate the relative values of the energy outputs, and is not directly comparable to any performance metric representing separate power and thermal generation

CHP electric effectiveness ε_{EE} is a new, single metric that recognizes and adequately values the multiple outputs of CHP systems and allows direct comparison of system performance to the conventional electric grid and competing technologies. This more closely balances the output values of CHP systems and allows CHP system development to be evaluated over time.

CHP electric effectiveness views the CHP system as primarily providing thermal energy, with electricity as a by-product. It is then defined as net electrical output divided by incremental fuel consumption of the CHP system above the fuel that would have been required to produce the system's useful thermal output by conventional means. This approach credits the system's fuel consumption to account for the value of the thermal energy output, and measures how effective the CHP system is at generating power (or mechanical energy) once the thermal needs of a site have been met. This metric is most effective when used on a consistent and standardized basis, meaning

- The metric measures a single point of performance (design point)
- The design point for power generation is measured at ISO conditions (for combustion turbines, microturbines, and fuel cells, 59°F, 60% rh, sea level, per ISO *Standard* 3977-2; for reciprocating engines, 77°F, 30% rh, and 14.5 psia per ISO *Standard* 3046-1)
- The performance evaluates fuel input and CHP outputs at design point only
- HHV is used because it measures the true values of performance in relation to fuel use and fuel cost (HHV is more commonly used to compare energy systems, is the basis of fuel purchases, and is the basis of emissions regulation)

Power and Heating Systems

For CHP systems delivering power and heating (steam and/or hot water, or direct heating), the CHP electric effectiveness is defined as

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)}$$

where α is the efficiency of the conventional technology that otherwise would be used to provide the useful thermal energy output of the system (for steam or hot water, a conventional boiler); see Table 2.

Examples 1 to 5 demonstrate how to apply this metric. The basis for comparison is a 25% HHV efficient electric power source. Performance values for larger combustion turbines, reciprocating engines, and fuel cells vary significantly.

Example 1. Separate Power and Conventional Thermal Generation. A facility supplies its power and thermal requirements by two separate systems: a conventional boiler for its thermal needs and a power-only generator for electricity.

Conventional Boiler: 100 units of fuel are converted into 80 units of heat and 20 units of exhaust energy as shown in Figure 3.

Power-Only Generator: A 25% HHV efficient electric generator consumes 160 units of fuel and produces 40 units of electricity and 120 units of exhaust energy (Figure 4).

The performance metrics for this separate approach to energy supply are as follows:

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 80}{160 + 100} = 0.46$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)} = \frac{40}{260 - (80/0.80)} = 0.25$$

Example 2. Combined Power and Thermal Generation (Hot Water/Steam). A CHP system is used to meet the same power and thermal requirements as in Example 1, with a 25% HHV efficient generator and a 67% efficient heat recovery heat exchanger (e.g., a 600°F airstream reduced to 240°F exhaust and yielding 200°F hot water). The performance parameters for this combined system are shown in Figure 5.

Table 2 Values of α for Conventional Thermal Generation Technologies

Fuel	α
Natural gas boiler	0.80
Biomass boiler	0.65
Direct exhaust*	1.0

*Direct drying using exhaust gas is analogous to using flue stack gas for the same purpose; therefore, a direct one-to-one equivalence is best for comparison.

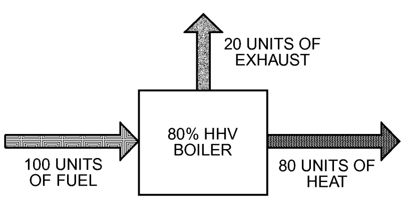

Fig. 3 Conventional Boiler for Example 1

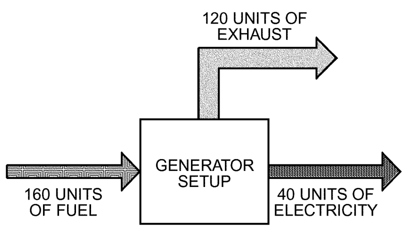

Fig. 4 Power-Only Generator for Example 1

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 80}{160} = 0.75$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)} = \frac{40}{160 - (80/0.80)} = 0.67$$

Note that η_E for both systems (Example 1's separate generation and Example 2's CHP) is the same, but the CHP system uses less fuel to produce the required outputs, as shown by the differences in overall efficiency (η_O = 75% for CHP versus η_O = 46% for separate systems); see Figure 6. However, this metric does not adequately account for the relative values of the thermal and electric outputs. The electric effectiveness metric, on the other hand, nets out the thermal energy, leaving an ε_{EE} of 67% for the CHP system.

Example 3. Combined Power and Thermal Generation (Direct Exhaust Heat). In some cases, exhaust gases are clean enough to be used for heating directly (e.g., greenhouses and drying where microturbine and gas turbine exhaust is used). For these cases, the thermal recovery efficiency is the difference between exhaust gas temperature and ambient temperature, where delivered exhaust gas is divided by the difference between exhaust gas temperature and outdoor ambient temperature. Direct exhaust gas delivery is a direct-contact form of heating; thus, heat transfer losses are minimal. For a 25% efficient electric generator

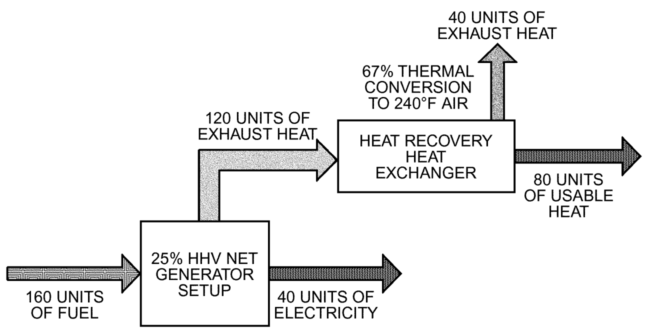

Fig. 5 Performance Parameters for Combined System for Example 2

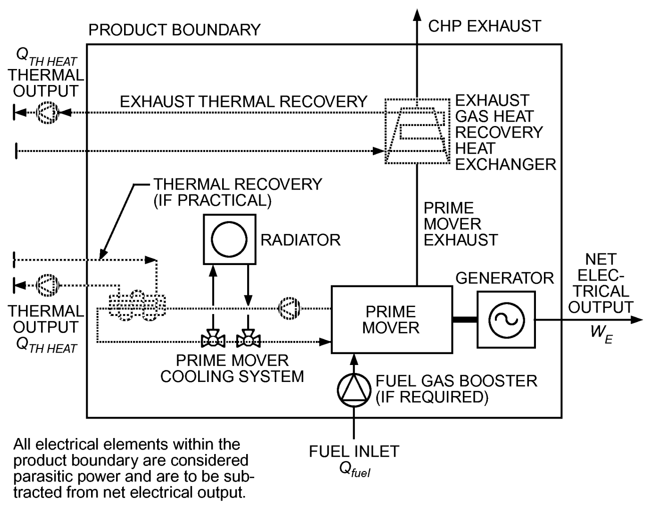

Fig. 6 CHP Power and Heating Energy Boundary Diagram for Example 2

exhausting into a greenhouse with an internal temperature of 100°F, thermal recovery efficiency = (600°F – 100°F)/(600°F – 59°F) = 92% (note that 59°F is the ISO rating condition for microturbines per ISO *Standard* 3977-2).

Performance parameters for this combined system are shown in Figure 7, and system boundaries are shown in Figure 8.

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 110}{160} = 0.94$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{160 - (110/1.00)} = 0.80$$

Example 4. Combined Power and Thermal Generation (Combustion Turbine [CT] Without Cofired Duct Burner). In this example, exhaust gas from the 25% efficient electrical combustion turbine generator setup is assessed first without any exhaust enhancement, and then the same system is assessed with temperature and energy content enhancement using cofiring of additional fuel in a duct burner placed in the exhaust and using a heat recovery steam generator (HRSG). The basis for this example is using fuel input and steam output data from a simple cycle 12,000 Btu/h gas turbine, as shown in Figures 9 and 10.

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 69}{160} = 0.68$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{160 - (69/0.80)} = 0.54$$

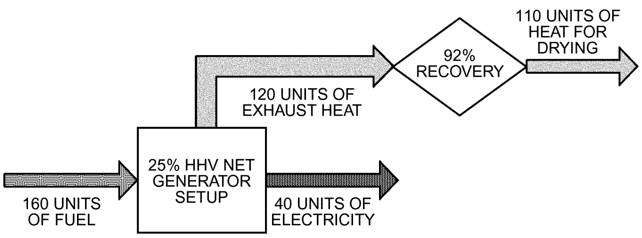

Fig. 7 Performance Parameters for Example 3

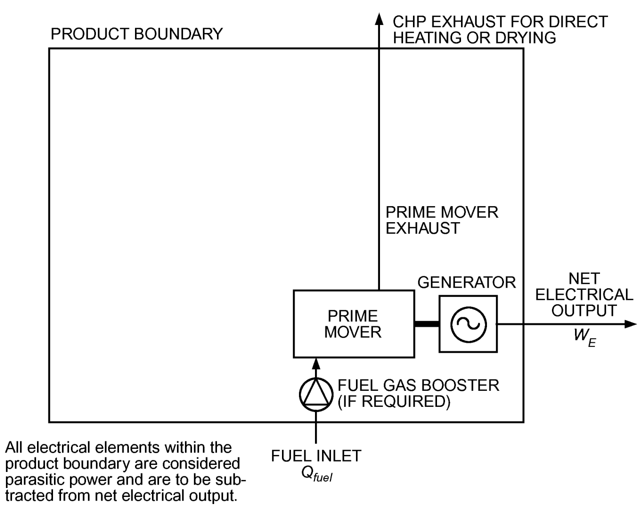

Fig. 8 CHP Power and Direct Heating Energy Boundary Diagram for Example 3

Note that, based on this system approach, cofiring has no effect on η_E because no fuel flows to the duct burner in power-only mode. However, ε_{EE} increases from 0.54 to 0.71 (Figure 11) as shown in Example 5.

Example 5. Combined Power and Thermal Generation (Combustion Turbine [CT] with Cofired Duct Burner) (Figure 12).

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 182}{284} = 0.78$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{284 - (182/0.80)} = 0.71$$

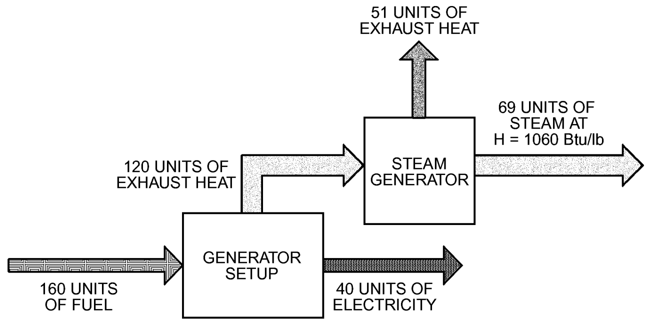

Fig. 9 Performance Parameters for Example 4

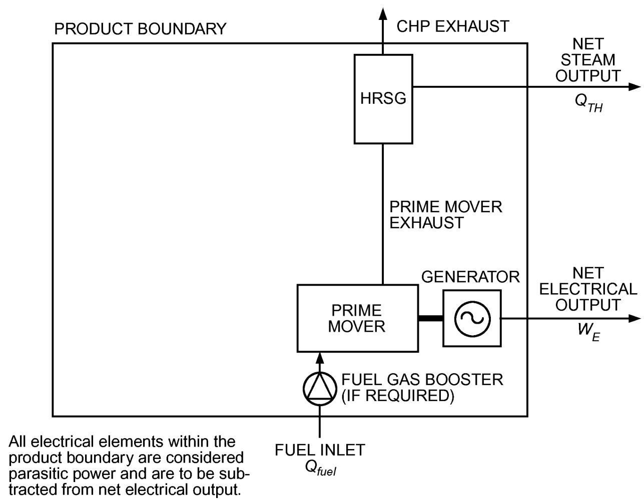

Fig. 10 CHP Power and HRSG Heating Without Duct Burner Energy Boundary Diagram for Example 4

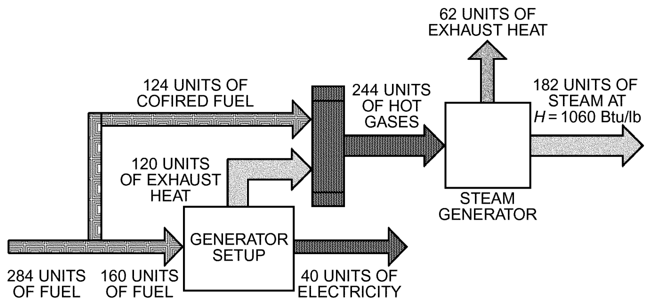

Fig. 11 Cofiring Performance Parameters for Example 4

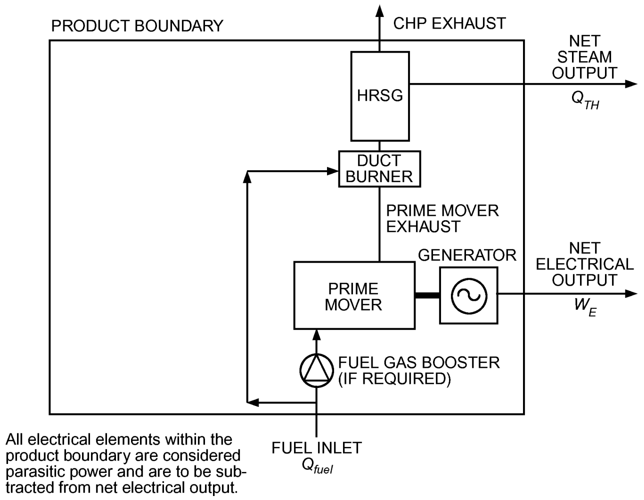

Fig. 12 CHP Power and HRSG Heating with Duct Burner Energy Boundary Diagram for Example 5

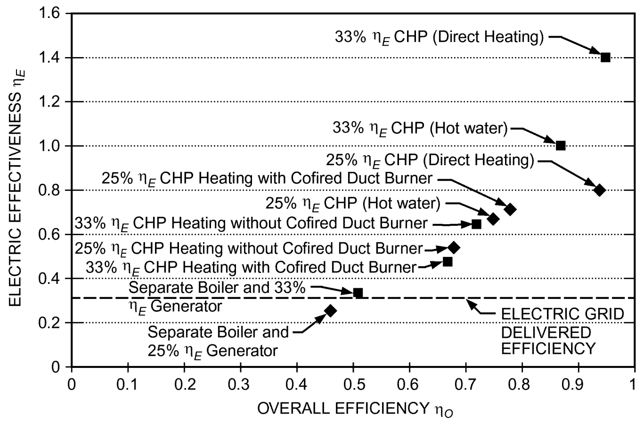

Fig. 13 Electric Effectiveness η_E Versus Overall Efficiency η_O

Table 3 Summary of Results from Examples 1 to 5

Example	System	η_E	η_O	ε_{EE}
1	Separate boiler and generator	0.25	0.46	0.25
2	Combined heat (hot water) and power	0.25	0.75	0.67
3	Combined heat (direct) and power	0.25	0.94	0.80
4	Heating (CT without cofired duct burner)	0.25	0.68	0.54
5	Heating (CT with cofired duct burner)	0.25	0.78	0.71

Table 4 Summary of Results Assuming 33% Efficient Combustion Turbine

Example	System	η_E	η_O	ε_{EE}
1	Separate boiler and generator	0.33	0.51	0.33
2	Combined heat (hot water) and power	0.33	0.87	1.00
3	Combined heat (direct) and power	0.33	0.95	1.40
4	Heating (CT without cofired duct burner)	0.33	0.72	0.64
5	Heating (CT with cofired duct burner)	0.33	0.67	0.47

Table 3 shows a summary of the performance metric results of Examples 1 to 5.

Demonstrating the effect of increasing efficiency, Table 4 presents the results from using a combustion turbine (CT) that delivers 33% efficiency. Notice that η_E is, as expected, higher in all examples, and η_O is higher in all but the cofired duct burner case. Cofiring uses exhaust gas heat very effectively, with a 75% recovery rate, accounting for the dominance of thermal recovery from a primary-energy basis and consequently lower η_O associated with the higher η_E system. ε_{EE}, like coefficient of performance (COP), can exceed 1.00, which demonstrates the primary-energy power CHP systems demonstrated in Examples 3 and 4. ε_{EE}, like η_O, exhibits the same pattern in Table 4 as in Table 3.

The implication is that greater use of the thermal energy results in a higher electric effectiveness (Figure 13). All of the example systems, except a low electrically efficient generator with separate boiler, are superior in electrical effectiveness to the delivered efficiency of the electric grid.

Table 5 Typical ψ Values

Electric Generation Source	Generator η LHV	Generator η HHV	T&D Losses*	ψ
EIA average national grid	—	—	—	0.32*
High-efficiency combined cycle combustion turbine (CT)	0.600	0.540	0.050	0.49
Simple-cycle CT	0.380	0.340	0.068	0.27

*Calculated from DOE/EIA (2005).

FUEL ENERGY SAVINGS

Electrical effectiveness provides a reasonable metric for CHP system comparison; however, it alone cannot provide a measure of fuel savings or emissions effects compared to separate, conventional generation of electric and thermal energy requirements. Understanding fuel savings or emissions effects requires further assumptions about the conventional, or reference, systems. Reference baselines for separate thermal heating are as previously developed. The reference baseline for separate electric generation is a conventional power plant's **electric generation efficiency** ψ. Typical values for ψ are listed in Table 5.

Fuel energy savings $\theta_{savings}$ then reflects fuel savings associated with generating the CHP system power and thermal output through CHP Q_{fuel} compared to using separate heating and electric power sources $\mathrm{FUEL}_{reference}$:

$$\mathrm{FUEL}_{reference} = \underbrace{\left\{\frac{W_E}{\psi}\right\}}_{\text{Separate electricity}} + \underbrace{\left\{\frac{\sum Q_{TH}}{\mathrm{HEAT}_\alpha}\right\}}_{\text{Separate thermal heating}}$$

$$\theta_{savings} = \frac{\mathrm{FUEL}_{reference} - Q_{fuel}}{\mathrm{FUEL}_{reference}}$$

Calculations using the system in Example 2 show a projected fuel savings of 29%, based on operation at the system design point:

$$\mathrm{FUEL}_{reference} = \underbrace{\left\{\frac{40}{0.32}\right\}}_{\text{Electricity}} + \underbrace{\left\{\frac{80}{0.80}\right\}}_{\text{Thermal heating}} = 225$$

$$\theta_{savings} = \frac{225 - 160}{225} = 0.289$$

Note the rated design point, to show the utility of the approach in comparing equipment performance on a consistent basis (i.e., for program management and performance metrics). The same methodology could be applied to different design points (e.g., part load, different ambient temperatures) as long as system outputs and fuel inputs are all determined on a consistent basis (e.g., power output, fuel input, thermal output, and recovery all estimated based on 100°F ambient temperature and 1000 ft altitude). Reference system performance should also be considered on the same basis (e.g., it would not be fair to compare CHP electric effectiveness or fuel savings as calculated on a 100°F day to combined cycle efficiencies calculated at ISO conditions). Similarly, the methodology could be applied to actual application performance if system outputs and utilization are considered on a consistent basis (e.g., evaluating actual system power output, fuel input, and thermal energy used over some specified time period).

Fuel savings from the same direct heating and power CHP system as in Example 3.

$$\mathrm{FUEL}_{reference} = \underbrace{\left\{\frac{40}{0.32}\right\}}_{\text{Electricity}} + \underbrace{\left\{\frac{110}{1.0}\right\}}_{\text{Thermal heating}} = 235$$

$$\theta_{savings} = \frac{235-160}{235} = 0.32$$

Applying the same process for the 25% power generator CHP systems (Examples 1 to 5) and using each of the three referenced electric comparisons gives the results presented in Table 6. Table 7 presents results obtained if a 33% generator is assumed.

Table 6 Summary of Fuel Energy Savings for 25% Power Generator in Examples 1 to 5

Example	CHP System	Simple Cycle Peaker	Grid Average	Adv. Combined Cycle
1	Separate boiler and generator	0.00	0.00	0.00
2	Combined heat (hot water) and power	0.35	0.29	0.12
3	Combined heat (direct) and power	0.38	0.32	0.17
4	Heating (CT without cofired duct burner)	0.32	0.24	0.05
5	Heating (CT with cofired duct burner)	0.24	0.19	0.08

Table 7 Summary of Fuel Energy Savings for 33% Power Generator in Examples 1 to 5

Example	CHP System	Simple Cycle Peaker	Grid Average	Adv. Combined Cycle
1	Separate boiler and generator	0.00	0.00	0.00
2	Combined heat (hot water) and power	0.47	0.41	0.26
3	Combined heat (direct) and power	0.50	0.44	0.31
4	Heating (CT without cofired duct burner)	0.41	0.33	0.13
5	Heating (CT with cofired duct burner)	0.23	0.16	(0.01)

FUEL-TO-POWER COMPONENTS

This section describes devices that convert fuel energy to useful power. These energy-conversion technologies provide an important by-product in useful thermal energy that, when harnessed, enables thermal-to-power and thermal-to-thermal devices to operate. These devices are combined with thermal-to-power and/or thermal-to-thermal devices to form CHP systems.

RECIPROCATING ENGINES

Types

Two primary reciprocating engine designs are relevant to stationary power generation applications: the spark ignition (SI) Otto-cycle engine and the compression ignition diesel-cycle engine. The essential mechanical components of Otto-cycle and diesel-cycle engines are the same. Both have cylindrical combustion chambers, in which closely fitting pistons travel the length of the cylinders. The pistons are connected to a crankshaft by connecting rods that transform the linear motion of the pistons into the rotary motion of the crankshaft. Most engines have multiple cylinders that power a single crankshaft.

The primary difference between the Otto and diesel cycles is the method of igniting the fuel. Otto-cycle [or spark-ignition (SI)] engines use a spark plug to ignite the premixed air-fuel mixture after it is introduced into the cylinder. Diesel-cycle engines compress the air introduced into the cylinder, raising its temperature above the auto-ignition temperature of the fuel, which is then injected into the cylinder at high pressure.

Reciprocating engines are further categorized by crankshaft speed (rpm), operating cycle (2- or 4-stroke), and whether turbocharging is used. These engines also are categorized by their original design purpose: automotive, truck, industrial, locomotive, or marine. Engines intended for industrial use are four-stroke Otto-cycle engines, and are designed for durability and for a wide range of mechanical drive and electric power applications. Stationary engine sizes range from 27 to more than 20,000 hp, including industrialized truck engines in the 270 to 800 hp range and industrially applied marine and locomotive engines to more than 20,000 hp. Marine engines (two-stroke diesels) are available in capacities over 100,000 hp.

Both the spark-ignition and the diesel four-stroke engines, most prevalent in stationary power generation applications, complete a power cycle involving the following four piston strokes for each power stroke:

Intake stroke—Piston travels from top dead center (the highest position in the cylinder) to bottom dead center (the lowest position in the cylinder) and draws fresh air into the cylinder during the stroke. The intake valve is kept 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. Fuel combustion 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.

Table 8 Reciprocating Engine Types by Speed (Available Ratings)

Speed Classification	Engine Speed, rpm	Stoichiometric/Rich Burn, Spark Ignition (Natural Gas)	Lean Burn, Spark Ignition (Natural Gas)	Dual Fuel	Diesel
High	1000 to 3600	13 to 2011 hp	201 to 4021 hp	1340 to 4692 hp	13 to 4692 hp
Medium	275 to 1000	None	1340 to 21,448 hp	1340 to 33,512 hp	670 to 46,917 hp
Low	60 to 275	None	None	2681 to 87,131 hp	2681 to 107,239 hp

Source: Adapted from EPA (2002).

The range of medium- and high-speed industrial engines and low-speed marine diesel reciprocating engines are listed in Table 8 by type, fuel, and speed.

Performance Characteristics

Important performance characteristics of an engine include 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 CHP system is most cost-effective when operating at its base load, the rating at prime power (i.e., when the engine is the primary source of power) is usually of greatest interest. This rating is based on providing extended operating life with minimum maintenance. When used for standby, the engine produces continuously (24 h/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, which 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, whereas turbocharged engines lose 2% per 1000 ft. Output decreases 1% per 10°F increase in ambient temperature, so it is important to avoid using heated air for combustion. In addition, an engine must be derated for fuels with a heating value significantly greater than the base specified by the manufacturer. For CHP applications, natural gas is the baseline fuel, although use of propane, landfill gas, digester gas, and biomass is increasing.

Natural gas spark-ignition engines are typically less efficient than diesel engines because of their lower compression ratios. However, large, high-performance lean-burn engine efficiencies approach those of diesel engines of the same size. Natural gas engine efficiencies range from about 25% HHV (28% LHV) for engines smaller than 50 kW, to 37% HHV (41% LHV) for larger, high-performance, lean-burn engines (DOE 2003).

Power rating is determined by a number of engine design characteristics, the most important of which is displacement; other factors include 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 14 illustrates the efficiency of typical SI natural gas engine/shaft power operating at the prime power rating (HHV).

Fuel consumption is the greatest contributor to operating cost and should be carefully considered during planning and design of a CHP 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 rate is the heat input per unit of power output based on either the low or high heating value of the fuel.

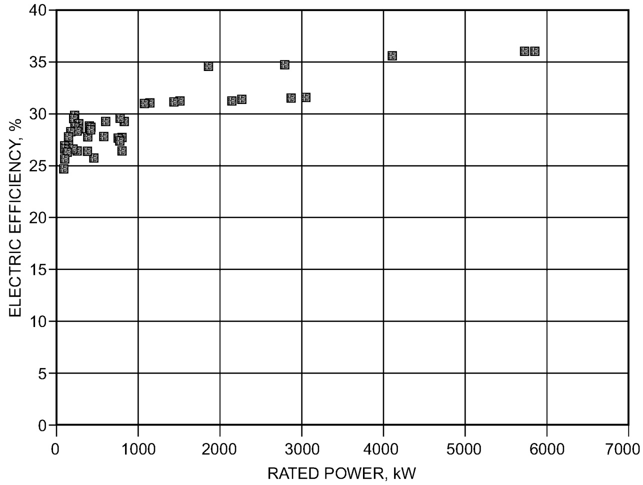

Fig. 14 Efficiency (HHV) of Spark Ignition Engines

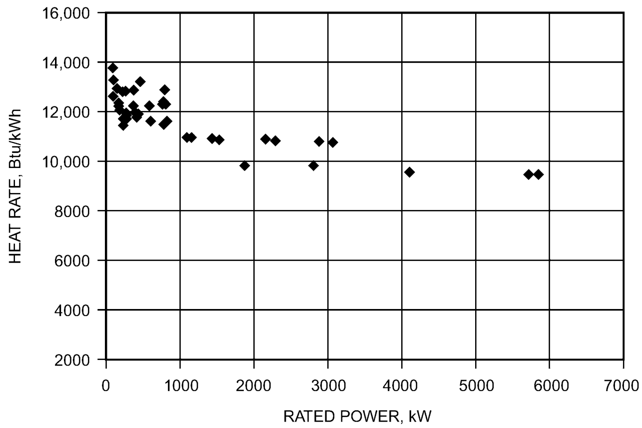

Fig. 15 Heat Rate (HHV) of Spark Ignition Engines

Heat rates of several SI engines are shown in Figure 15. The 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 because of 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 16 shows the thermal-to-electric ratio of the SI engines.

Ideally, a CHP 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. Figures 17 and 18 show the part-load heat rate and thermal-to-electric ratio as a function of load for 1430, 425, and 85 kW engines.

Fuels and Fuel Systems

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).

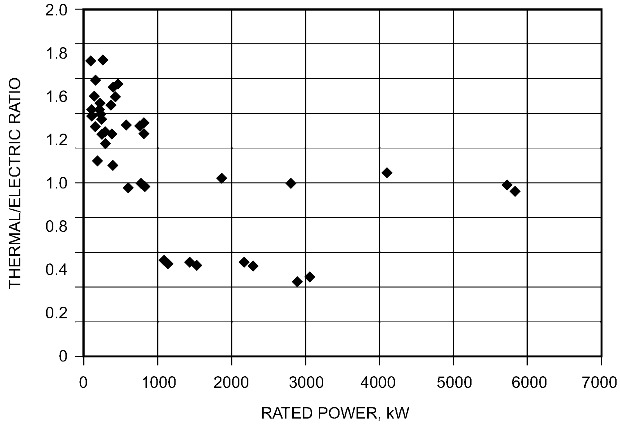

Fig. 16 Thermal-to-Electric Ratio of Spark Ignition Engines (Jacket and Exhaust Energy)

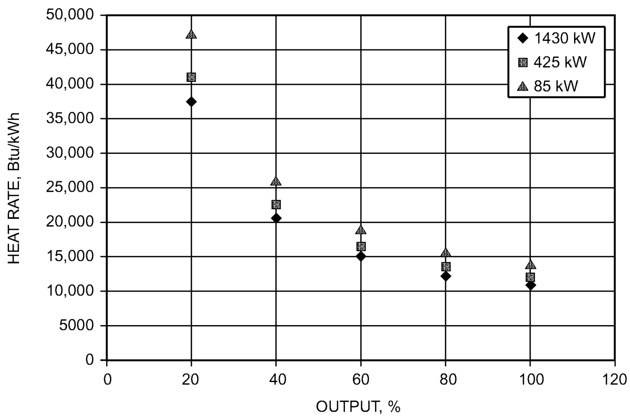

Fig. 17 Part-Load Heat Rate (HHV) of 1430, 425, and 85 kW Gas Engines

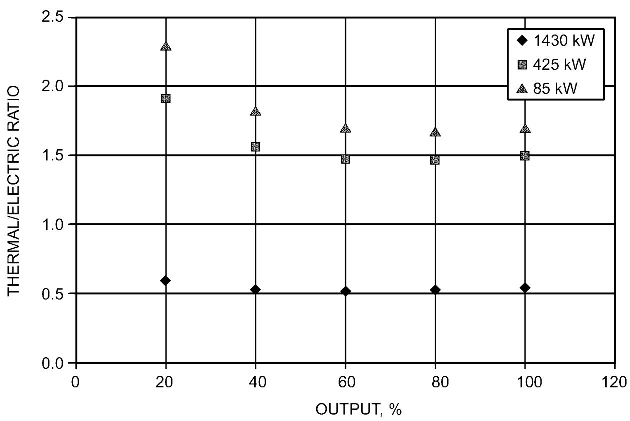

Fig. 18 Part-Load Thermal-to-Electric Ratio of 1430, 425, and 85 kW Gas Engines

Engines may be fueled with a wide variety of fuels and fuel blends, including natural gas, propane, landfill gas, digester gas, bio-oils, biodisel, diesel, or heavier oils. A small amount of diesel oil is used as the compression ignition agent when dual-fuel engines are operated in gaseous-fuel mode.

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

Methane-rich gas from wastewater treatment or anaerobic digesters can be used as a fuel for both engines and other heating services. The fuel must be dried and cleaned before injection into engines. Because methane-rich gas has a lower heat content (approximately 600 to 700 Btu/ft^3), it is sometimes mixed with natural gas. The large amount of hydrogen sulfide in the fuel requires using 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. The cost of natural gas is based on HHV. Most natural gases have an LHV/HHV factor of 0.9 to 0.95, ranging 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).

Fuel Oil Systems. Storage, handling, and cleaning of liquid fuels are covered in Chapter 30. Most oil-fueled CHP 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.

Atomization can be by 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 used:

- 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 (excess fuel is recirculated back to the tank through a cooler to remove heat)

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 wastewater or biomass, which may also require scrubbers and holding tanks. Gas system accessories include the following.

Line-Type Gas Pressure Regulators. Turbocharged (and aftercooled) engines, as well as many naturally aspirated units, are equipped with line regulators designed to control gas pressure to the engine regulator, as shown in Table 9. The same regulators (both line

Table 9 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 variation for individual installations.
[b]Also inlet to engine regulator.
[c]Turbocharger boost plus 7 to 10 in. of water.

and engine) used on naturally aspirated gas engines may be used on turbocharged equipment.

Line-type gas pressure regulators are commonly called service 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 allow 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 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.

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

Fuel Injectors. These electromechanical devices, which are essentially solenoids through which fuel is metered, spray atomized fuel directly into the combustion chamber. Electric current applied to the injector coil creates a magnetic field, which causes the armature to move upward and unseats a spring-loaded ball or pintle valve. Pressurized fuel can then flow out of the injector nozzle in a cone-shaped pattern (caused by the pintle valve's shape). When the injector is deenergized, the ball or valve reseats itself, stopping fuel flow.

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 used.

Dual-Fuel. Engines using gas with pilot oil for ignition (but that can also operate on 100% diesel) are commonly classified as *dual-fuel engines*. Dual-fuel engines operate either on full oil or on gas and pilot oil, with automatic online switchover when appropriate. 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.

Combustion Air

All internal combustion engines require clean, cool air for optimum performance. High humidity does not hurt performance, and may even help by slowing combustion and reducing cylinder pressure and temperature. Provisions must be made to silence air noise and provide adequate air for combustion.

Smaller engines generally use engine-mounted impingement filters, often designed for some silencing, whereas 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 common in HVAC systems are used, but they are designed specifically for engine use. Air piping is designed for low pressure drop (more important for naturally aspirated than for supercharged engines) to maintain high engine performance. 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.

Essentially, all modern industrial engines above 300 kW are turbocharged to achieve higher power densities. A turbocharger is basically a turbine-driven intake air compressor; hot, high-velocity exhaust gases leaving the engine cylinders power the turbine. Very large engines typically are equipped with two large or four small turbochargers. On a carbureted engine, turbocharging forces more air and fuel into the cylinders, increasing engine output. On a fuel-injected engine, the mass of fuel injected must be increased in proportion to the increased air input. Turbocharging normally increases cylinder pressure and temperature, increasing the tendency for detonation for both spark ignition and dual-fuel engines and requiring a careful balance between compression ratio and turbocharger boost level. Turbochargers normally boost inlet air pressure by a factor of 3 to 4. A wide range of turbocharger designs and models is used. Aftercoolers or intercoolers are often used to cool combustion air exiting the turbocharger compressor, to keep the temperature of air to the engine under a specified limit and to increase the air density. A recirculated water coolant recovery system is used for aftercooling.

The following factors apply to combustion air requirements:

- 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 engine air intake temperature, both in °R.
- 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 maximum total static pressure opposing the fan should be 0.35 in. of water. Table 10 gives sample ventilation air requirements.
- On large engines, intake air is taken from outside the building, enabling a reduction in the building HVAC system. An intake silencer is typically used to eliminate engine noise.

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 engine heat with the lubricating system. The engine's operating temperature may be significant

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.

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 allow 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 oil and jacket coolant is not significant. If oil temperature rises in one area (such as around the piston skirts), heat may be transferred to other engine oil passages and then removed by the jacket coolant. When engine jacket temperatures are much higher than 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. Consider the following factors in selecting an engine:

- High-quality lubricating oils are generally required for operation 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.
- Contact with copper can cause oil breakdown, so copper piping should be avoided in oil-side surfaces in oil coolers and heat exchangers.
- 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.

Starting 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 multiple compressors is usually engine-driven for a “black start.” The same procedure is used for fuel oil systems when the main storage tank cannot gravity-feed the day tank. However, 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 separate, smaller 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 repairs at the plant and at other buildings in the complex.

Cooling Systems

Jacket Water Systems. Circulating water and oil systems must be kept clean because the internal coolant passages of the engine are not readily accessible for service. Installation of piping, heat exchangers, valves, and accessories must include provisions for internally cleaning 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 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 benefits 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.

Size water piping according to the engine manufacturer’s recommendations, avoiding restrictions in the water pump inlet line. Piping must not be connected rigidly to the engine. Provide shutoffs to facilitate maintenance.

If the cost of pumping 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 39 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. This independence can be a disadvantage because unused engines are not maintained at operating temperature, as they are when all units are in a common circulating system. If idle machine temperature drops below combustion products’ dew point, 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 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 exposure to corrosion.

Still another concept for avoiding a total plant shutdown, and minimizing the risk 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 (e.g., pumps, heat exchangers). Valves must be installed to isolate any auxiliary that fails or is out for preventive maintenance, to allow continued operation.
- Header isolation valves to allow 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, using common headers 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 allows a warm start of any engine by circulation of a moderate flow of the hot jacket water through any idle engine.

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, range 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 exhaust piping. Exhaust gas temperature does not normally exceed 1200°F, but may reach 1400°F for short periods. An appropriate stainless steel connector may be used.
- Adequately support the exhaust system downstream from the connector. At maximum operating temperature, no weight should be exerted on the engine or its exhaust outlet.
- Minimize the distance between the silencer and engine.
- Use a 30 to 45° tailpipe angle to reduce turbulence.
- Specify tailpipe 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 (°R), and P is exhaust frequency (pulses per second). The value of P is calculated as follows:

$$P = \text{rpm}/120 = (\text{rev/s})/2 \text{ for four-stroke engines}$$

$$P = \text{rpm}/60 = \text{rev/s} \text{ for two-stroke engines}$$

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, because pulses of one branch will interfere with pulses from the other.
- Use an engine-to-silencer pipe length that is 25% of the tailpipe length.
- Install a separate exhaust for each engine to reduce the possibility of condensation in an engine that is not running.
- Install individual silencers to reduce condensation resulting 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 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 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 (over 2500 fpm) through a straight stack; in intermittently operated systems, protection can be maintained by installing drain-type stacks.

Drain-Type Stack. Drain-type stacks effectively eliminate rainfall entry into a vertical stack terminal without destroying the upward ejection velocity as a rain cap does. 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 (see Chapter 44 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on exhaust stack design). Bolts for inner stack fastening should be soldered, welded, or brazed, depending on the tack material.

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 and an annular clearance of 4 to 5 in. is used. Sufficient clearance is required between the flue terminal and rain cap on the pipe to allow flue venting. 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 engine, and insulate inside the muffler portion of the flue. Flue runs more than 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 (using 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 11 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.

Table 11 Exhaust Pipe Diameter*

Output Power, hp	Minimum Pipe Diameter, in. Equivalent Length of Exhaust Pipe 25 ft	50 ft	75 ft	100 ft
25	3	4	4	4
50	4	4	5	5
75	4	5	5	5
100	5	5	6	6
200	6	6	7	7
400	7	8	9	9
600	9	9	10	11
800	11	11	11	12
1000	12	12	12	13
1500	13	13	15	15
2000	15	16	17	17

*Minimum exhaust pipe diameter to limit engine exhaust back pressure to 8 in. of water.

Emissions

Exhaust emissions are the major environmental concern with reciprocating engines. The main pollutants are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs; unburned or partially burned nonmethane hydrocarbons). Other pollutants, such as oxides of sulfur (SO_x) and particulate matter (PM), depend on the fuel used. Emissions of sulfur compounds (particularly SO_2) are directly related to the fuel's sulfur content. Engines operating on natural gas or distillate oil, which has been desulfurized in the refinery, emit insignificant levels of SO_x. In general, SO_x emissions are an issue only in larger, lower-speed diesel engines firing heavy oils.

Particulate matter can be important for liquid-fueled engines. Ash and metallic additives in fuel and lubricating oil contribute to PM concentrations in exhaust.

NO_x emissions, usually the major concern with natural gas engines, are mostly a mixture of NO and NO_2. Measurements of NO_x are reported as parts per million by volume, in which both species count equally (e.g., ppmv at 15% O_2, dry). Other common units for reporting NO_x in reciprocating engines are specific output-based emission factors, such as g/hp·h and g/kW·h, or as total output rates, such as lb/h. Among the engine options without exhaust aftertreatment, lean-burn natural gas engines produce the lowest NO_x emissions; diesel-fueled engines produce the highest. In many localities, emissions pollutant reduction is mandatory. Three-way catalytic reduction is the general method for stoichiometric and rich-burn engines, and selective catalytic reduction (SCR) for lean-burn engines.

Instruments and Controls

Starting Systems. Start/stop control may include manual or automatic activation of the engine fuel supply, 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, allows 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 disables the governor. 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 caused by 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

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.

Because engine exhaust must be muffled to reduce ambient noise levels, most recovery units also act as silencers. Figure 19 illustrates a typical exhaust noise curve. Figure 20 shows typical attenuation curves for various silencers. Table 42 in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* lists acceptable noise level criteria for various applications.

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 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 (often a concrete block enclosure) between the prime mover and the listener; (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 doors.

Noise levels must meet legal requirements (see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC 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

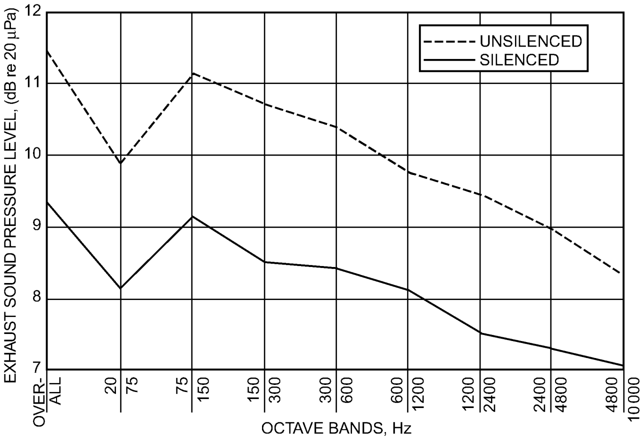

Fig. 19 Typical Reciprocating Engine Exhaust Noise Curves

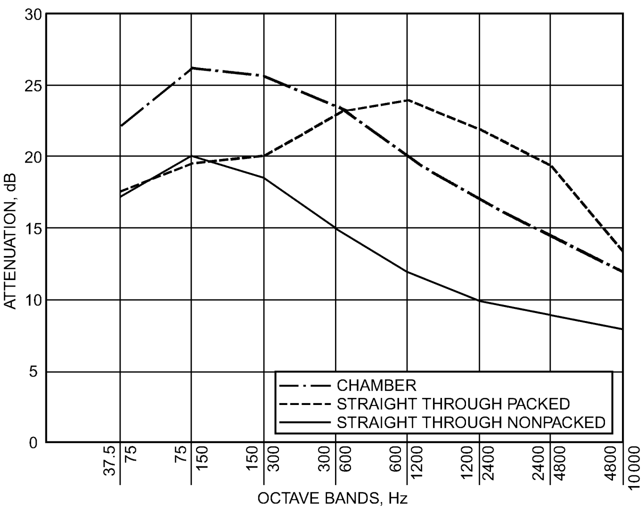

Fig. 20 Typical Attenuation Curves for Engine Silencers

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, if 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 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 vibration transmission, an engine base or foundation should never rest directly on natural rock formations. Under some conditions, such as shifting soil on which an outlying CHP 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 (see Chapters 47 and 54 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on vibration isolation and seismic design, respectively).

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 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 in the assembly. Encotech (1992) describes how changing the coupling's mass, stiffness, or damping 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 45 for further information on piping.)

Installation Ventilation Requirements

In addition to dissipating heat from the jacket water system, exhaust system, lubrication and piston cooling oil, turbocharger, and air intercooler, 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, 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 electrical systems with high ambient conditions. Heat can be removed by outside 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 vary 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 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 back-up systems located in unheated spaces.
- Positioning the engine so that the radiator face 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.
- Separation of units in a multiple-unit 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.

Sufficient ventilation must also be provided to protect against minor fuel supply leaks (not rupture of the supply line). Table 12's minimum ventilation air requirements may be used. Ventilation may be provided by a fan that induces the draft through a sleeve

Table 12 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.

surrounding the exhaust pipe. Slightly 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 short-circuiting of cool air directly to the exhaust while bypassing equipment; and (6) arranging equipment locations, when possible, to allow the desired 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.
- Allow the silencer and/or recovery device's hot surfaces to warm 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 dampers with a thermostat, which is reset by room temperature, at the inlet to the machine room.
- Cool hot combustion air with a cooling device downstream of the air filter to increase engine performance. This is particularly helpful with large, slow-speed, naturally aspirated engines. The Diesel Engine Manufacturers Association (DEMA 1972) 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.

Operation and Maintenance

Preventive Maintenance. One of the most important provisions for healthy and continuous plant operation is implementing 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. Keep an inventory of critical parts on site. (See Chapters 36 to 43 of the 2007 *ASHRAE Handbook—HVAC 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. Factors such 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 allow periods between procedures to be longer, catches incipient problems before they create outages or major repairs, and avoids unnecessary maintenance.

Table 13 Recommended Engine Maintenance

Procedure	Hours Between Procedures	
	Diesel Fueled Engine	Gaseous Fueled 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

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 undergoes 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 used for several hours in any week.

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 13 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 air, cleanliness of the engine room, engine manufacturer's recommendations, number of engine starts and stops, and lubricating conditions indicated by oil analysis. With some engines and some operating conditions, the intervals between procedures listed in Table 13 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 engine's physical condition 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 help interpret 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 38 to 644,000 hp and can burn a wide range of liquid and gaseous fuels. Turbines, and some dual-fuel engines, can shift from one fuel to another without loss of service. **Microturbine** systems typically have capacities below 670 hp, although this is a subjective cutoff between micro- and **miniturbine** (671 hp to about 1340 hp).

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. Simple-cycle combustion gas turbines have thermal efficiency levels of 25 to 32% HHV (28 to 36% LHV). Recuperative combustion gas turbines have thermal efficiency levels of 35% HHV (39% LHV).

In addition to these components, some turbines use regenerators and recuperators as heat exchangers to preheat combustion air with heat from the turbine discharge gas, thereby increasing machine efficiency. Most microturbine units are currently designed for continuous-duty operation and are recuperated to obtain higher electric efficiencies levels of 23 to 30% HHV (26 to 33% LHV) for sizes 335 hp and below. Unrecuperated engines or simple cycles have lower electric efficiencies but higher exhaust temperatures, which make them better suited for some CHP applications. Most turbines are the single-shaft type, (i.e., air compressor, turbine, and generator on a common shaft). However, dual-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 100,000 rpm and often need speed-reduction gearboxes to obtain shaft speeds suitable for generators or other machinery. Many microturbine designs directly couple a high-speed turbine to a wide-frequency alternator whose high-frequency ac power is converted to dc by a rectifier and then to 60 Hz ac by an inverter to utility voltage and frequency specifications. 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 in space, foundation requirements, and ease of start-up.

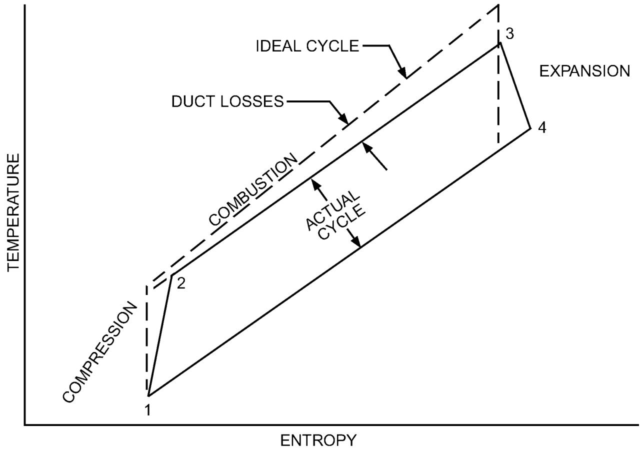

Fig. 21 Temperature-Entropy Diagram for Brayton Cycle

Combustion turbines have the following advantages and disadvantages compared to other internal combustion engine drivers:

Advantages

- 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
- No cooling water required
- Vibration-free operation
- Easy maintenance
- Low installation cost
- Clean, dry exhaust
- Lubricating oil not contaminated by combustion oil

Disadvantages

- Tend to be more complicated devices
- Lower electrical output efficiency compared to internal combustion engines
- Higher capital (first) costs

Gas Turbine Cycle

The basic gas turbine cycle (Figure 21) is the **Brayton cycle (open cycle)**, which consists of adiabatic compression, constant-pressure heating, and adiabatic expansion. Figure 21 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. Entropy increases 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 (ASME 2005).

Components

Figure 22 shows the major components of the gas turbine unit, which include the air compressor, combustor, and power turbine. Atmospheric air is compressed by the air compressor. Fuel is then injected into the airstream and ignited in the combustor, with leaving gases reaching temperatures between 1600 and 2500°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 22) and dual-shaft (Figure 23). The **single-shaft turbine** has the air compressor, gas-producer turbine, and power turbine on the same shaft. The **dual-shaft** or **split-shaft turbine** has the section required for air compression on one shaft and the section producing output power on a separate shaft. For a dual-shaft turbine, the portion that includes the compressor, combustion 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. Dual-shaft engines allow for variable speed at full load and 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.

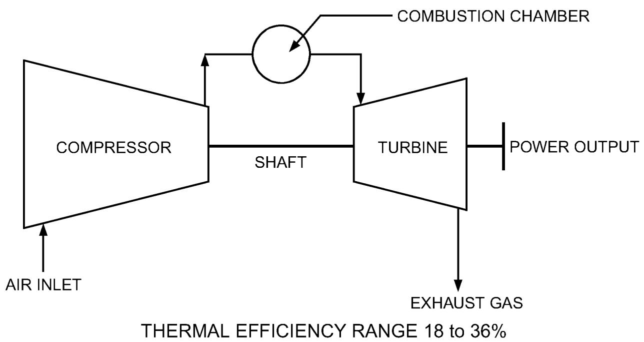

Fig. 22 Simple-Cycle Single-Shaft Turbine

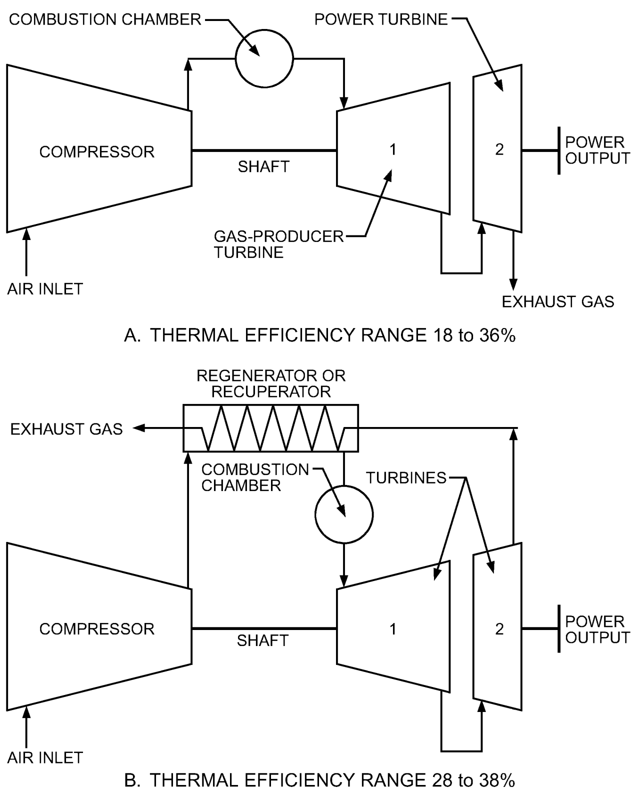

Fig. 23 Simple-Cycle Dual-Shaft Turbines

PERFORMANCE CHARACTERISTICS

A gas turbine's rating 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 recovery units, 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 air compressor's inlet flange and turbine's exhaust flange. Corrections for other conditions must be obtained from the manufacturer, because 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 considerations:

- Each 18°F rise in inlet temperature typically 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 waste heat recovery units, 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.

Figure 24 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 utilization efficiency can increase.

Figure 23B shows a regenerator that uses exhaust gas heat to heat air from the compressor before combustion. Overall shaft efficiency can be increased from 25 to 36% HHV (28 to 39% LHV) by using a regenerator or recuperator.

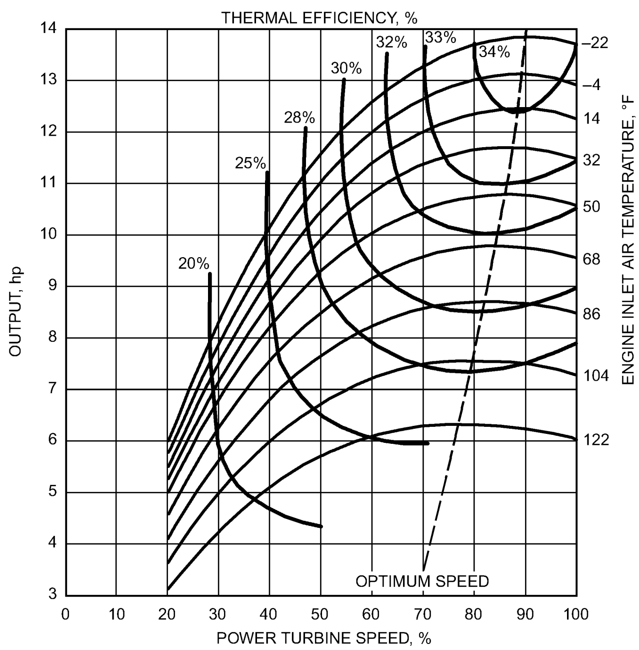

Fig. 24 Turbine Engine Performance Characteristics

If process heat is required, turbine exhaust can satisfy some of that heat, and the combined system is a CHP system. The exhaust can be used (1) directly as a source of hot air, to drive an exhaust-fired absorption chiller, regenerate desiccant materials in a direct-fired dehumidifier, activate a thermal enthalpy wheel, or operate a bottoming cycle (e.g., Rankine) to generate electric or shaft power; (2) in a large boiler or furnace as a source of preheated combustion air or cofiring an absorption chiller (exhaust typically contains 16% or higher oxygen levels); or (3) to heat a process or working fluid such as the steam system shown in Figure 25. Overall thermal efficiency around the CHP system boundary (Figure 26) is [(net electrical energy output + useful thermal energy output) × 100]/(energy input from fuel). Thermal efficiencies of these systems vary from 50% to greater than 70%. The exhaust of a gas turbine has about 4000 to 8000 Btu/h of available heat energy per horsepower output.

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

Fuels and Fuel Systems

The ability to burn almost any combustible fluid is a key advantage of the gas turbine. Natural gas is preferred 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. Low-heat-value fuels from waste gas sites such as landfills or wastewater treatment plants are beginning to be used by gas turbines, although significant fuel conditioning is often needed to remove moisture and contaminants such as siloxane.

Distillate oil is the most common liquid fuel and, except for a few installations where natural gas is not available, is primarily used as a back-up and alternative start-up fuel. Crude oils are common 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 can transfer from one fuel to the other any time after completing the starting sequence without interrupting operation.

Steam and demineralized water injection are sometimes 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 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.

Combustion Air

Combustion turbine inlet cooling (CTIC) systems increase the capacity of turbine-generators by increasing combustion air density. Because volumetric flow to most turbines is constant, increasing air density increases mass flow rate. As inlet air temperature increases, (e.g., on hot summer days), capacity decreases (MacCracken 1994).

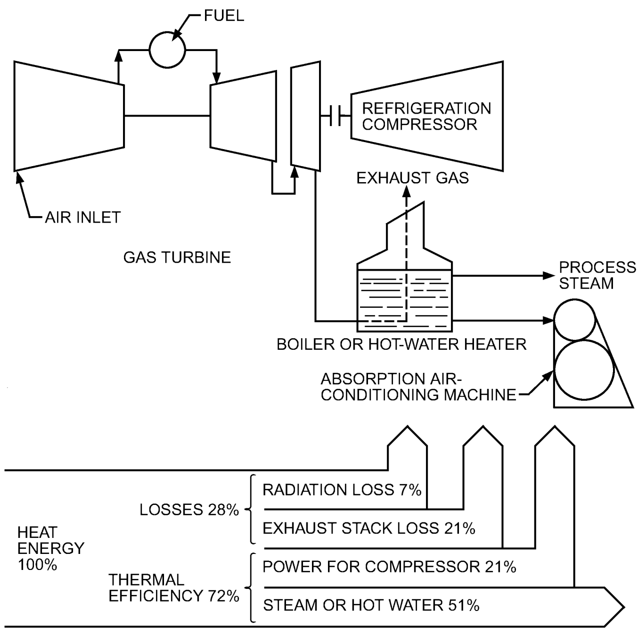

Fig. 25 Gas Turbine Refrigeration System Using Exhaust Heat

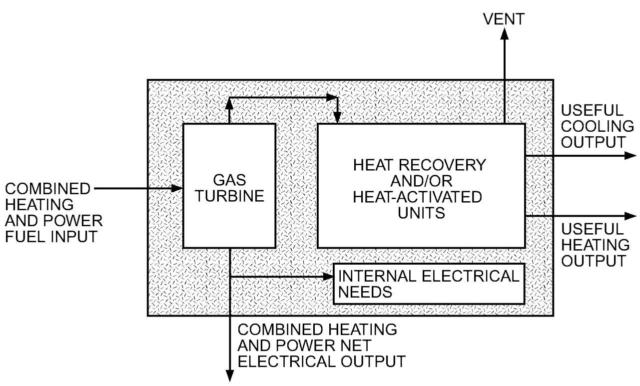

Fig. 26 CHP System Boundary

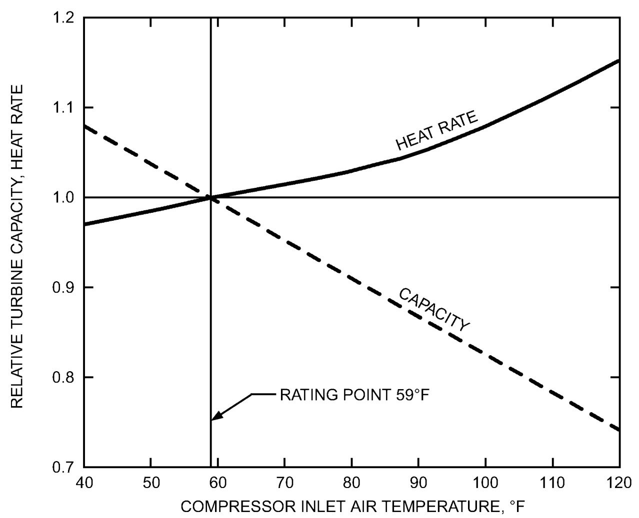

Fig. 27 Relative Turbine Power Output and Heat Rate Versus Inlet Air Temperature

Cooling inlet air increases the power and typically decreases the heat rate, after all parasitic cooling power usage is considered (Figure 27). Factors that affect CTIC installation and operation include 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. (More information is available in Chapter 17 and from sources such as the Turbine Inlet Cooling Association.)

Cooling Methods. Some CTIC designs are for turbines that operate only a few hours per year, to demonstrate power reserve or provide peak demand power. **Peaking** combustion turbines operate when utilities experience the greatest demand. Both inlet evaporative cooling systems (wetted media and fogging) and thermal energy storage (TES) systems allow CTIC 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 **base load** and need 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 (Brown and Somasundaram 1997).

The most prevalent CTIC system is **evaporative cooling** using wetted media, because of low installation and operating costs. Ideal evaporative cooling occurs at a constant wet-bulb temperature, cooling the air to near 100% rh. 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 coils) and evaporative cooling is used, sensible cooling should be used before 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, chilled-water, or low-temperature fluid TES (Andrepont 2001; Ebeling 1994). From a cost standpoint, the TES system is usually justified for turbines that operate relatively few hours per day or to increase reserve power. In systems designed for long periods of cooling per day, 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, required pumping, and 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 can reduce refrigeration equipment size and on-peak parasitic energy usage, which can decrease overall system capital costs. Parasitic loads for a TES system that operates only a few hours per day usually do not severely affect the economic value of a CTIC 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 peak cooling; therefore, it is larger than a TES system (ASHRAE 1997).

For base-load combined-cycle power plants under continuous, varying loads, a direct online chilled-water cooling system using low-pressure steam from back-pressure steam turbines for absorption chillers is an economical option during peak load operation. The incremental cost of low-pressure steam is very low because the original high-pressure steam has done major work already and only a small part of the steam enthalpy remains for use in absorption chillers.

Because all base-load plants operate under part-load conditions for many hours during the day, the CTIC cooling coil increases heat rate because of pressure drop, raising specific fuel consumption for the whole plant. This has discouraged use of CTIC for combined-cycle power plants, especially plants operating under low part-load conditions for extended periods.

One solution is to heat combustion inlet air (a patented process), using the same coil, piping, and pumps, to raise combustion turbine exhaust air temperature and reduce mass flow, for better waste heat recovery and thus higher CHP efficiency. This technique also either avoids the need for variable air inlet guide vanes, thus reducing pressure drop and heat rate, or complements them under very low part-loads. This combustion turbine inlet air conditioning can provide cooling during peak-load operation and heating during part-load operation, save considerable fuel costs, and reduce pollution because of higher CHP efficiencies (Abedin 2003).

Advantages of CTIC.

Capacity Enhancement. Using CTIC 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. CTIC 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 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 CTIC 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. The increased generation capacity provided by a CTIC system can delay addition of actual or reserve generation capacity.

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 CTIC 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.
- CTIC systems that reduce 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.
- CTIC systems are simple, energized only when required.
- Emissions can decrease because of increased overall efficiency.
- A CTIC system can match inlet air temperature to required turbine generating capacity, allowing 100% open inlet guide vanes, which eliminate inlet guide vane pressure loss penalties.

Disadvantages.

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

Lubricating Systems

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

The start-up/coast-down oil pump circulates oil until the gas turbine reaches a speed at which the primary pump can take over. If 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 pumps' full flow. Two filters are sometimes 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 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- or air-cooled, depending on cooling water availability.

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 motor directly to the gas engine for starting. All direct-coupled 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. To perform black starts, include some form of external energy storage or drive system in the starting system.

Exhaust Systems

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

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 (ASME 1994). 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.

Instruments and Controls

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

When operating the gas turbine engine at its maximum rating is desirable, the load is controlled based on the temperature of 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 ambient air temperature, the control system adjusts the load to maintain the set temperature value in the gas engine's turbine section. When 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.

Noise and Vibration

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, inlet silencer, and 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.

Operation and Maintenance

Industrial gas turbines are designed to operate for 12,000 to 40,000 h between overhauls, with normal maintenance, which includes checking filters and oil level, inspecting for leaks, etc., 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 (predictive maintenance) rather than by specific overhaul intervals. 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.

Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on operation and maintenance management.

FUEL CELLS

Types

Fuel cells convert 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. 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 reformer (processor), which generates hydrogen-rich gas from fuel; a power section (stack) where the electrochemical process occurs; and a power conditioner (inverter), which converts the dc power generated in the fuel cell into ac power. Most fuel cell applications involve interconnectivity with the electric grid; thus, the power conditioner must synchronize the fuel cell's electrical output with the grid (ASHRAE 2001; ASME *Standard* PTC 50). A growing number of fuel cell applications are grid independent to reliably power remote or critical systems.

Table 14 Overview of Fuel Cell Characteristics

	Phosphoric Acid (PAFC)	Solid Oxide (SOFC)	Molten Carbonate (MCFC)	Proton Exchange Membrane (PEMFC)
Commercially available	Yes	No	Yes	Yes
Size range	100 to 200 kW	1 kW to 10 MW	250 kW to 10 MW	500 W to 250 kW
Efficiency (LHV)	40%	45 to 60%	45 to 55%	30 to 40%
Efficiency (HHV)	36%	40 to 54%	40 to 50%	27 to 36%
Average operating temperature	400°F	1800°F	1200°F	200°F
Heat recovery characteristics	Hot water	Hot water/steam	Hot water, steam	140°F water

Source: Adapted from Foley and Sweetser (2002).

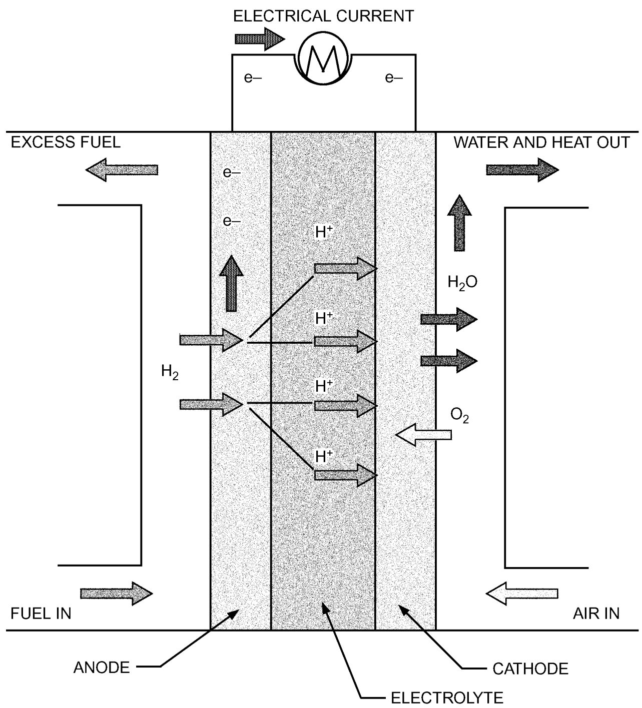

Fig. 28 PAFC Cell
(DOE 2007)

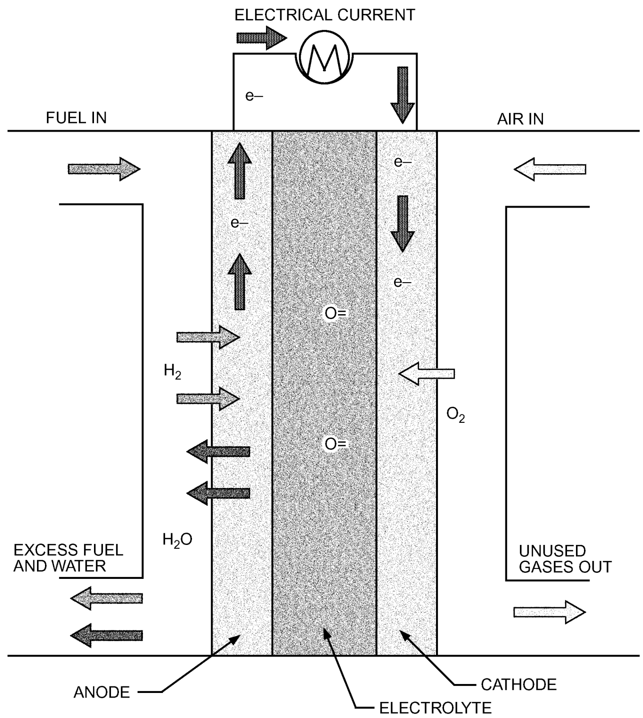

Fig. 29 SOFC Cell
(DOE 2007)

Most fuel cells have similar designs, but differ in the type of electrolyte used. The main types of fuel cells, classified by their electrodes, are (1) phosphoric acid (PAFC), (2) molten carbonate (MCFC), (3) solid oxide (SOFC), (4) proton exchange membrane (PEMFC), and (5) alkaline (AFC). PAFCs, MCFCs, and PEMFCs are commercially available, AFCs are used for space power, and SOFCs are in development and testing. The most significant research and development activities focus on PEMFC for automotive and home use and SOFC for stationary applications. Efficiencies of several types of fuel cells are shown in Table 14 (Foley and Sweetser 2002). Emissions from fuel cells are very low; NO_x emissions are less than 20 ppm. Large phosphoric acid fuel cells are commercially available.

Phosphoric Acid Fuel Cells (PAFCs). PAFCs are generally considered first-generation technology (Figure 28). They operate at about 400°F and achieve 40% LHV efficiencies. PAFCs use liquid phosphoric acid as an electrolyte. Platinum-catalyzed, porous-carbon electrodes are used for both the cathode and anode. For each type of fuel cell, the reformer supplies hydrogen gas to the anode through a process in which hydrocarbons, water, and oxygen react to produce hydrogen, carbon dioxide, and carbon monoxide. At the anode, hydrogen is split into two hydrogen ions (H^+) and two electrons. The ions pass through the electrolyte to the cathode, and the electrons pass through the external circuit to the cathode. At the cathode, the hydrogen, electrons, and oxygen combine to form water.

Solid-Oxide Fuel Cells (SOFCs). SOFCs operate at temperatures up to 1800°F, offering enhanced heat recovery performance (Figure 29). A solid-oxide system typically uses a hard ceramic material instead of a liquid electrolyte. The solid-state ceramic construction is a more stable and reliable design, enabling high temperatures and more flexibility in fuel choice. SOFCs can reach 54% HHV (60% LHV) efficiencies. Combined-cycle applications could reach system efficiencies up to 77% HHV (85% LHV).

SOFCs can use carbon monoxide as well as hydrogen as direct fuel. Hydrogen and carbon monoxide in the fuel stream react with oxide ions from the electrolyte, producing water and carbon dioxide, and releasing electrons into the anode. The electrons pass outside the fuel cell, through the load, and back to the cathode. At the cathode, oxygen molecules from the air receive the electrons and the molecules are converted into oxide ions. These ions are injected back into the electrolyte.

Molten-Carbonate Fuel Cells (MCFCs). MCFCs can reach 50% HHV (55% LHV) efficiencies (Figure 30). Combined-cycle applications could reach system thermal efficiencies of 77% HHV (85% LHV). MCFCs operate on hydrogen, carbon monoxide, natural gas, propane, landfill gas, marine diesel, and simulated coal gasification products. Operating temperatures are around 1200°F.

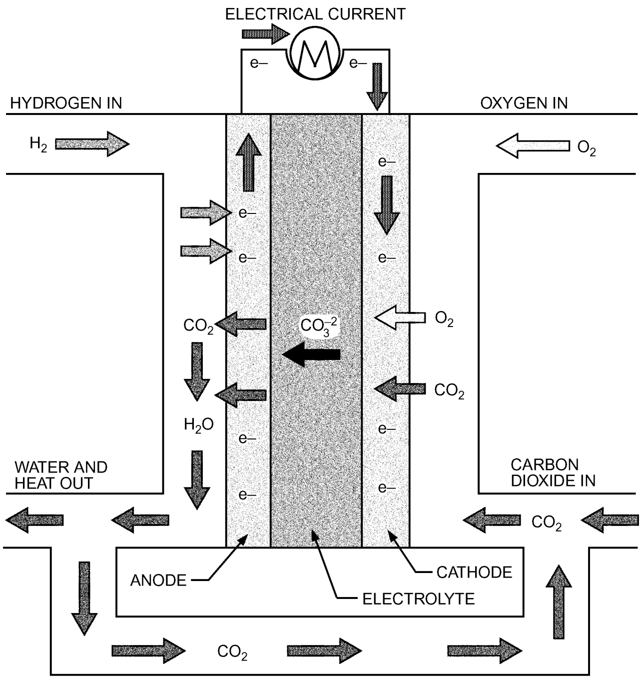

Fig. 30 MCFC Cell
(DOE 2007)

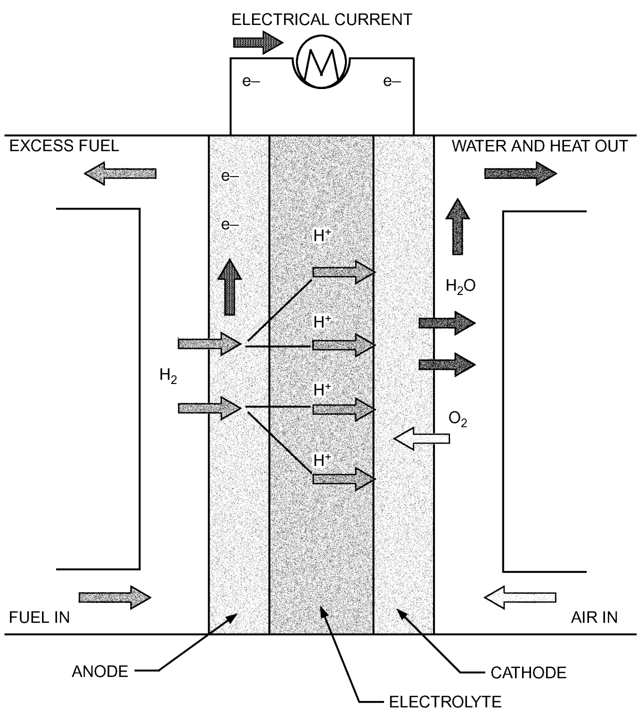

Fig. 31 PEMFC Cell
(DOE 2007)

This high operating temperature makes direct operation on gaseous hydrocarbon fuels (e.g., natural gas) possible. Natural gas can be reformed internally in MCFCs to produce hydrogen.

MCFCs use a molten carbonate salt mixture as an electrolyte; the electrolyte composition varies, but usually consists of lithium carbonate and potassium carbonate. The salt mixture is liquid and a good ionic conductor at the MCFC's high operating temperature. An electrochemical reaction occurs at the anode between the hydrogen

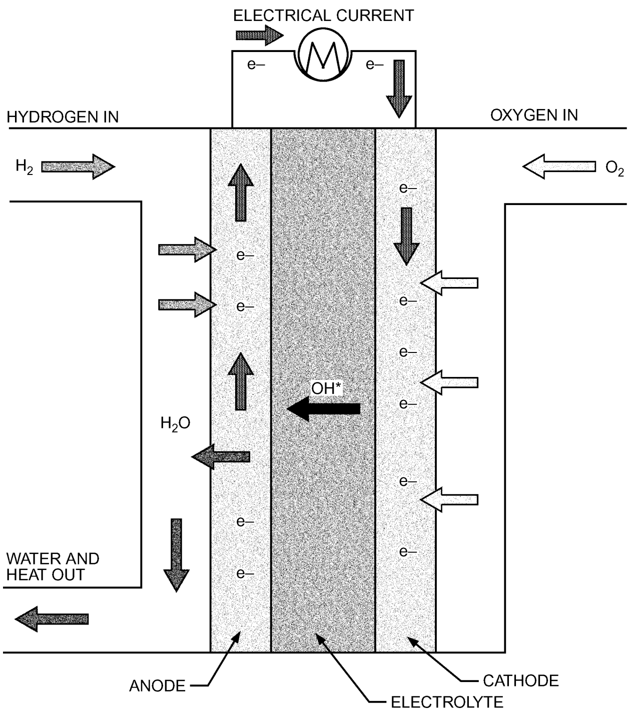

Fig. 32 AFC Cell
(DOE 2007)

fuel and carbonate ions from the electrolyte. This reaction produces water and carbon dioxide, and releases electrons to the anode. At the cathode, oxygen and carbon dioxide from the oxidant stream combine with electrons from the anode to produce carbonate ions, which enter the electrolyte.

Proton Exchange Membrane Fuel Cells (PEMFCs). PEMFCs can reach 36% HHV (40% LHV) efficiencies. PEMFCs contain a thin plastic polymer membrane through which hydrogen ions can pass. The membrane is coated on both sides with highly dispersed metal alloy particles (mostly platinum) that are active catalysts (Figure 31). Because the electrolyte is a solid polymer, electrolyte loss does not affect stack life. Using a solid electrolyte eliminates the safety concerns and corrosive effects associated with liquid electrolytes. PEMFCs operate at relatively low temperatures (approximately 200°F).

Electrode reactions in the PEMFC are analogous to those in the PAFC. Hydrogen ions and electrons are produced from the fuel gas at the anode. At the cathode, oxygen combines with electrons from the anode and hydrogen ions from the electrolyte to produce water. The solid electrolyte does not absorb the water, which is rejected from the back of the cathode into the oxidant gas stream (Hodge and Hardy 2002).

Hydrogen is delivered to the anode side of the membrane-electrode assembly (MEA), where it is catalytically split into protons and electrons. The protons move through the polymer electrolyte membrane to the cathode side, and the electrons travel to the cathode side along an external load circuit, creating the fuel cell's current output.

Meanwhile, oxygen is delivered to the cathode side of the MEA. The oxygen molecules react with protons and electrons coming from the anode side to form water molecules.

Alkaline Fuel Cells (AFCs). Alkaline fuel cells (AFCs) use a solution of potassium hydroxide in water as the electrolyte and various nonprecious metals as a catalyst at the anode and cathode (Figure 32). High-temperature AFCs operate between 212 and 482°F, although, newer designs operate at roughly 74 to 158°F.

AFCs' high performance derives from the rate at which chemical reactions take place in the cell. They have demonstrated efficiencies near 60% in space applications. AFC stacks maintain stable operation for 8000 operating hours, which limits their commercial viability.

THERMAL-TO-POWER COMPONENTS

These devices convert thermal energy to useful power. They do not convert the fuel source directly to power; this separation of energy conversion enables these devices to operate with an enormous variety of fuels and waste heat.

STEAM TURBINES

Steam turbines, which are among the oldest prime mover technologies, convert thermal energy from pressurized steam into useful power. They require a source of high-pressure steam produced in a boiler or HRSG. Most electricity in the United States is generated by conventional steam turbine power plants. Steam turbines are made in a variety of sizes, ranging from small 1 hp units to several hundred megawatts. In CHP applications, low-pressure steam from the turbine can be used directly in a process, or for district heating, or can be converted to other useful thermal energy.

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 33 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 extracting 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 extracting 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 that can either exhaust or admit a supplemental flow of steam through an intermediate port in the casing, thereby maintaining a process heat balance. Extraction and induction-extraction turbines 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 through the steam's velocity, pressure energy, 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 steam velocity into the blades with little or no steam pressure drop across them, and causes wheel rotation. These combinations of nozzles and velocity-powered wheels are characteristic of an **impulse turbine**.

Reaction 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, and steam pressure drops through both.

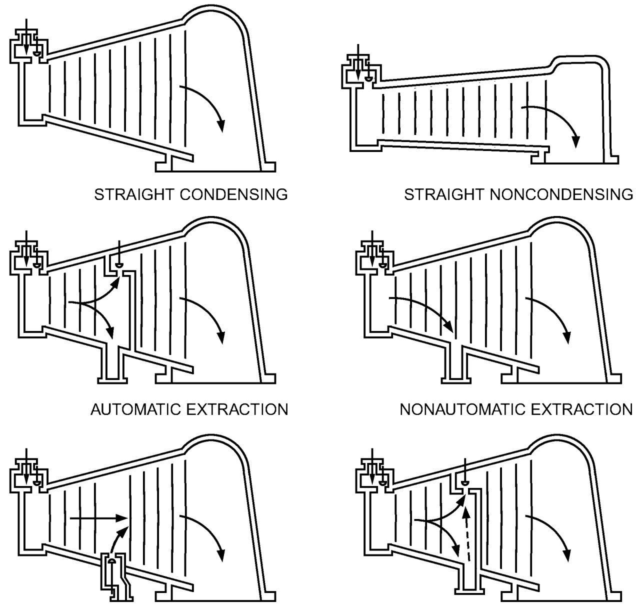

Fig. 33 Basic Types of Axial Flow Turbines

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 impulse- or reaction-type blading. 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 in the turbine casing to prevent damage from condensate accumulation. Carbon rings or closely fitted labyrinths prevent steam leakage 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 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 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.

Performance Characteristics

The topping CHP-cycle steam turbine is typically either a back-pressure or 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 processes. 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 pre-heater 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., 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 directly to the absorber has a 75% fuel-to-steam efficiency, which is 15% lower than the 90% efficiency of a high-pressure boiler used in the CHP 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 the 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 34 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, non-reversible 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 actual exhaust condition of saturated steam h_a at 30 psia, 250°F, 1164 Btu/lb; the actual turbine efficiency is

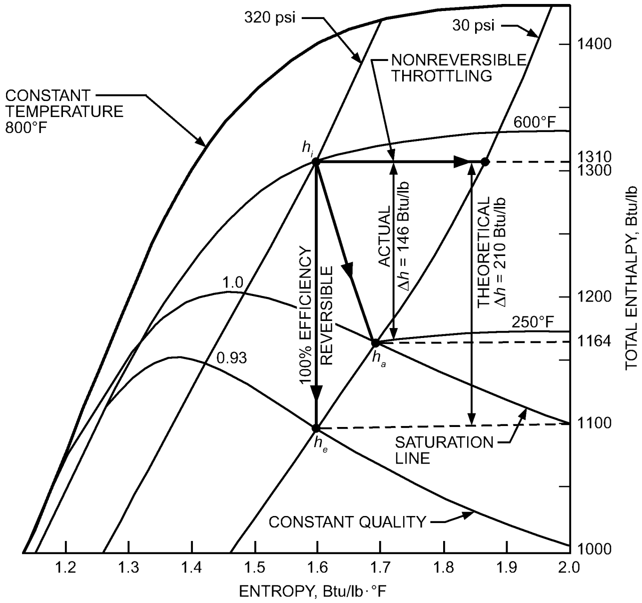

Fig. 34 Isentropic Versus Actual Turbine Process

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

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,

$$E_a = \frac{1310 - 1164}{1310 - 1100} = 0.70 \text{ or } 70\%$$

As a CHP 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

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

The incremental turbine heat rate to generate this power is only

$$\frac{146 \times 45{,}000}{1923} = 3417 \text{ Btu/kWh}$$

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 conditions. Often, one or more of these factors can be improved with only a nominal increase in initial capital cost. CHP applications range, with equal flow turbines, from approximately 100 to 10,000 hp and from

3000 to 10,000 rpm, with high speeds generally associated with lower power outputs, and low speeds with higher power outputs. (Some typical characteristics of turbines driving centrifugal water chillers are shown in Figures 35 to 39.)

Initial steam pressures for small turbines commonly fall in the 100 to 250 psig range, but 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.

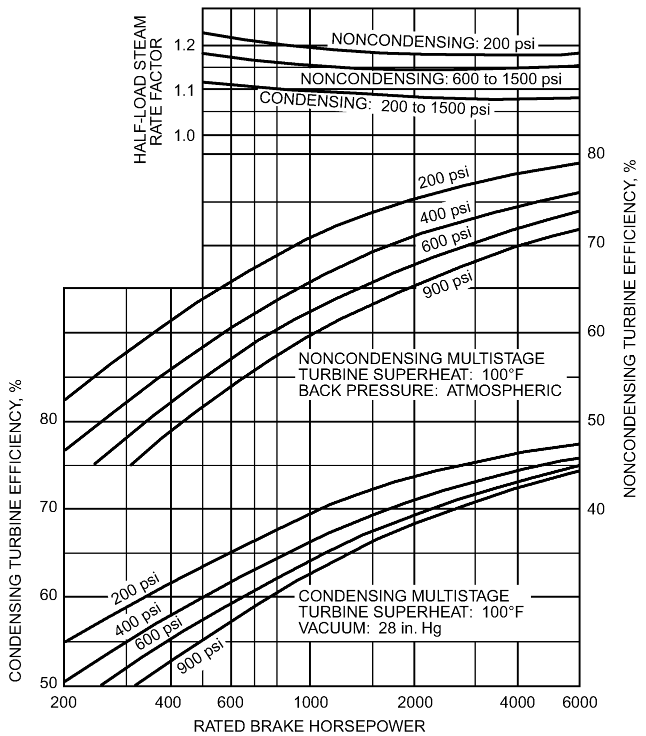

Fig. 35 Efficiency of Typical Multistage Turbines

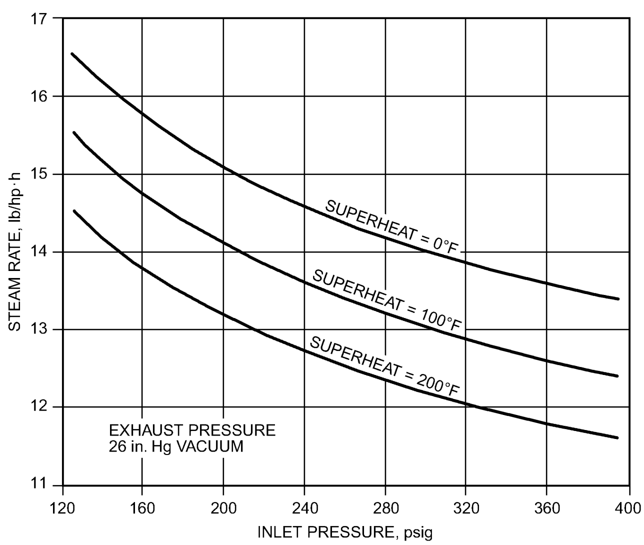

Fig. 36 Effect of Inlet Pressure and Superheat on Condensing Turbine

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 15 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.

Using the entire exhaust steam flow from a base-loaded back-pressure turbine achieves the maximum efficiency of a steam turbine CHP cycle. However, if the facility's thermal/electrical load

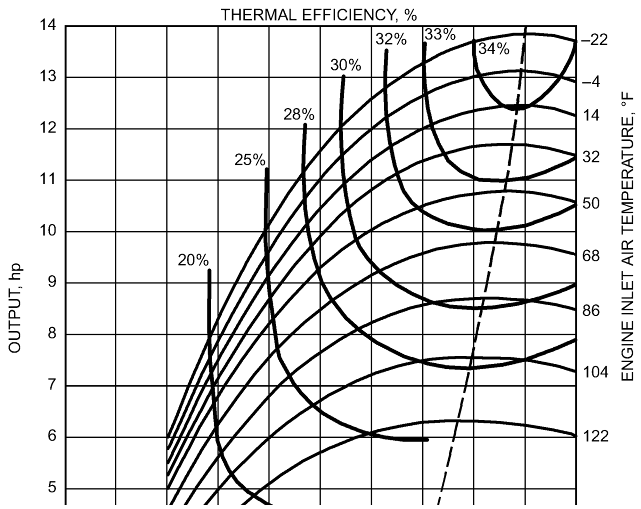

Fig. 37 Effect of Exhaust Pressure on Noncondensing Turbine

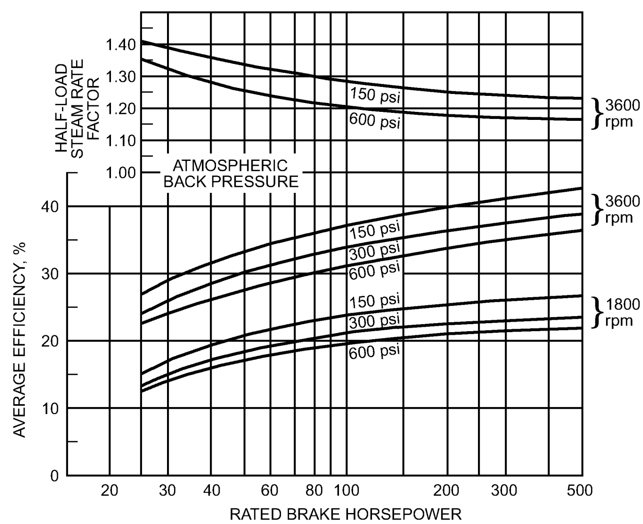

Fig. 38 Single-Stage Noncondensing Turbine Efficiency

Table 15 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 (absolute)	10.52	10.01	9.07	7.37	7.09	6.77	6.58	6.28
4 in. Hg (absolute)	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	33.60	28.05	22.94	15.23	13.62	12.75	11.80	11.07
50 psig	46.00	36.00	28.20	17.57	15.36	14.31	13.07	12.21
60 psig	53.90	40.40	31.10	18.75	16.19	15.05	13.66	12.74
70 psig	63.50	45.60	34.10	19.96	17.00	15.79	14.22	13.25
75 psig	69.30	48.50	35.80	20.59	17.40	16.17	14.50	13.51

cannot absorb the fully loaded output of the turbine, whichever profile is lower can be tracked, and the power output or steam flow is reduced unless the output remaining is exported. 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 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.

Because steam cycles operate at pressures above those allowed by ASME and local codes for unattended operation, their use in CHP 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 39 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, 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 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 have no useful thermal output. Therefore, with overall plant efficiencies no better than their shaft efficiencies, they are unsuitable for topping CHP cycles.

At maximum extraction, the heat/power ratio of extraction turbines is relatively high. This 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 in Figure 39 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:

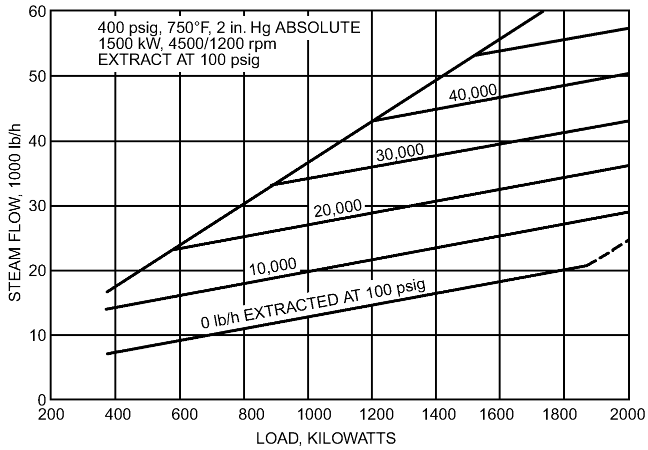

Fig. 39 Effect of Extraction Rate on Condensing Turbine

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

$$\text{Actual } \Delta h = \frac{3412 \text{ Btu/kWh}}{\text{Actual steam rate (lb/kWh)}}$$

Actual Δh = 3412/11.67 = 292 Btu/lb
Plant shaft efficiency = (100 × 292 × 0.85)/1022
= 24.3% (electric output to fuel energy input)

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 turbine efficiencies of 70 to 80%, compared with single-stage axial turbines spinning at only 10,000 rpm with efficiencies of up to 40%.

Fuel Systems

Unlike gas turbines and reciprocating engines, steam turbines normally generate electricity as a by-product of heat generation. These devices do not directly convert fuel to useful power; instead, thermal energy is transferred from the boiler or HRSG to these turbines. This separation of energy conversion enables steam turbines to operate with an enormous variety of fuels such as coal, oil, or natural gas or renewable fuels like wood or municipal waste.

Lubricating Oil Systems

Small turbines often have only simple oil rings to handle bearing lubrication, but most turbines for CHP service have a complete pressure lubrication system. Basic components include a shaft-driven oil pump, oil filter, oil cooler, means of regulating oil pressure, reservoir, and interconnecting piping. Turbines with a hydraulic governor may use oil from the lubrication circuit or, with some types of governors, use a self-contained governor oil system. To ensure an adequate supply of oil to bearings during acceleration and deceleration, 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 driven compressor are integrated. Proper oil pressure, temperature, and compatibility of lubricant qualities must be maintained.

Power Systems

Steam turbines are used for power only or in CHP applications. In CHP applications, low-pressure steam leaving the turbine is used directly in a process or for district heating, or converted to other forms of thermal energy. In power-only applications, condensing turbines are generally used to maximize electricity production by using fuels that otherwise would go to waste. Condensing turbines are also used in bottoming cycles for gas turbines to produce additional electricity in a combined cycle (gas turbine exhaust is used in a HRSG to generate high-pressure steam to drive the steam turbine) and improve the cycle's electric efficiency.

Exhaust Systems

Steam turbines can be driven with a boiler firing with various fuels, or they can be used in a combined cycle with a gas turbine and HRSG. Boiler emissions depend on the fuel used to generate the high-pressure steam to drive the steam turbine.

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 dual shafting arrangements are started gradually 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 allows a governor to be ideally matched to the characteristics of the driven machine and 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 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 temperature of fluid leaving the water-chilling heat exchanger (evaporator). To avoid compressor surge and 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. External signals may be needed 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 most common speed governor for 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 (Figure 40), 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 41 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 42). Its principal application is in larger turbines where a single governor-controlled steam admission valve would be too large to allow sensitive control. The greater power required to actuate the multiple-valve mechanism dictates using 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

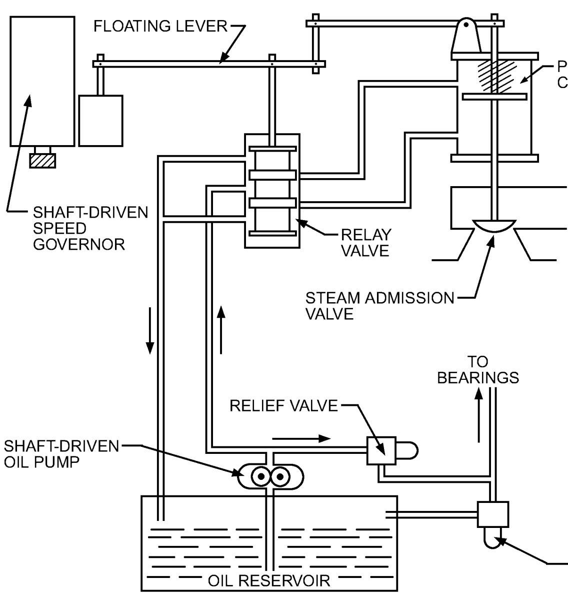

Fig. 40 Oil Relay Governor

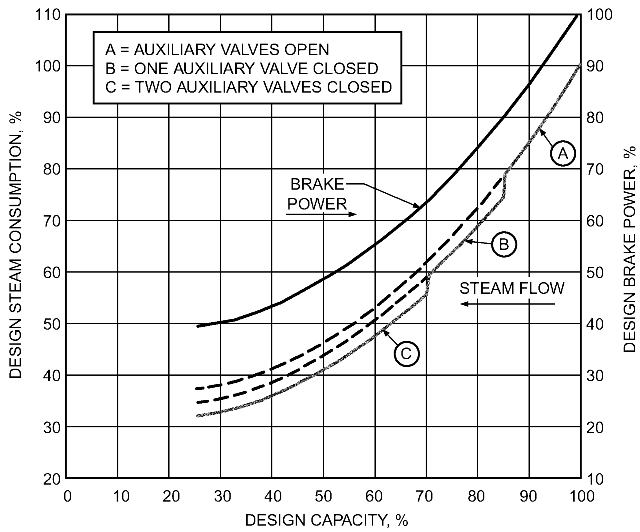

Fig. 41 Part-Load Turbine Performance Showing Effect of Auxiliary Valves

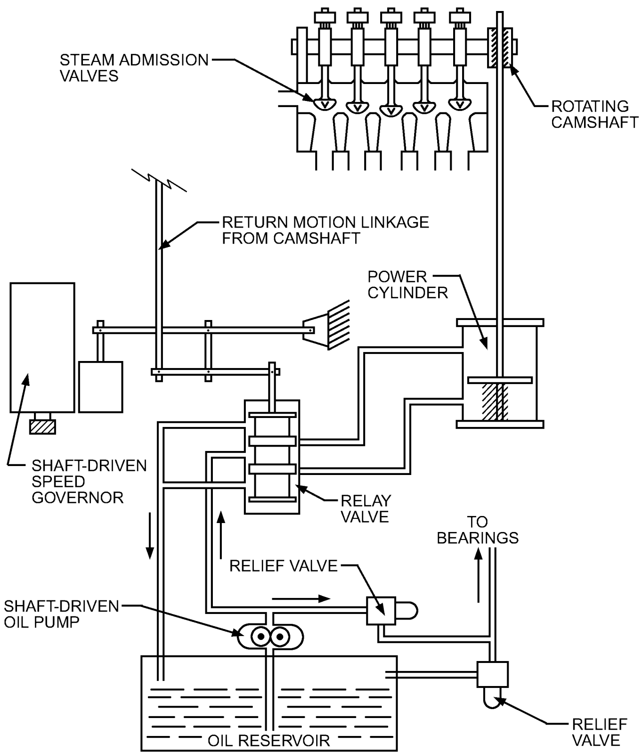

Fig. 42 Multivalve Oil Relay Governor

oil flow by a needle valve. The upper limit of a speed changer's capability should not exceed the rated turbine speed. These speed changers, though usually mounted on the turbine, may sometimes be remotely located at a central control point.

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 adjusting 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.

Table 16 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.

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) *Standard* SM 24 classifies steam turbine governors as shown in Table 16. **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 control of the speed governor and 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:

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

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 open to allow free flow of steam to the governor-controlled valve(s).

The steady-state speed regulation 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:

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

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 speed-governing system's ability 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. A governing system's ability to minimize sustained oscillations 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.

Operation and 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. These include erosion of high-pressure turbine nozzle and blades by solid particles transported from steam lines, superheaters, or reheaters, especially during cycling operation, that weakens rotating blades; fouling of high-pressure turbine with copper deposits (in case of mixed metallurgy of feedwater train); and stress corrosion cracking and corrosion fatigue of low-pressure turbine. The latter failures occur during both steady-state and cycling operation and are attributed to formation of early condensate in the so-called phase transition region where steam changes from superheated to saturated condition. The concentration of impurities in the early condensate is higher than in the steam that enters the turbine. Early condensate droplets may precipitate on blade surface and form liquid films, which can evaporate during flow and form the highly concentrated solutions on the surfaces inside turbines. The level of corrosive impurities in these films may be much higher than in the early condensate. In addition, steam moisture may result in water droplet erosion and, in combination with other parameters, flow-accelerated corrosion.

The best way to minimize both corrosion and erosion in steam turbines is to maintain proper feedwater boiler and steam chemistry. The requirements to cycle chemistry are more stringent with steam pressure. To protect steam turbines from unnecessary damage, the level of impurities in the steam from the boiler or HRSG must be kept within the limits specified by corresponding cycle chemistry guidelines.

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.

Turbine seals, glands, and bearings are also common areas of deterioration and maintenance. Bearings require frequent examination, especially in cyclical duty systems. 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 blades, blade material erodes or corrodes; as mechanical tolerance of bearing surfaces increases, nonuniform rotation increases 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.

ORGANIC RANKINE CYCLES

Organic Rankine cycle (ORC) technology is based on the Rankine process, except that instead of water/steam, an organic working fluid is used. ORCs can convert high- or low-quality thermal energy into useful mechanical or electrical energy and thus improve the efficiency of integrated energy systems by using waste heat from DG equipment. One problem with ORCs is that their inherent thermal conversion (Carnot) efficiency is very low for lower-quality thermal sources. However, this disadvantage is offset by the fact that the fuel is energy that would otherwise be wasted.

ORCs began to be studied and applied as energy conversion engines over 100 years ago to use low-grade heat from geothermal waters, solar energy from liquid and vapor collectors, industrial waste heat, biomass combustion, etc.

ORCs are used for low-temperature waste heat recovery, efficiency improvement in power stations, and recovery of geothermal and solar heat. Small-scale ORCs have been used commercially or as pilot plants. About 30 commercial ORC plants were built before 1984 with an output of 100 kW. ORCs allow efficient exploitation of low-temperature heat sources to produce electricity in a wide range of power outputs, from a few kilowatts to 3 MW. Several organic compounds have been used in ORCs (e.g., CFCs, halocarbons, isopentane, ammonia) to match the temperature of available waste heat sources. Efficiency is estimated to be between 10 and 30%, depending on operating temperatures.

EXPANSION ENGINES/TURBINES

Expansion turbines use compressed gases available from gas-producing processes to produce shaft power. One application expands 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, 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, 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.

This process consumes no fuel, but it does provide shaft energy, as well as expansion refrigeration, which fits the definition of CHP. Similarly, a back-pressure steam turbine, used instead of a pressure-reducing valve, generates productive shaft power and allows 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.

STIRLING ENGINES

Stirling-cycle engines were patented in 1816 and were commonly used before World War I. They were popular because they had a better safety record than steam engines and used air as the working fluid. As steam engines improved and the competing compact Otto-cycle engine was invented, Stirling engines lost favor. Recent attention to distributed energy resources (DER), used by the space and marine industries, has revived interest in Stirling engines and increased research and development efforts.

Stirling engines, classified as external combustion engines, are sealed systems with an inert, reusable working fluid (either helium or hydrogen). Rather than burning fuel inside the cylinder, the engine uses external heat to expand the gas contained inside the cylinder and push against its pistons. The engine then recycles the same captive working fluid by cooling and compressing it, then reheating it to expand and drive the pistons, which in turn drive a generator. These engines are generally found in small sizes (1 to 55 kW). The maximum Carnot efficiency based on the second law of thermodynamics $[1-(T_{cold}/T_{hot})]$ for a heat source of 1600°F (2060°R) and a cold sink of 77°F (537°R) is approximately 74%. Overall, electrical efficiency of these engines are usually in the range of 12 to 30% (difference is due to friction losses and heat transfer ineffectiveness); overall efficiency in a combined heat and power mode is around 80%.

Types

Stirling engines can be divided into two major types: kinematic and free-piston engines (NREL 2003).

Kinematic Stirling Engines. Pistons of these engines are connected by rods and crankshaft or with a wobble plate (also known as a swash plate). The wobble plate allows for variable output while maintaining thermal input, allowing faster response to changes in demand (Figure 43).

Kinematic Stirling engines can be designed as either single- or double-acting engines. Single-acting kinematic Stirling engines can have one or more cylinders with varying designs for mechanical linkage. Double-acting kinematic engines require several cylinders. The working fluid in a double-acting system operates on opposite sides of the piston.

Free-Piston Stirling Engines. These engines do not have any mechanical coupling between the piston and the power output. The pistons are mounted in flexures and oscillate freely (Figure 44). Flexures are springs that are flexible in the axial direction, but very stiff in the radial direction. A fluid or a linear alternator receives the reciprocating output via pneumatic transfer. Free-piston Stirling engines are single-acting, with the working fluid only operating on one side of a piston. Free-piston configurations are limited to sizes less than 12.5 kW.

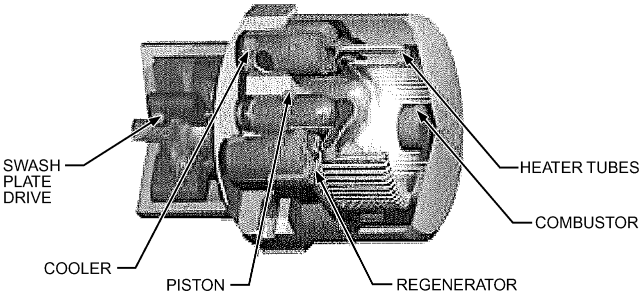

Fig. 43 Cutaway Core of a Kinematic Stirling Engine

Performance Characteristics

Most manufacturers promoting Stirling engines for distributed generation do not yet offer a standard product line. Although Stirling engines work on simple principles, designers are faced with challenges in determining design tradeoffs between complexity, cost, durability/reliability, and efficiency. The main difficulties with recent Stirling engines are their long-term durability/reliability and high cost. Durability concerns for Stirling engines include the following:

- Shaft seals separating the high-pressure hydrogen space from lubrication in the mechanical drive train
- Low-leakage piston rings and bearings for operation in the unlubricated working engine space
- Minimizing material stress and corrosion in the high-temperature, high-pressure heater head, which must operate at internal conditions of >2000 psi and 1300°F.
- Blockage of fine-meshed heat matrices used in regenerator assemblies by particles generated by rubbing piston rings

Although these problems have delayed adoption of Stirling engine technology, run times are beginning to approach acceptable lengths for some distributed power applications. The electrical efficiency of current engines is approximately 27% HHV (30% LHV).

Fuel Systems

Stirling engines absorb heat from a wide range of fuel sources (e.g., natural gas, propane, oil, biomass, waste fuels) and convert it to electricity with minimal emissions and low maintenance requirements. Typical heat sources include standard gaseous and liquid fuels, with options to accept low-heating-value landfill and digester gas, petroleum flare gas, and other low-grade gaseous, liquid, and solid waste fuels. Raw heat from solar concentrators or flue gas stacks can also be converted to electricity, with no fuel costs and no incremental emissions.

Stirling engines usually require a heat source ranging between 1500 and 1800°F.

Contaminants are a concern with some waste fuels, particularly acid gas components such as hydrogen sulfide (H_2S content should be less than 7%).

Power Systems

Fuel flexibility of these engines makes them candidates for use as heat recovery units in industrial processes. These engines generally

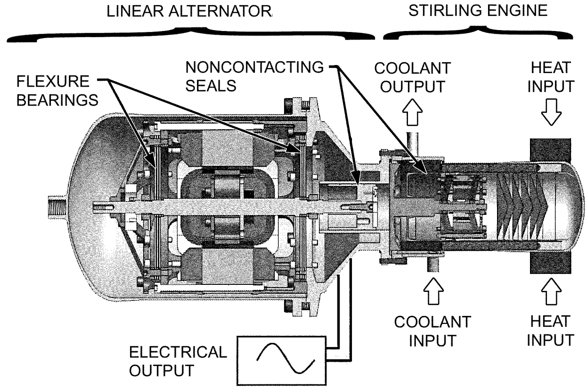

Fig. 44 Cutaway Core of a Free-Piston Stirling Engine
(Courtesy Infinia Corporation)

require a heat source between 1500 and 1800°F, noncorrosive, and relatively free of particulates. Their electric efficiencies are approximately 30% (based on LHV), leaving approximately 70% of the heat input to be rejected in the cooling system and exhaust gases. As with any combustion system, there performance drops as elevation increases, because of density changes in the fuel and combustion air.

Part-load operation is a design trade-off issue. Devices designed for distributed generation applications are usually capable of a 3:1 turn-down.

Exhaust Systems

The combustor of a Stirling engine can be customized to achieve low emissions. With a hot-end operating temperature of 1600°F, the combustor needs to operate only a few hundred degrees hotter, which puts it in the lower end of the thermal NO_x formation range. NO_x-reducing techniques are available, such as lean burn, staged combustion, or flue gas recirculation. NO_x emissions improve at part load because of lower combustor temperatures. When firing with natural gas, SO_x should not be an issue.

Coolant Systems

Stirling engines usually convert approximately 30% of heat input to electric power, with almost 50% going to the coolant and the rest dissipated as exhaust or other losses. Because Stirling engines are liquid-cooled, it is relatively easy to capture the low-temperature waste heat for CHP applications (thermal output of approximately 140°F) such as water heating or other low-temperature heating applications. The addition of a heat exchanger (with related controls) is the only difference between a power-only unit and CHP unit. Power-only units are usually equipped with a radiator and fan to dissipate coolant heat into the environment.

Operation and Maintenance

Maintenance procedures and associated costs have not yet been developed because of the lack of long-term operating experience. The engine is usually designed for a 40,000 h life. The engine would be replaced at the end of this period. Periodic maintenance after 8000 h usually includes oil and oil filter and air filter changes. Another important maintenance item is working fluid makeup: the engine operates at a relatively high working pressure, and the small molecular size of the working fluid means it can leak through the engine seals. Occasional working fluid injection may be required. Economic life of these systems is estimated to be around 10 years.

THERMAL-TO-THERMAL COMPONENTS

THERMAL OUTPUT CHARACTERISTICS

By definition, CHP systems use the fuel energy that the prime mover does not convert into shaft energy. If site heat energy requirements can be met effectively by the thermal by-product at the level it is available from the prime mover, this salvaged heat reduces the normal fuel requirements of the site and increase overall plant efficiency. The ability to use prime mover waste heat determines overall system efficiency and is one of the critical factors in economic feasibility.

This section describes devices that convert prime mover "waste" thermal energy into energy streams suited to meet typical thermal loads. The section begins with a description of the thermal output characteristics and heat recovery opportunities of key prime movers.

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.

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 Figures 47 to 51).

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 balances for naturally aspirated (Figure 45) and turbocharged gas engines (Figure 46) 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 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.

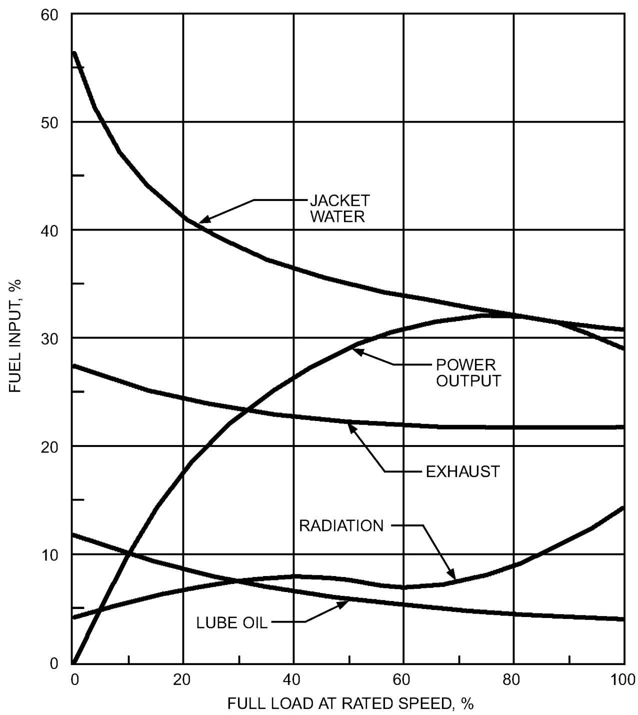

Fig. 45 Heat Balance for Naturally Aspirated Engine

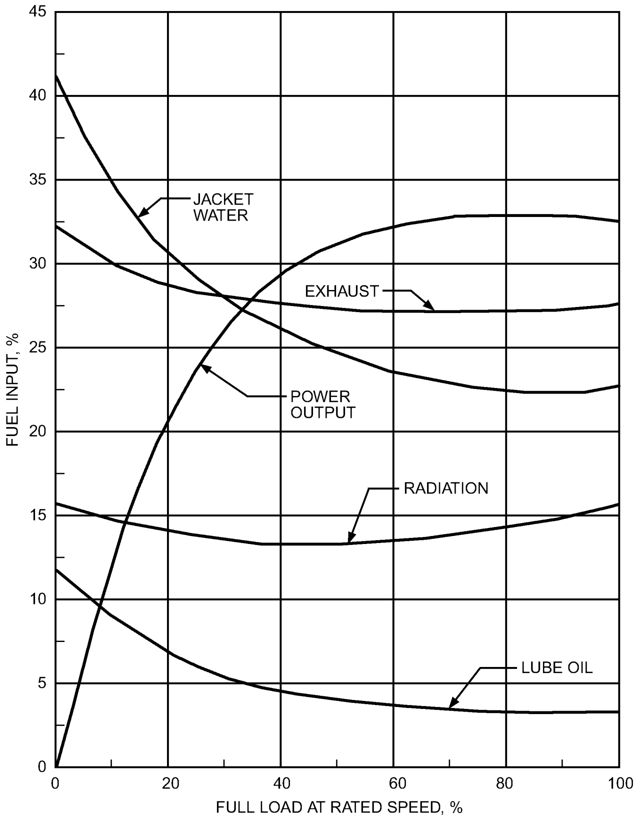

Fig. 46 Heat Balance for Turbocharged Engine

Combustion Turbines

In the gas turbine cycle, the average fuel-to-electrical-shaft efficiency ranges from approximately 12% to above 35%, with the rest 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 50% and higher. Because gas turbine exhaust contains a large percentage of excess air (high O_2 content), afterburners or boost burners may be installed in the exhaust to create a supplementary boiler system or to cofire an absorption chiller. This system can provide additional steam, or level the steam production or cooling during reduced turbine loads. Absorption chillers able to operate directly off turbine exhausts (exhaust-fired absorption chillers) with coefficients of performance (COPs) of 1.30 or more are also available. The COP is the cooling energy output divided by energy input. The conventional method of controlling steam or hot-water production in a heat recovery system at part load is to bypass some exhaust gases around the boiler tubes and out the exhaust stack through a gas bypass valve assembly.

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 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. Keeping temperature rise through the jacket to less than 15°F helps avoid thermal stress. Keep flow rates within engine manufacturers' design limits to avoid erosion from excessive flow or inadequate distribution in the engine from low flow rates.

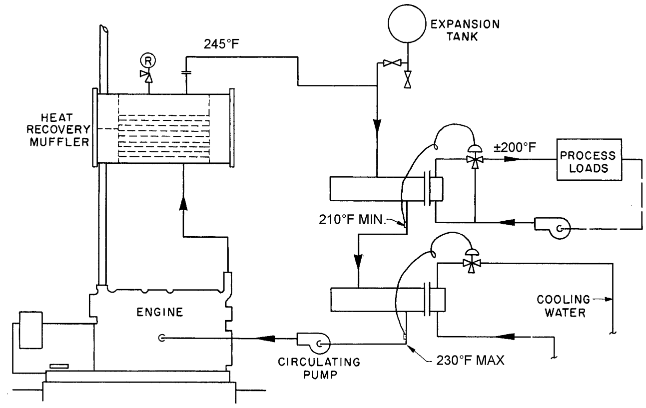

Fig. 47 Hot-Water Heat Recovery

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 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 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 loads, including water-heating systems for comfort and process loads, absorption refrigeration chillers, desiccant dehumidifiers, and service water heating. Engine jacket coolant distribution 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 salvaged heat to another circuit that serves several engines through 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 47). This approach is highly reliable, 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 overloading the system and seriously reducing the engines' operating temperature, which could cause the engine casings to crack. A heat storage tank is an excellent buffer, because it can provide a high rate of heat for short periods to protect machinery serving the heat loads. The heat level can be controlled with supplementary input, such as an auxiliary boiler.

A limited distribution system that distributes steam through nearby heat exchangers can salvage heat without contaminating the main engine cooling system (Figures 48 and 49). 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.

Lubricant Heat Recovery. Lubricant heat exchangers should keep oil temperatures at 190°F, with the highest coolant temperature consistent with economical use of the salvaged heat. Engine manufacturers usually size oil cooler heat transfer surfaces based on 130°F

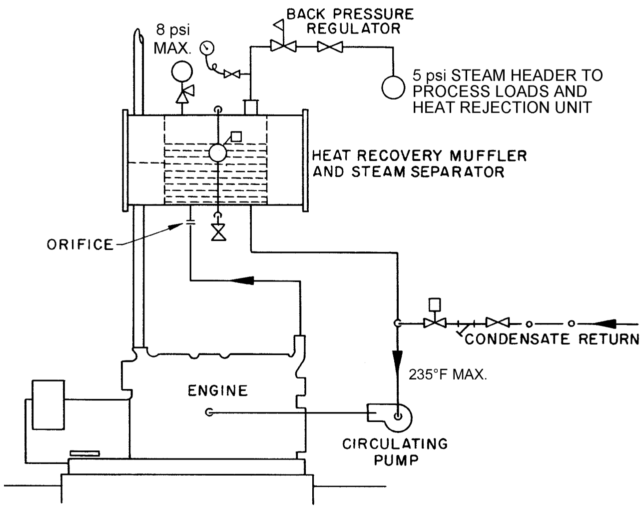

Fig. 48 Hot-Water Engine Cooling with Steam Heat Recovery (Forced Recirculation)

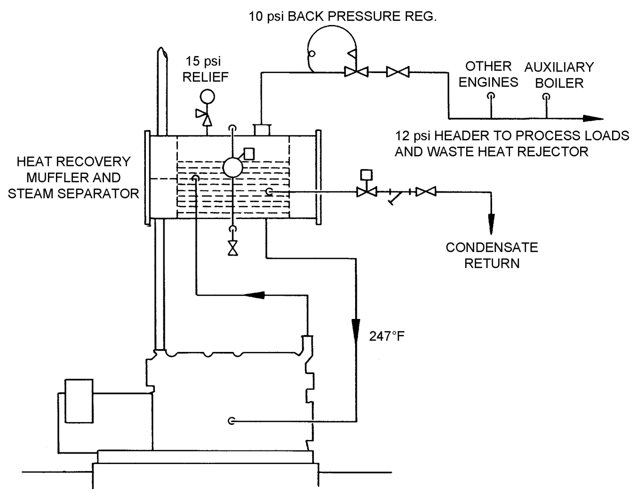

Fig. 49 Engine Cooling with Gravity Circulation and Steam Heat Recovery

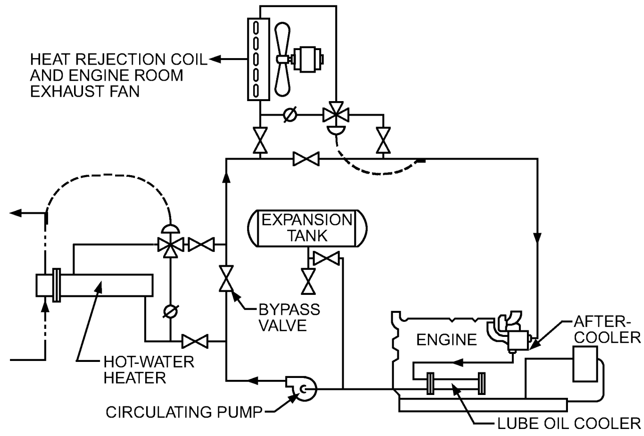

Fig. 50 Lubricant and Aftercooler System

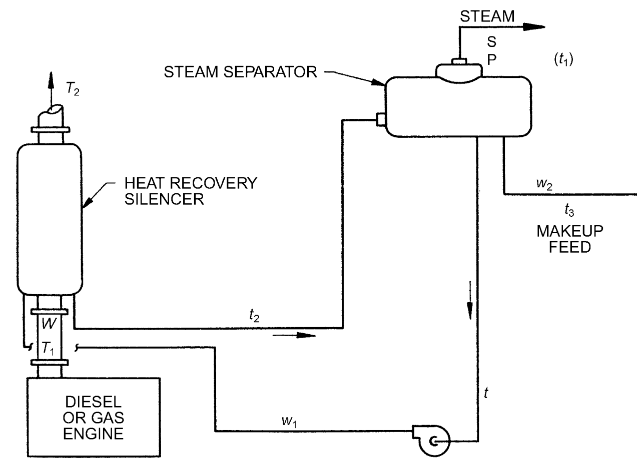

Fig. 51 Exhaust Heat Recovery with Steam Separator

entering coolant water, without provision for additional lubricant heat gains from high engine operating temperatures. Compare the cost of a reliable supply of lower-temperature cooling water with that of increasing the oil-cooling heat exchanger size and using cooling water at 165°F. If engine jacket coolant temperatures are above 220°F and heat at 155 to 165°F can be used, heat from the lubricant can be recovered 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 aftercooler water on diesel engines can be run in series with the oil cooler (Figure 50). A double-tube heat exchanger can also be used to prevent contamination from a leaking tube.

Turbocharger Heat Recovery. Natural gas engine turbochargers need medium fuel gas pressure (12 to 20 psi) and 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 usual sources are raw domestic water and evaporative cooling systems such as cooling towers. Aftercooler water as warm as 135°F can be used, although this derates the engine. Using domestic water may be expensive because (1) coolant is continuously needed while the engine is running, and (2) available heat exchanger designs require a lot of water even though the load is less than 200 Btu/hp·h. A cooling tower can be used, but it can increase initial costs; it also requires 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.

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 because of the limitations of heat transfer equipment and the need to prevent flue gas condensation (Figure 51).

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 for generator sets that frequently operate at part load. Depending on the initial exhaust temperature, about 50 to 60% of the available exhaust heat can be recovered.

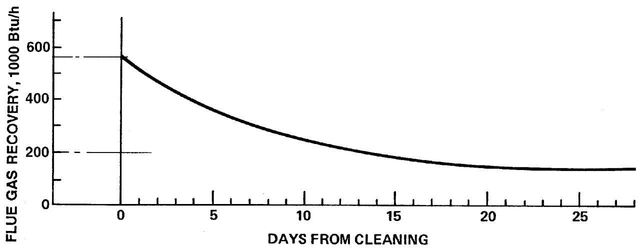

Fig. 52 Effect of Soot on Energy Recovery from Flue Gas Recovery Unit on Diesel Engine

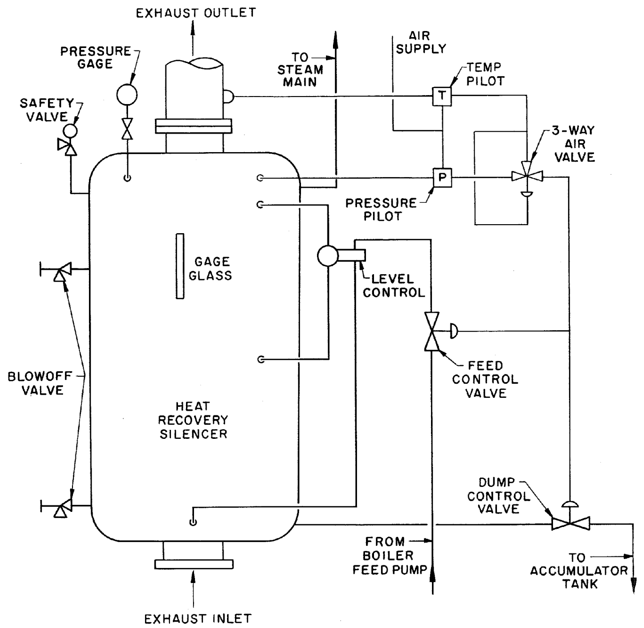

Fig. 53 Automatic Boiler System with Overriding Exhaust Temperature Control

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 have a means of soot removal, because soot deposits can quickly reduce heat exchanger effectiveness (Figure 52). 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 53 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 boiler output to maintain a preset minimum

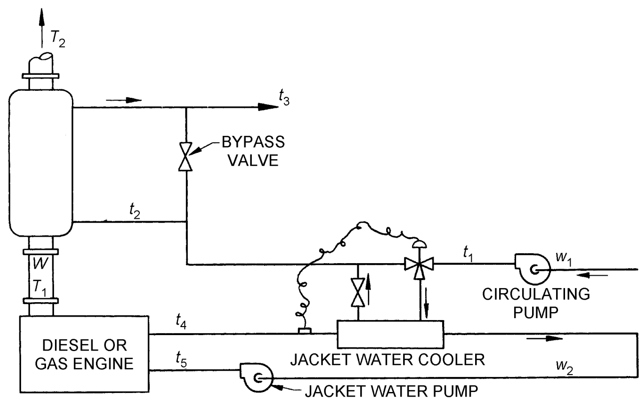

Fig. 54 Combined Exhaust and Jacket Water Heat Recovery System

exhaust temperature at the outlet. This type of automatic control is limited to vertical boilers because the ASME (1998) *Boiler and Pressure Vessel Code* does not allow 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 exhaust gas to achieve automatic modulation with heat load demand (Figure 54). If 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. Heat recovery equipment should not adversely affect the primary function of the engine to produce work. Therefore, design of waste heat recovery boilers should begin by determining the back pressure imposed on 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 55 illustrates the effect of lowering 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 28% surface increase. Similarly, a reduction from 300°F to 100°F increases heat recovery 29% but requires a 120% surface increase. Therefore, the cost of heat transfer surface must be considered when determining the final temperature.

Another factor to consider is water vapor condensation and acid formation if 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. Condensates formed at low exhaust temperatures can be highly acidic. Sulfuric acid from diesel fuels and carbonic acid from natural gas fuels can cause

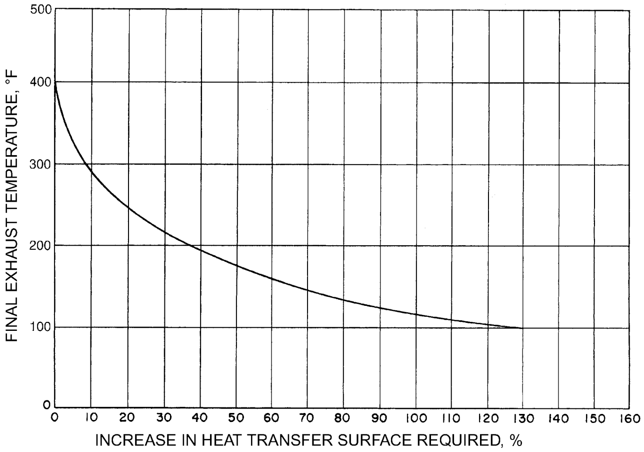

Fig. 55 Effect of Lowering Exhaust Temperature below 300°F

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

Exhaust recovery equation 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 shaft power is known but engine data are not, 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 flow rate is 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.

Normal procedure is to design and fabricate heat recovery boilers to the ASME (1998) *Boiler and Pressure Vessel Code* (Section VIII) for the working pressure required. Because temperatures in most exhaust systems are not excessive, it is common to use flange or firebox-quality steels for 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.

Table 17 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 18 Full-Load Exhaust Mass Flows and Temperatures for Various Engines

Type of Four-Cycle Engine	Mass Flow, lb/bhp·h	Temperature, °F
Naturally aspirated gas	9	1200
Turbocharged gas	10	1200
Naturally aspirated diesel	12	750
Turbocharged diesel	13	850
Gas turbine, nonregenerative	18 to 48*	800 to 1050*

*Lower mass flows correspond to more efficient gas turbines.

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 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 17 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 18. 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, 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 they require higher airflows to purge their cylinders, two-cycle engines have lower temperatures than four-cycle engines, and thus 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 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 steam is exhausted at atmospheric pressure or higher. They 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 56 illustrates the steam path for a noncondensing turbine. The back-pressure steam turbine operates on the enthalpy difference between steam inlet and exhaust conditions.

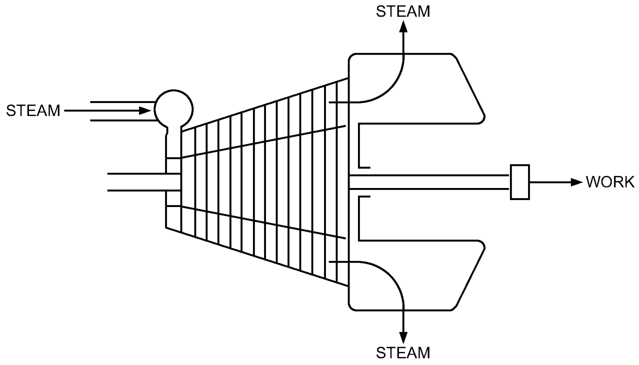

Fig. 56 Back-Pressure Turbine

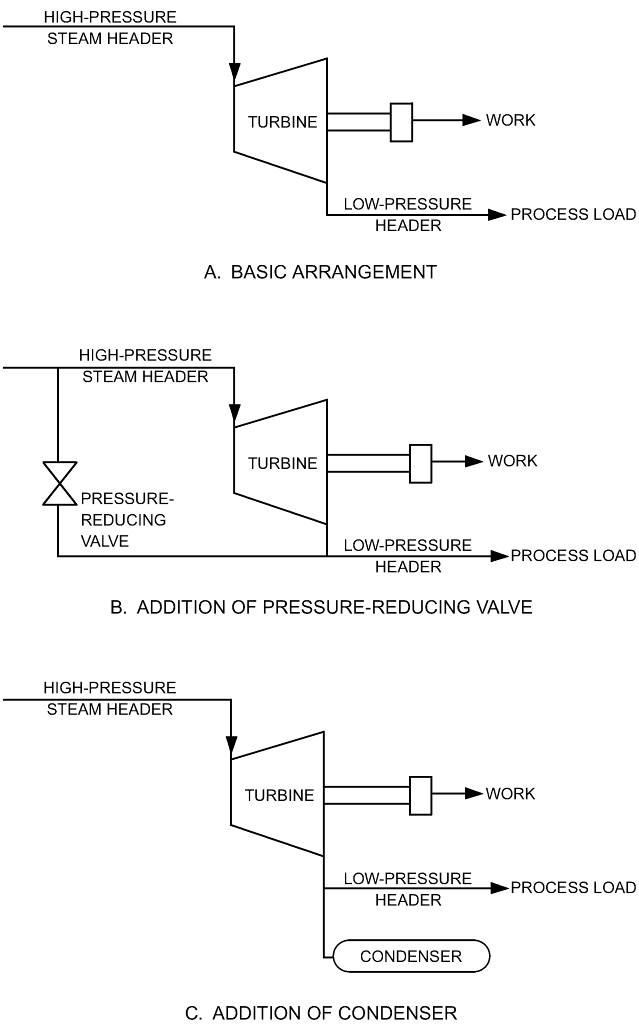

Fig. 57 Integration of Back-Pressure Turbine with Facility

The noncondensing turbine's Carnot cycle efficiency tends to be lower than is possible with other turbines because the difference between 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 CHP system process or total energy efficiency can be very high. One 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).

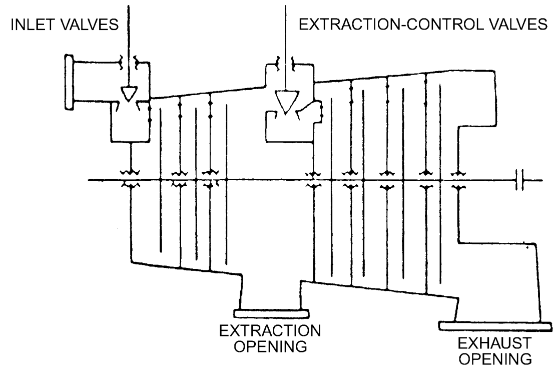

Fig. 58 Condensing Automatic Extraction Turbine

The back-pressure turbine has one major disadvantage in CHP: 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. Direct linkage of 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 57 illustrates several ways to achieve flexible performance when electrical and thermal loads do not match the back-pressure turbine's capability. Figure 57A shows the basic arrangement of a noncondensing steam turbine and its relationship to the facility. Figure 57B 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 turbine's capability, the additional steam can be provided through the PRV. Figure 57C illustrates use of a load condenser to allow electricity generation, even when there is no process steam demand. These techniques to match thermal and electrical loads are 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 58 illustrates the internal arrangement of an automatic extraction turbine that exhausts steam at one or more stations along the steam flow path. Conceptually, the extraction turbine is a hybrid of condensing and noncondensing turbines. Its advantage 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 CHP cycle to 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- and low-pressure sections. All the steam passes through the high-pressure section just as it does in a single back-pressure turbine. A throttle, controlled by the pressure in the process steam line, controls steam flow into the low-pressure section. If pressure drops, the throttle closes, allowing more steam to the process. If pressure rises, the throttle opens to allow steam to flow through the low-pressure section, where additional power is generated.

An automatic extraction turbine (Figure 59) 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

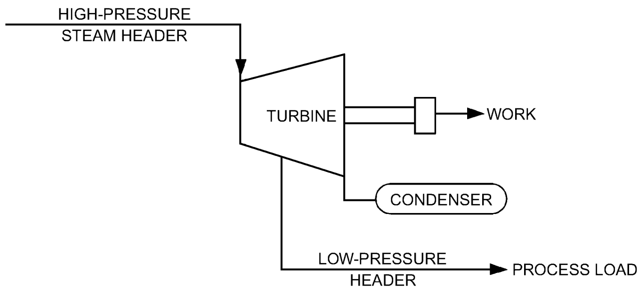

Fig. 59 Automatic Extraction Turbine CHP System

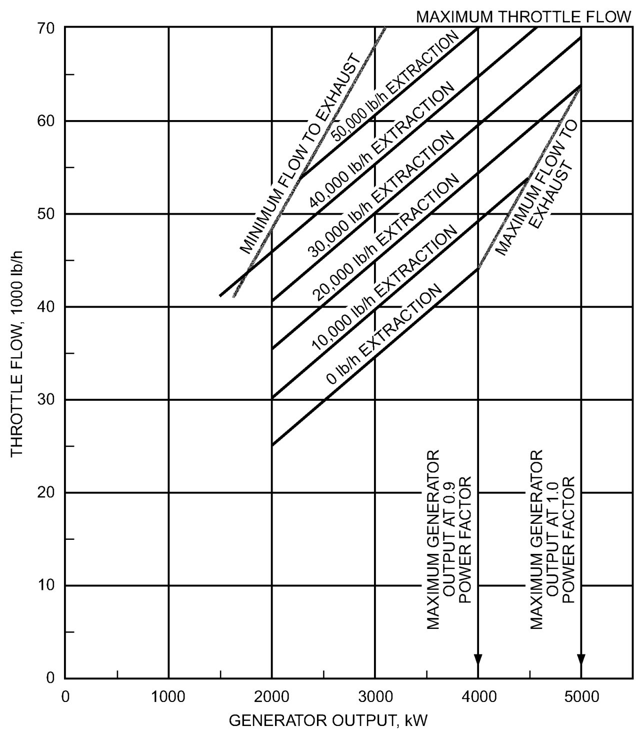

Fig. 60 Performance Map of Automatic Extraction Turbine

(1945) can be used to estimate performance. The product of such an analysis is a performance map similar to that shown in Figure 60 for a 5000 kW generator. The performance map provides steam flow to the turbine as a function of generator output with extraction flow as a parameter.

THERMALLY ACTIVATED TECHNOLOGIES

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 0.7 or less; 18,000 Btu/h of recovered heat can produce about 1 ton of cooling (12,000 Btu/h). Note that the following equations are based on full-load operation. Part-load output can be calculated by substituting the part-load COPs provided by the manufacturer for part-load COPs given here.

Hot water at 190 to 220°F, recovered from the cooling jacket loop of the prime mover, is used to produce chilled water in an indirect-fired, single-stage absorption machine. The equation to estimate the cooling produced from the heat recovered from the water is

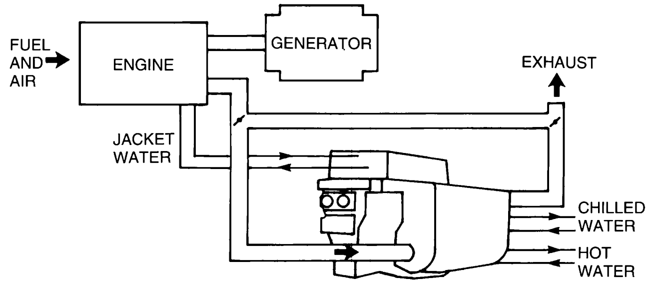

Fig. 61 Hybrid Heat Recovery Absorption Chiller-Heater

$$q = \frac{\text{COP} \times \dot{m}_w (c_p)_w (t_1 - t_2)}{12{,}000}$$

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]
COP = coefficient of performance (typically 0.7)
12,000 = Btu/h·ton

For engines under 2000 kW, jacket water heat and exhaust energy are usually combined to maximize cooling output from the system. In this case, jacket water passes through the engine and picks up available waste heat, and is then directed to an exhaust-to-water heat exchanger where its temperature is elevated by the exhaust energy. Flow rate is determined by the engine manufacturer and should be set to maximize the temperature differential as much as possible; the temperature differential across the engine loop is typically 15°F, and the exhaust energy can add a further 7 to 10°F, for a total differential of 25°F.

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 or steam-turbine-driven chillers can also be considered. The full-load COP at standard conditions of a steam-fired two-stage absorption chiller is typically 1.20. The steam input required is 10 lb/ton·h. This compares to 18 lb/ton·h for the single-stage absorption machine using 15 psig steam.

Steam turbine drive chillers can accept steam pressures exceeding 200 psi. The turbine drive centrifugal chiller typically incorporates a variable expansion valve which allows full-load operation at low cooling tower temperatures down to 50°F.

Providing cooling with engine exhaust heat is described in the section on Exhaust Gas Heat Recovery, and some absorption machines have been designed specifically for heat recovery in CHP applications. These units use both the jacket water and exhaust gas directly (see Figure 61).

Another type of absorption chiller uses gas engine or turbine exhaust directly (exhaust-fired absorption chiller). If a direct-exhaust, two-stage absorption chiller is used, the equation to estimate cooling produced from recoverable heat is

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

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
COP = coefficient of performance (typically 1.3)
0.97 = connecting duct system efficiency
12,000 = Btu/h·ton

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

Desiccant Dehumidification

Desiccant dehumidification is another essential thermally activated technology to convert low-grade waste heat into useful dehumidification that helps maximize energy savings and economic return. Both solid and liquid desiccant units have been successfully applied for many years for humidity control. Desiccants are particularly well suited to CHP because their operation requires mainly thermal energy, which can be provided from most prime movers. The energy required to remove humidity can be relatively low compared to the heat available from a prime mover, so desiccants can work in conjunction with other thermally activated technologies. Their temperature requirements are also relatively low (160 to 250°F), so they can be configured for a bottoming cycle. For more information, see Chapter 22.

Hot Water and Steam Heat Recovery

CHP heat recovery equipment encompasses all forms of heat exchangers that capture or recover waste heat or exhaust gas of a prime mover, such as a combustion turbines or natural gas or diesel engines, to generate steam or hot water. Heat recovery steam generators (HRSGs) are used to recover energy from the hot exhaust gases in power generation. Water is pumped and circulated through tubes heated by the exhaust gases and can be held under high pressure and temperature, and boiled to produce steam. HRSGs are found in may combined-cycle power plants. They were originally designed to produce steam at one pressure level; modern HRSGs may have up to three pressure levels with superheat and reheat, and may be once-through or recirculating.

Thermal Energy Storage Technologies

Thermal energy storage (TES) can decouple power generation from the production of process heat, allowing 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 (though there can be added value from the resulting increase in peaking capacity). As a result, evaluation of economic benefits of adding TES to a conventional CHP system must consider the increased cost of the combined system and the value of peaking capacity.

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 2007 *ASHRAE Handbook—HVAC Applications* has more information on thermal storage.

TES systems and technologies for power generation applications can be categorized by storage temperature. High-temperature storage can be used to store thermal energy from sources (e.g., gas turbine exhaust) at high temperatures (**heat storage**). Storage options such as oil/rock, molten nitrate salt, and combined molten salt and oil/rock are well developed and commercially available (Somasundaram et al. 1996). Medium-temperature storage with hot water can also be used.

Low-temperature TES technologies store thermal energy below ambient temperature (**cool storage**) and can be used for cooling air entering gas turbines, or for storing cooling captured from waste heat by absorption and/or steam-turbine-driven chillers. Examples include commercially available options such as diurnal ice, chilled-water, and low-temperature-fluid storage, as well as more advanced schemes represented by complex, compound chemisorption TES systems.

ELECTRICAL GENERATORS AND COMPONENTS

GENERATORS

Criteria for selecting alternating current (ac) generators for CHP 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.

For prime movers coupled to a generator, generator speed is a direct function of the number of poles and the output frequency. For 60 Hz output, speed can range from 3600 rpm for a two-pole machine to 900 rpm for an eight-pole machine. There is a wide latitude 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.

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.

Prime movers that use **alternators** (to convert high-frequency ac to dc) and **inverters** (dc to ac) to produce 60/50 Hz ac power rely on the inverter design both to meet electrical specifications and for protection by the power conversion controls. The generator's rotating speed is irrelevant because the inverter creates the appropriate ac power to either follow the utility electric system voltage and frequency characteristics when operating grid-parallel, or meet the requirements of local loads when operating grid-isolated during system outages. Note that in these situations, the unit is clearly separated from the utility system by breakers or switches.

The combined efficiency of the generator, alternator, and inverter is a nonlinear function of the load and is usually maximized at or near the rated load (Figure 62).

The rated load estimate should include a safety factor to cover transient conditions such as short-term peaking and equipment start-up. Because combustion turbine engines can produce more power under cold ambient conditions than their ISO rating, generators are typically sized to match the engine's expected maximum power output. For combustion turbines driving alternator and inverter systems, the maximum power capability of the inverter typically limits system output. The inverter design may not support power output above the engine's ISO ratings because of the expense of the power electronics (UL 1999). 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 and other loads from the three-phase generator presents the best phase balance, assuming that power factor requirements are met. Motor start-up can be a problem for inverter-based microturbines because they are current-limited to protect the inverter. 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.

In grid-isolated operation, voltage for synchronous generators is regulated by using static converters or rotating dc generators to excite the generator. Voltage regulation should be within 0.5% of nominal voltage and frequency regulation within 0.3 Hz from full load to no load during steady-state conditions. Good electronic three-phase voltage sensing is necessary to control the system response to load changes and 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 CHP 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.

U.S. *National Electrical Code*® (NFPA *Standard* 70), Article 250, discusses grounding neutral connection of on-site generators. Section 250-5 covers emergency generators in electrical systems, with a four-pole transfer switch that prevents a solid neutral connection from service equipment to generator. For "separately derived systems," grounding requirements of Section 250-5(d) apply only where the generator has no direct electrical connection, including a solidly grounded circuit conductor to the normal service. The rule of grounding applies to a generator that feeds its load without any tie-in through a transfer switch to any other system, but does not apply to one with a solidly connected neutral from it to the service through a three-pole, solid neutral transfer switch.

Section 250-27(b) requires that a neutral that might function as an equipment grounding conductor have cross-sectional area of at least 12.5% of the cross-sectional area of the largest phase conductor of the generator circuit to the transfer switch.

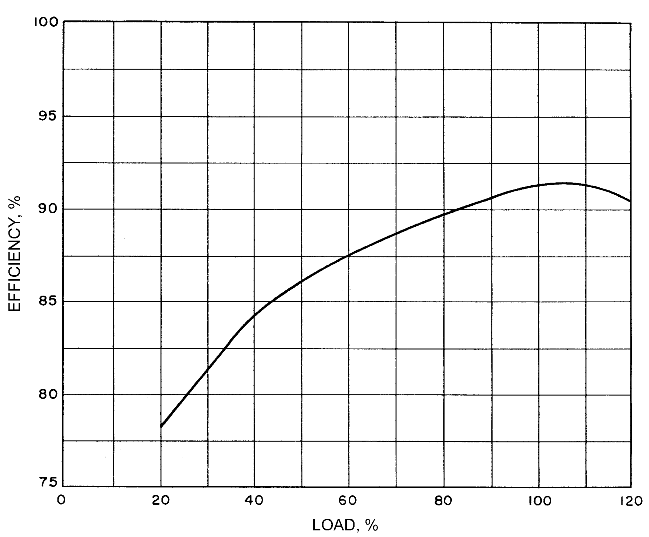

Fig. 62 Typical Generator Efficiency

Control requirements for systems that provide electricity and heat for equipment and electronic processors differ, depending on the number of energy sources and 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 19 shows the control functions required for systems isolated from and in parallel with the utility grid, with single or multiple prime movers. Frequency and voltage are directly controlled in a single-engine isolated system. 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. Power output is determined by the throttle setting, which responds to 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 way to control 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) are typically air-type, and circuit breakers for medium voltage (up to 12,000 V) should be vacuum-type. The Institute of Electrical and Electronics Engineers' (IEEE) *Standard* 1547 defines interconnection requirements for a paralleling local generator and the utility's electrical power system, and describes specific design, operation, and testing requirements for interconnecting generators below 10 MW and typical radial or spot network primary or secondary electrical distribution systems.

In grid-isolated operation, 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 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.

Table 19 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)

SYSTEM DESIGN

CHP ELECTRICITY-GENERATING SYSTEMS

Good CHP planning responds to the end user's requirements and strives to maximize use of the equipment and the energy it produces. For CHP 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.

Thermal Loads

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 back-up 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.

Gas turbines can provide a larger quantity and better quality of heat per unit of power, whereas extraction steam turbines can provide even greater flexibility in both quantity and quality (temperature and pressure) of heat delivered.

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.

Prime Mover Selection

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 condition seldom occurs over extended periods.

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

- Match the prime mover's heat/power ratio to that of the user's hourly load profiles.
- Store excess power as chilled water or ice when thermal demand exceeds coincident power demand.

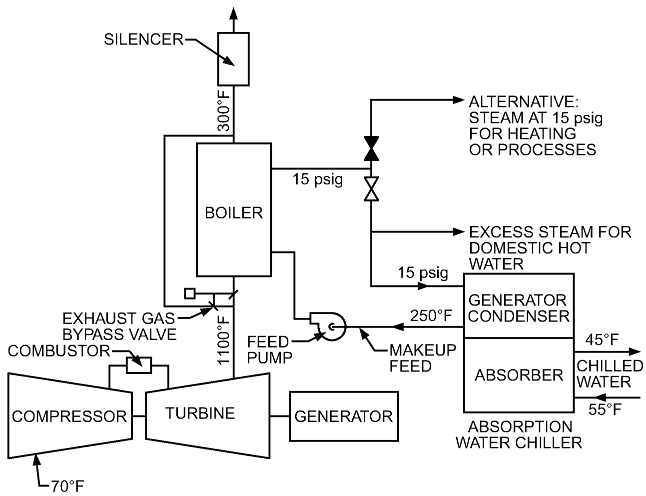

Fig. 63 Typical Heat Recovery Cycle for Gas Turbine

- Store excess thermal production as heat when power demand exceeds 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 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 be preferred. Figure 63 shows a typical heat recovery cycle.

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/LHV 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 temperatures/pressures are available, but only from an exhaust gas recovery system, separated 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 27 to 34% efficient, with 66 to 73% of their fuel energy released into the exhaust. For a typical case with 70% of the fuel energy in the exhaust, assuming that approximately 55% of this energy is recoverable (with 300°F final exhaust gas temperature), net result is an overall thermal efficiency of 69% (excluding power required to drive auxiliary equipment such as fans and pumps to recover and use the exhaust heat).

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.

Air Systems

Large, central air handlers with deep and/or suitably circuited coils that operate with a large cooling and heating Δt (e.g., 24°F) are available to reduce distribution piping and pumping costs. These units serve multiple 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 CHP 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 intermediate seasons. Chilled water is available to cool any zone. Zones needing heat can cut off the coil and turn on 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 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 CHP 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 CHP 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. See Chapter 12 for more information on hydronic system design; information on various types of terminals and systems may be found in Chapter 5. 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 a CHP design standpoint, all these applications are relevant, but some are not HVAC applications. Each application may offer an opportunity to improve the CHP 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 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 CHP. 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 CHP 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 heat pump is even more effective if there is 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 CHP plant has 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.

Service Water Heating

Service hot-water systems can be a major and preferred user of thermal energy from prime movers, and 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 variable in a 24 h weekday or 48 h weekend profile; heat storage allows 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.

District Heating and Cooling

The high cost of a central plant and distribution system generally mandates that significant 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.

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 grid-parallel with the utility is determined by either heat recovery requirements or power requirements, whichever govern, and its exciter current, which is set by the reactive power flow through the interconnection. When 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. The Institute of Electrical and Electronics Engineers (IEEE) *Standard* 1547 establishes requirements for operating the distributed generation (DG) in parallel with the grid both for normal and fault conditions. Depending on the design, the system may be able to continue operating while isolated from the grid, providing power to some or all of a facility's electrical loads as an intentional island. A key concern of this standard is to prevent **islanding** (a situation in which generator closely matches the local and/or neighboring load). Anti-islanding precautions are both for safety and protection.

With 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, reverse power, 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 which point, it either speeds up or slows down. This allows the over- or underfrequency device to sense loss of source and 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.

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.

Output Energy Streams

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 CHP 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 CHP 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 Utility Interfacing 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 than for 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 equipped with supplemental firing 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. Chapters 10 and 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 or desiccant dehumidifiers 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 CHP plant. If the buildings already have one or more types of chillers in good condition, CHP and chilled water distribution may have diminished economic prospects unless applied on a small scale to individual buildings.

If chilled-water distribution is feasible, central CHP 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; Becker 1975; Mannion 1988).

Cost-reduction concepts for variable-flow water systems include the following:

- Let the main pump(s) and primary distribution system flow rate match the instantaneous sum of the demand flows from all cooling coils served by the primary loop. 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 is necessary for successful implementation of these concepts. The *Air-Conditioning Systems Design Manual* (Lorsch 1993) has further design information.

Hot Water. CHP thermal output is well adapted to low-, medium-, or high-temperature water (LTW, MTW, HTW) distribution systems. The major difference between chilled- 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, beyond 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 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. Case-by-case analysis of applying heat in other thermally activated technologies is helpful, because usefulness depends on specific economic conditions.

For both hot- and chilled-water distribution systems, a common approach is to lower the hot-water supply temperature as ambient temperature rises and to raise the 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 effect of increased pumping costs is often overlooked.

Below some part-load condition in both hot- and chilled-water systems, the cost of reducing flow by varying pump speed and air volume may be more than the energy savings. 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 factors need to be examined in determining the part-load condition at which hot- and chilled-water system scheduling might be advantageously modified.

CHP SHAFT-DRIVEN HVAC AND REFRIGERATION SYSTEMS

Engine-Driven Systems

HVAC Equipment. Engine-driven chillers are rated according to ARI *Standard* 550/590 conditions. Manufacturers offer performance curves for other conditions. As with any chiller, performance is largely a function of design conditions for the condenser and chilled-water supply temperatures. The engine size required for a given chiller capacity is typically 1 hp per ton of cooling.

Table 20 Coefficient of Performance (COP) of Engine-Driven Chillers

Heat Recovery Option	COP at Full Load
No heat recovery	1.2 to 2.0
Jacket water heat recovery	1.5 to 2.25
Jacket water and exhaust heat recovery	1.7 to 2.4

Table 20 provides a typical range of engine-driven chiller coefficient of performance (COPs) with and without heat recovery. The COP is cooling energy output divided by fuel energy input. Engine fuel input is based on HHV of the fuel.

Heat recovered from the jacket coolant and exhaust gas is added to the cooling load produced by the chiller, increasing useful thermal output and the COP. Because no standards exist for calculating the COP of an engine-driven chiller when considering heat recovery, most manufacturers present COPs with and without heat recovery.

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 low operating cost and high flexibility by combining speed variation with cylinder unloading. Part-load capacity is controlled by modulating engine speed down to about 30 to 50% of rated speed, which improves fuel economy. Some reciprocating engine chillers also use cylinder unloading to reduce capacity further. 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., 4 out of 12) unload; in this case, a bypass valve must be installed for a fully unloaded start. The engine first reaches one-half or two-thirds of full speed. Then, a gradual cylinder load is added, and 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 64 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.

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 packages because of their compact size and mass. These units may be equipped with manual or automatic start/stop systems and engine speed controls.

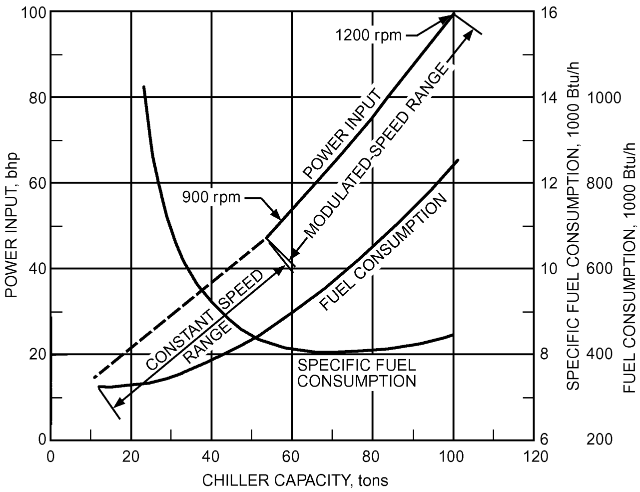

Fig. 64 Performance Curve for Typical 100 Ton, Gas-Engine-Driven, Reciprocating Chiller

Larger open-drive centrifugal chillers are usually field-assembled and include a compressor mounted on an individual base and coupled by 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 affect the best compromise between the initial cost of the equipment (engine, couplings, and transmission) and maintenance cost, engine speeds between 900 and 1200 rpm are generally used. Engine output can be modulated by reducing engine speed. If 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 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.

Heat Pumps. An additional economic gain can result from operating an engine-driven refrigeration cycle as a heat pump, if 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 CHP 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.

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 coefficient of performance is really defined for the cycle that captures jacket and exhaust heat.

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 engine speed and adjusting the slide valve on the compressor. Units have an ECOP near 1.45 at rated cooling load. Chapter 37 has more information on compressors.

Because refrigeration equipment operates at low evaporator temperatures (20 to –70°F), refrigerants such as ammonia and other cycles that improve efficiency over single-stage cycles are used. Besides the standard, single-stage vapor compression cycle, a multistage or cascade refrigeration cycle may be chosen. The multistage cycle is the most common cycle used to efficiently provide refrigeration from –10 to –70°F. The section on Compression Refrigeration

Table 21 Typical Efficiency of Engine-Driven Refrigeration Equipment (Ammonia Screw Compressor)

	Saturation Suction Temperature/ Saturation Discharge Temperature		
	–40/95°F	–12/95°F	20/95°F
Electric COP	1.32	2.66	4.62
Engine-driven COP without heat recovery	0.44	0.78	1.32
Engine-driven COP with jacket water heat recovery	0.74	1.08	1.62
Engine-driven COP with jacket water and exhaust heat recovery	0.89	1.23	1.77

Source: AGCC (1999).

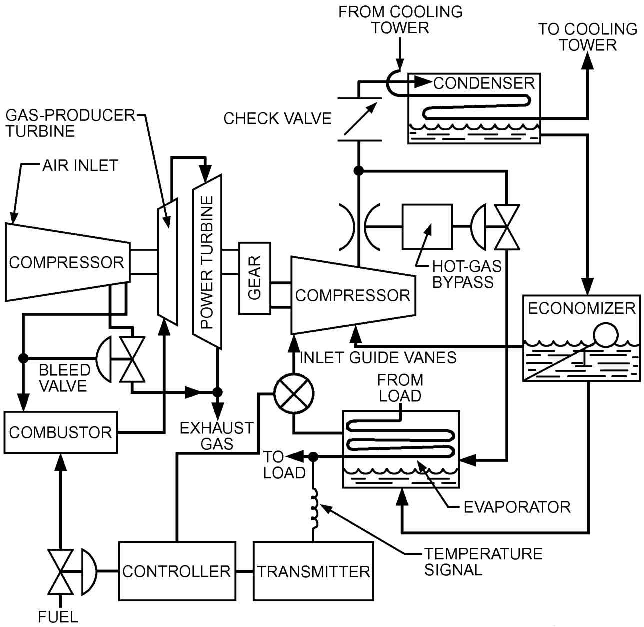

Fig. 65 Typical Gas Turbine Refrigeration Cycle

Cycles in Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals* describes this cycle.

As with engine-driven chillers, heat recovery from the jacket coolant and exhaust gas boosts overall energy use and efficiency. Table 21 lists the coefficient of performance and effect of heat recovery for a range of conditions.

Combustion-Turbine-Driven Systems

The gas turbine has become increasingly important 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 65 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, manufacturers of split-shaft engines can provide the power turbine with either direction of rotation.

At low loads, both the gas turbine unit and 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 using a hot-gas bypass valve.

The turbine manufacturer normally includes automatic surge protection, either as a bleed valve that bypasses some 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 turbine's output shaft runs at the speed required by the refrigeration compressor. The temperature of chilled water or brine leaving the cooler of the refrigeration machine controls the fuel. See also the section on Fuels and Fuel Systems.

Steam-Turbine-Driven Systems

Steam turbines in air conditioning and refrigeration are mainly used to drive centrifugal compressors, which are usually part of a water or secondary-coolant chilling system using one of the newer or halogenated hydrocarbon refrigerants. In addition, many industrial processes use turbine-driven centrifugal compressors with various other refrigerants such as ammonia, propane, 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 CHP 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

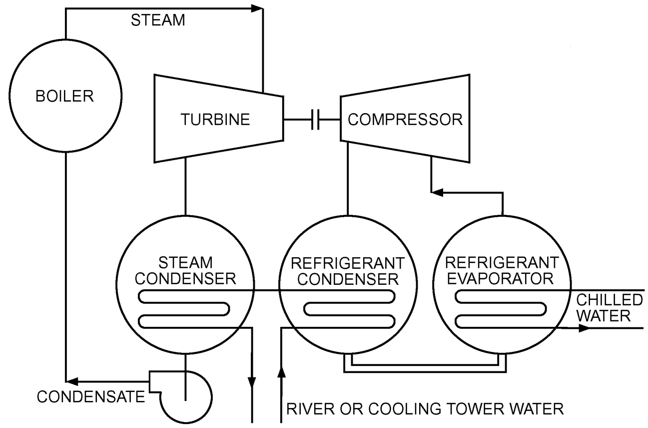

Fig. 66 Condensing Turbine-Driven Centrifugal Compressor

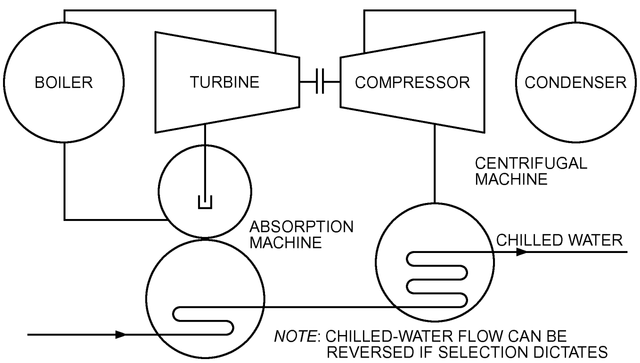

Fig. 67 Combination Centrifugal-Absorption System

multitude of buildings with heating and cooling also use steam-turbine-driven equipment.

Most steam turbines driving centrifugal compressors for air conditioning are multistage condensing turbines (Figure 66), which provide 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. As an example of the wide range of application of this turbine, one industrial firm 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 67). 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 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 design of combined systems is balancing turbine exhaust steam flow with 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.

CODES AND INSTALLATION

GENERAL INSTALLATION PARAMETERS

Structural support for the equipment must be adequate for the operating weight and any inertial forces.

Depending on the fuel(s) chosen for the system, it may be important to address location and capacity of **fuel supply** and storage.

Ventilation is required to supply adequate combustion air and prevent plant overheating.

Exhaust systems must be constructed of materials resistant to corrosion by exhaust gases, and condensate drains are usually required. Locate exhaust outlets carefully to avoid noise and pollution problems.

Monitoring systems should allow operators to monitor CHP performance for maintenance and economic purposes. As a minimum, the following should be monitored:

- CHP run time
- Fuel consumption
- Electrical power generated in each period (minus parasitic losses)
- Usable heat produced
- Duration and cause of any plant failure
- Cost of fuels, maintenance, etc.

Acoustic attenuation is often necessary for CHP units. Packaged plants usually come in a purpose-built acoustic enclosure. In addition, silencers must be fitted to the exhaust system. Antivibration mountings and couplings are usually standard.

Commissioning and **testing** are key. The plant must be tested under various loads, according to the manufacturer's instructions. Pumps, electrical switchgear, and controls need careful commissioning to ensure they operate as designed. Test that electrical output can be synchronized, paralleled, and disconnected safely. Where installed, heat rejection systems should be tested across a range of loads to establish that sufficient heat can be rejected and overheating avoided. Also, ensure that the CHP unit interacts correctly with the existing heating system.

At project completion, proof of commissioning and operating instruction documentation must be provided. In particular, the CHP system must be included in the building log book required by building regulations. Permits and approvals also should be handed over, and appropriate staff trained to monitor the plant. Most of this is usually done by the CHP supplier, but is ultimately the client's responsibility.

It is important to monitor the immediate and ongoing system performance. As soon as operation begins, detailed monitoring is essential to confirm efficiency and economic viability. Continuous monitoring is required to maintain long-term performance, particularly as fuel costs and electricity prices change.

UTILITY INTERCONNECTION

Requirements for interconnection with public utilities' grids vary by cogenerator and by the individual electric utility, depending on generation equipment, size, and host utility systems. Interconnection equipment requirements increase with generator size and voltage level. Generally, complexity of the utility interface depends on the mode of transition between paralleling and stand-alone operation. The plant connection to the electric grid must have an automatic utility tie-breaker and associated protective relays. When utility power is lost, this tie-breaker opens and isolates the generator and its loads from the utility. Protective relays at the entrance and generator must be coordinated so that the utility tie-breaker opens before the generator's breaker. An automatic load control system is needed to shed noncritical loads to match generator's capacity. When utility power returns, the generator must be synchronized with the utility across the utility tie-breaker, whereas under normal start-up conditions the generator is synchronized across the generator circuit breaker. The synchronizing equipment must accommodate both situations.

When a CHP system is integrated into the utility system, the following issues must be understood and managed:

- Control and monitoring
- Metering
- Protection
- Stability
- Voltage, frequency, synchronization, and reactive compensation for power factors
- Safety
- Power system imbalance
- Voltage flicker
- Harmonics
- Grounding

Many jurisdictions have developed standard interconnections requirements, most of which are based on IEEE *Standard* 1547 and UL *Standard* 1741. IEEE *Standard* 1547 addresses performance, operation, testing, and safety of interconnection products and services, such as hardware and software for distributed power control and communication. It also discusses product quality, interoperability, design, engineering, installation, and certification, and provides a platform for standardized interconnection throughout the United States.

AIR PERMITS

Basic air permitting and emission control requirements are as follows:

- De minimis exemptions
- State minor source permitting
- Major source permitting
- Emergency generators

Most states have a **de minimis exemption**, a threshold below which units are either small enough or have emissions low enough that they do not have to apply for a permit of any kind. Requirements and conditions vary by state. Even when the CHP system does not require a construction permit, it may be advisable to file an information copy with the permitting authority.

Sources that are not exempted must obtain a permit. An important factor in determining how a source is permitted is its **potential to emit**, which is the measure of its maximum possible emissions if operated at full capacity for 8760 h per year. If a source's potential emissions exceed certain thresholds, it is considered a major source and, in the United States, is subject to federal new source review permitting. The trigger threshold depends on local air quality.

Intermediate-level sources are generally subject to state minor source permitting.

Both minor and major source permits are likely to require some kind of emission limitations or controls, which could be anything from raising a unit's stack height to installing the most stringent control technologies available. The permitting process also can range from a simple application to a complex cost-based technology evaluation. Requirements vary by state and the type of unit proposed.

In addition, most states have special treatment for emergency back-up generators. The EPA recommends calculating emergency units' the potential to emit based on 500 h of operation per year.

Note that air permits are required before any construction can begin.

BUILDING, ZONING, AND FIRE CODES

U.S. building design and construction codes (except for federal buildings and facilities) are typically developed at the state and local level. Because power generation in the United States has typically been regulated from within the utility industry, few provisions exist in standards, codes, and building construction regulations for CHP systems; in fact, many codes explicitly exempt facilities used exclusively for generating electricity. As a result, there is very little regulatory guidance for power generation technologies and installations, and most code officials are not experienced in design, construction, installation, or operation of power plants, and particularly generators serving commercial buildings.

Important considerations include the following.

Zoning

Zoning regulations are locally implemented and enforced, but typically are based on national standards. Topics addressed range from noise regulations to visual impact. Blanket prohibitions on electricity generation, limits on operation of back-up generators, and height restrictions on towers for wind generators, are not uncommon. In these cases, the developer must obtain an exception, which increases the expense and time needed for approval.

Building Code/Structural Design

The site must structurally support the system load and vibration. If a unit is installed within a building, the whole building may need to be reclassified for occupancy. The unit cannot block egress from the building.

Mechanical/Plumbing Code

Some localities do not allow installation of gas-fired units not listed as gas appliances by an approved agency. In these cases, a registered engineer must certify that the installation meets all applicable standards and is in safe operating condition, and must seek an exemption. Codes also regulate use of gas meters, piping and emergency shutoff valves.

Fire Code

Fire prevention and firefighters' access to equipment during an emergency are strongly regulated by code rules. The type of fuel (natural gas, oil, propane, etc.) used must be determined, and its flammability and combustibility analyzed. The fire department must know a unit's location and how to disconnect it in case of a fire. A fire suppression system may be necessary for buildings with interior CHP systems.

Applicable codes and standards of the National Fire Protection Association (NFPA) include the following:

- *Standard* 850, Electric Generating Plants
- *Standard* 37, Installation and Use of Stationary Combustion Engines and Gas Turbines, which also covers temporary, portable supplementary engines
- *Standard* 54, National Fuel Gas Code
- *Standard* 70, National Electrical Code®

Electrical Connection

The most important electrical decision is the point of connection. Connection on the line side creates the fewest design problems. When connecting on the load side, the service equipment's ampacity, ground fault protection, and system neutral bonding must be considered.

ECONOMIC FEASIBILITY

ECONOMIC ASSESSMENT

The economics of CHP systems is evaluated based on the costs of equipment, installation, operation (fuel and maintenance), and capital compared to the cost of grid electricity and fuel for meeting thermal loads. With this information, CHP can be compared to a conventional system on the basis of energy cost. Once a utilization factor has been established, the CHP system can be evaluated on the basis of simple payback. For systems with dual-mode capability (grid parallel and grid islanding), the value of avoided power outages should also be included.

Some CHP analysis can be performed with as little as the last 12 months' utility bills, but the more site information that can be obtained, the more accurate the analysis.

Understanding the facility and its needs is essential. For a new building, this includes the building design and energy model. For an existing building, a site visit is necessary to gather information on energy usage, utility costs, operating schedules, electric service, and existing heating and cooling equipment size, configuration, and location. During a site walk-through, the following questions may be relevant:

- Can important infrastructure issues (e.g., lack of a centralized, facility-wide distribution system; lack of space for CHP equipment near the central plant; presence of many electricity meters) be handled at reasonable expense?
- Are there other justifications for a CHP system beyond cost savings by reducing energy bills? Examples include the need to add back-up power, and an unreliable existing power supply.
- Are there any specific financial opportunities that make CHP more attractive? Examples include a 12-month heating load that is large compared to the heating output of the potential generating package, or a planned retirement or upgrade of the existing heating and cooling equipment.

Two project assessment levels are discussed here: simple CHP screening, and feasibility analysis.

Simple CHP Screening Analysis. This level may be adequate to provide a quick understanding of whether a site is suitable for CHP, but investment decisions are not normally made as a result.

Energy consumption and utility rate information gathered during the walk-through are used in a rule-of-thumb evaluation, including the type and approximate size of the CHP system, a rough first cost of the CHP system, and the estimated range of annual energy savings from CHP. Equipment costs are estimates, and their accuracy often depends on the analyst's level of experience.

After obtaining estimates of the energy savings and first cost, an estimated simple payback for the CHP system can be calculated. A packaged screening analysis tool, such as a computer program or a spreadsheet using an established methodology, is recommended. This analysis should be inexpensive. Averaged costs for electricity, particularly if there are demand charges, ratchet and/or time of day rates, should not be used in economic evaluations.

CHP Feasibility Analysis. This more detailed level of analysis includes three steps: energy analysis, conceptual development, and financial pro-forma analysis.

The **energy analysis** should be based on engine or turbine net useful heat output. Generally, most CHP systems cannot recover more than 60 to 70% of the engine's net useful output for most commercial applications (although hospitals and some industrial recovery may be up to about 85%). Note that operating CHP systems when off-peak rates apply (i.e., night, weekends, and designated holidays) may not be beneficial. Usually, operation should focus on on-peak utility rate hours.

Consider the size of the CHP system compared to the facility's peak load. Find the shortest CHP paybacks by running repeated financial analyses across practical system size ranges. In general, for commercial-class buildings applications, the best payback occurs when the CHP system is 40 to 60% of the facility's peak electric load. A 100% (island) system should only be considered if needed by the owner.

The analysis should be run both with and without using heat rejected from the CHP system for water and space heating and for absorption cooling, unless the application has a specific difficulty with any of these technologies. Desiccant dehumidification should also be considered.

Normalize the load (electric and thermal) characterization to the local average weather year. This is particularly important for commercial buildings with loads dominated by space heating and cooling, and if the load was developed using only the previous year's utility consumption and bills, which may not reflect usage in an average year and could introduce error in extended (five- to seven-year) forecasts.

Provide a detailed list of all assumed first costs, as well as costs for electricity and natural gas (both currently, and reasonable projections over the next five to seven years). Standby charges from the local electric utility should also be addressed, and are usually based on the CHP system's total capacity or that of the largest single generator in the CHP system, unless another solution for engine outages is used.

The **conceptual development analysis** should provide one-line and block diagrams of electrical and mechanical layouts, as well as the major planned components and issues noticed during the walk-through. Specifics, such as type and size of equipment, should be included.

The **financial pro forma analysis** should show the following, on a year-by-year basis:

- Initial and additional out-year investments, if any.
- Engine generator maintenance allocations, with charges based on projected equipment operating hours.
- Energy cost savings.
- Capital repayment and carrying charges.
- Depreciation.
- Tax effects (can be zeroed out for nontaxable owners)
- Internal rate of return (IRR) on the investment, without leverage; this includes all annual energy cost savings, maintenance costs, and tax savings, as well as the initial cost of the CHP system (as a single lump sum outflow in year zero). It should not include principal or interest payments on financing.

Do not use fuel or electricity price escalators or leveraged rates of return in the financial analysis. Projected rates need to be considered, particularly when their rates diverge. Most financial decision makers require this analysis.

PRELIMINARY FEASIBILITY BIN ANALYSIS EXAMPLES

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

First Estimates

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

The economic coefficient of performance (ECOP) is a methodology that enables each energy stream to be valued on a comparable economic basis and at prevailing rates, with 1 kWh of electrical energy taken as 3412 Btu and fuel input as the high heating value (HHV) in Btu. Then, a direct comparison 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 \$5.50 per million Btu (HHV of 1000 Btu/ft^3), 1000 Btu of electrical energy costs \$0.023 (0.08/3.412), and 1000 Btu of natural gas costs \$0.0055. 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 (fuel input based on HHV), again converted in terms of economic costs of each energy stream. For this example, the electrical energy costs (0.023/0.0055) = 4.18 times more than the equivalent energy from natural gas. This ratio is used to calculate the ECOP in the following examples.

Example 6. 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:

$$\text{On-site fuel input} = 19(1164 - 180)/0.8 = 23{,}400 \text{ Btu/ton}\cdot\text{h}$$

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

$$\begin{aligned}\text{Off-site electrical input} &= (0.746 \text{ kW/hp} \times 0.025 \text{ hp/ton})/0.87\\ &= 0.0214 \text{ kW input/ton}\\ &= 3412 \times 0.0214 = 73 \text{ Btu/ton}\cdot\text{h}\end{aligned}$$

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

$$23{,}400 + (73 \times 4.18) = 23{,}705 \text{ Btu/ton}\cdot\text{h(equivalent energy)}$$

Thus, the ECOP for 12,000 Btu/ton·h output and 23,705 Btu/ton·h equivalent input (at the preceding power costs) is

$$12{,}000/23{,}705 = 0.506$$

Example 7. 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). Heat rate in this example is based on HHV.

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 6, is 73 Btu/ton·h. The equivalent total input energy is

$$8600 + (73 \times 4.18) = 8905 \text{ Btu}$$

Thus, the ECOP for above is 14,227/8905 = 1.598, which is more than three times that of the conventional system covered in Example 6.

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.

Load Duration Curve Analysis

A much more comprehensive energy analysis, combined with an economic analysis, must be used to select a CHP system that maximizes 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 facility's electrical and thermal loads, as well as the size and fuel consumption of the prime mover.

Accurate analysis is especially important for commercial and institutional CHP applications because of the large time-dependent changes in magnitude of load and the noncoincident 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 HVAC system performance is the bin method. The basic tool for sizing and evaluating power system performance is the load duration curve, which contains the same information as bins, but arranges the load data 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 spreadsheets or databases. The load duration curve produces a visually intuitive tool for sizing CHP systems and for accurately estimating system performance.

Figure 68 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 69 is a load duration curve (a descending-order sort) of the steam load data in Figure 70. 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 base load and peak load can be readily identified. Note that the practical base load 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.

Sizing a cogenerator at 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 CHP plant might be sized somewhat larger than base load to minimize payback time 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. CHP systems sized this way produce high equipment utilization and depend on the utility to serve peak loads above 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 equations for rectangles and triangles or an appropriate curve analysis program. For a thermally tracked plant sized at the 21,000 lb/h baseload, the hours of operation are about 93 h per week. Therefore, based on the area under the rectangle, the total steam produced by the CHP plant is

$$\begin{aligned}\text{Cogenerated steam} &= (93 \text{ h per week})(21 \times 10^3 \text{ lb/h}) \\ &= 1.953 \times 10^6 \text{ lb per week}\end{aligned}$$

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

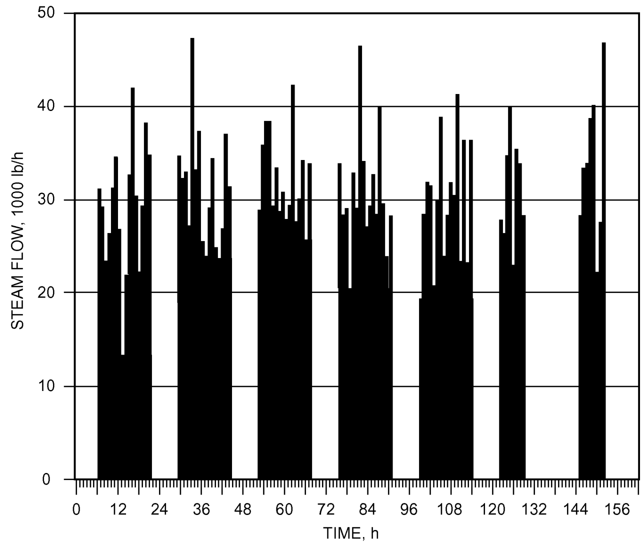

Fig. 68 Hypothetical Steam Load Profile

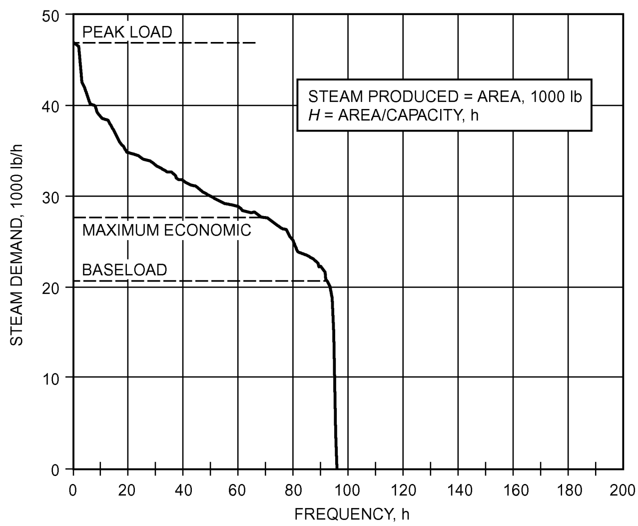

Fig. 69 Load Duration Curve

the facility's steam load profile is unable to absorb the steam produced at continuous full power, electrical output can be reduced accordingly to avoid steam waste.

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 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

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

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

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 CHP production in this base-load sizing and operating mode is 93 h per week, 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

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

For this larger size, there is a higher electrical and lower boiler production, but some 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 above the baseload is not as simple as for the base-load design because changes in the prime mover's performance at part load must be considered. In this case, average value estimates of 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 70 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 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.

The advantages of multiple units are offset by 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 CHP because heat recovery is rarely justified. Figure 71 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 control system can anticipate when facility demand will exceed 1400 kW in time to bring the generator up to operating condition. The peak shaver should not 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's monthly peak occurs, the ratchet is set for a year hence, and the demand savings potential of the 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 facility's standby requirements. 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 as when 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 (i.e., it operates in electrical tracking mode).

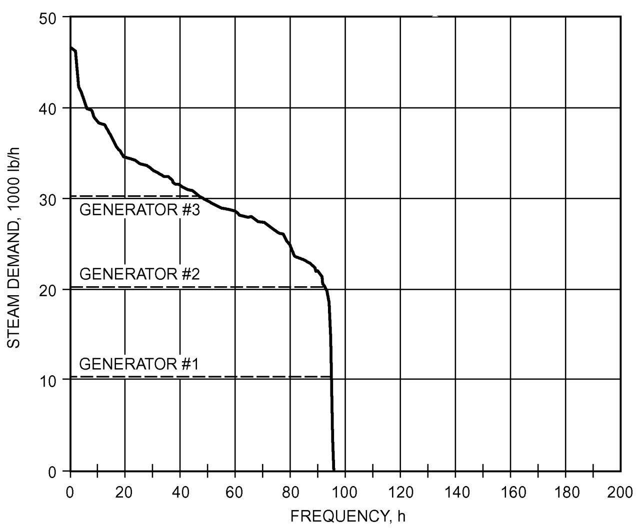

Fig. 70 Load Duration Curve with Multiple Generators

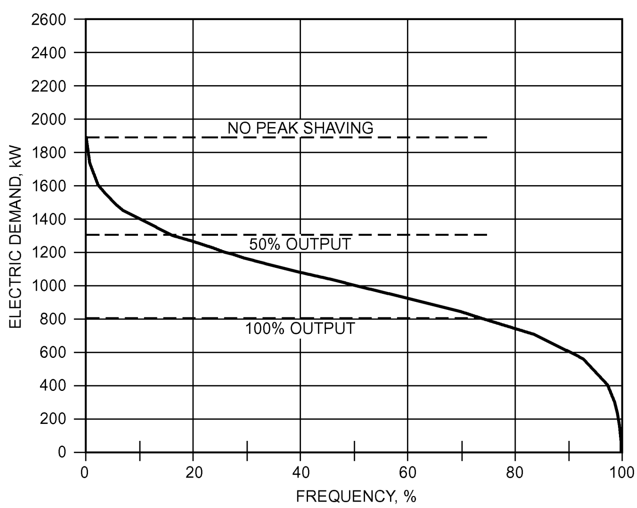

Fig. 71 Hypothetical Peaking Generator

To develop the two-dimensional load duration curve method for simultaneous loads, either the electrical or steam demand must be broken into discrete periods defined by the number of hours per year when the 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 72 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 72 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 analysis becomes burdensome when there are large load swings and several rate periods, the calculations are run by computers.

Analysis by Simulations

The load duration curve is a convenient, intuitive graphical tool for preliminary sizing and analysis of a CHP 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: 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.

The fact that prime movers rarely operate at full rated load further complicates evaluation of commercial systems; 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.

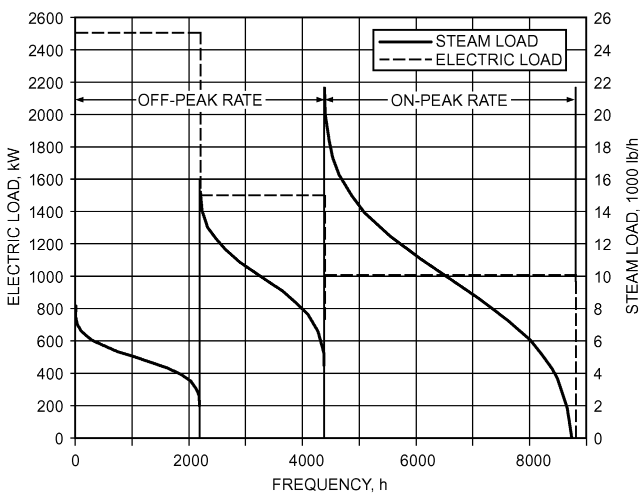

Fig. 72 Example of Two-Dimensional Load Duration Curve

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 CHP system may include absorption chillers that use jacket and exhaust heat to produce chilled water at two different COPs. Some commercial 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 screening programs that analyze CHP systems are readily available. Many of these programs emphasize the financial aspects of CHP with elaborate rate structures, energy price forecasts, and economic models. Other programs emphasize equipment part-load performance, load schedules, and other technical characteristics. Although these programs can be useful, they do not remove the need for detailed analysis before undertaking a project.

The primary consideration in selecting a computer program for analyzing commercial CHP 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

CHP simulation using hourly averaged values for load representation provides the greatest accuracy; however, the 8760 values for each type of load can make managing 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; accuracy is greatest 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 developing hourly values for each load over a typical average day of each month. Usually, two typical days are considered: working and nonworking. 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 CHP computer programs. Pedreyra (1988) and Somasundaram (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 CHP analysis if more than two loads must be considered.

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