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A Cooling, Heating, And Power For Buildings (Chp-B) Instructional Module

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A COOLING, HEATING, AND POWER FOR BUILDINGS

(CHP-B) INSTRUCTIONAL MODULE

By

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An emerging category of energy systems, consisting of power generation equipment coupled with thermally-activated components, has evolved as Cooling, Heating, and Power (CHP). The application of CHP systems to buildings has developed into a new paradigm – Cooling, Heating, and Power for Buildings (CHP-B). This instructional module has been developed to introduce undergraduate engineering students to CHP-B. In the typical ME curriculum, a number of courses could contain topics related to CHP. Thermodynamics, heat transfer, thermal systems design, heat and power, alternate energy systems, and HVAC courses are appropriate for CHP topics. However, the types of material needed for this mix of courses vary. In thermodynamics, basic problems involving a CHP flavor are needed, but in an alternate energy systems course much more CHP detail and content would be required. This series of lectures on CHP-B contains both a stand-alone CHP treatment and a compilation of problems/exercises.

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NOMENCLATURE

W_{Otto} = net work accomplished by the Otto cycle
m = mass
u = specific internal energy
T = temperature
c_v = constant volume specific heat
Q_{in} = heat addition during the Otto cycle
η_{Otto} = thermal efficiency of the Otto cycle
V = volume
r = compression ratio
P = pressure
k = ratio of specific heats
h = specific enthalpy
$Q_{D,in}$ = heat addition during the Diesel cycle
$Q_{D,out}$ = heat rejection during the Diesel cycle
W_{Diesel} = net work accomplished by the Diesel cycle
c_p = constant pressure specific heat
r_c = cutoff ratio
η_{Diesel} = thermal efficiency of the Diesel cycle
W_c = compressor work accomplished during the Brayton cycle
W_t = turbine work accomplished during the Brayton cycle
Q_s = heat addition during the Brayton cycle
W_{net} = net work accomplished during the Brayton cycle
η = thermal efficiency of the Brayton cycle
η_c = isentropic compressor efficiency
η_t = isentropic turbine efficiency
W_{comp} = compressor work accomplished during the gas turbine cycle
W_{turb} = turbine work accomplished during the gas turbine cycle
Q_{out} = heat rejected in the condenser
Q_{in} = heat added in the evaporator
W_{in} = compressor work
W'_{in} = pump work
Q'_{in} = heat added in the generator
Q'_{out} = heat rejected in the absorber
$\dot{m}$ = mass flow rate
i = enthalpy
x = concentration
$\dot{q}$ = rate of heat exchange
ε = effectiveness of heat exchange
$\dot{W}_P$ = power requirement for pump operation
p = pressure
η_p = pump efficiency
ν = specific volume
COP = coefficient of performance

CHAPTER 1

INTRODUCTION TO COOLING, HEATING, AND POWER FOR BUILDINGS

The traditional model of electric power generation and delivery is based on the construction of large, centrally-located power plants. "Central" means that a power plant is located on a hub surrounded by major electrical load centers. For instance, a power plant may be located close to a city to serve the electrical loads in the city and its suburbs or a plant may be located in the midpoint of a triangle formed by three cities.

Power must be transferred from a centrally-located plant to the users. This transfer is accomplished through an electricity grid that consists of high-voltage transmission systems and low-voltage distribution systems. High-voltage transmission systems carry electricity from the power plants to substations. At the substations, the high-voltage electricity is transformed into low-voltages and distributed to individual customers.

Inefficiencies are associated with the traditional method of electric power generation and delivery. Figures 1.1 and 1.2 illustrate the losses inherent to the generation and delivery of electric power in traditional power plants and in combined cycle power plants. Traditional power plants convert about 30 % of the fuel’s available energy into electric power, and highly efficient, combined cycle power plants convert over 50 % of the available energy into electric power. The majority of the energy content of the fuel is lost at the power plant through the discharge of waste heat. Further energy losses occur in the transmission and distribution of electric power to the individual user. Inefficiencies and pollution issues associated with conventional power plants provide the impetus for new developments in "onsite and near-site" power generation.

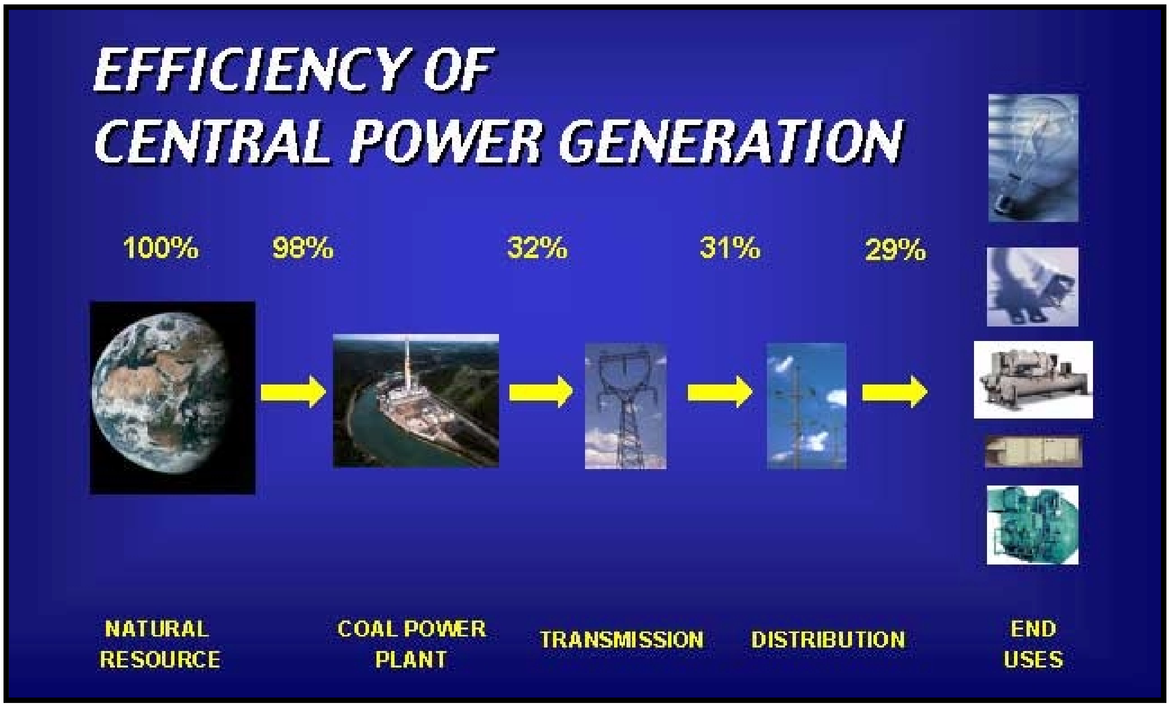


Figure 1.1: Efficiency of Central Power Generation (www.bchp.org)

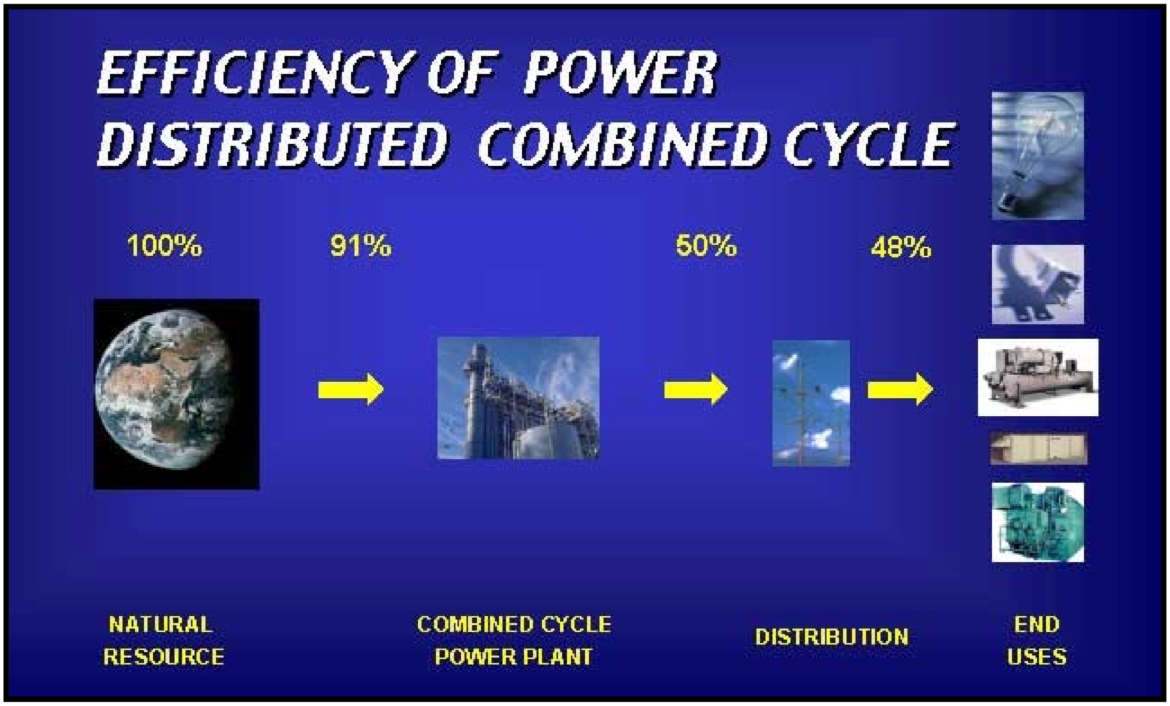


Figure 1.2: Efficiency of Power Distributed Combined Cycle (www.bchp.org)

The traditional structure of the electrical utility market has resulted in a relatively small number of electric utilities. However, today's technology permits development of smaller, less expensive power plants, bringing in new, independent producers. Competition from these independent producers along with the re-thinking of existing regulations have affected the conventional structure of electric utilities.

The restructuring of the electric utility industry and the development of new "onsite and near-site" power generation technologies have opened up new possibilities for buildings, building complexes, and communities to generate and sell power. Competitive forces have created new challenges as well as opportunities for companies that can anticipate technological needs and emerging market trends.

Historically research, development, and commercialization efforts have been focused on individual systems (cooling, thermal storage, ventilation air, and power). A new category consisting of power generation equipment coupled with thermally-activated components has evolved as Cooling, Heating, and Power (CHP). A successful CHP system requires a need for both generated electric/shaft power and thermal energy. An operation that does not have a need for both electric/shaft power and thermal energy will not likely benefit from CHP. CHP is especially beneficial to buildings, which typically use electric power and can have thermally-activated HVAC system components.

The application of CHP systems to buildings has developed into a new paradigm – Cooling, Heating, and Power for Buildings (CHP-B). CHP-B focuses on onsite fuel conversion, combining power generation and HVAC system optimization and integration with other innovative building technologies. CHP-B systems consist of integrated power generation equipment (gas turbines, microturbines, internal combustion engines, and fuel cells), thermal systems (water chillers, absorption chillers, air conditioners and refrigeration – electric or engine driven, boilers, and thermal storage), ventilation/IAQ systems (desiccant, enthalpy, and other energy recovery devices), and building control and system integration technologies.

Cooling, Heating, and Power for Buildings has the potential to reduce carbon and air pollutant emissions and to increase resource energy efficiency dramatically. CHP-B produces both electric or shaft power and useable thermal energy onsite or near site, converting as much as 80 % of the fuel into useable energy. A higher efficiency in energy conversion means less fuel is necessary to meet energy demands. Also, onsite power generation reduces the load on the existing electricity grid and infers better power quality and reliability. Figure 1.3 illustrates the increase in efficiency of CHP-B systems over the power plant efficiencies seen in Figures 1.1 and 1.2. Additionally, CHP-B systems include values such as variable fuel requirements, enhanced energy-security, and improved indoor air quality. CHP-B holds answers to the efficiency, pollution, and deregulation issues that the utility industry currently faces. The majority of information presented in this introduction and in the following chapter on the CHP-B system originated at www.bchp.org.

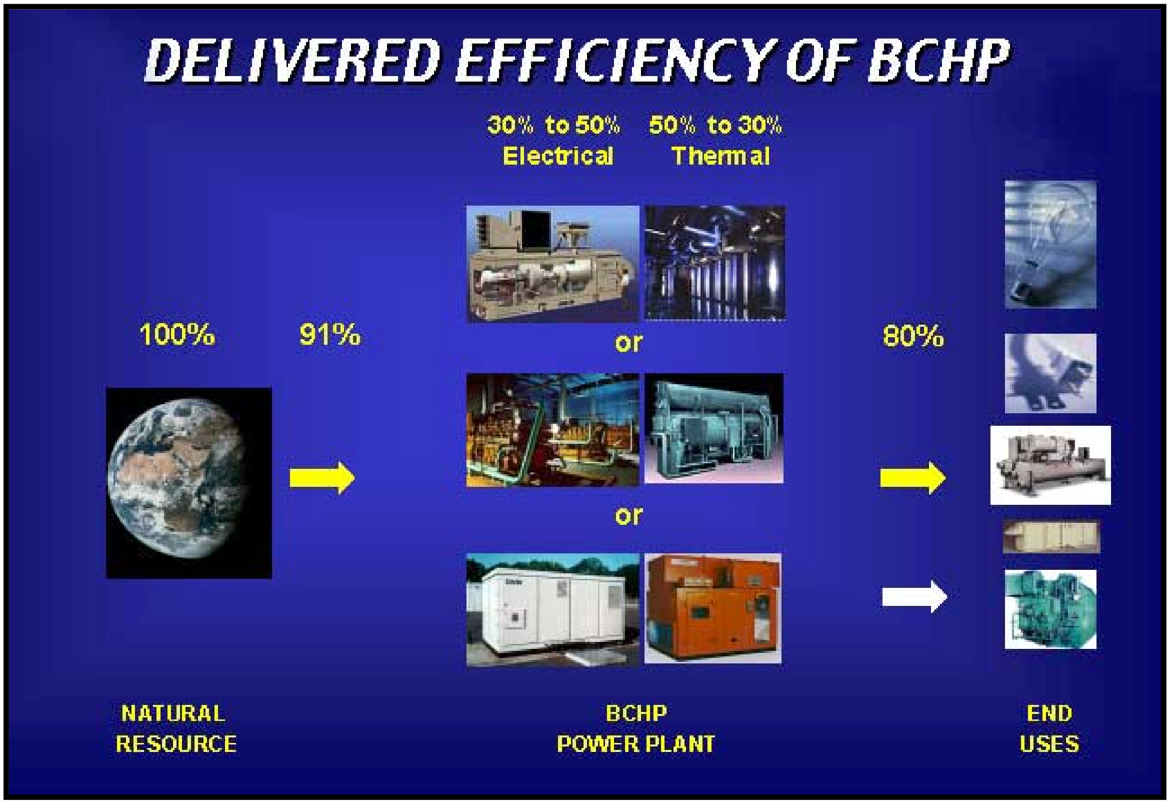


Figure 1.3 Efficiency of CHP-B Systems (www.bchp.org)

CHAPTER II

THE CHP-B SYSTEM

Cooling, Heating, and Power for Buildings combines distributed power generation with thermally-activated components to meet the cooling, heating and power needs of buildings. Technological advances in both power generation and thermally-activated systems have contributed to the development of diverse CHP-B applications. Specific types of distributed power generation and thermally-activated technologies will be introduced and briefly discussed.

Distributed Power Generation

A number of technologies are commercially available for generating electric power or mechanical shaft power onsite or near the site where the power is used. The three major categories for distributed generation are combustion turbines, engines, and fuel cells.

Combustion Turbines

Combustion turbines are based on gas turbines and use a variety of fuels, including natural gas, fuel oil, or bio-derived fuels. Combustion turbines can also use recuperators to recover thermal energy in the turbine exhaust streams for preheating the air/fuel mixtures for the combustor sections.

The efficiency of electric power generation for combustion turbine systems, operating in a simple-cycle mode (i.e., without external use of heat recovery in the turbine exhaust), ranges from 21 % to 40 %. Combustion turbines produce high quality heat that can be used to generate steam or hot water for thermally-activated applications, including heating and cooling.

Utilization of thermal energy in a combustion turbine exhaust stream significantly enhances fuel efficiency. Maintenance costs per unit of power output for combustion turbines are among the lowest of all power-generating technologies. Since the output capacity of combustion turbines decreases with increases in ambient air temperature, in hot weather climates or on hot days, cooling of inlet air has been found to be cost effective for many power plants.

Two types of combustion turbines are commercially available:

- Industrial turbines
- Microturbines

Industrial turbines represent a well-established technology for power generation. These turbines also represent the "high" end of power generating capacity equipment. Industrial turbines can provide 1 MW to more than 100 MW of electric power. Most CHP systems need capacities below 20 MW, enough for large office buildings, hospitals, or small campuses of offices and commercial buildings. The thermal efficiency of industrial gas turbines for power generation ranges from 25 % to 40 %. A picture of the rotating spool of an industrial turbine is shown in Figure 2.1.

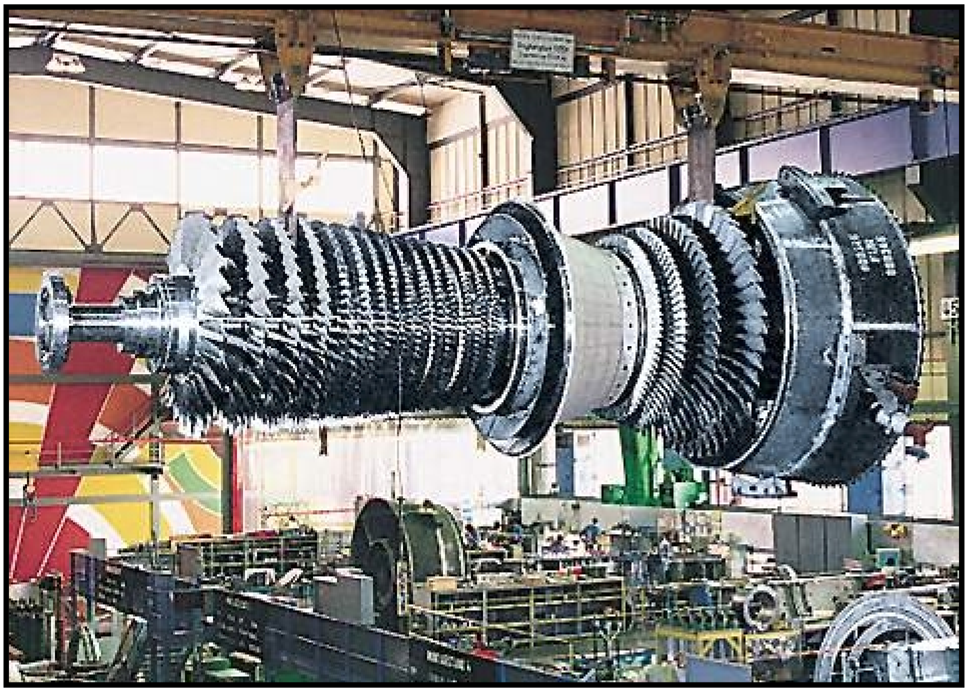


Figure 2.1: Industrial Turbine by Siemens Westinghouse
(www.siemenswestinghouse.com)

Microturbines are a new generation of smaller turbines. The capacities of microturbines range from 25 kW to 500 kW. Figure 2.2 pictures a 30-kilowatt microturbine by Capstone Turbine, Inc.

Figure 2.2: 30-kilowatt Microturbine by Capstone (www.capstone.com)

Microturbines can use natural gas, propane, and bio-derived gases produced from landfills, sewage treatment facilities, and animal waste processing plants as a primary fuel. The fuel source versatility of microturbines allows their application in rural as well as urban areas. Microturbines evolved from automotive and truck turbochargers, auxiliary power units for airplanes, and small jet engines used on remotely piloted military aircraft. Because microturbines have fewer moving parts than conventional generating equipment of similar capacity, microturbines have the potential to significantly reduce maintenance and operating costs as compared to traditional distributed-energy prime movers. By using recuperators, microturbine systems are capable of energy efficiencies for power generation in the 25-30 % range. These turbines have a significant potential for onsite power generation for CHP systems.

Internal Combustion Engines

A reciprocating engine, either four-cycle internal combustion or diesel, is used for producing mechanical shaft power. Shaft power can be used to drive a generator to produce electric power or to operate other equipment including air compressors for process or vapor compression systems for space conditioning. These applications of reciprocating engines are very well established and widespread. Engines can use natural gas, propane or diesel fuel and are available in capacities ranging from 5 kW to 10 MW. A diesel fuel engine generator is pictured in Figure 2.3.

Figure 2.3: Engine-Generator Set by Caterpillar (www.caterpillar.com)

Reciprocating engines used for power generation have low capital cost, easy startup, proven reliability, good load-following characteristics, and significant heat recovery potential. Reciprocating engines are the most widespread distributed-generation technology in the world today. Existing engines achieve generation efficiencies in the range of 25 % to over 40 %. The incorporation of exhaust catalysts and better combustion design and control has significantly reduced pollutant emissions over the past few years.

Thermal energy in the engine exhaust gases and from the engine cooling system can be employed to provide space heating, hot water, or to power thermally-activated equipment. Emissions of engines tend to be higher than those of microturbines and fuel cells. In some locations, depending on local air quality standards, engine emissions may limit reciprocating engine applications for CHP systems.

In a gas engine-driven chiller, the engine produces mechanical shaft power that is used for operating a chiller compressor. Such chillers are very similar to conventional electric-driven chillers. The only difference is that the electric motor that drives the compressor in an electric chiller is replaced with a reciprocating engine.

Fuel cells

Fuel cells produce electric power by electrochemical reactions, generally between hydrogen and oxygen, without the combustion processes. Unlike turbine- and engine-generator sets, fuel cells have no moving parts and, thus, no mechanical inefficiencies.

Phosphoric acid fuel cells (PAFCs) are commercially available. More than two hundred PAFC units, most on the order of 200kW, are operating worldwide. PAFCs are realizing efficiencies of up to 40 %. The only byproducts of PAFC operation are water and heat. However, enriched hydrogen fuel must be produced by subjecting hydrocarbon resources (natural gas or methanol) to a reforming or gasification process. This process results in chemical reactions that produce carbon dioxide and other environmental emissions.

Like a battery, a fuel cell produces direct current (DC). However, fuel cells come in a complete package in which the fuel cell stack is integrated with an inverter to convert direct current to alternating current (AC) and a reformer to provide the hydrogen-rich fuel. A complete fuel cell system thus includes a fuel reformer, a fuel cell stack, and a power conditioner. A 200-kW PAFC unit by United Technologies Company is illustrated in Figure 2.4.

Figure 2.4: PC-25™ Fuel Cell by United Technologies (www.utcfuelcells.com)

There are other types of fuel cells; proton exchange membranes (PEMFC), molten carbonate (MCFC), and solid oxide (SOFC) are the most promising. These fuel cells are at various stages of technology demonstration and are not commercially available. Each type of fuel cell has its own "preferred" range of capacities and waste heat temperatures that determine where they can be used to best advantage in CHP systems.

Distributed power generation (DPG) is a required component of a CHP-B system. Internal combustion (IC) engines, combustion turbines, and fuel cells are the current prime movers that have the most potential for DPG. Characteristics of these DPG technologies are compared in Table 2.1. The characteristics such as electric efficiency, power output, and cost presented in Table 2.1 will be discussed in greater detail for each technology.

Table 2.1: Comparison of DPG Technologies
(http://www.eren.doe.gov/der/chp/pdfs/chprev.pdf)

Comparison of DPG Technologies					
	Diesel Engine	**Natural Gas Engine**	**Gas Turbine**	**Microturbine**	**Fuel Cells**
Electric Efficiency (LHV)	30-50 %	25-45 %	25-40 % (simple) 40-60 % (combined)	20-30 %	40-70 %
Power Output (MW)	0.05-5	0.05-5	3-200	0.025-0.25	0.2-2
Footprint (ft^2/kW)	0.22	0.22-0.31	0.02-0.61	0.15-1.5	0.6-4
CHP installed cost ($/kW)	800-1,500	800-1,500	700-900	500-1,300	>3,000
O&M cost ($/kW)	0.005-0.010	0.007-0.015	0.002-0.008	0.002-0.01	0.003-0.015
Availability (uptime)	90-95 %	92-97 %	90-98 %	90-98 %	>95 %
Hours between Overhauls	25,000-30,000	24,000-60,000	30,000-50,000	5,000-40,000	10,000-40,000
Start up time	10 sec	10 sec	10 min-1 hr	60 sec	3 hrs-2 days
Fuel pressure (psi)	<5	1-45	125-500 (may require compressor)	40-100 (may require compressor)	0.5-45
Fuels	Diesel and residual oil	Natural gas, biogas, propane	Natural gas, biogas, propane, distillate oil	Natural gas, biogas, propane, distillate oil	Hydrogen, natural gas, propane
Noise	Moderate to high (requires building enclosure)	Moderate to high (requires building enclosure)	Moderate (enclosure supplied with unit)	Moderate (enclosure supplied with unit)	Low (no enclosure required)
NO_X emissions (lb/MW-hr)	3-33	2.2-28	0.3-4	0.4-2.2	<0.02
Uses for Heat Recovery	Hot water, LP steam, district heating	Hot water, LP steam, district heating	Direct heat, hot water, LP-HP steam, district heating	Direct heat, hot water, LP steam	Hot water, LP-HP steam
CHP output (Btu/kWh)	3,400	1,000-5,000	3,400-12,000	4,000-15,000	500-3,700
Useable Temp for CHP (8F)	180-900	300-500	500-1,100	400-650	140-700

Heat Recovery

In most CHP applications, the exhaust gas from a prime mover is ducted to a heat exchanger to recover the thermal energy in the gas stream. Generally, these heat exchangers are air-to-water heat exchangers, where the exhaust gas flows over some form of tube-and-fin heat exchanger surface and the heat from the exhaust gas is used to make hot water or steam. The hot water or steam is then used to provide process energy and/or to operate thermally-activated equipment, such as absorption chillers or desiccant dehumidifiers. Many of the thermal-recovery technologies used in building CHP systems require hot water, some at moderate pressures of 15 to 150 psig. In the cases where additional steam or pressurized hot water is needed, supplemental heat can be added to the exhaust gas with a burner in the exhaust gas duct.

In some applications, air-to-air heat exchangers can be used. If the emissions from the generation equipment are low enough, the hot exhaust gases can be mixed with make-up air and vented directly into the heating system for building space heating.

In the majority of installations, a flapper damper or "diverter" valve is employed to vary the flow across the heat exchanger to maintain a specific design temperature of the hot water or a specific steam generation rate. In some CHP designs, the exhaust gases are used to activate a thermal enthalpy wheel or a desiccant dehumidifier. A thermal wheel uses the exhaust gas to heat a rotating wheel with a medium that absorbs the heat and then transfers the heat into the incoming airflow into which the wheel is rotated.

Thermally-Activated Devices

Thermally-activated devices are based on technologies that use thermal energy, preferably heat from the exhaust gases of power generation equipment, instead of electric energy for providing heating, cooling, or humidity control for buildings. The two primary components for thermally-activated devices for application in CHP systems are absorption chillers and desiccant dehumidifiers.

Absorption chillers

Absorption chillers use heat as the primary source of energy for driving an absorption refrigeration cycle. These chillers require very little electric power (0.02 kW/ton) compared to electric chillers that need 0.47 to 0.88 kW/ton, depending on the type of electric chiller. Absorption chillers have fewer and smaller moving parts and are quieter during operation than electric chillers. These chillers are also environmentally friendly in that they use non-CFC refrigerants.

Commercially available absorption chillers can utilize the following sources of heat:

- Steam
- Hot water
- Exhaust gases
- Direct combustion

Absorption chillers, except those that use direct combustion, are excellent candidates for providing some or all of the cooling load in a CHP system for a building. Modern absorption chillers can also provide heat during winter and feature electronic controls that provide quick start-up, automatic purge, and better partial-load operation than many electric chillers. Maintenance contracts and extended warranties are also available for absorption chillers at costs similar to those for electric chillers. Many facilities across the U.S. are already benefiting from the use of absorption chillers, such as the one pictured in Figure 2.5.

Figure 2.5: Absorption Chiller by Broad Air Conditioning (www.broad.org)

Two types of absorption chillers are commercially available: single effect and multiple effect. Compared to single-effect chillers, multiple-effect absorption chillers cost more (higher capital cost) but are more energy efficient and are, thus, less expensive to operate (lower energy cost). The overall economic attractiveness of each chiller depends on many factors, including the cost of capital and the cost of energy.

Desiccant dehumidifiers

There are two separate aspects of space conditioning for comfort cooling,

- Lowering the temperature of the air (sensible cooling), and

- Reducing humidity in the air (latent cooling)

The humidity level should remain below 60 % Relative Humidity (RH) to prevent growth of mold, bacteria and other harmful microorganisms in buildings and to prevent adverse health effects.

Traditionally, temperature and humidity control have been accomplished using a single piece of equipment that reduces the air temperature below its dew point temperature. Moisture in the incoming air condenses on the outside of a cooling coil over which the air passes and cooler air, containing less moisture, is sent to the space being conditioned. Reducing humidity in the air by cooling often requires lowering the air temperature below a comfortable level and may necessitate reheating of the dehumidified air to achieve comfort.

Desiccant dehumidifiers reduce humidity in the air by using solid desiccant materials or liquid desiccant materials to attract and hold moisture. Desiccant dehumidifiers can operate independently of chillers and can be operated in series or parallel with chillers. Recoverable heat from the exhaust gases of turbines and engines for power generation or engine-driven chillers is used for regenerating saturated desiccant material in these dehumidifiers.

In some CHP systems, the moisture content of the air is reduced using a desiccant dehumidifier and the dehumidified air is then cooled using conventional cooling equipment. By reducing the moisture content of the air, desiccant dehumidifiers satisfy the latent cooling load and, thus, reduce the load of the chillers to only the sensible cooling (reducing the temperature). Alternatively, a desiccant dehumidifier can be used to further dehumidify and partially reheat cool, saturated air leaving a conventional cooling coil. By positioning the desiccant dehumidifier after the cooling coil, the dehumidification performance of the desiccant is enhanced. This allows the use of moderate or lower temperatures, typical of CHP systems, for regenerating the desiccant. A typical, commercially available desiccant system is shown in Figure 2.6.

Figure 2.6: Desiccant Dehumidifier (www.bchp.org)

CHAPTER III

INTERNAL COMBUSTION ENGINES

Technology Overview

Internal combustion (IC) engines are widespread in their application. IC engines require fuel, air, compression, and a combustion source to function. The four-stroke reciprocating IC engine, illustrated in Figure 3.1, is comprised of an ignition source, intake and exhaust valves, a cylinder, a piston, a connecting rod, and a crankshaft. Depending on the ignition source, IC engines generally fall into two categories: (1) spark-ignited engines, typically fueled by gasoline or natural gas, or (2) compression-ignited engines, typically fueled by diesel fuel.

Spark Plug

Exhaust Valve

Cylinder

Piston

Connecting Rod

Crankshaft

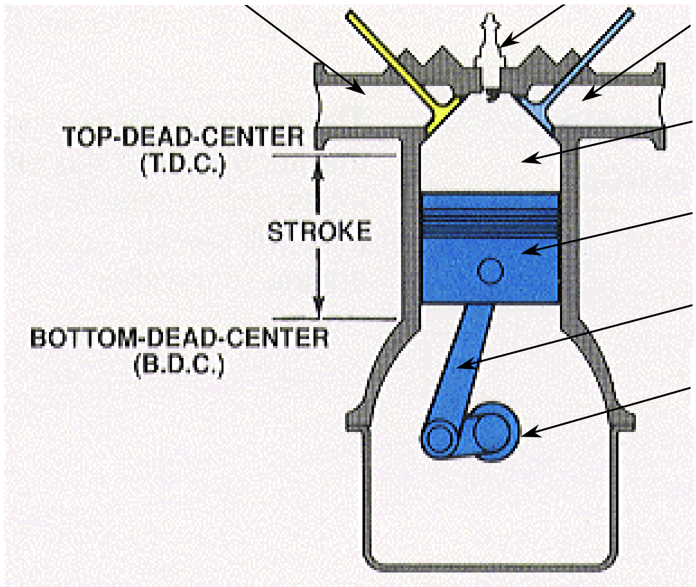


Figure 3.1: Four-stroke Reciprocating IC Engine (www.personal.washtenaw.cc.mi.us)

A spark-ignited reciprocating engine operates on the basis of the Otto cycle. The most fundamental model of an IC engine is the air-standard Otto cycle, which assumes that heat is added instantaneously, that the heat is rejected at a constant volume, and that the working fluid is air. The ideal Otto cycle consists of four internally reversible processes. First, as the piston moves from bottom-dead-center to top-dead-center, air is isentropically compressed. Second, combustion begins when heat is added at a constant volume to the compressed working fluid. Third, during the process known as the power stroke, the working fluid expands isentropically and forces the piston to move to bottom-dead-center. Finally, heat is rejected at a constant volume. The pressure-specific volume (pv) and temperature-entropy (Ts) diagrams for the ideal Otto cycle are presented in Figure 3.2.

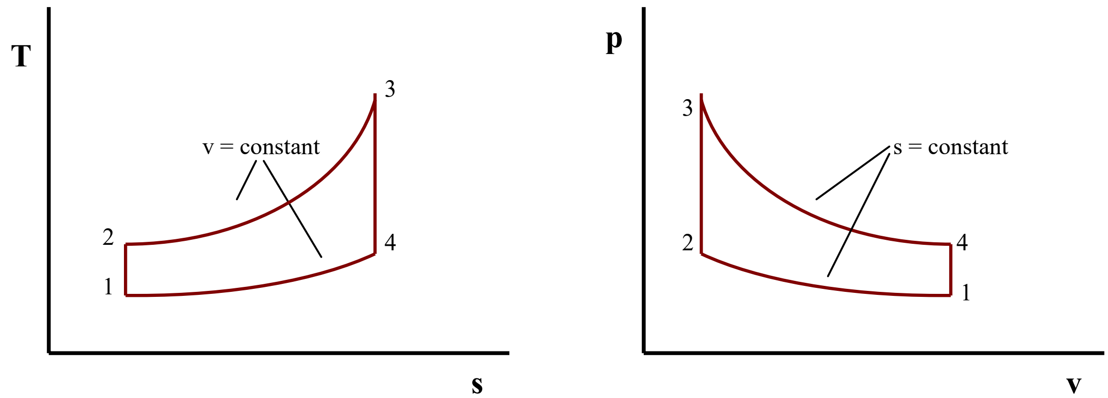


Figure 3.2: Pressure-Specific Volume and Temperature-Entropy Diagrams for the Ideal Otto Cycle

Application of the first law to the ideal Otto cycle determines the expressions for the net work accomplished by the cycle (W_{Otto}), the heat added (Q_{in}), and the thermal efficiency (η_{Otto}). Applying the air-standard assumptions of constant specific heats and ideal gas properties, the following relationships are employed.

$$\begin{aligned} W_{Otto} &= m\cdot(u_{34}-u_{12}) \\ &= m\cdot(u_3-u_4)-m\cdot(u_2-u_1) \\ &= c_v\cdot m\cdot(T_3+T_2-T_4-T_1) \end{aligned} \tag{3-1}$$

$$\begin{aligned} Q_{in} &= m\cdot(u_3-u_2) \\ &= c_v\cdot m\cdot(T_3-T_2) \end{aligned} \tag{3-2}$$

$$\begin{aligned} \eta_{Otto} &= \frac{W_{Otto}}{Q_{in}} \\ &= \frac{c_v\cdot m\cdot(T_3+T_2-T_4-T_1)}{c_v\cdot m\cdot(T_3-T_2)} \\ &= 1-\frac{T_1}{T_2}\cdot\left(\frac{T_4/T_1-1}{T_3/T_2-1}\right) \end{aligned} \tag{3-3}$$

For the isentropic compression and expansion processes, the compression ratio (r) is defined as the ratio of the volume of the working fluid when the piston is at bottom-dead-center to the volume of the working fluid when the piston is at top-dead-center. Noting from Figure 3.2 that $V_2 = V_3$ and $V_1 = V_4$, the expression for compression ratio appears in the following form:

$$r = \frac{V_1}{V_2} = \frac{V_4}{V_3} \tag{3-4}$$

An air-standard analysis provides the following isentropic relationships for pressure, temperature, and volume.

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^k = \frac{1}{r^k} \quad and \quad \frac{P_3}{P_4} = \left(\frac{V_4}{V_3}\right)^k = r^k \tag{3-5a,b}$$

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \frac{1}{r^{k-1}} \quad and \quad \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} = \frac{1}{r^{k-1}} \tag{3-6a,b}$$

Where k is the ratio of specific heats (c_p/c_v). For air-standard analysis, k is 1.4. From Equations 3-6a and 3-6b, $T_4/T_1=T_3/T_2$. The Otto cycle thermal efficiency then becomes

$$\eta_{Otto} = 1 - \frac{T_1}{T_2} \tag{3-7}$$

The efficiency can, therefore, be expressed in terms of compression ratio as

$$\eta_{Otto} = 1 - \frac{1}{r^{k-1}} \tag{3-8}$$

For a working fluid with constant specific heats, the thermal efficiency will increase with an increase in the compression ratio. Figure 3.3 shows the ideal Otto cycle thermal efficiency as a function of the compression ratio of the engine. The figure clearly illustrates that a change in compression ratio from 1 to 10 has the most dramatic effect on the thermal efficiency of an Otto engine.

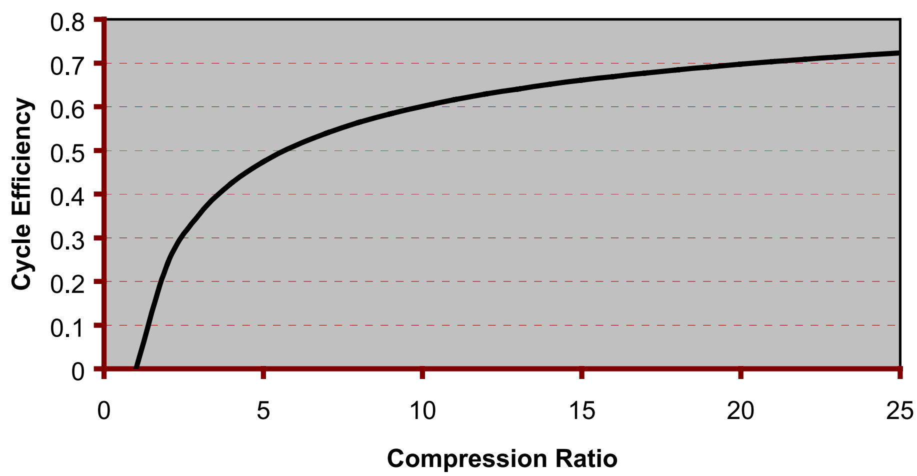
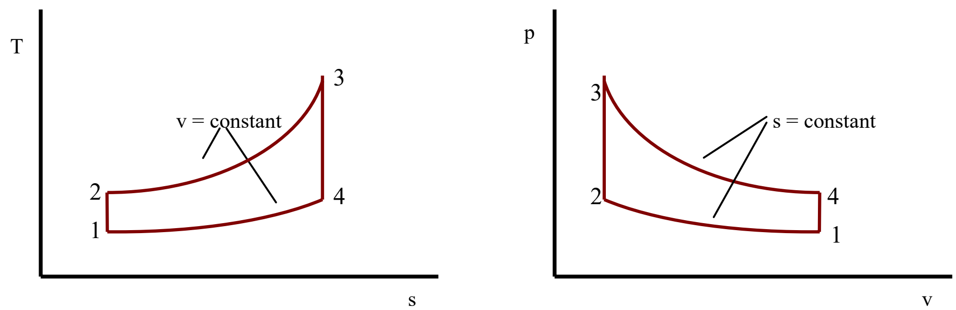


Figure 3.3: Otto Cycle Thermal Efficiency as a Function of Compression Ratio

The expressions for work, heat addition, and thermal efficiency presented in Equations 3-1, 3-2 and 3-8 are useful for analyzing the Otto cycle under ideal, air-standard conditions. In an actual Otto cycle, the ratio of fuel to air and the ratio of combustion gases to air remain small enough to apply the properties of air to the working fluid. However, reversibility and constant volume heat rejection and addition are assumptions that affect the accuracy of an actual Otto cycle model. Detailed modeling of the combustion processes, the irreversibilities and the heat transfers associated with an actual Otto engine normally involves computer simulation and is beyond the scope of this introduction. An example IC engine problem involving the Otto cycle is presented in Figure 3.4.

Problem 3-1:
The temperature at the beginning of the compression process of an air-standard Otto cycle with compression ratio of 9 is 550 R, the pressure is 1 atm, and the cylinder volume is 0.03 ft^3. The maximum temperature during the cycle is 3400 R. The air in the cycle maintains a c_v = 0.171 Btu/lb-R and a c_p = 0.24 Btu/lb-R. Determine (a) the temperature and pressure at the end of each process of the cycle, (b) the thermal efficiency, (c) the net work per cycle, (d) the amount of rejected heat per cycle, and (e) the maximum temperature and pressure of the rejected heat.

Solution:
Schematic:



Given Data:

$T_1 := 550R$	Temperature before compression stroke
$T_3 := 3400R$	Maximum temperature
$r := 9$	Compression ratio
$P_1 := 1atm$	Pressure before compression stroke
$V_1 := 0.03ft^3$	Cylinder volume
$c_v := 0.171\frac{BTU}{lb\cdot R}$	Constant volume specific heat
$c_p := 0.24\frac{BTU}{lb\cdot R}$	Constant pressure specific heat

Assumptions:
1. The air in the piston-cylinder assembly is a closed system.
2. The compression and expansion processes are adiabatic.
3. All processes are internally reversible.
4. The air is modeled as an ideal gas with a cold air-standard analysis (k = 1.4).
5. Kinetic and potential energy effects are negligible.

Figure 3.4: IC Engine Example Problem

(a) Determine the temperature, pressure, and specific internal energy (u) at each principal state in the cycle.

Using T_1,

$u_1 := c_v \cdot T_1$

$u_1 = 94.05 \frac{BTU}{lb}$ Internal energy at 1

For isentropic compression (Process 1-2), Equation 3-6a gives:

$T_2 := T_1 \cdot \left(r^{k-1}\right)$ $T_2 = 1.325\times 10^3\, R$ Temperature at 2

Equation 3-5a gives:

$P_2 := P_1 \cdot r^k$ $P_2 = 21.674 atm$ Pressure at 2

Using T_2,

$u_2 := c_v \cdot T_2$

$u_2 = 226.494 \frac{BTU}{lb}$ Internal energy at 2

Process 2-3 occurs at constant volume. Using the ideal gas equation:

$P_3 := P_2 \cdot \frac{T_3}{T_2}$ $P_3 = 55.636 atm$ Pressure at 3

Using T_3,

$u_3 := c_v \cdot T_3$

$u_3 = 581.4 \frac{BTU}{lb}$ Internal energy at 3

For isentropic compression (Process 3-4), Equation 3-5b gives:

$T_4 := \frac{T_3}{r^{k-1}}$ $T_4 = 1.412\times 10^3\, R$ Temperature at 4

Equation 3-5b gives:

$P_4 := P_3 \cdot \frac{1}{r^k}$ $P_4 = 2.567 atm$ Pressure at 4

Using T_4,

$u_4 := c_v \cdot T_4$

$u_4 = 241.423 \frac{BTU}{lb}$ Internal energy at 4

Figure 3.4 (continued)

(b) The thermal efficiency is determined based on the compression ratio.

Equation 3-8 gives the thermal efficiency as

$\eta_{Otto} := 1 - \frac{1}{r^{k-1}}$ $\quad \eta_{Otto} = 58.476\%$ Thermal efficiency

(c) The net work produced per cycle can be calculated now that the internal energy is known at each point in the cycle.

The mass (m) is calculated using the ideal gas law.

$$m_1 := \frac{P_1 \cdot V_1}{\left(\frac{R_g}{M}\right) \cdot T_1} \qquad m_1 = 2.164 \times 10^{-3}\,\text{lb}$$

Equation 3-1 expresses the net work per cycle as

$W_{Otto} := m_1 \cdot \left[\left(u_3 - u_4\right) - \left(u_2 - u_1\right)\right]$ $\quad W_{Otto} = 0.449\text{BTU}$ Net work per cycle

(d) The heat rejected per cycle is

$Q_{rej} := m_1 \cdot \left(u_4 - u_1\right)$ $\quad Q_{rej} = 0.319\text{BTU}$ Amount of rejected heat

(e) The pressure and temperature of the rejected heat is maximum at the beginning of the heat rejection process.

$P_4 = 2.567\text{atm}$ Maximum pressure of Q_{rej}

$T_4 = 1.412 \times 10^3\,\text{R}$ Maximum temperature of Q_{rej}

Figure 3.4 (continued)

Compression-ignition reciprocating engines operate on the basis of the Diesel cycle. The ideal, air-standard Diesel cycle is similar to the model of the ideal, air-standard Otto cycle presented above. The Ts and pv diagrams for the ideal Diesel cycle are illustrated in Figure 3.5. The major difference in these two cycles occurs in process 2-3 when heat is added to the compressed working fluid. In the Diesel cycle, heat is not added to the working fluid at constant volume as in the Otto cycle, but heat is added at constant pressure. The pv diagram demonstrates that the heat addition process and isentropic expansion process in the ideal Diesel cycle both occur as part of the power stroke.

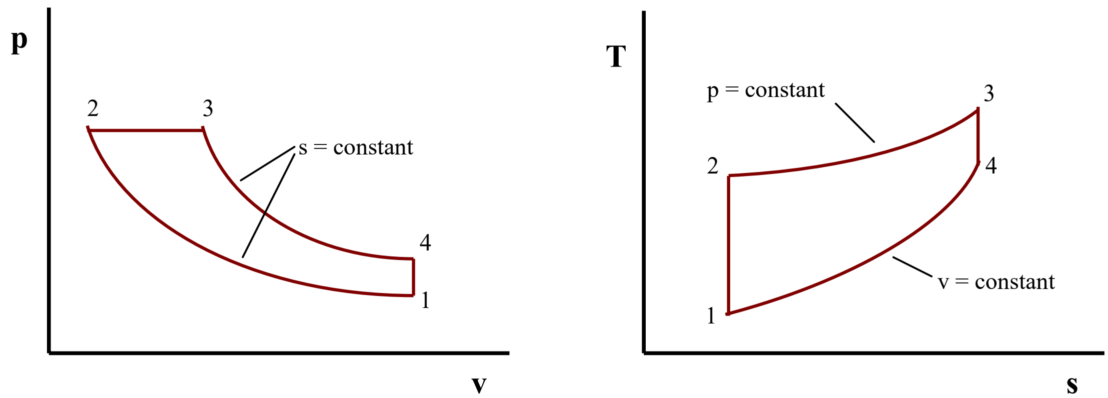


Figure 3.5. Pressure-Specific Volume and Temperature-Entropy Diagrams for the Ideal Diesel Cycle

Unlike a spark-ignition, which compresses a fuel/air mixture, a compression-ignition requires the working fluid to be compressed to a high pressure and temperature before the fuel is added. The addition of fuel to the high pressure, high temperature working fluid initiates combustion. Diesel engines can achieve compression ratios as high as 25:1 and are, therefore, able to perform at better thermal efficiencies than Otto engines, which are limited to compression ratios below 12:1. Application of the first law to the ideal Diesel cycle determines the expressions for the net work accomplished by the cycle (W_{Diesel}), the heat added ($Q_{D,in}$), the heat rejected ($Q_{D,out}$), and the thermal efficiency (η_{Diesel}).

$$\begin{aligned} Q_{D,in} &= m \cdot (h_3 - h_2) \\ &= c_p \cdot m \cdot (T_3 - T_2) \end{aligned} \tag{3-9}$$

$$Q_{D,out} = m \cdot (u_4 - u_1) \\ = c_v \cdot m \cdot (T_4 - T_1) \tag{3-10}$$

$$W_{Diesel} = Q_{D,in} - Q_{D,out} \\ = c_p \cdot m \cdot (T_3 - T_2) - c_v \cdot m \cdot (T_4 - T_1) \tag{3-11}$$

$$\eta_{Diesel} = \frac{W_{Diesel}}{Q_{D,in}} \tag{3-12}$$

For the Diesel cycle, both a compression ratio (r) and cutoff ratio (r_c) are defined.

$$r = \frac{V_1}{V_2} \tag{3-13}$$

$$r_c = \frac{V_3}{V_2} \tag{3-14}$$

Note that the compression ratio for the Diesel cycle is based only on the isentropic compression and not on the isentropic expansion. Applying the compression and cutoff ratios and air-standard assumptions, the thermal efficiency of the ideal Diesel cycle can be expressed in the following form:

$$\eta_{Diesel} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_c^k - 1}{k(r_c - 1)} \right] \tag{3-15}$$

The thermal efficiency of the Diesel cycle differs from the thermal efficiency of the Otto cycle by the bracketed term in Equation 3-15. Diesel engines always have a cutoff ratio greater than unity ($r_c > 1$). Otto engines have a higher thermal efficiency than Diesel engines at the same compression ratio. However, Diesel engines achieve better overall thermal efficiencies since they can operate at higher compression ratios than Otto engines. A comparison of the thermal efficiencies of the Otto and Diesel cycles is presented in Figure 3.6.

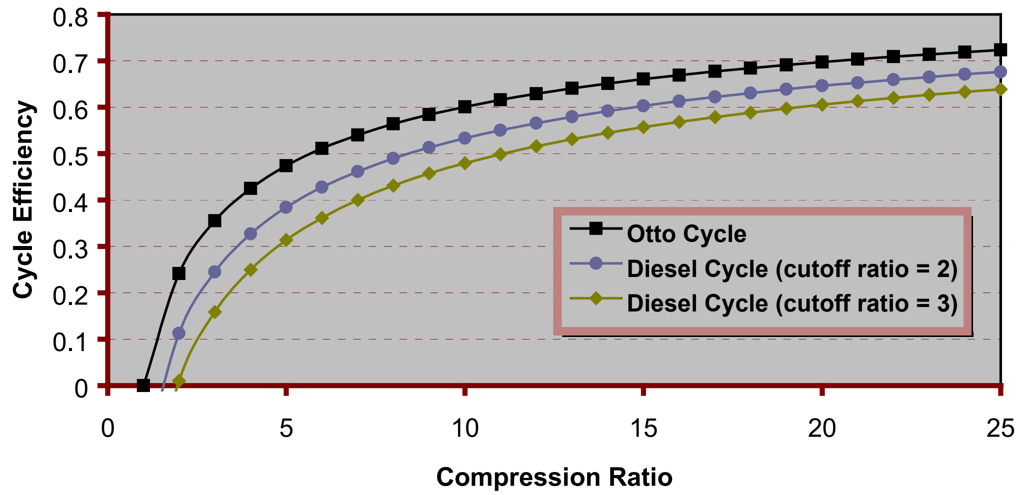


Figure 3.6: Otto and Diesel Cycle Thermal Efficiencies as Functions of Compression Ratio

Application

Combustion engines are the most mature prime mover among the distributed power generation (DPG) technologies. Commercially available reciprocating engines for power generation range from 0.5-kW to 6.5-MW. Reciprocating engines can be used in a variety of applications due to their small size, low unit costs, and useful thermal output. Applications for reciprocating engines in power generation include continuous- or prime-power generation, peak shaving, back-up power, premium power, remote power, standby power, and mechanical drive use. An overview of reciprocating engine characteristics is presented in Table 3.1.

Table 3.1: Overview of Reciprocating Engine Technology (www.energy.ca.gov/distgen/)

Reciprocating Engine Overview	
Commercially Available	Yes
Size Range	0.5 kW – 6.5 MW
Fuel	Natural gas, diesel, landfill gas, digester gas
Efficiency	25-45 % (primarily size dependent)
Environmental	Emission controls required for NO_x and CO

Stationary reciprocating engines represent 7 % of the world's electric generating capacity and 7 % of the United States' capacity. Engine-driven generators accounted for 46 % of cogeneration installations in the United States in 1998 but accounted for only 1.5 % of the total cogeneration capacity. In comparison with other DPG systems, engine generator systems dominate the market for capacities below 1 MW. The availability of reciprocating engines with capacities below 1 MW has promoted their installation in small-scale CHP applications.

Comparatively low installation costs, suitability for intermittent operation, and high temperature exhaust make combustion engines very attractive for CHP applications. Combustion engines have an existing sales and support structure, generally inexpensive and readily available parts, and service technicians with experience in maintenance and repair. In most aspects, combustion engines are more developed than other DPG technologies (Borbely and Kreider, 2001).

Heat Recovery

Traditional-fuel-based, large-scale electric power generation is typically about 39 % efficient, while separate boilers are about 50 % efficient. In either case, the reject heat is simply lost. Engine-driven CHP systems recover heat from the engine exhaust, the jacket water, and the lubricating oil. Steam or hot water can be generated from recovered heat and is can be used for space heating, reheat, domestic hot water, and absorption cooling.

Heat in the engine jacket coolant accounts for up to 30 % of the energy input and is capable of producing 200°F hot water. Some engines, such as those with high pressure or ebullient cooling systems, can operate with water jacket temperatures up to 265°F. The engine exhaust heat accounts for 10-30 % of the energy input. Exhaust temperatures of 850°F-1200°F are typical; however, only a portion of the exhaust heat can be recovered since exhaust gas temperatures are generally kept above condensation thresholds. Most heat recovery units are designed for a 300-350°F exhaust outlet temperature to avoid the corrosive effects of condensation in the exhaust. Exhaust heat is typically used to generate hot water to about 230°F or low-pressure steam (15 psig). By recovering heat in the jacket water and exhaust, 70 % to over 80 % of the fuel's energy can be effectively utilized.

In current CHP configurations, engine-driven electric generation efficiencies range from 34 % in small CHP units to 41 % in larger installations. The efficiencies of thermally-activated components typically range from 40 % to 50 %; thus, total CHP system efficiency approaches 90 % (Borbely and Kreider, 2001).

Cost

Reciprocating IC engines are the traditional technology for emergency power all over the world. They have the lowest first costs among DPG technologies. The capital cost of a basic gas-fueled generator set ranges from \$300-\$900/kW, depending on size, fuel type, and engine type. Overall engine cost (\$/kW) increases with size. The total installed cost can be 50-100 % more than the engine itself. Additional costs include balance of plant (BOP) equipment, installation fees, engineering fees, and other owner or miscellaneous costs. CHP projects using reciprocating engines are typically installed at a cost between \$800 - \$1,500/kW. The pie chart in Figure 3.7 shows a breakdown of the total installed costs of a 550-kW natural gas IC engine (www.energy.ca.gov/distgen/).

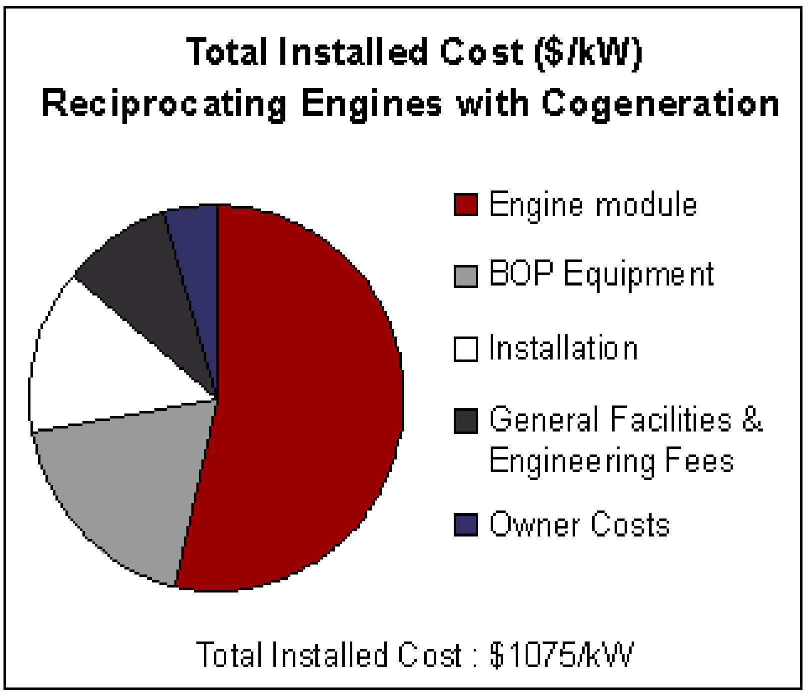


Figure 3.7: Total Installed Cost of a 550-kW Natural Gas IC Engine (www.energy.ca.gov/distgen/)

Maintenance costs over the life of IC engines are significant. Engine maintenance is comprised of routine inspections/adjustments and periodic replacement of engine oil, coolant, and spark plugs every 500-2,000 hours. Additionally, periodic overhauls are recommended by the manufacturer. On average, maintenance costs range from $0.01 – $0.015/kWhr (http://www.eren.doe.gov/der/chp/pdfs/chprev.pdf).

IC Engines and CHP-B

Many of the characteristics of combustion engines make them attractive to many CHP applications, specifically CHP-B applications. High initial costs are one of the major obstacles to the installation of CHP-B systems. Reciprocating engines are economically favorable since they are generally less expensive than other competing DPG technologies. Also, building power demands can fluctuate in a relatively short period of time. Reciprocating engines can quickly meet fluctuating power demands with start-up times as low as ten seconds, compared to some emerging technologies that may take several hours to reach steady-state operation.

Through years of technology advancements, reciprocating engines have climbed in efficiency from under 20 % to over 30 %. Today's most advanced natural gas-fueled IC engines have electrical efficiencies (based on lower heating value, LHV) close to 45 %. Such high efficiencies rival other DPG technologies. Further, the technology's many years of experience has given internal combustion engines a reliability that is unmatched by any competing CHP-B distributed-generation technology. Another strength of reciprocating engines is their ability to operate on a wide variety of fuels.

One disadvantage associated with integrating reciprocating IC engines into CHP-B systems is the high levels of the emissions. Many locations enforce air quality standards that IC engines cannot meet. Reciprocating engines produce a high noise level due to the combustion process and moving parts. Thus, noise is another disadvantage, especially since the application of CHP-B is to buildings where high noise levels may not be permitted. A final disadvantage of IC engine application to CHP-B may be the most imposing disadvantage. The high number of moving parts in a reciprocating IC engine imposes more frequent maintenance intervals resulting in higher operating costs and more frequent downtimes. Over the life of a reciprocating engine, maintenance becomes a major expense of operation. Many of the advantages and disadvantages associated with reciprocating engines are listed in Table 3.2.

Table 3.2: Advantages and Disadvantages of Reciprocating IC Engines (www.energy.ca.gov/distgen/)

IC Engine Problems

For air-standard analysis c_v = 0.171 Btu/lb-R and c_p = 0.24 Btu/lb-R.

1. A piston-cylinder arrangement has a cylinder diameter (bore) of 98 mm and performs a 125 mm stroke. If the clearance volume, the volume above top-dead-center, is 100 cm^3, what is the compression ratio.

2. A four-cylinder, four-stroke IC engine has a bore of 5.5 in. and a stroke of 4.7 in. The clearance volume is 14 % of the cylinder volume when the piston is at bottom-dead center. What are (a) the compression ratio, (b) the cylinder displacement (volume displaced by cylinder), and (c) the engine displacement?

3. An air-standard analysis is to be performed on a spark-ignited IC engine with a compression ratio of 9.0. At the start of compression, the pressure and temperature are 100 kPa and 300 K, respectively. The heat addition per unit mass of air is 1400 kJ/kg. Determine (a) the net work, (b) the thermal efficiency, (c) the heat rejected during the cycle, and (d) the maximum temperature and pressure of the rejected heat.

4. An Otto cycle operates on an air-standard basis. The properties of the air prior to compression are p_1 = 1 bar, T_1 = 310 K, and V_1 = 600 cm^3. The compression ratio of the cycle is 8 and the maximum temperature is 1800 K. Determine (a) the heat addition in kJ, (b) the net work in kJ, (c) the thermal efficiency, and (d) the availability transfer from the air accompanying the heat rejection process, in kJ, for T_0 = 310 K, p_0 = 1 bar.

5. Consider a four-stroke, spark-ignited engine with four cylinders. Each cylinder undergoes a process like the cycle in Problem 2. If the engine operates at 1800 rpm, determine the net power output in kW.

6. Prior to the compression stroke in an Otto cycle, the pressure and temperature are 14.7 psi and 540 R, respectively. The compression ratio is 6, and the heat addition per unit mass of air is 550 Btu/lb. Find (a) the maximum temperature of heat rejection, in R, (b) the maximum pressure of heat rejection, in psi, and (c) the thermal efficiency if the cycle is modeled on a cold-air standard basis with specific heats evaluated at 540 R.

7. A four-stroke two-cylinder Otto engine has a bore of 24 in, a stroke of 20 in, and a compression ratio of 7.2:1. The intake air is at 14.7 psi and 40°F with compression and expansion processes that are reversible and adiabatic. If 1400 Btu/lbm air is added and the engine speed is 340 rpm, determine (a) the net work per cycle, (b) the work per minute, and (c) the rate of heat rejection in Btu/hr.

8. A four-cylinder, four-stroke IC engine has a bore of 3.7 in. and a stroke of 3.4 inches. The clearance volume is 16 % of the cylinder volume when the piston is at bottom-dead center and the crankshaft rotates at 2100 rpm. The processes in the cylinders are modeled as a cold air-standard Otto cycle with a pressure of 14.7 psi and a temperature of 80°F at the beginning of compression. The maximum temperature in the cycle is 5200 R. Based on this model, calculate (a) the net work per cycle, (b) the power developed by the engine, (c) the amount of heat rejected, (d) the maximum temperature and pressure of heat rejection, and (e) the thermal efficiency of the cycle.

9. A diesel cycle is modeled on a cold air-standard basis with specific heats calculated at 300 K. At the beginning of compression, the pressure and temperature are 125 kPa and 270 K, respectively. After heat addition, the pressure is 8.2 MPa and the temperature is 2300 K. Determine (a) the compression ratio, (b) the cutoff ratio, (c) the heat rejected by the cycle, and (d) the thermal efficiency of the cycle.

10. An air standard diesel cycle has a compression ratio of 16 and a cutoff ratio of 2. Before compression, p_1 = 14.7 psi, V_1 = 0.7 ft^3, and T_1 = 560 R. Calculate (a) the heat added, (b) the heat rejected, (c) the thermal efficiency, (d) the maximum temperature and pressure of the rejected heat, and (e) the availability transfer from the air accompanying the heat rejection process for $T_0 = T_1$ and $p_0 = p_1$.

11. The maximum temperature of a cold air-standard diesel cycle is 1900 K. The pressure and temperature before compression are 84 kPa and 290 K, respectively. With an air mass of 9 grams and compression ratios of 14, 19, and 21, calculate (a) the net work of the cycle, (b) the heat rejected during the cycle, and (b) the thermal efficiency of the cycle.

Manufacturers

There are a large number of companies worldwide that manufacture reciprocating engines and/or complete generator sets for large and small distributed energy resource (DER) applications. Six of these companies are listed as follows: (www.energy.ca.gov/distgen/)

- Caterpillar, headquartered in Peoria, Illinois, manufactures reciprocating engines and generator sets, like the engine-generator set illustrated in Figure 3.8, for a variety of fuels and with power outputs ranging from 40-kW to 10-MW.

Figure 3.8: Engine-Generator Set by Caterpillar (www.caterpillar.com)

- Cummins, based in Columbus, Indiana, is a manufacturer of engines and power generators. Generator sets include a prime mover fueled by diesel, natural gas or propane, and an alternator that provides power at 50 Hz or 60Hz. Figure 3.9 pictures a generator from Cummins' line of 7-kW to 2-MW diesel engine-generator sets.

Figure 3.9: Cummins Diesel Engine-Generator Set (www.cummins.com)

- Deutz Corporation, North American headquarters located in Norcross, Georgia, is an international supplier of reciprocating gas- and diesel-fueled engine/generator systems. The Deutz diesel engine shown in Figure 3.10 is capable of producing from 1,460 to 2,017 kW of electricity.

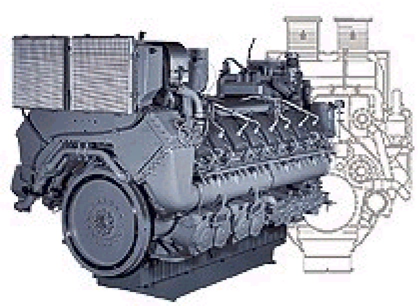


Figure 3.10: Diesel-Fueled Engine by Deutz Corporation (www.deutzusa.com)

- Generac Power Systems, located in Waukesha, Wisconsin, manufactures diesel- and gaseous-fueled engine generator systems, as pictured in Figure 3.11, with power outputs of 3.kW to 2-MW for residential, commercial, industrial, mobile, recreational vehicle, and communications applications.

Figure 3.11: 50-kW Natural Gas Fueled Generator by Generac Power System (www.generac.com)

- Honda Power Equipment of Alpharetta, Georgia is a manufacturer of engines and generators that operate on various fuels for DER applications with power output up to 20-kW. Figure 3.12 presents a small electricity generator by Honda.

Figure 3.12: Honda 11.5-kW Gas Generator (www.hondapowerequipment.com)

- Kohler, based in Kohler, Wisconsin, manufactures 6-kW to 20-kW engines and 8.5-kW to 2-MW onsite power generators. An onsite power generator by Kohler is shown in Figure 3.13.

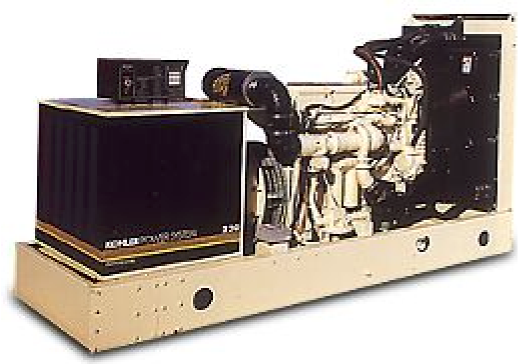


Figure 3.13: 200-kW Natural Gas Generator by Kohler (www.kohler.com)

- Waukesha Engine of Waukesha, Wisconsin is a manufacturer of gaseous-fueled reciprocating engines up to multi-megawatt power outputs. An example of a Waukesha engine is pictured in Figure 3.14.

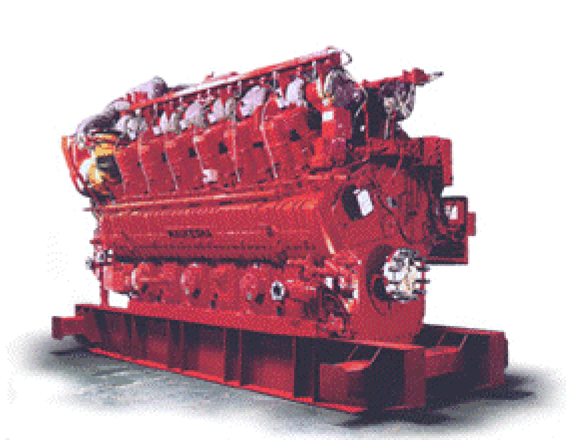


Figure 3.14: Gas Reciprocating Engine by Waukesha (www.waukeshaengine.com)

CHAPTER IV

COMBUSTION TURBINES

Technology Overview

The simplest model for gas turbines is the air-standard Brayton cycle. The air-standard model is comprised of isentropic compression and expansion processes, and reversible, constant-pressure heat addition and rejection. The working fluid is air and is modeled as an ideal gas with constant specific heat throughout the cycle. Figure 4.1 presents the temperature-entropy (Ts) and pressure-specific volume (pv) diagrams for the air-standard Brayton cycle.

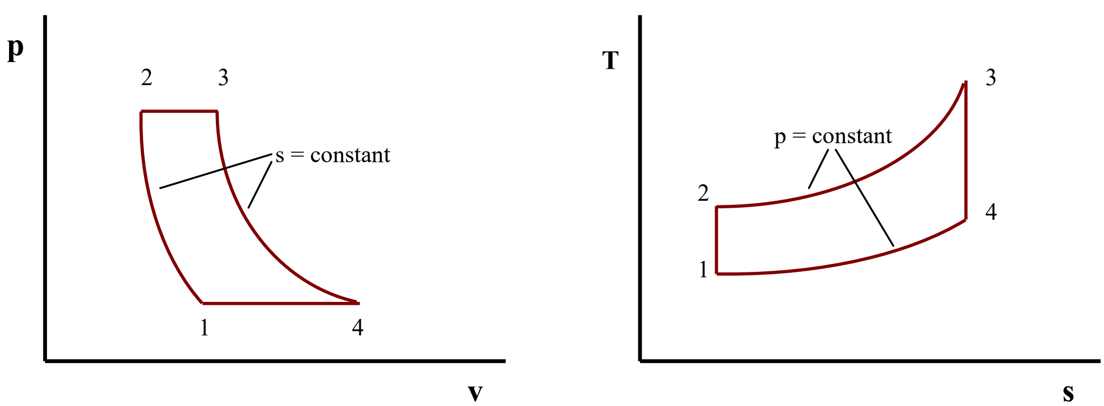


Figure 4.1: Pressure-Specific Volume and Temperature-Entropy Diagrams for the Ideal Brayton Cycle

Application of the first law to the model determines the expressions for turbine work (W_t), compressor work (W_c), net work (W_{net}), and cycle efficiency (η).

$$\begin{aligned} W_c &= h_1 - h_2 \\ &= c_p(T_1 - T_2) \end{aligned} \tag{4-1}$$

$$\begin{aligned} W_t &= h_3 - h_4 \\ &= c_p(T_3 - T_4) \end{aligned} \tag{4-2}$$

$$\begin{aligned} Q_S &= h_3 - h_2 \\ &= c_p(T_3 - T_2) \end{aligned} \tag{4-3}$$

$$W_{net} = W_c + W_t \tag{4-4}$$

$$\begin{aligned} \eta &= \frac{W_{net}}{Q_S} \\ &= \frac{(T_1 - T_2) + (T_3 - T_4)}{T_3 - T_2} \end{aligned} \tag{4-5}$$

Since the specific heats are constant, then

$$\frac{T_4}{T_1} = \frac{T_3}{T_2} \tag{4-6}$$

and

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{k-1/k} \tag{4-7}$$

Where k is the ratio of specific heats (c_p/c_v). A similar relationship can be written for T_3 and T_4 with P_3 and P_4. The cycle efficiency becomes

$$\eta = 1 - \frac{1}{\left(\frac{P_2}{P_1}\right)^{k-1/k}} \tag{4-8}$$

The air-standard Brayton cycle includes the assumptions that the compressor and turbine operate isentropically and that there is no pressure drop during heat addition and rejection. These assumptions are not realistic in a real gas turbine cycle. In a real gas turbine cycle, irreversibilities in the compression and expansion processes cause the working fluid to increase in specific entropy. Also, the heat addition in the combustor and the heat rejection between the turbine exhaust and the compressor inlet do not occur at constant pressure. However, the irreversibilities in the heat exchange process are often ignored since they are less significant than those in the compression and expansion processes. The Ts diagram in Figure 4.2 illustrates the effects of the irreversibilies in an actual gas turbine cycle.

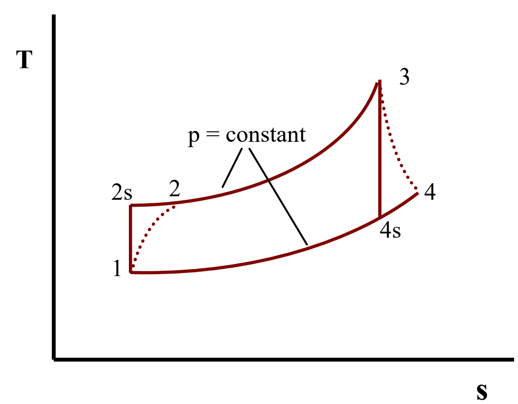


Figure 4.2: Temperature-Entropy Diagram of a Real Gas Turbine Cycle

The isentropic compressor efficiency is defined as the ratio of the isentropic compressive work to the actual compressive work, when both are compressed through the same pressure ratio. The isentropic turbine efficiency is defined as the ratio of the actual work of expansion to the isentropic work of expansion, when both expand from the same initial state to the same final pressure. The expressions for isentropic compressor efficiency (η_c) and isentropic turbine efficiency (η_t) are presented below.

$$\eta_c = \frac{h_{2S} - h_1}{h_2 - h_1} \quad (4\text{-}9)$$

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4S}} \quad (4\text{-}10)$$

For constant specific heats, the isentropic compressor and expansion efficiencies become

$$\eta_c = \frac{\left(\frac{P_2}{P_1}\right)^{k-1/k}}{\frac{T_2}{T_1} - 1} \quad (4\text{-}11)$$

$$\eta_t = \frac{1 - \frac{T_4}{T_3}}{1 - \left(\frac{P_4}{P_3}\right)^{k-1/k}} \quad (4\text{-}12)$$

Equation 4-8 shows that the thermal efficiency of the Brayton cycle is a function of the pressure ratio (P_2/P_1). A real gas turbine cycle is primarily described by the pressure ratio, the turbine inlet temperature, the compressor inlet temperature, the isentropic turbine efficiency, and the isentropic compressor efficiency. Equations 4-13 and 4-14 express the compressor work (W_{comp}) and the turbine work (W_{turb}) for a gas turbine cycle.

$$W_{comp} = \frac{c_p}{\eta_c} \cdot T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{k-1/k} \right] \tag{4-13}$$

$$W_{turb} = \eta_t \cdot c_p \cdot T_3 \left[1 - \left(\frac{P_4}{P_3} \right)^{k-1/k} \right] \tag{4-14}$$

where the constant pressure specific heat is that of air (c_p = 1.004 kJ/kg-K) for the compressor and that of the combustion products (c_p = 1.148 kJ/kg-K) for the turbine. Further, the ratio c_p/c_v is k = 1.4 for the compressor and k = 1.333 for the combustion products in the turbine. The constant pressure specific heat for the combustion process is generally taken as the average of the compressor and turbine c_p values, and the constant pressure specific heat in the heat rejection process is generally assumed to be that of air.

For a gas turbine power plant with prescribed pressure ratios, the work required of the compressor is a function of the inlet temperature (T_1). As T_1 decreases, the work required of the compressor decreases and the net work accomplished by the power plant is enhanced. However, a decrease in T_1 will result in a lower inlet temperature to the combustor (T_2), hence, the fuel/air ratio supplied at the combustor will have to increase to meet the prescribed turbine inlet temperature (T_3). An increase in T_3 will enhance both the turbine work and the net work of the power plant. T_3 is limited by the thermal properties of the materials used in construction of the turbine. An example simple-cycle gas turbine problem is presented in Figure 4.3.

Example 4-1:
A simple-cycle gas turbine has a turbine inlet temperature of 1,500 F, a compressor ratio of 14, and compressor and turbine efficiencies of 82% and 89%, respectively. The ambient conditions are 90 F and 1 atm. Sketch (a) the T-s diagram for this engine and determine (b) the net work for the cycle, (c) the engine thermal efficiency, and (d) the available waste heat rejected to the ambient air.

Solution:

Given information:

$T_1 := (460 + 90)R$ Compressor inlet temperature

$T_3 := (460 + 1500)R$ Turbine inlet temperature

$\eta_c := 0.82$ Compressor efficiency

$\eta_t := 0.89$ Turbine efficiency

$PR := 14$ Pressure ratio

Assumptions:
The following property values are used for the working fluid.

$c_{pc} := 1.004\frac{kJ}{kg \cdot K}$ Specific heat of air

$c_{pt} := 1.148\frac{kJ}{kg \cdot K}$ Specific heat of combustion gas

$c_{pAve} := \frac{c_{pc} + c_{pt}}{2}$ Average specific heat of combustion gas and air

$k_c := 1.4$ c_p/c_v for air

$k_t := 1.333$ c_p/c_v for combustion gas

(a) Sketch the T-s diagram for this engine.

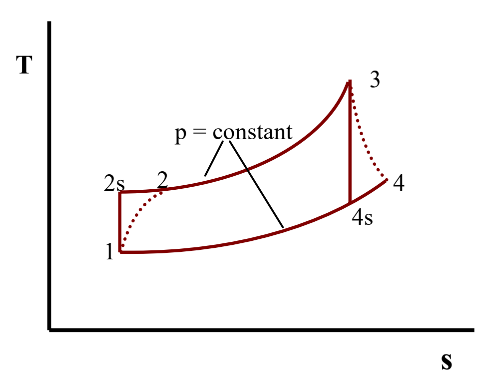


Figure 4.3: Simple-Cycle Gas Turbine Example Problem

(b) Determine the net work of the engine.

Compressor and turbine exit temperature:

$$T_2 := T_1 \cdot \left[1 + \eta_c^{-1} \cdot \left[(PR)^{\frac{k_c - 1}{k_c}} - 1\right]\right] \qquad T_2 = 1.305 \times 10^3\, R$$

$$T_4 := T_3 \cdot \left[1 - \eta_t \cdot \left[1 - \left(\frac{1}{PR}\right)^{\frac{k_t - 1}{k_t}}\right]\right] \qquad T_4 = 1.118 \times 10^3\, R$$

Compressor work and turbine work

$$W_c := c_{pc} \cdot (T_1 - T_2) \qquad W_c = -181.031 \frac{BTU}{lb}$$

$$W_t := c_{pt} \cdot (T_3 - T_4) \qquad W_t = 230.912 \frac{BTU}{lb}$$

The net work of the turbine can be found as follows:

$$W_{net} := W_c + W_t \qquad W_{net} = 49.881 \frac{BTU}{lb}$$ Net work

(c) Deternine the thermal efficiency of the turbine.

$$Q_s := c_{pAve} \cdot (T_3 - T_2) \qquad Q_s = 168.354 \frac{BTU}{lb}$$

$$\eta := \frac{W_{net}}{Q_s} \qquad \eta = 29.629\%$$ Thermal efficiency

(d) What is the amount of waste heat available from the engine.

$$Q_{waste} := c_{pc} \cdot (T_1 - T_4)$$

$$Q_{waste} = -136.172 \frac{BTU}{lb}$$ Available waste heat

Figure 4.3 (continued)

The basic combustion or gas turbine includes a compressor, a combustor, and a turbine, and operates on a single shaft. The schematic in Figure 4.4 illustrates the components of a simple gas turbine. A single shaft connects the compressor and the turbine, and a power turbine produces the net work of the system. The working fluid flows from the inlet of the compressor to the combustion chamber, where fuel is added and burned. The combustion gas then flows to the turbine and power turbine and is then exhausted.

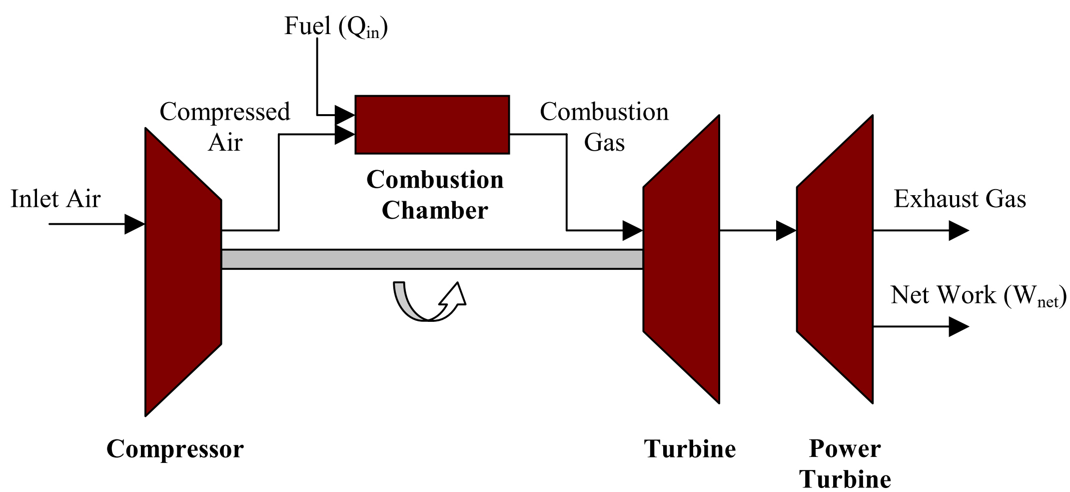


Figure 4.4: Gas Turbine Cycle

There are many variations of the basic combustion turbine cycle. One variation is the addition of a recuperator or regenerator. Turbine exhaust gases normally have temperatures well above ambient conditions, and these gases can be utilized for their energy content. A recuperator is a heat exchanger, which uses turbine exhaust gases to preheat compressed air before the air enters the combustor. Raising the temperature of air entering the combustor reduces the amount of fuel that must be burned, thus, increasing the overall efficiency of the power plant. A combustion turbine schematic with regeneration is presented in Figure 4.5.

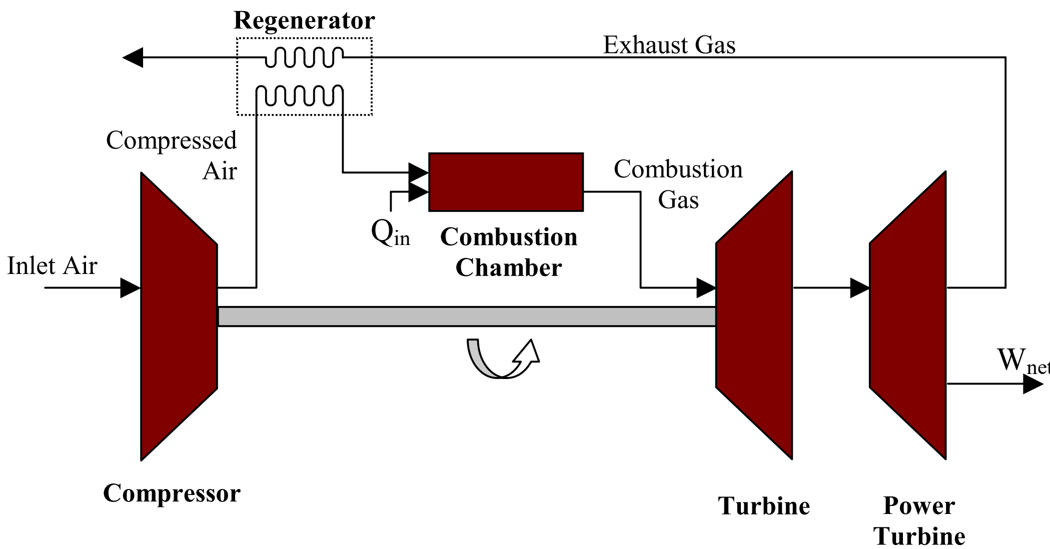


Figure 4.5: Gas Turbine Cycle with Regeneration

Another variation of the basic combustion turbine cycle is achieved with the addition of an intercooler. Adiabatic compression requires a significant amount of work but removing heat from the working fluid before compression can reduce this work. An intercooler reduces the temperature of the air during the compression cycle, thus, reducing the amount of work required of the compressor. However, heat transfer rates high enough to significantly reduce the compressor work are hard to achieve. Intercooling is sometimes achieved by implementing multiple intercoolers into several stages of the compressor. Figure 4.6 illustrates a combustion turbine with an intercooler between the low-pressure and the high-pressure compressor stages.

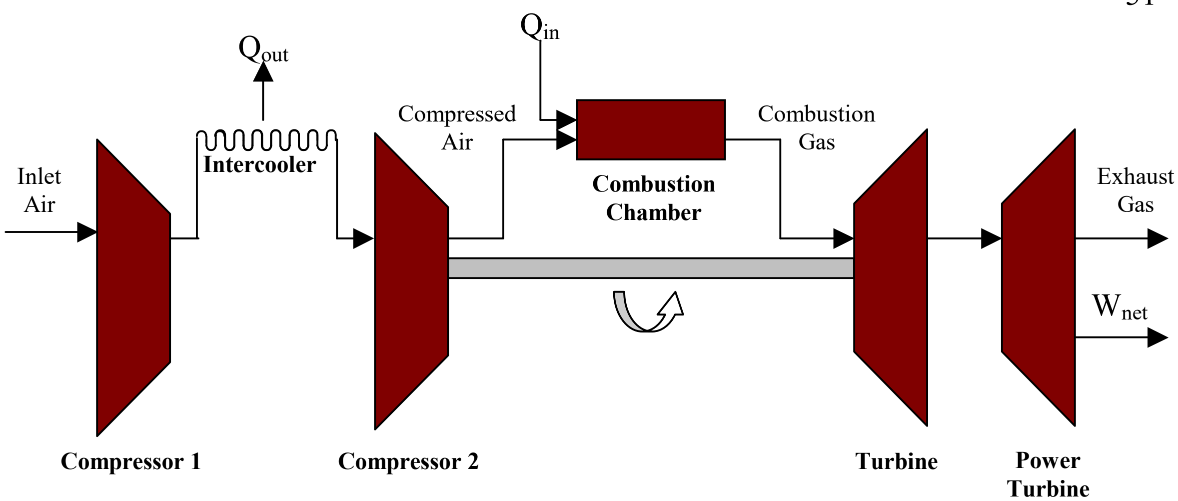


Figure 4.6: Gas Turbine Cycle with Intercooler and Two-Stage Compression

Another variation of the basic combustion turbine cycle is a two-stage combustion turbine with reheat. As previously discussed, the temperature of combustion gases entering the turbine is limited due to the turbine blading material properties. Excess air, combined with combusted air to control the temperature during expansion, oxygenates the turbine exhaust. The oxygen-enriched turbine exhaust can be used to attain an additional combustion process. This addition of energy in a staged turbine is called reheat. A reheat combustor raises the temperature of the incoming exhaust gases for the downstream turbine section. These combustion gases are then expanded through an additional turbine stage. Gas turbines with reheat have higher exhaust gas temperatures exiting the primary turbine than gas turbines without reheat; therefore, a gas turbine with reheat has greater potential for regeneration. Gas turbines with reheat, like the one illustrated in Figure 4.7, require additional fuel but produce a greater amount of work at higher overall efficiencies than gas turbines without reheat.

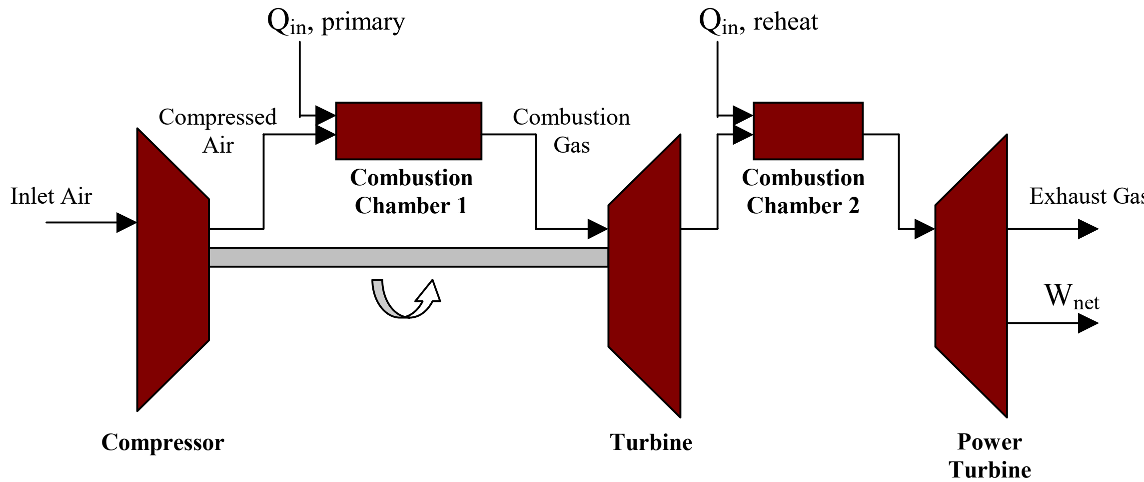


Figure 4.7: Gas Turbine Cycle with Reheat

Regeneration and reheat are variations of the basic combustion cycle that utilize the energy content of turbine exhaust. Turbine exhaust gases not only can enhance efficiency and power production in combustion turbines, but can also be used for thermally-activated systems.

Industrial Turbines

Single-shaft combustion turbines, like the one illustrated in Figure 4.8, are commonly used in industry for power generation. An inlet section (1), a compressor section (2), a combustion system (3), a turbine section (4), and an exhaust system are standard components in industrial turbines. The combustion turbine in Figure 4.8 operates similar to the combustion turbine cycle presented in Figure 4.4. An industrial combustion turbine does not include a separate power turbine; instead, the excess shaft power is harnessed to produce electricity or mechanical power.

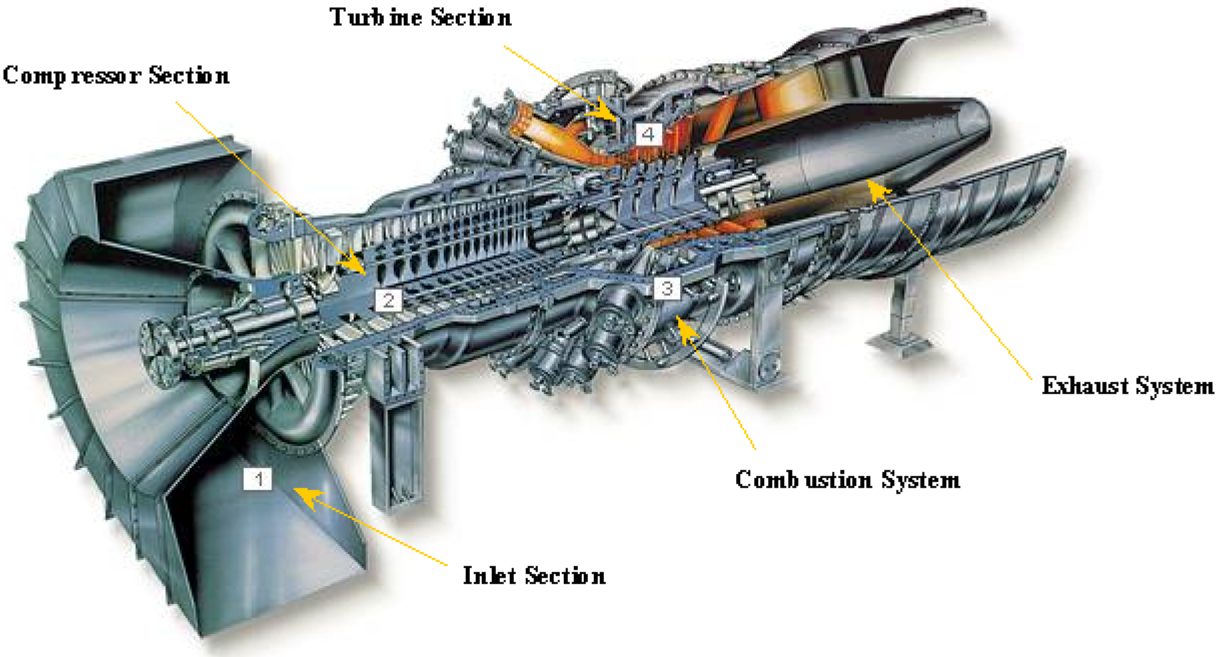


Figure 4.8: Single-shaft Industrial Combustion Turbine by Siemens Westinghouse (www.siemenswestinghouse.com)

Application

Nearly all new power plants are combined cycle (reheat) combustion turbines. Industrial turbines are available in capacities from 0.5-MW to 250-MW. This broad range of power generation capacity meets the electrical demand of most institutional, commercial, and industrial end-users. End-users often require lower capacity turbines for their distributed power generation (DPG) applications. Small combustion turbines are found in a broad array of applications including mechanical drives, base-load grid-connected power generation, and remote off-grid applications. Table 4.1 presents an overview of combustion turbine characteristics.

Table 4.1: Overview of Industrial Turbines (www.energy.ca.gov/distgen//distgen/)

Combustion Turbine Overview	
Commercial Status	Widely Available
Size Range	500 kW – 250 MW
Fuel	Natural gas, liquid fuels
Single-Cycle Efficiency **(Aeroderivative Turbines)**	26-45 % (primarily size dependent)
Single-Cycle Efficiency **(Industrial or Frame Turbines)**	20-34 % (primarily size dependent)
Combined Cycle Efficiency	Approaches 60 % in larger units
Environmental Effects	Very low when controls are used
Other Features	Cogen (steam)

Aeroderivative gas turbines are an additional class of turbines used for stationary power. These turbines are adapted from their aircraft jet engine counterpart and are light weight with higher simple-cycle efficiencies than the more rugged industrial turbines. Aeroderivative turbines are limited to capacities below 55-MW and are, therefore, less common in industrial, institutional, and commercial applications. However, since the power requirements of CHP-B applications are usually below 55-MW, aeroderivative turbines remain a viable component in CHP-B applications.

In an increasingly-competitive electricity market, installation of small industrial and aeroderivative gas turbines is a cost-effective alternative to grid power. Industrial turbines generally operate for longer periods between overhauls than do aeroderivative turbines and are especially suited for continuous base-load operation.

Heat Recovery

Simple-cycle gas turbines are the least efficient configuration since there is no recovery of heat from the exhaust gas. Simple-cycle efficiencies typically range from 20

% to 45 % depending on the capacity and the type of the turbine. The hot exhaust gas can be used directly in a process or used to generate steam or hot water with a heat recovery steam generator (HRSG). As with the other DPG technologies, steam and hot water produced from recovered heat can be used for space heating, reheat, domestic hot water, absorption cooling, and desiccant regeneration.

Combined-cycle gas turbines, typically for larger installations, can achieve up to 60 % electric generation efficiencies using the most advanced utility-class turbines. Waste heat from these combined cycle turbines can be recovered in an HRSG, similar to that in the simple cycle. Since a gas turbine exhaust is oxygen rich, the gas can support additional combustion through supplemental firing. A duct burner is usually fitted within the HRSG to increase the exhaust gas temperature. With the additional thermal energy captured from exhaust gases, combined cycle turbines can achieve overall thermal efficiencies greater than 90 %. (http://www.eren.doe.gov/der/chp/pdfs/chprev.pdf)

Cost

Gas turbines are relatively inexpensive compared with other DPG technologies. The capital cost of combustion turbines ranges from $300-$1000/kW and, generally, increases with decreasing power output. Combustion turbines tend to cost more than internal combustion (IC) engines of smaller capacities and cost less than larger capacity IC engines. These costs have remained fairly stable in recent history, showing less than a 5 % increase over the past three years. (www.energy.ca.gov/distgen/)

Installation costs, balance of plant (BOP) equipment costs and other owner or miscellaneous costs can be expected to increase initial capital costs by 30-50 %. A natural gas compressor is an example of the BOP equipment required. Natural gas compressors are needed to meet the high gas pressure requirements of combustion turbines, unless high-pressure cross-country pipelines are accessible. Natural-gas compressors increase the first costs of a combustion turbine system by 5-10 %. Also, adding heat recovery capabilities increases the capital cost by $100-$200/kW. Including other BOP components, the typical installed cost of a mid-sized combustion turbine with a heat recovery unit will be in the $1,000-$1,200/kW range. The pie chart in Figure 4.9 shows an example breakdown of the total installed cost of a 15-MW combustion turbine. (www.energy.ca.gov/distgen/)

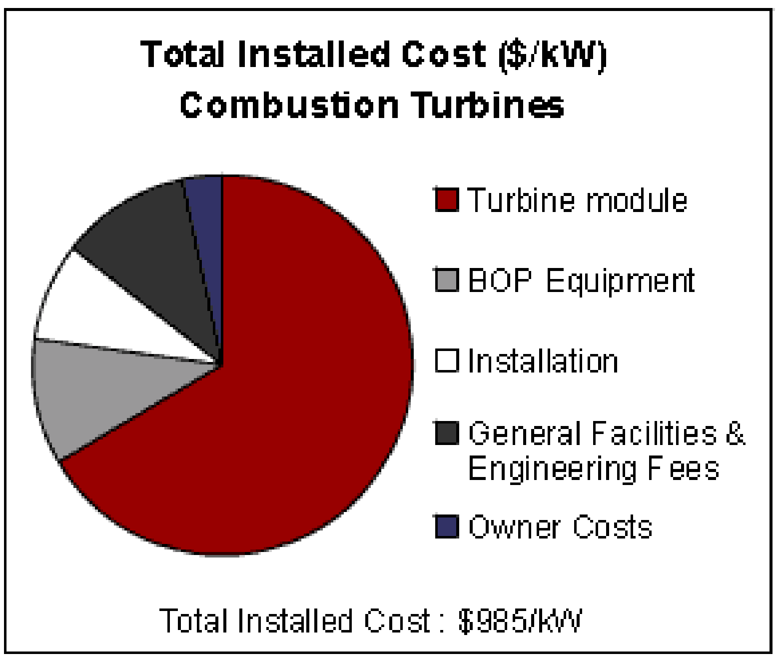


Figure 4.9: Total installed cost of a 15-MW Natural Gas Combustion Turbine (www.energy.ca.gov/distgen/)

Gas turbines can be cycled; however, maintenance costs can triple for a turbine that is cycled every hour versus a turbine that is operated for intervals of 1000 hours. The number of inspections and overhauls substantially increase when turbines are operated over their rated design capacity for significant periods of time. Maintenance costs of a turbine operating on fuel oil are approximately three times that of a turbine operating on natural gas. Maintenance costs associated with industrial turbines are much less significant than the maintenance costs associated with IC engines. Typical maintenance costs for an industrial turbine fired by natural gas are \$0.003 - \$0.005/kWhr. (http://www.eren.doe.gov/der/chp/pdfs/chprev.pdf)

Industrial Turbines and CHP-B

The use of combustion turbines at industrial facilities, institutional campuses, and commercial buildings has proven to be a successful method for meeting distributed power demands. Current practice with combustion turbines demonstrates that these turbines can supply electrical, mechanical, and thermal energy that end users need to power diverse operations. Combustion turbines are implemented to provide power for manufacturing processes and to meet facility and process electrical and heating requirements. To be a viable component for CHP-B, combustion turbines must be available to meet smaller power demands and to produce a useful thermal energy.

Compared to IC engines, industrial and aeroderivative combustion turbines have lower costs at higher capacities. Combustion turbines are available over a much wider rage of power output than other DPG technologies and have the capability of producing high-temperature steam from the exhaust heat. The advantages of industrial-size combustion turbines suggest that turbines may be effective in meeting the cooling, heating, and power requirements of large buildings. As power requirements exceed a megawatt, building owners may experience lower initial costs and lower operating and maintenance costs with a combustion turbine than with other DPG prime movers. Like IC engines, industrial combustion turbines are widely available, have a well-established marketing and customer-service networks, and have reputations for reliability.

The cost per unit of output power for acquiring an industrial-size combustion turbine increases significantly as the turbine size decreases. Additionally, the efficiency of industrial turbines decreases with decreasing power output. Since many CHP-B applications are intended for buildings requiring power levels at the low end of the industrial turbine power range, industrial turbines may not be the preferred DPG component for these applications.

Another disadvantage to using combustion turbines for CHP-B is the reduced efficiency of turbines at part loads. Most buildings will experience fluctuating power requirements due to seasonal operation, reduced workloads at nights and weekends, and seasonal heating and cooling requirements. Combustion turbines installed to meet peak power requirements will experience reduced efficiencies as these power requirements fluctuate. However, installation of multiple turbines and connection to the electricity grid can counteract many of these disadvantages.

Combustion turbines are very sensitive to ambient conditions. Ambient pressure and temperature affect the efficiencies of compression and combustion in a turbine system. If ambient conditions change significantly, a combustion turbine may not be able to meet peak-power requirements. This disadvantage of combustion turbines can be negated if the turbine is contained in an area where ambient air conditions are controllable. When acceptable noise levels can be established, CHP-B systems driven by combustion turbines can be housed inside a building where air temperature, pressure, and humidity can be maintained near ideal conditions. Table 4.2 lists advantages and disadvantages of combustion turbines as a DPG technology.

Table 4.2: Advantages and Disadvantages of Combustion Turbines as a DPG Technology (www.energy.ca.gov/distgen/)

Manufacturers

There are over a dozen manufacturers of combustion turbine products around the world. The following list, originating from (www.energy.ca.gov/distgen/), identifies some of these manufacturers.

- General Electric Power Systems has a distributed power division in Schenectady, New York. GE Power Systems manufactures heavy duty and aeroderivative turbines used in mechanical and electrical power production, like the gas turbine pictured in Figure 4.10. GE also manufactures steam turbines systems, hydropower systems, and combined-cycle systems.

Figure 4.10: 28-kW Gas Turbine by GE Power Systems
(www.gepower.com)

- Pratt & Whitney, a division of United Technologies Corporation, produces combustion turbines for power generation in the range of 20-MW to 50-MW. Figure 4.11 presents a 25-kW turbine by Pratt & Whitney.

Figure 4.11: 25-kW Natural Gas Turbine by Pratt & Whitney
(www.pratt-whitney.com)

- Rolls-Royce North America, in Chantilly, VA, manufactures gas turbines with power output ranging from 2.2-MW to 51.2-MW. An aeroderivative gas turbine by Rolls-Royce is shown in Figure 4.12.

Figure 4.12: 51.2-MW Aeroderivative Gas Turbine by Rolls-Royce
(www.rolls-royce.com)

- Siemens Westinghouse Power Corporation has many locations in North America and offers gas turbines in the range of 67-MW to 265-MW. An industrial gas turbine by Siemens Westinghouse is presented in Figure 4.13.

Figure 4.13: 157-MW Gas Turbine by Siemens Westinghouse (www.siemenswestinghouse.com)

- Solar (owned by Caterpillar), located in San Diego, California, is a leader in the design and manufacture of gas turbine power generation systems up to 15-MW. Figure 4.14 illustrates the Titan 130, a 14 MW gas turbine-generator set by Solar.

Figure 4.14: 14-MW Gas Turbine Generator Set by Solar (http://esolar.cat.com)

- Vericor Power Systems, located in Alpharetta, Georgia, offers products, like the natural gas turbine generator in Figure 4.15, and support services in the 0.5-MW to 50-MW power range.

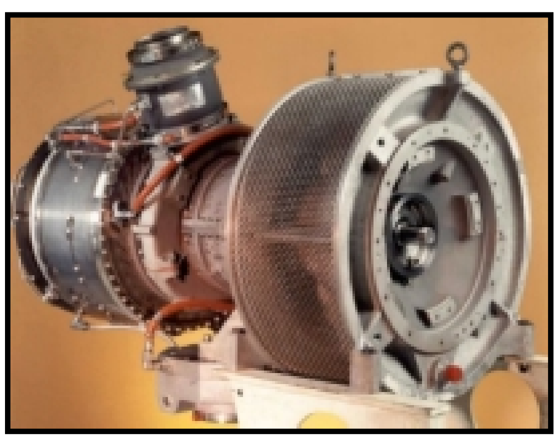


Figure 4.15: 3-MW Natural Gas Turbine Generator by Vericor Power Systems (www.vericor.com)

Microturbines

The design of a microturbine is similar to the design of an industrial turbine, except that most microturbine designs incorporate a recuperator to recover part of the exhaust heat for preheating the combustion air (see Figure 4.5). Microturbines, like industrial turbines, have an inlet, a compressor, a combustor, a turbine, and an exhaust outlet. The microturbine generator configuration shown in Figure 4.16 includes these standard combustion turbine components as well as a recuperator.

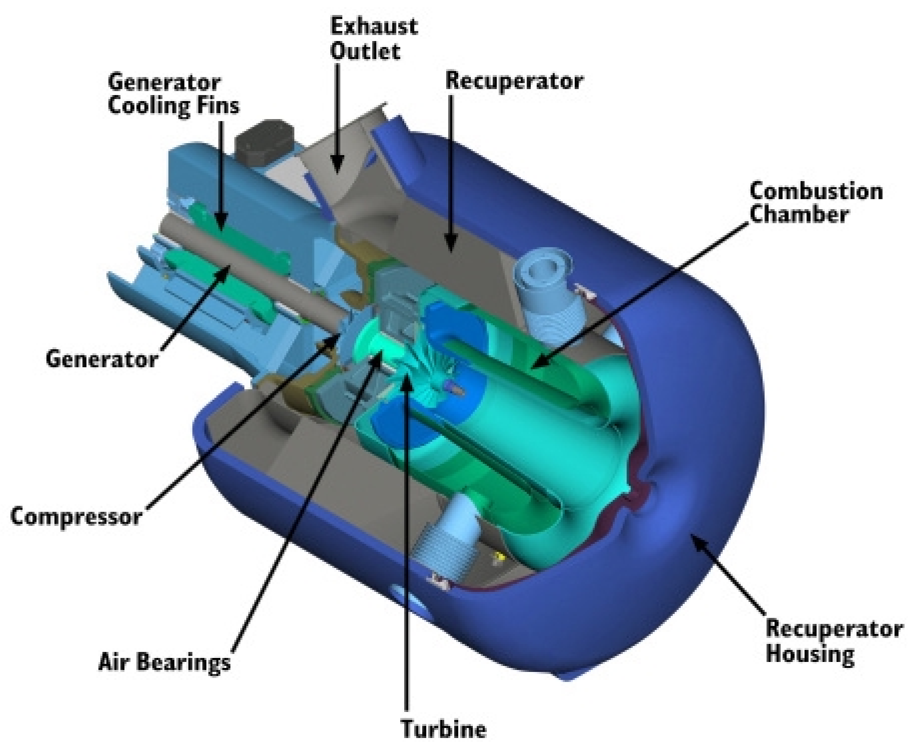


Figure 4.16: Natural Gas Microturbine Generator by Capstone (www.capstone.com)

Unrecuperated microturbines produce electricity from natural gas at efficiencies around 15 %. However, most microturbines are equipped with recuperators, which result in electrical generation efficiencies that range from 20 % to 30 %. Recuperated microturbines can produce 30 % to 40 % fuel savings by preheating the incoming combustion air. Combined with thermal efficiencies acquired through cogeneration applications, overall system efficiencies of microturbine generator systems coupled with thermally-activated technologies can reach 85 %. As compared to industrial turbines, microturbines have very low emissions. Three Capstone microturbines in series are pictured in Figure 4.17.

Figure 4.17: Microturbine generator System by Capstone (www.capstone.com)

Microturbines have much smaller physical dimensions than industrial turbines. With power outputs ranging from 25-kW to 500-kW, many commercially available microturbines are similar in size to a refrigerator. One benefit of smaller component size is lubrication-free air bearings. The shaft of a microturbine can be supported by air instead of lubricating oil. Using air instead of oil reduces maintenance requirements and, thus, maintenance costs.

Most microturbine manufacturers produce single-shaft microturbines that operate at speeds of up to 120,000 rpm. Dual-shaft designs are available to meet electrical and mechanical power demands. Dual-shaft microturbines have an additional power turbine and gear for mechanical drive applications, operating at speeds of up to 40,000 rpm. Table 4.3 lists an overview of microturbine technology. An example problem involving a microturbine is presented in Figure 4.18.

Table 4.3: Overview of Microturbines (www.energy.ca.gov/distgen/)

Microturbine Overview	
Commercial Status	Limited Availability
Size Range	25 kW – 500 kW
Fuel	Natural gas, hydrogen, propane, diesel
Unrecuperated Efficiency	15 %
Recuperated Efficiency	20-30 %
Environmental Effects	Very low (<9-50 ppm) NOx
Other Features	Cogen (50-80ûC water)

Example 4-2:
A microturbine operates in an office building with an indoor temperature of 70 F. The turbine inlet temperature is 1,850 F and the compressor and turbine have efficiencies of 0.82 and 0.85 respectively. If the compressor pressure ratio is 9, determine (a) the work c the compressor and the turbine, (b) the net work of the turbine, and (c) the thermal efficiency of the microturbine. The gas flow rate is 0.05 lb/sec, and the air flow is 0.6 lb/s.

Solution:

Given information:

$T_1 := (460 + 70)R$	Compressor inlet temperature
$T_3 := (460 + 1350)R$	Turbine inlet temperature
$\eta_c := 0.82$	Compressor efficiency
$\eta_t := 0.85$	Turbine efficiency
$m_{gas} := 0.05\frac{lb}{s}$	Mass flow rate of gas
$m_{air} := 0.6\frac{lb}{s}$	Mass flow rate of air
$PR := 9$	Pressure ratio

Assumptions:
The microturbine operates as a simple gas turbine cycle with the following properties for the working fluid.

$c_{pc} := 1.004\frac{kJ}{kg \cdot K}$	Specific heat of air
$c_{pt} := 1.148\frac{kJ}{kg \cdot K}$	Specific heat of combustion gas
$c_{pAve} := \frac{c_{pc} + c_{pt}}{2}$	Average specific heat of combustion gas and air
$k_c := 1.4$	c_p/c_v for air

Figure 4.18: Microturbine Example Problem

(a) Determining the work of the compressor and turbine requires the temperatures at the compressor exit and turbine exit to be calculated.

$$T_2 := T_1 \cdot \left[1 + \eta_c^{-1} \cdot \left[(PR)^{\frac{k_c - 1}{k_c}} - 1 \right] \right] \qquad T_2 = 1.095 \times 10^3 R$$

$$T_4 := T_3 \cdot \left[1 - \eta_t \cdot \left[1 - \left(\frac{1}{PR} \right)^{\frac{k_t - 1}{k_t}} \right] \right] \qquad T_4 = 1.16 \times 10^3 R$$

Determine the mass flow rate into the turbine

$$m_{comb} := m_{air} + m_{gas} \qquad m_{comb} = 0.65 \frac{lb}{s}$$

Compressor work and Turbine Work

$$W_c := m_{air} \cdot c_{pc} \cdot (T_1 - T_2) \qquad W_c = -85.699 kW$$ Compressor work

$$W_t := m_{comb} \cdot c_{pt} \cdot (T_3 - T_4) \qquad W_t = 122.203 kW$$ Turbine work

(b) The net work of the microturbine can be found as follows,

$$W_{net} := W_c + W_t \qquad W_{net} = 36.504 kW$$ Net work

(c) Deternine the thermal efficiency of the turbine.

$$Q_s := m_{comb} \cdot c_{pAve} \cdot (T_3 - T_2) \qquad Q_s = 126.096 kW$$

$$\eta := \frac{W_{net}}{Q_s} \qquad \eta = 28.949\%$$ Thermal efficiency

Figure 4.18 (continued)

Application

Markets for microturbines include commercial and light industrial facilities. These customers often pay more for electricity than larger end-users and can potentially utilize microturbines as a cost-effective alternative to the electricity grid. Microturbines have a relatively modest heat output that is ideally matched to end-users with low pressure steam or hot water requirements. Electric generation applications suited to microturbines are standby power, peak shaving, and base loaded operation with and without heat recovery. Single-shaft microturbines are optimal for these electric generation applications. Two-shaft turbines are used to supply electrical power and can drive chillers or air compressors.

Microturbines are being developed to utilize a variety of fuels and are being used for resource recovery and landfill gas applications. Since microturbines produce between 25-kW and 500-kW of power, they are well-suited for small commercial building establishments such as restaurants, hotels/motels, small offices, and retail stores.

Heat Recovery

Most microturbine designs incorporate a recuperator, which limits the amount of heat available for CHP applications. Recuperators are utilized as a method of heat recovery to significantly increase the operating efficiency of a microturbine generator system. The remaining hot exhaust gas from the turbine section is available for CHP applications. Heat can be recovered from microturbine exhaust for hot water heating and for low-pressure steam used in process and space heating applications. An 80-kW microturbine generator system with heat recovery manufactured by Bowman Power Systems is illustrated in Figure 4.19.

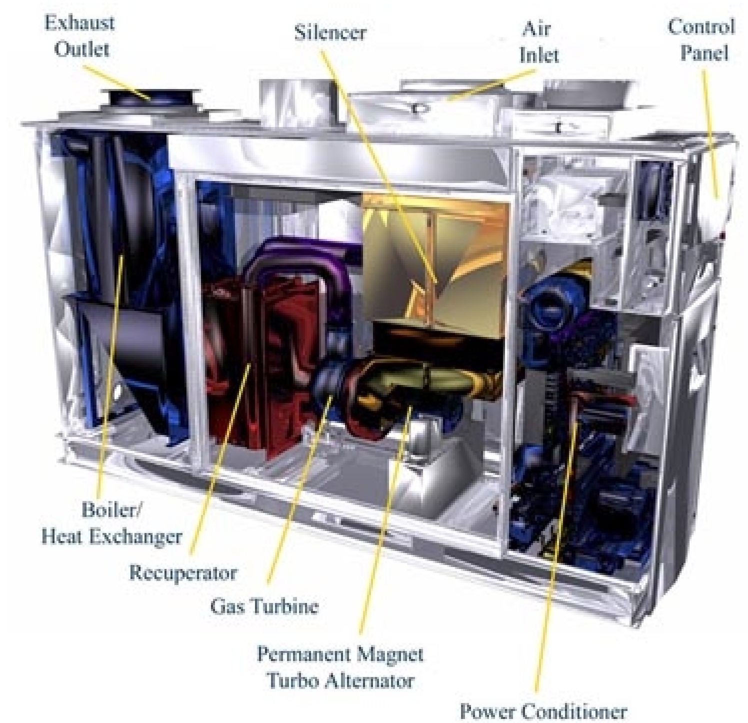


Figure 4.19: TG80 Microturbine generator System with Heat Recovery
(www.bowmanpower.com)

A system performance chart for the Bowman TG80 is presented in Figure 4.20. The chart illustrates the importance of recuperation to system efficiency as well as the loss of available thermal energy with the addition of a recuperator. With recuperation, the TG80 microturbine generator system yields an electric generation efficiency of 26 % leaving 150-kW of available thermal energy. Without recuperation, the TG80 system yields an electric generation efficiency of 14 % with 420-kW of available thermal energy. There is a significant increase in the overall efficiency of both the recuperated and unrecuperated systems when the thermal energy is recovered.

TG80 System Performance

RECUPERATOR STATE		90% EFFECTIVE RECUPERATOR	NO RECUPERATOR SIMPLE CYCLE
Thermal*	kW(th)	150	420
Electrical Output Power	kW(e)	80	80
Generating Set Efficiency	%	26	14
Overall System Efficiency	%	74	>80

Figure 4.20: TG80 System Performance Chart (www.bowmanpower.com)

Cost

Microturbine capital costs range from \$700 - \$1,100/kW. These costs include all hardware, associated manuals, software, and initial training. Adding heat recovery components increases the capital cost by \$75 - \$350/kW. Installing the system typically increases the capital cost by 30-50 %.

Targeted microturbine costs appear to be attainable if the market expands and sales volumes increase. With fewer moving parts than reciprocating engines, microturbines provide higher reliability than conventional reciprocating generation technologies. Manufacturers expect microturbines to require a once-a-year maintenance schedule when the technology matures and are targeting maintenance intervals of 5,000-8,000 hours. Forecasted maintenance costs for microturbine units are \$0.005-\$0.016 per kWh, similar to those of small reciprocating engine systems.

(www.energy.ca.gov/distgen/)

Microturbines and CHP-B

Low-level power outputs well suit microturbine generators to CHP-B applications, especially in commercial and light industrial facilities. Restaurants, hotels/motels, hospitals, small offices, retail stores, and other establishments that could benefit from DPG technologies require power components that have compact sizes, low noise levels, and low emissions. IC engines can supply equivalent levels of power output as microturbines, but don't share advantages such as low noise levels and low emissions that make microturbines desirable to commercial end-users. Microturbines benefit light industrial facilities by providing stand-by power, peak-power shaving, and power for base load operation.

Microturbines have fewer moving parts in comparison with IC engines. Fewer moving parts and low lubrication requirements provide microturbines with long maintenance intervals. Both commercial and industrial end-users that have a need for microturbine waste heat benefit from system cogeneration efficiencies of over 80 %. Another advantage of microturbines is their potential to operate on waste fuels such as recovered resources and landfill gas.

Similar to larger combustion turbines, microturbines suffer a loss of power output and efficiency as the ambient temperature and elevation increase. Higher initial costs are a disadvantage of microturbines when compared with the initial costs of IC engines. With recuperation, microturbines reach electric generation efficiencies near 30 %. This fuel to electricity efficiency is lower than the efficiencies of competing DPG technologies. There are many advantages that microturbines have over other DPG components; however, low efficiencies and high costs are two disadvantages that currently hinder widespread acceptance of the technology. Table 4.4 lists advantages and disadvantages of microturbines in distributed power generation.

Table 4.4: Advantages and Disadvantages of Microturbines as a DPG Technology (www.energy.ca.gov/distgen/)

Manufacturers

There are more than twenty companies worldwide that are involved in the development and commercialization of microturbines for distributed energy resource applications. Five of the leading microturbine manufacturers are listed here.

- Bowman Power Systems is an U.K. company that develops 80-kW microturbine power generation systems for distributed energy resource and mobile power applications. The Bowman TG80 microturbine is shown in Figure 4.21.

Figure 4.21: TG80 Microturbine generator System by Bowman Power Systems (www.bowmanpower.com)

- Capstone Turbine Corporation, based in Chatsworth, California, is a leader in the commercialization of low-emission, high-reliability microturbine power generators. The company offers 30-kW and 60-kW systems for distributed energy resource applications. A series of Capstone Turbine microturbines are pictured in Figure 4.22.

Figure 4.22: Microturbine generator Sets by Capstone Turbine (www.capstone.com)

- Elliot Energy Systems, located in Stuart, Florida, develops and manufactures 80-kW microturbines. Figure 4.23 presents an Elliot microturbine.

Figure 4.23: 80-kW Microturbine by Elliot Energy Systems (www.elliot-turbo.com)

- Ingersoll Rand Energy Systems of Portsmouth, New Hampshire develops the PowerWorks™ line of microturbine generators with an output of 70-kW, like the microturbine-generator pictured in Figure 4.24.

Figure 4.24: 70-kW PowerWorks Microturbine Generator by Ingersoll Rand (www.ingersoll-rand.com)

- Turbec AB is a Swedish company jointly owned by ABB and Volvo Aero. The company offers a 100-kW microturbine power generator for commercial distributed energy resource applications. Figure 4.25 shows a Turbec microturbine.

Figure 4.25: 100-kW Microturbine Generator System by Turbec AB (www.turbec.com)

Combustion Turbine Problems

1. An ideal Brayton cycle is modeled on an air-standard basis. The compression ratio of the cycle is 11with air entering the compressor at T1 = 70°F, p1 = 14.7 psi, with a mass flow rate of 85,000 lb/hr. If the turbine inlet temperature is 1,750°F, determine (a) the thermal efficiency, (b) the net power developed, and (c) the amount of waste heat available for recovery.

2. The net power developed by an ideal air-standard Brayton cycle is 3.4×10^7 Btu/hr. The pressure ratio for the cycle is 10, and the minimum and maximum temperatures are 520°R and 2800°R, respectively. Calculate (a) the thermal efficiency of the cycle, (b) the mass flow rate of the air, and (c) the amount of waste heat available for recovery.

3. The minimum and maximum temperatures of an ideal Brayton cycle are 290 K and 01650 K respectively. The pressure ratio is that which maximizes the net work developed by the cycle per unit mass of air-flow. Using a cold air-standard analysis, determine (a) the compressor work per unit mass of air-flow, (b) the turbine work per unit mass of air-flow, and (c) the thermal efficiency of the cycle.

4. A simplified analysis is performed on a gas turbine engine cycle with a compression ratio of 8 to 1. The compressor and turbine inlet temperatures are 320 K and 1450 K, respectively. The compressor has an efficiency of 0.84 and the turbine has an efficiency of 0.88. Calculate the thermal efficiency of (a) the ideal Brayton cycle, (b) the actual gas turbine cycle, and (c) the amount of waste heat available for recovery.

5. A gas turbine power plant produces 5000 kW of shaft power from inlet air at 97 kPa and 30°C. The compressor has a compression ratio of 5.5 and an isentropic efficiency of 0.84. In the combustion chamber, there is a pressure loss equal to 3 percent of the inlet air pressure and the outlet temperature is 1000°C. If the turbine has an isentropic efficiency of 0.88 and an exhaust pressure of 100 kPa, determine the air flow rate and power plant thermal efficiency.

6. A stationary gas turbine has compressor and turbine efficiencies of 0.85 and 0.90, respectively, and a pressure ratio of 20. Determine the work of the compressor and the turbine, the net work, the turbine exit temperature, and the thermal efficiency for 80°F ambient and 1900°F turbine inlet temperatures.

7. A 25-kW microturbine meets all of the electricity requirements of an office building. The set point for the thermostat remains at 68°F. The turbine inlet temperature is 1,800°F and the compressor and turbine have efficiencies of 0.72 and 0.80, respectively. The shaft work produced by the turbine is completely converted into electrical power. If the compressor pressure ratio is 8, determine the compressor and turbine work in kJ/kg, the net work in kJ/kg, and the thermal efficiency of the microturbine.

8. Determine the required COP of an absorption chiller that must provide 20 tons of cooling for the office building in Problem 7. The exhaust heat of the microturbine is transferred to the absorption chiller by a direct pipe that is 90 % effective and the exhaust leaves the chiller at a temperature of 350°F. (1 ton = 12,000 Btu/hr)

9. The exhaust of a 2.5-MW gas turbine operating at a 900°C turbine inlet temperature with a compressor pressure ratio of 9 transfers heat without loss to operate an absorption chiller with a COP of 0.85. The exhaust gas leaves the chiller at a temperature of 110°C. The compressor inlet conditions are 105 kPa and 35°C, and the compressor and turbine isentropic efficiencies are 82 % and 87 % respectively. How many tons of refrigeration can be produced by the chiller?

CHAPTER V

FUEL CELLS

Technology Overview

A fuel cell is an electrochemical energy conversion device that converts hydrogen and oxygen into electricity and heat. Fuel cells are similar to batteries in that they both produce direct current (DC) through an electrochemical process without the direct combustion of a fuel source. However, whereas a battery delivers power from a finite amount of stored energy, fuel cells can operate indefinitely provided that a fuel source is continuously supplied. The input fuel passes over an anode where a catalytic reaction splits the fuel into ions and electrons. The ions pass from the anode, through an electrolyte, to an oxygen-rich cathode. Electrons pass through an external circuit to serve an electric load. An individual fuel cell produces between 0.5 – 0.9 volts of DC electricity. Fuel cells are combined into "stacks" like a battery to obtain usable voltage and power output. A single fuel cell "stack" is illustrated in Figure 5.1.

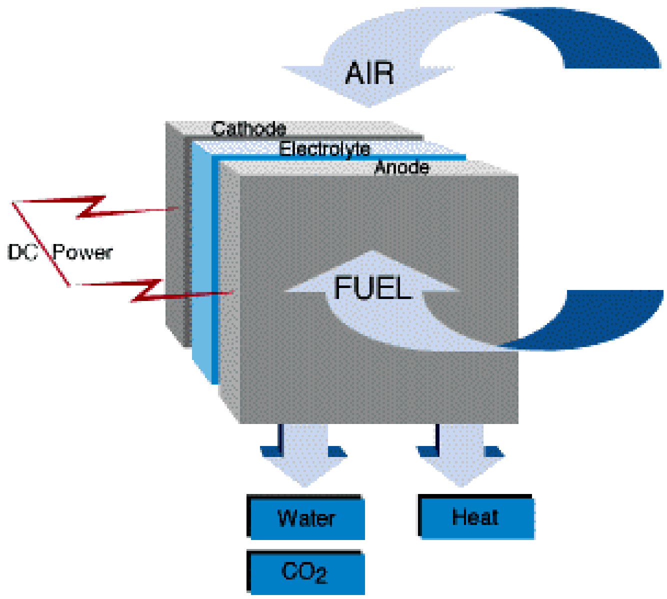


Figure 5.1: Single Stack of a Fuel Cell (www.dodfuelcell.com)

A fuel cell system consists of several major components including 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 direct current (DC) electricity generated in the fuel cell into alternating current (AC) electricity at the appropriate grid voltage. 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, meeting specific voltage and frequency requirements. A fuel cell system schematic is presented in Figure 5.2.

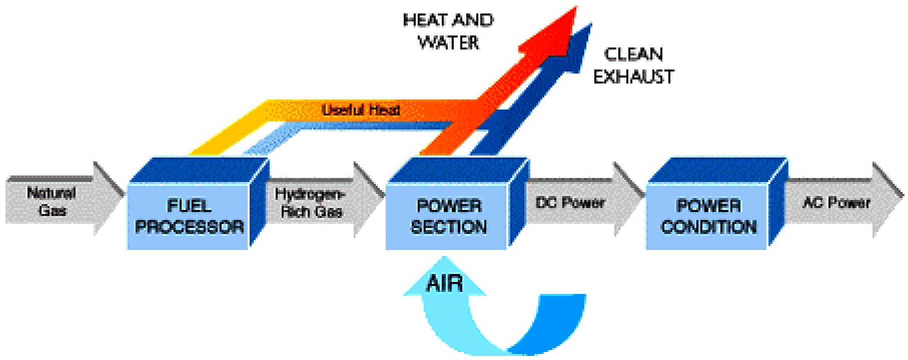


Figure 5.2: Fuel Cell System Schematic (www.dodfuelcell.com)

Natural gas (methane) is considered to be the most readily available fuel and the cleanest fuel (next to hydrogen) for powering fuel cells; thus, most research and development is focused on natural-gas-powered fuel cells. However, since fuel cells require hydrogen gas to operate, they can also be powered by propane, diesel, fuel oil, bio-derived gas, and other fuels.

Reforming the fuel "frees" the hydrogen and removes contaminants that would otherwise poison the catalytic electrodes. Storage of the reformed, hydrogen-rich mixture is usually unnecessary since the fuel is processed at the point of use. Fuel reforming can occur externally or internally depending on the fuel cell's operating temperature.

Most fuel cells have a similar design to the system illustrated in Figure 5.1, but differ with respect to the type of electrolyte used. The five main types of fuel cells, as classified by their electrolytes, are alkaline, phosphoric acid (PAFC), molten carbonate (MCFC), solid oxide (SOFC), and proton exchange membrane (PEMFC) fuel cells. Alkaline fuel cells require very pure hydrogen that is expensive to produce and for this reason are not considered as a major contender for distributed power generation technology.

PAFC

Phosphoric acid fuel cells are generally considered "first generation" technology. These fuel cells operate at about 200°C (400°F) and achieve 40 % to 45 % fuel-to-electricity efficiencies on a lower heating value basis (LHV). The PAFC is commercially available with over 245 units in operation.

The PAFC uses liquid phosphoric acid as an electrolyte. Platinum-catalyzed, porous-carbon electrodes are used for both the cathode and anode. The PAFC reactions are:

$$\text{Anode Reaction}: H_2 \rightarrow 2H^+ + 2e^-$$
$$\text{Cathode Reaction}: \frac{1}{2}O_2 + 2H^2 + 2e^- \rightarrow H_2O$$

For each type of fuel cell, the reformer supplies hydrogen gas (H_2) to the anode through a process in which hydrocarbons (CH_2), water (H_2O), and oxygen (O_2) react to produce hydrogen (H_2), carbon dioxide (CO_2), and carbon monoxide (CO).

At the anode, hydrogen is split into two hydrogen ions (H^+) and two electrons. As illustrated in Figure 5.3, the hydrogen 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.

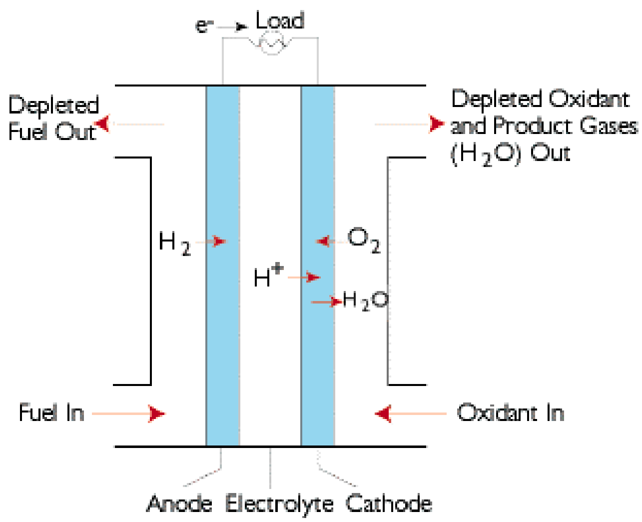


Figure 5.3: PAFC (www.dodfuelcell.com)

MCFC

Molten carbonate fuel cells have the potential to reach 50 % to 60 % LHV fuel-to-electricity efficiencies. Combined-cycle applications could reach system thermal efficiencies of 85 % LHV. MCFCs have operated on hydrogen, carbon monoxide, natural gas, propane, landfill gas, marine diesel, and simulated coal gasification products. Operating temperatures for MCFCs are around 650°C (1,200°F). The high operating temperature of the MCFC makes direct operation on gaseous hydrocarbon fuels, such as natural gas, possible. Natural gas can be reformed internally in MCFCs to produce hydrogen.

The molten carbonate fuel cell uses a molten carbonate salt mixture as an electrolyte. The composition of the electrolyte 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. The MCFC reactions are:

$$\begin{aligned} \textit{Anode reaction}: H_2 + CO_3^{2-} &\rightarrow H_2O + CO_2 + 2e^- \\ CO + CO_3^{2-} &\rightarrow 2CO_2 + 2e^- \\ \textit{Cathode reaction}: O_2 &\rightarrow 2CO_2 + 4e^- \rightarrow 2CO_3^{2-} \end{aligned}$$

An electrochemical reaction occurs at the anode between the hydrogen fuel and carbonate ions (CO_3^{2-}) from the electrolyte. This reaction produces water and carbon dioxide (CO_2) and releases electrons to the anode. At the cathode, oxygen and CO_2 from the oxidant stream are combined with electrons from the anode to produce carbonate ions, which enter the electrolyte. The reactions that occur in an MCFC are depicted in Figure 5.4.

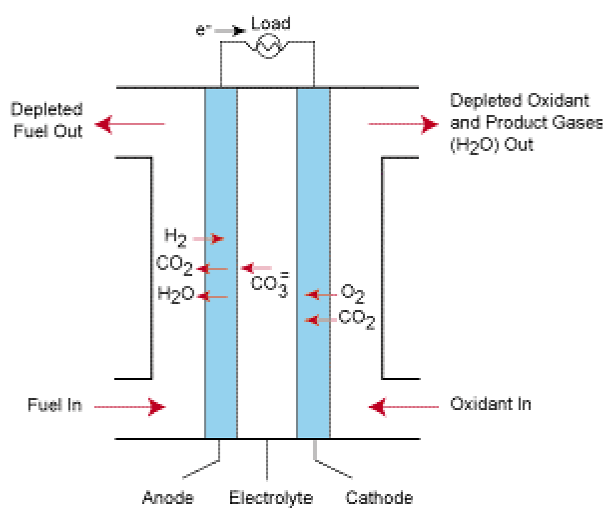


Figure 5.4: MCFC (www.dodfuelcell.com)

SOFC

Solid oxide fuel cells operate at temperatures of up to 1,000°C (1,800°F), offering enhanced combined-cycle performance. 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 are capable of fuel-to-electricity efficiencies of 50 % to 60 % LHV and total system thermal efficiencies up to 85 % LHV in combined-cycle applications.

SOFCs can use CO as well as hydrogen as direct fuel. The reactions occurring in an SOFC include

$$
\begin{aligned}
\textit{Anode reaction}: & H_2 + O^{2-} \rightarrow H_2O + 2e^- \\
& CO + O^{2-} \rightarrow CO_2 + 2e^- \\
& CH_4 + 4O^{2-} \rightarrow 2H_2O + CO_2 + 8e^- \\
\textit{Cathode reaction}: & O_2 + 4e^- \rightarrow 2O^{2-}
\end{aligned}
$$

In the SOFC reactions, hydrogen or carbon monoxide (CO) in the fuel stream react with oxide ions (O^{2-}) from the electrolyte. These respective reactions produce water and CO_2, and deposit 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. The oxide ions are injected back into the electrolyte. SOFC reactions are illustrated in Figure 5.5.

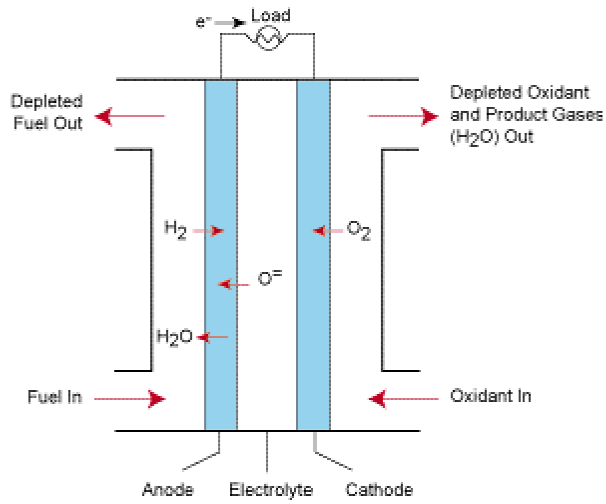


Figure 5.5: SOFC (www.dodfuelcell.com)

PEMFC

Proton exchange membrane fuel cells 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. Since the electrolyte in a PEMFC is a solid polymer; electrolyte loss is not an issue with regard to stack life. The use of a solid electrolyte eliminates the safety concerns and corrosive effects associated with liquid electrolytes. PEMFCs operate at relatively low temperatures (about 200°F). The reactions that occur in the PEMFC are as follows:

$$\textit{Anode}\ \mathrm{Re}\,\textit{action}: H_2 \rightarrow 2H^+ + 2e^-$$
$$\textit{Cathode}\ \mathrm{Re}\,\textit{action}: O_2 \rightarrow 4H^+ + 4e^- \rightarrow 2H_2O$$

The electrode reactions in the PEMFC are analogous to those in the PAFC, as indicated in Figure 5.6. 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; thus, the water is rejected from the back of the cathode into the oxidant gas stream.

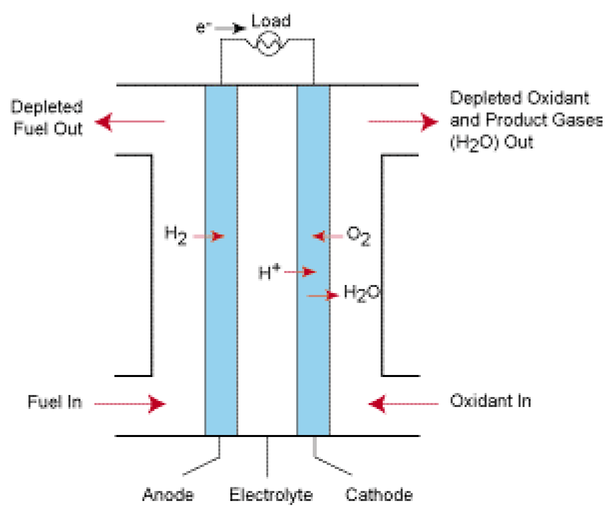


Figure 5.6: PEMFC (www.dodfuelcell.com)

The four types of fuel cells discussed above are similar in function and design. However, different physical compositions and operating characteristics affect the performance of each type of fuel cell. Table 5.1 gives an overview of fuel cell characteristics for the PAFC, SOFC, MCFC, and PEMFC.

Table 5.1: Overview of Fuel Cell Characteristics (www.energy.ca.gov/distgen/)

	PAFC	SOFC	MCFC	PEMFC
Commercial Status	Limited Availability	Not Available	Not Available	Not Available
Size Range	100-200 kW	1 kW - 10 MW	250 kW - 10 MW	3-250 kW
Average Operating Temperature	400°F (200°C)	1800°F (1000°C)	1200°F (650°C)	200°F (90°C)
Fuel	Natural gas, landfill gas, digester gas, propane	Natural gas, hydrogen, landfill gas, fuel oil	Natural gas, hydrogen	Natural gas, hydrogen, propane, diesel
Efficiency	40-45 %	50-60 %	50-60 %	40-50 %
Reforming	External	External/Internal	External/Internal	External
Environmental	Nearly zero emissions	Nearly zero emissions	Nearly zero emissions	Nearly zero emissions

Application

Fuel cells are being developed for stationary power in small commercial and residential markets and as peak shaving units for large commercial and industrial end-users. 200-kW phosphoric acid fuel cells have been installed at medical, industrial, and commercial facilities throughout the country. The commercially available PC-25™, a 200-kW PAFC by United Technologies Company, provides highly reliable electric power and cogeneration applications. However, the PC-25™ is the only commercially available fuel cell with significant operating experience. The SOFC, MCFC, and PEMFC are still in development stages; but future applications of these fuel cells are of great interest. Some of the applications being considered for MCFCs, SOFCs, and PEMFCs are discussed below.

Molten carbonate fuel cells have high efficiencies and high operating temperatures that make them most attractive for base-loaded power generation with or without cogeneration. The wide range of MCFC power output, 250-kW to 10-MW, mades this type of fuel cell attractive for base-load power generation for industrial and governmental facilities, universities, and hospitals.

Solid oxide fuel cells are being considered for a wide variety of applications in the 5-kW to 250-kW range. SOFCs have potential application for residential cogeneration and for power for small commercial buildings and industrial facilities. Due to the very high exhaust temperatures and to an all-solid ceramic construction, many experts believe that the SOFC will be the dominant technology for stationary power applications. SOFC designs in the multi-megawatt capacity range are being considered for base-loaded utility applications.

The development of proton exchange membrane fuel cells has been driven in large part by the automotive sector. PEMFCs have a high power density and short startup time, which is an advantage over other fuel cell technologies. PEMFCs are currently being developed for a wide range of applications including:

- Automotive
- Residential (<10 kW), with/without cogeneration functionality
- Commercial (10 – 250 kW), with/without cogeneration functionality
- Light industrial ([250 kW), with/without cogeneration functionality
- Portable power (several kW and smaller). (www.energy.ca.gov/distgen/)

Heat Recovery

Significant heat is released in a fuel cell during electrical generation. The PAFC and PEMFC operate at low temperatures (see Table 5.1) and can produce low-pressure steam and hot water that is suitable for commercial and industrial CHP applications. The MCFC and SOFC operate at much higher temperatures and produce heat that is sufficient to generate additional electricity with a steam turbine or a microturbine combined cycle. In a microturbine combined cycle, the combustion chamber of the microturbine is replaced with a SOFC. The SOFC generates both electricity and high-pressure, high-temperature exhaust suitable for expansion in a microturbine.

Cost

The first cost of fuel cells is very high compared to those of other DER technologies. The 2002 cost of the PC-25™ unit was approximately $4,000/kW. The installed cost of the unit approaches $1.1 million. At a rated output of 200 kW, the total cost with installation is about $5,500/kW. Efforts are underway to reduce the cost of phosphoric acid fuel cells. The Department of Energy (DOE) is helping to promote fuel cell technologies by offering a $1000/kW federal subsidy to reduce the cost to the purchaser. Since fuel cell types are under development, their costs are difficult to estimate.

Fuel cells are expected to have minimum maintenance requirements. The fuel supply systems and reformer system may need periodic (about once a year) inspection and maintenance. The fuel cell stack itself will not require maintenance until the end of the fuel cell's service life. The maintenance and reliability of fuel cell systems still need to be proven in large-scale, long-term demonstrations. Maintenance costs of a fuel cell are expected to be comparable to that of a microturbine, ranging from $0.005-$0.010/kWh (based on an annual inspection visit to the unit). (http://www.energy.ca.gov/distgen//distgen/equipment/fuel_cells/cost.html)

Fuel Cells and CHP-B

Fuel cells are expected to deliver electrical conversion efficiencies in the range of 40 to 60 %. Fuel cells used in cogeneration applications could realize energy conversion efficiencies of 80 to 90 % for the overall system. CHP-B applications can benefit from the high electrical conversion efficiencies of fuel cells and the even higher overall energy conversion efficiencies of fuel cells used in combined cycles and fuel cells coupled with thermally activated components. Fuel cells produce electricity without combustion; therefore, combustion products will not pollute populated CHP-B installation locations. Residential and commercial buildings will also benefit from the quiet and reliable operation of fuel cells.

Despite the many advantages that fuel cells offer for CHP-B practices, there are overwhelming obstacles that hinder the widespread availability and installation of fuel cells. Fuel cells are much more expensive than the other distributed power generation technologies that are being considered for CHP-B applications. The PC-25™ units are only used in high value, "niche" markets where reliability is a premium and in areas where electricity prices are very high and natural gas prices are low. The PC-25™ phosphoric acid fuel cell is the only commercially available fuel cell currently compiling operating experience. This limited operating experience gives fuel cells a disadvantage when compared with the operating experience of competing DPG technologies. Some of the advantages and disadvantages of each type of fuel cell are listed in Table 5.2.

Table 5.2: Advantages and Disadvantages of Fuel Cells (www.energy.ca.gov/distgen/)

PEMFC	
Advantages	**Disadvantages**
Quiet	Limited field test experience
Low emissions	Low temperature waste heat may limit cogeneration potential
High efficiency	
Synergy with automotive R&D	High Cost

Manufacturers

The following list of manufacturers and developers was obtained at http://www.energy.ca.gov/distgen//distgen/equipment/fuel_cells/vendors.html.

Phosphoric Acid

- UTC Fuel Cells, formerly ONSI, located in South Windsor, Connecticut, offers the 200-kW PC25™ PAFC power plant. The PC25™ is pictured in Figure 5.7.

Figure 5.7: 200-kW PC25™ PAFC by United Technologies Company (www.utcfuelcells.com)

Two other companies that are currently developing PAFCs are listed below.

- The fuel cell business department of Japan's Fuji Electric Company, Ltd. manufactures and sells the FP-100, a 100-kW PAFC power plant.

- Mitsubishi Electric Corporation plans to commercialize 200-kW PAFC systems. The company conducts PAFC work at the Advanced Technology R&D Center in Japan.

Molten Carbonate

There are few molten carbonate fuel cell developers worldwide. Three well-known MCFC companies are as follows:

- Fuel Cell Energy of Danbury, Connecticut is regarded as the leading developer of MCFC technology. The company plans to offer its Direct Fuel Cell™ power plants with power outputs ranging from 250 kW to 3 MW.
- The Power & Industrial Systems R&D Division of Hitachi, Ltd. and the Hitachi Works Fuel Cell Development Center, both located in Japan, develop and design MCFC structures and stacks.
- Ansaldo Ricerce Srl of Genova, Italy plans to manufacture and commercialize the "Series 500" MCFC power plant, which has an output of 500 kW and is based entirely on European technology.

Proton Exchange Membrane

Throughout the world, there are more than forty companies involved in the development of proton exchange membrane fuel cell systems for stationary and automotive applications. The following is a list of eight of the leading North American developers of PEMFC systems for stationary distributed energy resource applications:

- Avista Labs of Spokane, Washington, has patented a modular "hot-swap" PEMFC cartridge and plans to commercialize PEMFC systems for residential and small commercial applications.
- Ballard Generation Systems, based in Burnaby, British Columbia, is a leading developer of 1-kW and 250-kW PEMFC systems for stationary power applications.
- Dais-Analytic Corporation, with offices in Odessa, Florida, plans to commercialize a 3-kW residential fuel cell system based on PEMFC technology.
- H Power, located in Belleville, New Jersey, is a developer of 3- to 4.5-kW residential cogeneration units based on PEMFC technology.
- IdaTech of Bend, Oregon intends to offer a 3-kW PEMFC system for residential DER applications.
- UTC Fuel Cells, headquartered in South Windsor, Connecticut, plans to release a 7.5-kW residential PEMFC system.
- Nuvera Fuel Cells, based in Cambridge, Massachusetts, develops stationary PEMFC systems for applications in the 1-kW to 50-kW range.
- Plug Power of Latham, New York manufactures a 7-kW residential fuel cell system. General Electric is the master distributor of this system throughout the world, except for in Michigan, Indiana, Ohio, and Illinois where DTE Energy has distribution rights.
- Proton Energy Systems, with offices in Rocky Hill, Connecticut, develops regenerative fuel cells, utilizing PEMFC technology and electrolyzers.

Solid Oxide

There are more than twenty developers of solid oxide fuel cell technology. Four of the leading North American companies are delineated as follows:

- Global Thermoelectric, based in Calgary, Alberta, is a developer of planar (flat, square or rectangular plates, like Figure5.1) SOFC systems for residential and small commercial applications.
- Siemens-Westinghouse Power Corporation in Pittsburgh, Pennsylvania takes the lead in the development of tubular (single cell per tube with cell built up in layers from air electrode) SOFC systems. The company presently offers a 250-kW power plant for commercial distributed energy resource applications.
- SOFCo, a division of McDermott Technology, Inc. located in Alliance, Ohio, has developed a simple, proprietary manufacturing method for planar SOFCs. The company plans to develop SOFC systems for commercial stationary applications.
- ZTEK Corporation of Waltham, Massachusetts plans to commercialize a 200-kW SOFC-Microturbine hybrid system and a 150-kW SOFC system to produce electricity, heating, ventilation, and air conditioning.

CHAPTER VI

HEAT RECOVERY

Electrical and shaft power generation efficiencies have attained maximum values of 50 % for IC engines, 60 % for combustion turbines (combined cycle), 30 % for microturbines, and 70 % for fuel cells. Most power generation components falling into these categories do not reach the upper level efficiencies of these technologies. Components such as microturbines, that convert 30 % of the input fuel into electrical or shaft power, fail to harness 70 % of the available energy source. Energy that is not converted to electrical power or shaft power is typically rejected from the process in the form of waste heat. The task of converting waste heat to useful energy is called heat recovery and is primarily accomplished through the use of heat exchanger devices such as heat recovery steam generators (HRSG).

Distributed generation components offer sources of waste heat than can be harnessed as useful energy. The characteristics of waste heat generated in combustion turbines, internal combustion engines, and fuel cells directly affects the efficacy with which useful energy is recovered for additional processes. Some of the characteristics of the waste heat generated by these distributed power generation technologies are presented in Table 6.1. Waste heat is typically produced in the form of hot exhaust gases, process steam, and process liquids/solids. In combustion turbines and IC engines, heat is rejected in the combustion exhaust and the coolant. Fuel cells reject heat in the form of hot water or steam.

Table 6.1: Waste Heat Characteristics of DPG Technologies
(http://www.eren.doe.gov/der/chp/pdfs/chprev.pdf)

	Usable temp. for CHP (8F)	CHP output (Btu/kW-hr)	Uses for heat recovery
Diesel Engine	180-900	3,400	Hot water, LP steam, District heating
Natural Gas Engine	300-500	1,000-5,000	Hot water, LP steam, District heating
Gas Turbine	5,00-1,100	3,400-12,000	Direct Heat, Hot water, LP-HP steam, District heating
Microturbine	400-650	4,000-15,000	Direct heat, Hot water, LP steam
Fuel Cell	140-700	500-3,700	Hot water, LP steam

Shah (15) classifies recovered heat as low-temperature (< 2308C), medium-temperature (2308C - 6508C), or high-temperature (> 6508C). Recovered heat that is utilized in the power generation process is internal heat recovery, and recovered heat that is used for other processes is external heat recovery. Combustion pre-heaters, turbochargers and recuperators are examples of internal heat recovery components. Heat recovery steam generators, absorption chillers, and desiccant systems are examples of external heat recovery components.

Technology Overview

Because combustion exhaust cannot be used directly in many applications heat exchangers are used to facilitate the transfer of waste heat thermal energy to heat recovery applications. Heat exchangers classification is based on component and fluid characteristics. This study will focus on transmural, recuperative heat exchangers. The continuous transfer of heat between two fluids (recuperative) that are separated by a wall (transmural) characterizes this type of heat exchanger. Three prominent heat exchanger geometries that fit this classification are double-pipe heat exchangers, shell-and-tube heat exchangers, and cross-flow heat exchangers.

A detailed analysis of the various geometries of heat exchangers will not be performed herein. General heat transfer textbooks and specialized heat exchanger references contain information about double-pipe heat exchangers, shell-and-tube heat exchangers, and cross-flow heat exchangers. Each of these types of heat exchangers is useful in waste heat recovery. The selection of the best heat exchanger geometry depends on both the prime mover and the thermally-activated component.

The simplest type of heat exchanger is a double-pipe heat exchanger, which consists of one fluid flowing through an inner pipe and a second fluid flowing in an outer annulus. The two fluids can flow in a parallel-flow or counterflow arrangement. Extended surfaces (fins) are often added to the inside or outside surface of the inner pipe in order to increase the rate of heat transfer. Double-pipe heat exchangers are easy to manufacture, but have limited heat transfer capability due to small surface areas. Parallel-flow and counterflow double-pipe heat exchangers are illustrated in Figure 6.1.

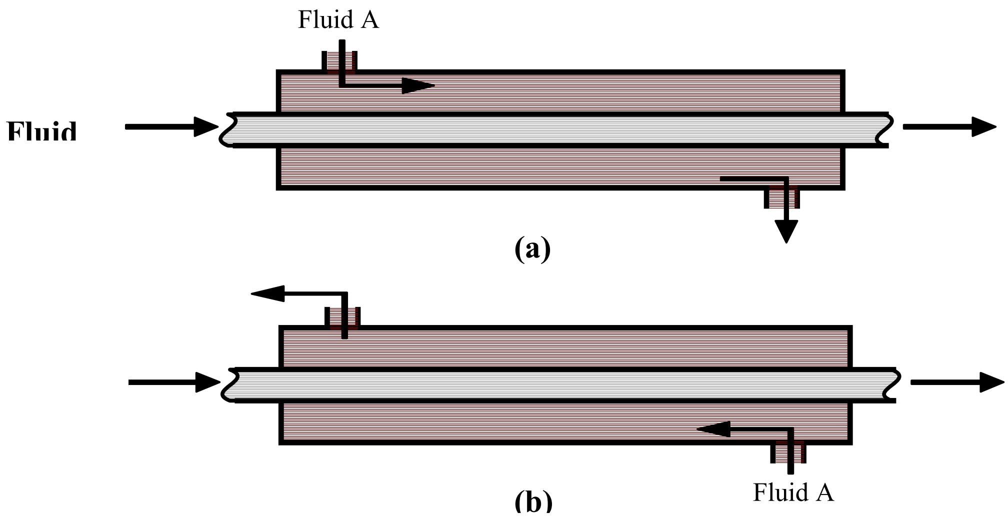


Figure 6.1: (a) Parallel-flow and (b) Counterflow Double-Pipe Heat Exchangers

Shell-tube-heat exchangers consist of a cylindrical or rectangular shell enclosing a bundle of tubes. One fluid flows through the tubes, and a second fluid flows across the tubes in the shell. Baffles are often used to control the flow of the shell-side fluid. If the tube-side fluid flows the length of the heat exchanger only one pass, the device is considered a one-tube pass shell-and-tube heat exchanger. Multiple-tube pass shell-and-tube heat exchangers are very common. Shell-and-tube heat exchangers have large ratios of heat transfer surface area to weight and volume and are manufactured in various sizes. A schematic of a shell-and-tube heat exchanger with water flowing through the shell and steam flowing through the tubes is presented in Figure 6.2.

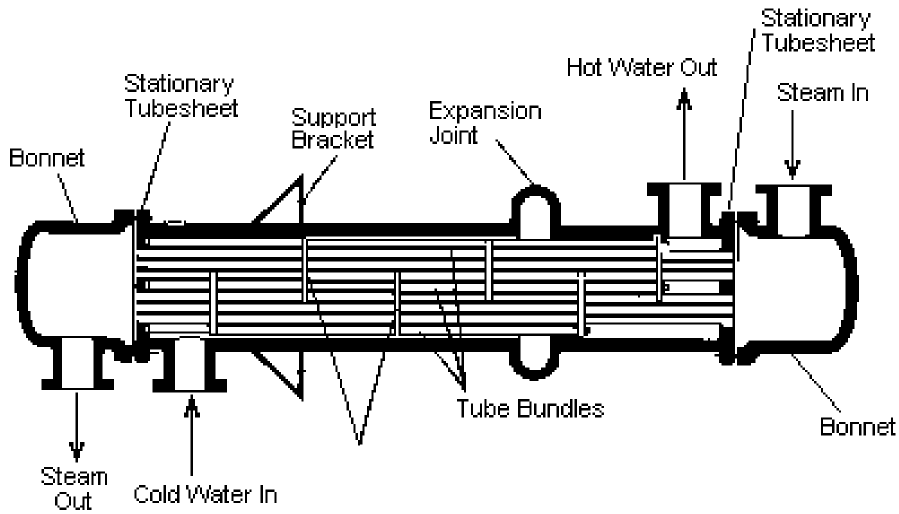


Figure 6.2: Shell-and-tube heat exchanger
(http://blt.colorado.edu/images/gif/heat_exa.gif)

Cross-flow heat exchangers consist of two fluids flowing in mutually perpendicular directions. A typical cross-flow heat exchanger consists of one fluid flowing through an array of tubes and a second fluid flowing across the tubes. The flow of fluid in the tubes is always considered unmixed; however, the flow across the tubes is considered mixed if there are no plate-fins attached and unmixed if plate-fins are attached. These two types of cross-flow heat exchangers are illustrated in Figure 6.3. Hodge and Taylor (9) suggest that the arrangement of cross-flow heat exchangers offers some advantages in terms of compactness and effectiveness and some disadvantages in terms of fabrication and maintenance.

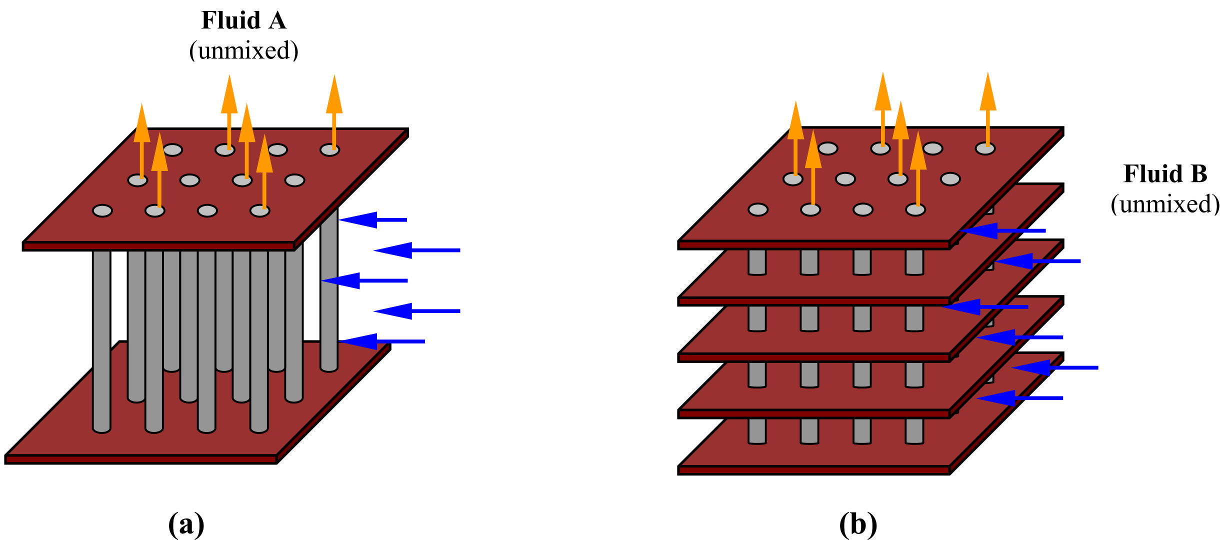


Figure 6.3. Cross-flow heat exchangers (a) unmixed-mixed (b) unmixed-unmixed

Heat Exchanger Analysis

Specific analysis methods and their formulations will not be covered in this study since most general heat transfer textbooks cover the fundamental of heat exchanger design and analysis. To determine the heat transfer rate (duty) in a heat exchanger, both an overall heat transfer coefficient (which relates the heat flux between two fluid streams) and a temperature difference must be known. The overall heat transfer coefficient can be determined based on the heat exchanger geometry and local heat transfer coefficients.

Conductive, convective, and radiative heat transfer may each contribute to the overall heat transfer coefficient of a heat exchanger. Convection and conduction are always present in waste heat recovery and will always be considered in heat exchanger design and analysis. Radiation is a significant mode of heat transfer in metallic radiation recuperators. Determining the convective heat transfer coefficient, (h), is accomplished using established and verified correlations that are based on experimental data. General heat transfer textbooks provide convective correlations for many regimes and arrangements.

Two methods are available for heat exchanger design and analysis. The log mean temperature difference (LMTD) method was the first method developed and is still widely used. A more recent method is the number of transfer units (NTU) approach (Hodge and Taylor, 1999). The NTU method will be discussed in brief, since this method is preferred for general-purpose design and analysis.

The NTU method uses the concept of effectiveness. The heat-exchanger effectiveness is defined as the ratio of the actual rate of heat transfer in a heat exchanger to the maximum possible rate of heat transfer in the exchanger as limited by the Second Law of Thermodynamics.

$$\xi = \frac{Actual\ Rate\ of\ Heat\ Transfer}{Maximum\ Possible\ Rate\ of\ Heat\ Transfer} \qquad (6\text{-}1)$$

The heat transfer rate (duty) in the heat exchanger may be computed by calculating either the energy lost by the hot fluid or the energy gained by the cold fluid. In a CHP-B system with a given waste heat stream, the heat exchanger's effectiveness determines the amount of energy available for the thermally-activated component.

Especially in waste heat recovery, the pressure drops that occur in a heat exchanger are very important selection criteria. The pressure drop in a heat exchanger determines the pumping power requirements. Excessive pressure drop can negate the energy savings from thermal energy recovery. Also, many thermally-activated components require as inlet conditions a specific fluid temperature and pressure.

Performance data are available for heat exchangers that fit a wide variety of scenarios. However, the uniqueness of the CHP-B paradigm presents many new waste heat recovery scenarios.

Application

Waste heat recovery heat exchangers are classified as gas-to-gas, gas-to-liquid, or liquid-to-liquid. These categories of heat exchangers are useful in high-, medium-, and low-temperature waste heat applications.

Gas-to-Gas Heat Exchangers

Gas-to-gas waste heat recovery exchangers are often used as recuperators for preheating combustion air in IC engines and combustion turbines. The use of recuperators on microturbines, industrial turbines, and IC engines depends on the thermal and electrical load characteristics of a CHP-B application. Metallic radiation recuperators, convection recuperators, and runaround coils are three types of heat exchangers used for gas-to-gas waste heat recovery. Metallic radiation recuperators consist of two metallic tubes in a double-pipe heat exchanger arrangement. Hot flue gas flows through the inner tube and transfers heat to air that flow in the outer tube. The majority of the heat transferred from the flue gas to the inner wall is accomplished by radiation, and the heat transfer from the wall to the air in the outer tube is accomplished by convection. Metallic radiation recuperators usually has an effectiveness of 40 % or lower. Although metallic radiation recuperators may be desirable for some CHP-B applications, the use of this exchanger is limited by the relatively low effectiveness.

Convection recuperators are cross-flow heat exchangers with flue gas flowing normal to a bundle of tubes containing air. Convection recuperators can be used in low-temperature applications such as space heating, return air heating in a desiccant dehumidification system, or direct-fired absorption chillers. Runaround coils are also cross-flow heat exchangers. A runaround coil consists of two connected coils that circulate a working fluid. The working fluid is heated by the waste gas and is used to heat a stream of cool air. Runaround coils are commonly used in HVAC applications and can be coupled with distributed generation components to produce warm air for district heating, to heat return air in a desiccant system, or to directly fire an absorption chiller. Some other types of heat exchangers used for gas-to-gas waste heat recovery are plate-fin and prime surface heat exchangers, heat pipes, and rotary generators. (Shah, 1997)

Gas-to-Liquid Heat Exchangers

Gas-to-liquid waste heat recovery exchangers include medium- to high-temperature waste heat recovery devices such as heat recovery steam generators, fluidized-bed heat exchangers, and heat pipes, as well as low- to medium-temperature waste heat recovery devices such as economizers and thermal fluid heaters. HRSG units, also called waste heat boilers, are used to generate steam from turbine exhaust gas. These steam generators are very important in CHP-B applications and will be discussed in greater detail in the following section. A fluidized-bed heat exchanger is comprised of water, steam, or a heat transfer fluid being heated by waste heat gases that flow over a bed of finely-divided solid particles. When the waste heat fluid reaches a critical velocity, the particles in the bed will float, and the resulting mixture acts like a fluid. The advantage of the fluidized bed is an increase in the heat transfer coefficient of the waste heat fluid when small solid particles are suspended in the fluid. Fluidized-bed heat exchangers are used for space heating, heating boiler feedwater, heating process fluids, and hot water services. Gas-to-liquid heat pipes are similar to those used in gas-to-gas applications. Flue gas flows through one tube and heat is transferred from the gas to a working fluid that flows through the other tube.

Two types of low- to medium-temperature gas-to-liquid waste heat recovery exchangers are economizers and thermal fluid heaters. Economizers are cross-flow heat exchangers that consist of water flowing in individually finned tubes with hot gas flowing normal to the tubes. Economizers are often used with the boiler flue gases to preheat the boiler feedwater. These waste heat recovery exchangers are also used to heat water, process liquids, and to superheat steam. Thermal fluid heaters are double-pipe heat exchangers that use waste heat gases to heat a high-temperature organic heat transfer fluid. This fluid is circulated throughout a plant and used for heating and cooling. Thermal fluid heaters can operate on waste heat or be fired by gas (they often use combustion gases). (Shah, 1997)

Liquid-to-Liquid Heat Exchangers

Liquid-to-liquid waste heat recovery exchangers are typically used in industrial applications. Shell-and-tube heat exchangers are generally used for this type of waste heat recovery. Liquid coolant systems in combustion turbines and IC engines offer liquid-to-liquid heat recovery opportunities from hot oil and other liquid coolants.

However, liquid-to-liquid waste heat recovery may not be as significant in small-scale CHP-B applications as gas-to-gas and gas-to-liquid waste heat recovery. (Shah, 1997)

Heat Recovery Steam Generators

In many waste heat recovery applications steam or hot water is required for thermally-activated components, process equipment, and district heating systems. Heat recovery steam generators, also known as heat recovery boilers (HRB) and waste heat boilers (WHB), are the most common heat exchanger device utilized to convert waste heat into hot water or steam. HRSGs are can be unfired, partially-fired, or fully-fired.

Unfired HRSGs use the hot exhaust gases alone to heat or boil water. As discussed in an earlier section, a duct burner can be installed in the exhaust duct of a turbine to burn the oxygen-rich exhaust. If a HRSG uses a duct burner upstream of the boiler to increase the exhaust gas temperature, the HRSG is considered partially-fired. In a fully-fired HRSG, the combustion exhaust is used like preheated air and is completely fired upon entering the HRSG.

In an HRSG, energy from exhaust gas is used to vaporize water and to superheat steam. Figure 6.4 demonstrates this process. According to Caton and Turner (3), the top line in the figure represents the temperature of the exhaust gas decreasing from left to right as thermal energy is removed from the gas to the water. The lower line represents the water increasing in temperature from right to left as heat is added from the exhaust gas. Figure 6.4 illustrates that an HRSG unit includes an economizer, an evaporator, and a superheater. The water is raised to saturation conditions by the low-temperature exhaust in the economizer. In the evaporator, the water is vaporized into saturated steam using the intermediate-temperature exhaust. Finally, the saturated steam is superheated by the high-temperature exhaust in the superheater. The pinch point is located where the water first starts to vaporize. The temperature difference between the exhaust gas and water at this point is called the pinch point temperature difference. The pinch point temperature difference is the smallest temperature difference in the HRSG and may limit the device's overall performance.

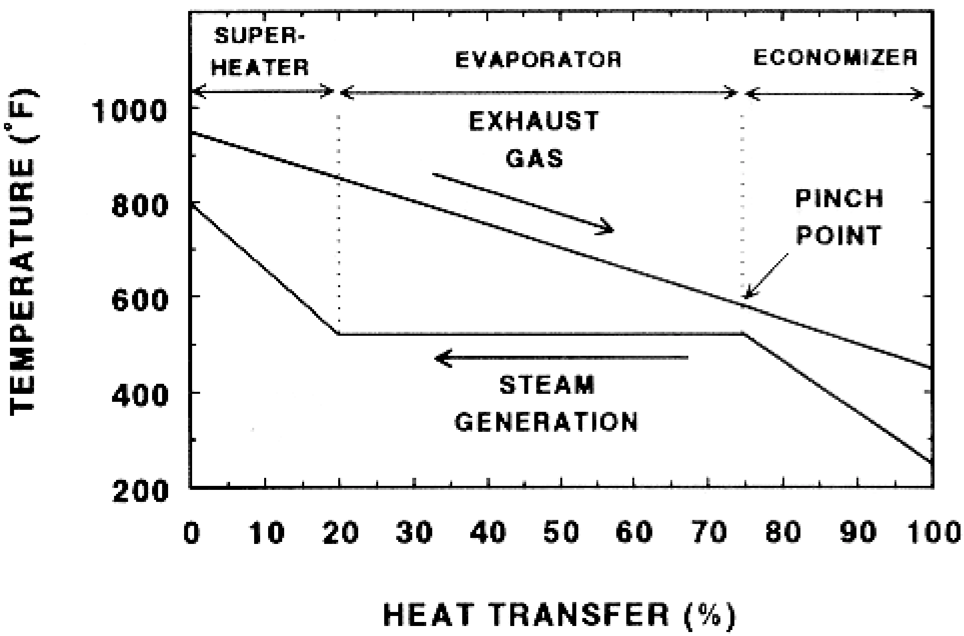


Figure 6.4: Exhaust gas and water/stream temperatures as a function of the heat transfer in an HRSG (Caton and Turner, 1997)

Selecting the proper HRSG unit depends upon the prime mover, the steam conditions required by the thermally-activated components or process, and other independent factors. The overall system performance can be improved by selecting an HRSG with low "back-pressure." Units with low back-pressure have higher efficiency as well as higher costs. Another consideration in selecting the appropriate HRSG is the outlet stack gas temperature, which affects acid formation. Finally, the temperature and pressure of the outlet steam in an HRSG is governed by the inlet conditions necessary to run a specific thermally-activated component or process. HRSG units are important to CHP-B applications since many thermally-activated components and processes require hot water and steam.

CHAPTER VII

ABSORPTION CHILLERS

Technology Overview

Absorption technology is one of a group of technologies classified as heat pumps. Heat pump technologies can be heat-driven or work-driven and transfer heat from a low temperature to a high temperature. Absorption technology is heat-driven, transferring heat from a low temperature to a high temperature using only heat as the driving energy. Absorption technology operates on the basis of the absorption cycle. The absorption cycle is similar to the more familiar vapor compression cycle, which will be examined in order to draw analogies between the two cycles.

Most commercial and residential air conditioning is accomplished by vapor compression systems. The vapor compression cycle is a work-driven cycle that consists of a fluid refrigerant flowing through a system of components as in the cycle illustrated in Figure 7.1. In the vapor compression cycle, shaft work is supplied (to the compressor) to compress the refrigerant vapor to a high pressure and a high temperature. At state 2 in Figure 7.1, the high-pressure refrigerant's condensation temperature is higher than the ambient temperature. The high-pressure, high-temperature refrigerant vapor enters the condenser where heat is rejected to the ambient air and the vapor condenses to a liquid. The high-pressure liquid at state 3 passes through an expansion valve, reducing the pressure and temperature. The low-pressure refrigerant at state 4 has a boiling temperature below the ambient air temperature. The refrigerant removes heat from the ambient air when boiling occurs in the evaporator. This evaporation results in a low-pressure refrigerant vapor (state 1) entering the compressor as the vapor compression cycle is completed.

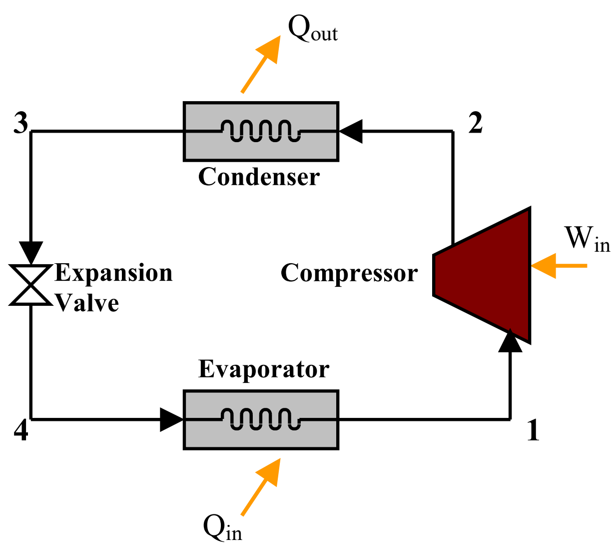


Figure 7.1: Vapor-Compression Cycle

The heat-driven absorption cycle has many of the same components as the vapor compression cycle. Figure 7.2 presents a cycle diagram for the basic absorption cycle. Much like in the vapor compression cycle, refrigerant in the absorption cycle flows through a condenser, expansion valve, and an evaporator (left of line Z-Z in Figure 7.2). However, the absorption cycle uses different refrigerants and a different method of compression than the vapor compression cycle.

The absorption cycle operates on the principle that some liquids (absorbents) have an affinity for other liquids or vapors and will absorb them under certain conditions. One example of an absorbent is lithium bromide, which readily absorbs water vapor. In an absorption system that uses water as the refrigerant and lithium bromide as the absorbent, water vapor that exits the evaporator is absorbed in liquid lithium bromide. The water/lithium bromide solution is pumped to the condenser pressure where the water is separated from the solution.

An absorption system does not mechanically compress refrigerant vapor from a low evaporator pressure to a higher condenser pressure. Rather, the absorption cycle depends on a "thermal compressor" to move low-pressure refrigerant vapor to higher pressure. The components to the right of line Z-Z in Figure 7.2, the generator, absorber, expansion valve, and pump, comprise the thermal compressor. Refrigerant vapor exits the evaporator at low-pressure and enters the absorber. In the absorber, the refrigerant vapor is dissolved in a liquid absorbent and rejects the heat of condensation and the heat of mixing. The refrigerant/absorbent solution is pumped from the low evaporator pressure to the high condenser pressure. Much less mechanical work is required in pumping the refrigerant/absorbent solution to the condenser pressure than the amount of mechanical work required to compress a refrigerant to the condenser pressure. Heat is added to the solution in the generator to vaporize the refrigerant, removing the refrigerant from solution. The liquid absorbent has a higher boiling temperature than the refrigerant and, thus, stays in solution while most of the refrigerant exits the generator and flows to the condenser. The liquid absorbent exits the generator with a low concentration of refrigerant still in solution. This low concentration solution flows through an expansion valve where the pressure is decreased to the evaporator pressure before the solution returns to the absorber.

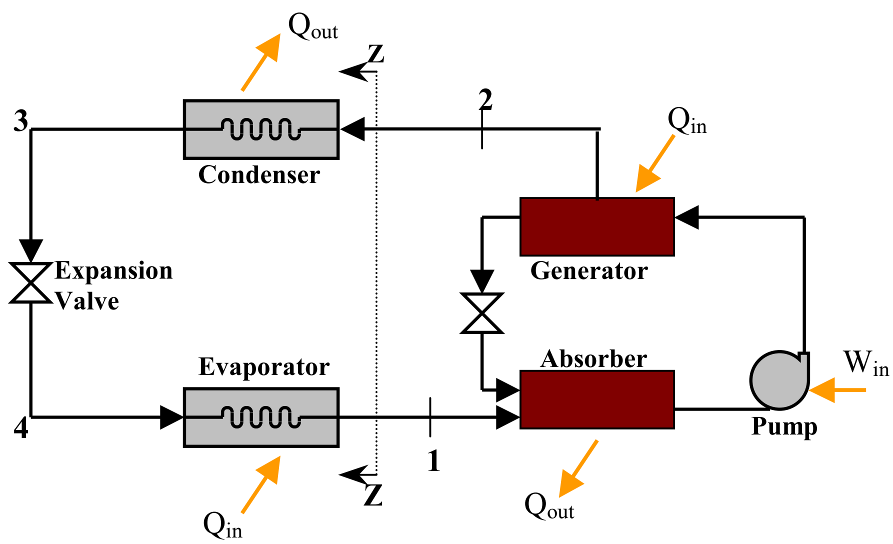


Figure 7.2: Basic Absorption Cycle

Refrigerant-Absorbent Selection

Though all absorption chillers operate on the basic cycle presented in Figure 7.2, each chiller design is dependent on the refrigerant-absorbent selection. Two common refrigerant-absorbent combinations are water-lithium bromide and ammonia-water. As previously mentioned, water-lithium bromide absorption chillers utilize water as the refrigerant and lithium bromide as the absorbent. Since lithium bromide is relatively non-volatile, this combination of refrigerant and absorbent is advantageous in areas where toxicity is a concern. Water-lithium bromide chillers are limited to refrigeration applications above 0°C due to the freezing point of water. Chillers using this fluid are very common and are available in a wide range of sizes from 10 tons to 2,600 tons (http://www.bchp.org/owner-equip.html) with a coefficient of performance (COP) ranging from 0.7 to 1.2 (Herold, Radermacher, and Klein, 1996).

Ammonia/water absorption chillers are not as widely used as water/lithium bromide chillers. In an ammonia/water chiller, ammonia is the refrigerant and water is the absorbent. An advantage of this type of chiller is the low freezing point of ammonia (-77.7°C) as compared to that of water, resulting in lower refrigeration temperatures than those of a water/lithium bromide system. However, the toxicity of ammonia is a disadvantage of this absorption technology. In commercial and residential building applications where there is insufficient ventilation, emissions from ammonia/water absorption chillers could be harmful to occupants. These machines have capacities ranging from 3 to 25 tons with a COP typically around 0.5 (Herold, Radermacher, and Klein, 1996). A schematic of an absorption cycle of ammonia/water is shown in Figure 7.3. The addition of a heat exchanger is common in all absorption chillers to increase the efficiency of the thermal compressor. Hot solution leaving the generator is used to preheat the cooler refrigerant/absorbent solution entering the generator. In an ammonia/water absorption chiller, ammonia vapor that leaves the generator often includes a low concentration of water vapor. Water vapor that travels through the condenser will freeze in the expansion valve. A rectifier is included in ammonia/water chillers to prevent water from entering the condenser.

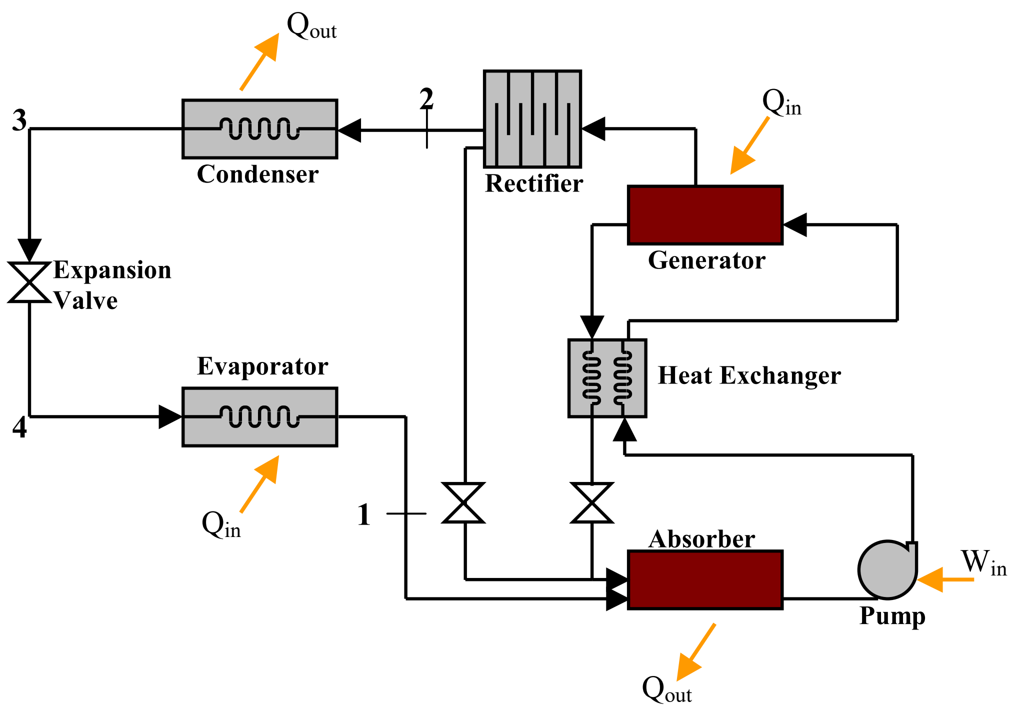


Figure 7.3: Ammonia/Water Absorption Cycle

Types of Absorption Chillers

Double-effect (two-stage) absorption chillers can be used when a high-temperature heat source is available. These chillers contain two stages of generation, as shown in Figure 7.4. The first stage generator separates refrigerant vapor from solution at a high temperature. The refrigerant/absorbent solution in the second stage generator is at a lower temperature than the solution in the first stage generator. The refrigerant vapor from the first stage generator flows through the second generator where some of the refrigerant remains in the vapor phase and some of the refrigerant condenses back into liquid. Additional refrigerant is vaporized in the second stage generator by the high-temperature and the heat of vaporization supplied by the refrigerant from the first stage generator. Refrigerant vapor from both generators flows to the condenser while the absorbent solution returns to the absorber. Double-effect chillers yield higher COPs (1.0 to 1.2 for water/lithium bromide chillers) than single-effect chillers (Herold, Radermacher, and Klein, 1996).

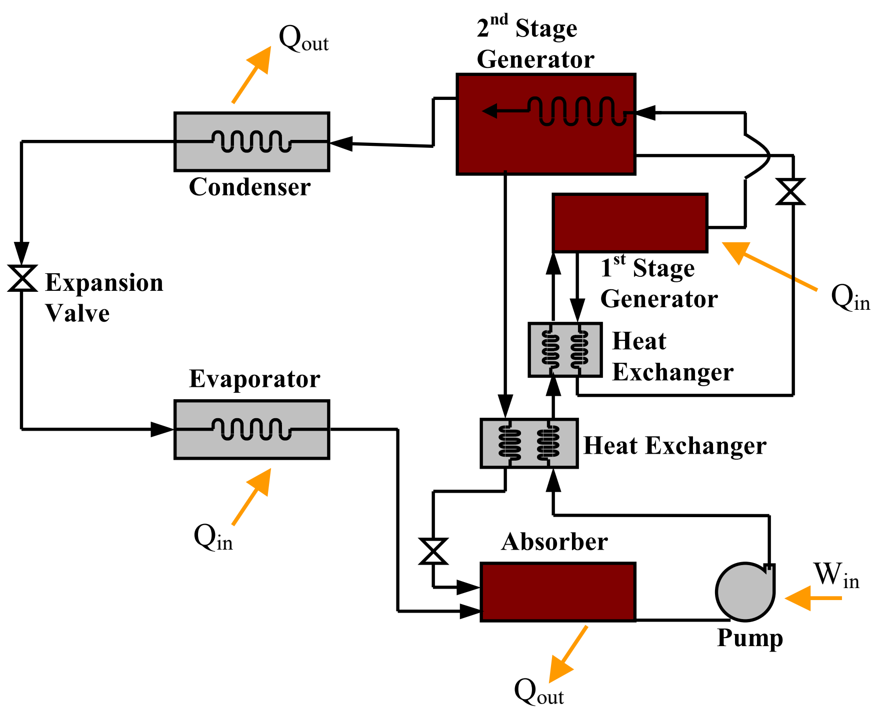


Figure 7.4: Double-Effect Water/Lithium Bromide Absorption Chiller

Some additional absorption cycles that may play a role in CHP-B systems are currently being researched and developed. Triple effect water/lithium bromide chillers can attain higher coefficients of performance than their single- and double-effect predecessors. Triple effect chillers have achieved cooling COPs that exceed 1.6.

Generator-absorber heat exchanger (GAX) absorption technology is a new entry into the residential and light-commercial chiller market. GAX chillers use an ammonia/water working fluid and are of particular interest to CHP-B applications for capacities as low as 3 tons. GAX absorption chillers have attained cooling COPs of approximately 0.7. (DeVault, Garland, Berry, and Fiskum, 2002)

System Analysis

An absorption refrigeration system consists of a working fluid undergoing a series of thermodynamic processes. Refrigerant and absorbent flow in a closed-loop system that includes adiabatic and non-adiabatic mixing, heating and cooling, pumping, and throttling. Each of these processes can be analyzed by energy and mass balances. The thermodynamic properties of the working fluids are essential for applying conservation of mass and energy relationships to a system.

Absorption systems use homogeneous binary mixtures as working fluids. McQuiston, Parker and Spitler (12) describe a homogeneous mixture as a uniform composition that cannot be separated into its constituents by pure mechanical means. The thermodynamic properties of the working fluid will change throughout the cycle as the fluid flows through components such as the generator, the separator, the evaporator, the expansion valves, and the heat exchangers.

Unlike pure substances, the thermodynamic state of a mixture cannot be determined by two independent properties. The concentration of a mixture (x) must be known along with two independent properties of the mixture to determine the thermodynamic state. The concentration of a mixture is defined as the ratio of the mass of one constituent to the mass of the mixture. For example, an ammonia/water mixture with a concentration of 0.3 contains 3 lbm. of ammonia for every 10 lbm. of ammonia/water mixture.

An enthalpy-concentration (i-x) diagram is a useful representation of the properties of binary mixtures. For working fluids such as ammonia/water and water/lithium bromide, the liquid and vapor regions of the (i-x) diagram are of interest. In an enthalpy-concentration diagram for a binary mixture, enthalpy and concentration are plotted on the vertical and horizontal axes, respectively. Lines of constant temperature as well as condensing and boiling lines for a range of constant pressures are graphed to aid in determining the thermodynamic properties at any point. McQuiston, Parker and Spitler (12) present a more detailed study of binary mixtures in absorption refrigeration.

With the enthalpy and concentration of the working fluid available from the (i-x) diagram, as well as from numerous software packages, analysis of the thermodynamic processes in the absorption cycle is possible. In the evaporator, condenser, absorber, rectifier, and generator one or more fluid streams enter a component, resulting in outgoing fluid and heat rejection. Figure 7.5 presents a schematic of a thermodynamic process that occurs in the absorber.

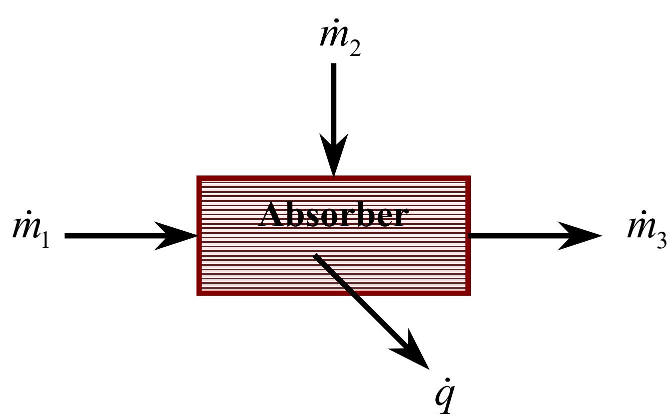


Figure 7.5: Absorber

Energy (7-1) and mass balance (7-2, 7-3) equations for this non-adiabatic process are

$$\dot{m}_1 i_1 + \dot{m}_2 i_2 = \dot{m}_3 i_3 + \dot{q} \tag{7-1}$$

$$\dot{m}_1 x_1 + \dot{m}_2 x_2 = \dot{m}_3 x_3 \tag{7-2}$$

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3 \tag{7-3}$$

Substitution and simplification of Equations 7-1, 7-2, and 7-3 yields the following equations for the enthalpy and the concentration (i_3, x_3) of stream three:

$$x_3 = x_1 + \frac{\dot{m}_2}{\dot{m}_3}\left(x_2 - x_1\right) \tag{7-4}$$

$$i_3 = i_1 + \frac{\dot{m}_2}{\dot{m}_3}\left(i_2 - i_1\right) - \frac{\dot{q}}{\dot{m}_3} \tag{7-5}$$

The last term in Equation 7-5 represents the loss of enthalpy during the process. Equations 7-4 and 7-5 can be used to identify the enthalpy and concentration of a fluid stream with unknown thermodynamic properties.

Heat exchange takes place in many locations in an absorption system. Heat exchangers, like the one illustrated in Figure 7.6, are used to increase the overall efficiency of the chiller system by transferring heat from hot fluid streams to cold fluid streams.

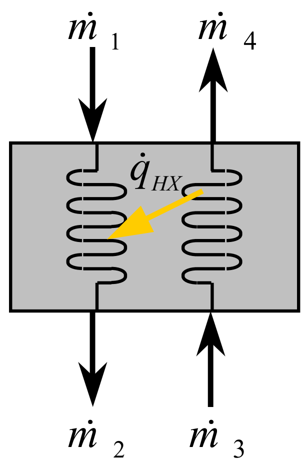


Figure 7.6: Heat Exchanger

For the heat exchanger, the following energy and mass balance relations apply.

$$\dot{m}_1(i_2 - i_1) = \dot{q}_{HX} = \dot{m}_3(i_3 - i_4) \tag{7-6}$$

$$\dot{m}_1 = \dot{m}_2 \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 \tag{7-7}$$

$$x_1 = x_2 \quad \text{and} \quad x_3 = x_4 \tag{7-8}$$

where

$$\dot{q}_{HX} = \varepsilon \cdot \dot{q}_{Max} \tag{7-9}$$

In Equation 7-9, ε is the effectiveness of the heat exchanger and q_{Max} is the maximum possible rate of heat exchange.

Another thermodynamic process encountered in absorption refrigeration is throttling. Throttling is accomplished by an expansion valve. In a throttling valve, stream one enters the expansion valve where evaporation occurs and the temperature and pressure of the working fluid are decreased. For example, when a refrigerant leaves the condenser in the absorption cycle and enters the throttling valve, the pressure and temperature of the refrigerant decrease before flowing into the evaporator. Clearly, a mass and energy balance for the throttling device will yield $m_1 = m_2$, $x_1 = x_2$, and $i_1 = i_2$. The changes in temperature and pressure that occur during expansion can be determined using the (i-x) diagram and graphical methods.

In the most basic absorption system, at least one pump is required to pump low-pressure ammonia/water solution to high-pressure ammonia/water solution from the absorber to the generator. The pump schematic in Figure 7.7 demonstrates the energy transfer required for moving fluid from low- to high-pressure.

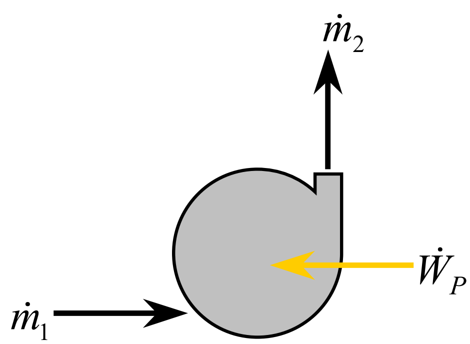


Figure 7.7: Solution Pump

The power requirement of the pump is minimal and is often neglected in an overall performance analysis. Equation 7-10 can be used to calculate the power requirement for pump operation.

$$\dot{W}_P = (p_2 - p_1)\frac{\nu \cdot \dot{m}_1}{\eta_P} \tag{7-10}$$

where p_1 and p_2 are the pressures entering and leaving the pump, respectively, ν is the specific volume of the solution, and η_P is the pump efficiency.

Application of the first and second laws of thermodynamics to the absorption cycle provides a relationship for coefficient of performance. The theoretical COP (Equ. 7-11) applies when all processes are reversible, and yields a value much larger than practical COPs.

(7-11)

$$(COP)_{\max} = \frac{T_e\left(T_g - T_o\right)}{T_g\left(T_o - T_e\right)} \qquad (41)$$

where T_e is the temperature of the refrigerated region associated with the evaporator, T_g is the temperature of the heat supplied in the generator, and T_o is the temperature of the ambient air associated with the heat rejections in the absorber and the condenser.

Equation 7-11 is based on a Carnot cycle which is not realistic. However, this equation shows that raising the evaporator or generator temperatures increase the efficiency of an absorption device. Neglecting the pump work, the actual COP for an absorption cycle is defined by Equation 7-12.

$$(COP)_{actual} = \frac{\dot{q}_{evaporator}}{\dot{q}_{generator}} \qquad (7\text{-}12)$$

While increasing the evaporator temperature may not be practical, increasing the generator temperature is very desirable. Multiple-effect absorption systems offer more cooling than single-effect systems, due to the addition of more heat in the generator through additional generator stages. The quality of waste heat available to an absorption chiller in a CHP-B system is vital to the success of overall system performance.

The following example problem, found in Figure 7.9, is derived from an example in *Absorption Chillers and Heat Pumps* (Herold, Radermacher, and Klein, 1996) and provides an overall analysis of an ammonia/water absorption system. Graphical methods, energy balance and mass balance equations, and COP relationships are implemented to determine the system performance.

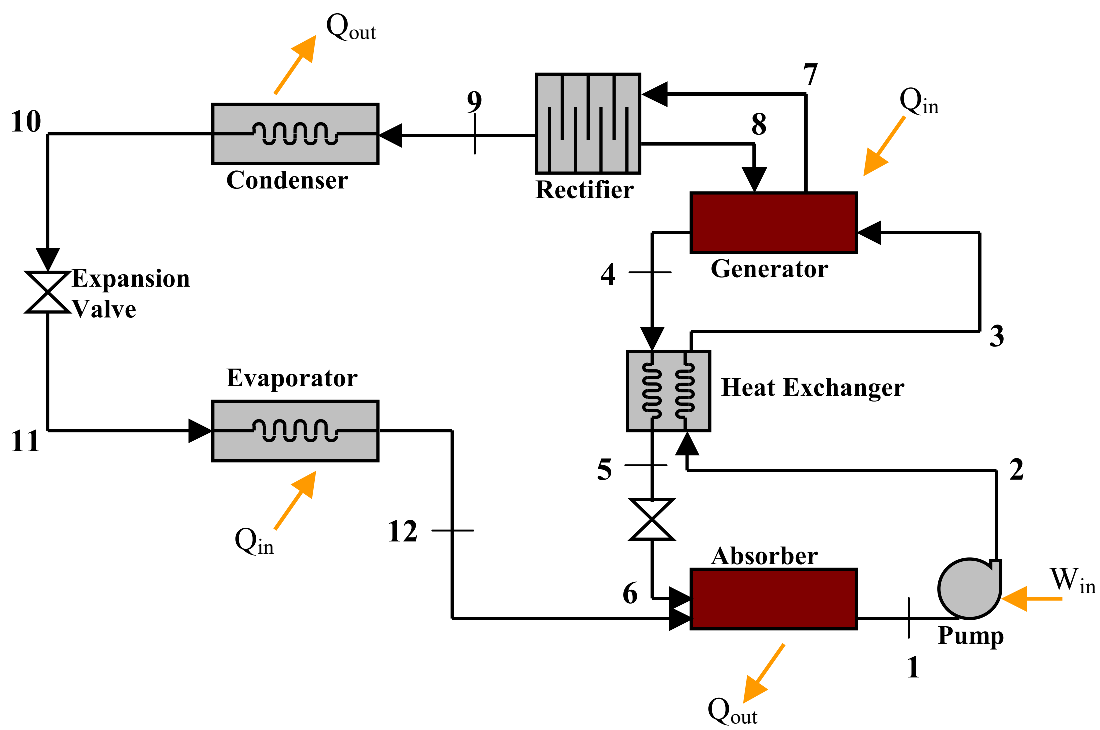


Figure 7.8: Single-stage ammonia/water chiller for Example 7-1

Example 7-1:
A single-stage ammonia/water absorption system is given as shown in Figure 7-8. The evaporator saturation temperature at the outlet is assumed to be -10 C with saturated vapor leaving. The mass flow rate of solution through the solution pump is 1 kg/sec. The temperature of the saturated liquid leaving the absorber and the condenser is 40 C. The difference in mass fraction of the two solution streams is given to be 0.10. The rectifier produces a vapor with a mass fraction of 0.999634 ammonia. The pump efficiency is assumed to be 100% and the effectiveness of the solution heat exchanger is 100?. Find the COP, the amounts of heat exchanged, and the work performed by the pump. (Note: "Q" and "m" are heat rate and mass flow rate, respectively.)

Solution:
Using binary mixture analysis methods along with a enthalpy-concentration chart for ammonia water (or using suitable software) the state points can be found for all conditions as shown in Table 7-1.

Based on the values in Table 7-1, and the equations specified in this chapter, the following detailed results are obtained.

The pump power, W_P is $$W_P = (p_2 - p_1)\frac{\nu \cdot m_1}{\eta_P} = 3.05\,\mathrm{kW}$$

with $\nu = 0.0012$ m^3/kg, the specific volume of the rich solution, and a pump efficiency of 1.0. Assuming a heat exchanger effectiveness of 1.0, the amount of heat exchanged in the solution heat exchanger can be found by writing an energy balance on either side.

$$Q_{shx} = m_2(i_3 - i_2) = -m_4(i_5 - i_4) = 346\,\mathrm{kW}$$

The heat losses and gains of the remaining components can be found by their respective energy balance equations.

$$Q_{absorber} = m_{12} \cdot i_{12} + m_6 \cdot i_6 - m_1 \cdot i_1 = 216\,\mathrm{kW}$$

$$Q_{rectifier} = m_7 \cdot i_7 - m_9 \cdot i_9 - m_8 \cdot i_8 = 51\,\mathrm{kW}$$

$$Q_{generator} = m_7 \cdot i_7 + m_4 \cdot i_4 - m_8 \cdot i_8 - m_3 \cdot i_3 = 268\,\mathrm{kW}$$

$$Q_{condenser} = m_9 \cdot (i_9 - i_{10}) = 151\,\mathrm{kW}$$

$$Q_{evaporator} = m_9(i_{12} - i_{11}) = 147\,\mathrm{kW}$$

The actual COP for this absorption system is evaluated as

$$COP_{actual} = \frac{Q_{evaporater}}{Q_{generator}} = 0.549$$

The system in this example is an ideal case. An actual absorption system that follows this example will realize pressure drops in the heat exchangers, less effective rectification, a less efficient pump, and a heat exchanger effectiveness less than 1.0.

Figure 7.9: Absorption Cycle Example Problem

Table 7.1: State points for the ammonia/water system in Figure 7.8

	i (J/g)	m (kg/s)	p (kPa)	Vapor Quality	T (C)	x (kg/kg)
1	(42.30)	1.00	240.20	-	40.00	0.37
2	(39.20)	1.00	1,555.00		40.50	0.37
3	306.80	1.00	1,555.00	0.02	110.70	0.37
4	401.60	0.86	1,555.00	-	131.00	0.27
5	0.90	0.86	1,555.00		40.50	0.27
6	0.90	0.86	240.20		40.70	0.27
7	1,547.00	0.15	1,555.00	1.00	108.00	0.94
8	264.70	0.01	1,555.00	-	108.00	0.37
9	1,294.00	0.14	1,555.00	1.00	44.00	1.00
10	190.10	0.14	1,555.00	0.00	40.00	1.00
11	190.10	0.14	240.20	0.20	(14.50)	1.00
12	1,264.00	0.14	240.20	0.99	(10.00)	1.00

Application

Absorption chillers can be directly fired or indirectly fired. Direct-fired absorption chillers contain a natural gas burner and can supply waste heat for a desiccant dehumidification system or for hot water. Direct-fired absorption chillers are used instead of vapor compression chillers in locations where electric rates are high and gas utilities offer low rates and rebate programs. Direct-fired chillers also operate in locations where chlorofluorocarbons are unfavorable.

Indirect-fired absorption chillers are of interest in CHP-B applications. Indirect-fired chillers utilize waste heat in the form of steam, hot water, hot process liquids and gases, and exhaust gases to separate the refrigerant from the absorbent in the generator. Combustion turbines, IC engines, or fuel cells can supply this waste heat. Like direct-fired absorption chillers, indirect-fired chillers may not utilize all of the input heat. Heat left over from an indirect-fired absorption chiller may be used in a desiccant dehumidification system or to produce hot water. An overview of absorption chiller technology is presented in Table 7.2.

Table 7.2: Overview of Absorption Chillers

Absorption Chiller Overview	
Water/Lithium Bromide	
Commercially Available	Yes
Size Range	10 Tons – 2,600 Tons
Fuel: *Direct-fired* *Indirect Fired*	Natural Gas Steam, hot water, hot process liquids and gases, and exhaust gases
COP_C	0.7 – 1.2
Environmental	Non-toxic working fluid, low emissions
Ammonia/Water	
Commercially Available	Yes
Size Range	3 Tons – 25 Tons
Fuel: *Direct-fired* *Indirect Fired*	Natural Gas Steam, hot water, hot process liquids and gases, and exhaust gases
COP_C	$\cong 0.5$
Environmental	Toxic working fluid, low emissions

Cost

The capital cost of installing an absorption chiller is generally more than installing an equivalent electric or engine-driven chiller. Table 7.3 compares the cost in dollars per ton for installing several different capacities of electric chillers, single-effect steam-heated absorption chillers, and double-effect direct-fired absorption chillers. These cost estimates are subject to change due to the development and deployment of new components. Costs will also be affected by the mode of energy input that is available from the waste stream of the prime mover. For example, the cost of the double-effect chiller presented in Table 7.3 is based on a direct-fired unit, and may be different for a chiller fueled by steam or exhaust gases.

Table 7.3: Installed costs of electrical and absorption chillers (http://www.bchp.org/owner-assessment.html)

	Installed Chiller Costs ($/ton)		
Electric Centrifugal	340	340	350
Single-Effect Steam-Heated Absorption	520	430	365
Double-Effect Direct-Fired Absorption	625	625	625

Maintenance costs for absorption chillers, like other technologies, is dependent on equipment capacity. Annual maintenance costs for absorption chillers range from $18 to $31 per ton of cooling. This cost varies for single-effect, double-effect, direct-fired, and indirect-fired chillers. The average maintenance cost of electric chillers ranges from $19 to $28 per ton of cooling capacity, depending upon whether the chiller uses a reciprocating, screw, or centrifugal compressor. (http://www.bchp.org/prof-assessment.html)

Absorption Chillers and CHP-B

The temperature of the waste heat available from a power source determines the appropriate absorption configuration. Table 7.4 matches waste heat temperatures typical of various power-generation components with appropriate absorption configurations. The matching is based solely on the temperature of the waste heat that could be used to generate refrigerant vapor in an absorption chiller cycle.

Table 7.4: Matching of Power Generation and Absorption Technology
(DeVault, Garland, Berry, and Fiskum, 2002)

Power Generation and Absorption Technology		
Power Source	**Temperature (°F)**	**Matching Technology**
Gas Turbine	>1,000	Triple-, double-, or single-effect
SOFC	~900	Triple-, double-, or single-effect
Microturbine	~600	Triple-, double-, or single-effect
PAFC	~250	Double-effect (preheat) or single-effect
IC Engine	~180	Single-effect
PEMFC	~140	Single-effect

Absorption chillers used in CHP-B applications offer many advantages over electric chillers. Table 7.5 lists many of these advantages, as well as some of the disadvantages that absorption chillers have in comparison to electric chillers. Lower annual operating costs that result from available waste heat streams produce shorter payback periods for absorption systems. Vapor compression systems commonly use refrigerants that are harmful to the ozone; however, newer refrigerants are more environmentally friendly. Absorption chillers, such as water/lithium bromide systems, use fluids that are not toxic and provide safe operation. Absorption systems do not have mechanical compressors and, therefore, have fewer moving parts. This gives absorption chillers lower maintenance activity, higher reliability, and quieter operation than equivalent electric and engine-driven chillers. Lower operating pressures are also an advantage for absorption chillers.

Currently, absorption chillers have higher initial costs than do electric or engine-driven chillers. Vapor compression systems are much more widely manufactured than absorption systems and, therefore, are much more widely available. Absorption chillers have seen resurgence in research and development since the early 1990's (Herold, Radermacher, and Klein, 1996). This resurgence has significantly increased the availability of absorption chillers that are suited to a broad range of applications and, more particularly, suited to CHP-B applications.

Table 7.5: Advantages of Absorption Chillers over Work-driven Heat Pumps

Absorption Chillers	
Advantages	**Disadvantages**
Lower operating costs, shorter payback period	Higher initial costs
No ozone-damaging refrigerants	Not as widely available
Safer, quieter operation	
Lower-pressure systems	
High reliability	
Low maintenance	

Absorption Refrigeration Problems

1. Two solution streams are adiabatically mixed at the same temperature. Assuming the process occurs at a constant pressure, determine the mass flow rate, quality, and enthalpy of the outlet state. Table 7.6 gives the properties of the inlet streams.

Table 7.6: Table for Problem 1

	m (kg/s)	x (%LiBr)	i (J/kg)
1	2.3	50	107.32
2	8.1	60	145.67

2. Consider a generator in an ammonia/water absorption cycle operating at steady state with the operating conditions specified in Table 7.7, where streams 1 and 2 are the inlet streams and stream 3 is the outlet stream. Find the required heat input that is to be supplied by a waste heat source assuming negligible pressure losses and a constant pressure throughout the system.

Table 7.7: Table for Problem 2

	m (kg/s)	x (%NH_3)	i (J/kg)
1	5.4	50	107.32
2			145.67
3	8.1	60	210.36

3. A double-effect water/lithium bromide absorption chiller has the operating conditions specified in Table 7.8, determine the enthalpy of the solution leaving the absorber and entering the pump. 875 kJ/s of energy is rejected during the absorption process. Assume negligible pressure losses and isobaric conditions.

Table 7.8: Table for Problem 3

	m (kg/s)	x (%NH_3)	i (kJ/kg)
From Evaporator	3.2	35	90.5
From Generator/HX	1.6		75.3
To Pump		40	

4. An absorption system like the one in Example 7-1 (Figure 7.8), has been analyzed and the data have been recorded for steady-state conditions. The generator pressure is 220 psi, and the evaporator pressure is 50 psi. Determine the pressure for all streams (1-12) in the system.

5. The absorption system in Problem 4 has a condenser mass flow rate of 45 lbm/min, a mass flow rate through the generator heat exchanger of 220 lbm/min, and a mass flow rate through the pump of 265 lbm/min. Determine the mass flow rate of all the streams in the system.

6. A microturbine supplies waste heat to a water/LiBr absorption chiller at 600°F. The ambient temperature is 75°F, and the chiller produces chilled water at 45°F. What is the maximum coefficient of performance this system can attain?

7. The mass flow rate of refrigerant vapor entering the absorber from the evaporator is 3.5 kg/s. The enthalpies of the streams on the inlet and outlet sides of the evaporator are 100 kJ/kg and 150 kJ/kg respectively. If an internal combustion engine provides 275 kJ/s of energy to the generator, determine the actual coefficient of performance of the absorption chiller.

Table 7.9: Table for Problems 8 – 10

	i (kJ/kg)	m (kg/s)	p (kPa)	T (C)	x (kg/kg)
1	-80.00	1.00	150.00	50.00	
2	-22.30			52.00	
3	256.30			125.00	
4			1,450.00	145.00	
5	5.60			52.00	0.25
6	5.60	0.80		55.00	
7	1,360.20			115.00	0.92
8		0.03		115.00	0.40
9				60.00	
10	1,150.00			50.00	
11				10.00	1.00
12	1,095.40	0.20		25.00	

8. Table 7.9 refers to an absorption system like the one analyzed in Example 1. Determine the mass flow rate, the pressure, and the concentration of all the streams in the system.

9. The absorption system above removes heat from the refrigerant in the condenser at a rate of 25 kJ/s. Determine (a) the rate of heat transfer in the evaporator, the absorber, and the generator, (b) the power required of the pump, and (c) the coefficient of performance.

10. Assuming that T_o = 55°C, T_g = 145°C, and T_e = 10°C, determine the maximum possible COP. What is the refrigerating efficiency of this chiller?

Manufacturers

A list of some absorption chiller manufacturers, along with brief product descriptions, is provided below.

- Carrier Corporation is a member of the United Technologies Corporation Family and is the world's largest supplier of air conditioning products. Carrier produces the 16JB absorption chiller, pictured in Figure 7.10, which ranges in capacity from 108 tons to 680 tons.

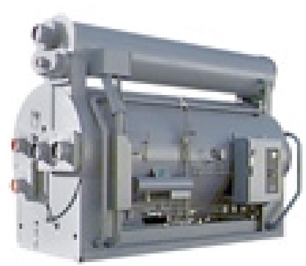


Figure 7.10: 16JB absorption chiller by Carrier Corporation (www.global.carrier.com)

- Robur Corporation, located in Evansville, IN, utilizes GAX technology in the production of ammonia/water absorption chillers. Robur manufactures 3- and 5-ton units that can be packaged together to meet larger capacity requirements up to 25 tons. Three 5-ton units are shown in Figure 7.11.

Figure 7.11: 15-Ton Chiller-Link by Robur Corp (www.robur.com)

- Trane, founded in La Crosse, Wisconsin, manufactures both single-stage and two-stage absorption chillers that are direct-, steam-, and water-fired. Figure 7.12 illustrates a Trane Horizon® Absorption Series single-stage hot water- or steam-fired absorption water chiller which is available in 500 - 1350 ton capacities.

Figure 7.12: Trane Horizon® Absorption Series Chiller (www.trane.com)

- Yazaki Energy Systems in Dallas, TX offers both direct-fired and water-fired absorption chillers. Direct-fired systems, like the one in Figure 7.13, are available in capacities ranging from 30 tons to 100 tons, and the water-fired systems are available in capacities ranging from 10 tons to 50 tons.

Figure 7.13: Gas-fired double-effect chiller-heater by Yazaki Energy Systems, Inc. (www.yazakienergy.com)

- York International, based in York, PA, produces 100 ton to 1,500 ton single-stage absorption chillers. York also manufactures two-stage absorption chillers than can be driven by natural gas, oil, propane, high-pressure steam, or waste-heat energy sources. Figure 7.14 illustrates a York single-stage absorption chiller.

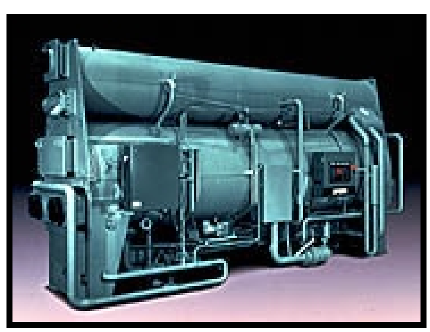


Figure 7.14: Single-stage Absorption Chiller by York International (www.york.com)

- Broad Air Conditioning is a privately owned Chinese company headquartered in Changsha, the capital of Hunan Province. Broad manufactures chillers ranging in capacities from 100 tons to 2,600 tons. Broad's primary product is a direct-fired lithium bromide absorption chiller/heater, however, indirect steam and hot water or waste heat driven units are also available.

Figure 7.15: Spectrum Absorption Chiller by Broad Air Conditioning (www.broadusa.com)

CHAPTER VIII

DESICCANT DEHUMIDIFIERS

This chapter presents an overview of desiccant dehumidifiers. Portions of the *Desiccant Dehumidification Curriculum Module* (17) have been selected. Topics include the principles of both sub-cooling and desiccant systems, types of desiccant systems, solid desiccant systems, and cost considerations for choosing desiccant systems. Desiccant technology is relevant to CHP-B systems, since the regeneration process in desiccant systems provides an excellent use for waste heat.

Sub-cooling Systems vs. Desiccant Systems

Summary of Principles of Sub-cooling Systems

In traditional cooling systems, dehumidification is achieved by cooling a moist air stream below its dew point so that liquid water condenses out of the air. This process is familiar to anyone who has seen moisture condense on a glass of ice water on a humid day. The approximate process is illustrated on a psychometric chart in Figure 8.1. The process shown is for air being cooled and dehumidified from conditions of 95ºF dry bulb (db), 75ºF wet bulb (wb) to about 77ºFdb, 58 grains/lbm$_{da}$. The resulting air state lies at the center of the ASHRAE Summer Comfort Zone in Figure 8.2. Initially, the dry bulb temperature of the moist air decreases, while the moisture content remains constant. The dry bulb te mperature continues to decrease as moisture begins to condense out of the air onto the cooling coil. In order to deliver air at 77ºFdb, 58 grains/lbm_{da}, reheat must be used. The reheat process path is also illustrated in Figure 69. In this example, the total net cooling load is 10.8 BTU/lbm_{da}, and of this, 6.4 BTU/lbm_{da}, or about 59 % is latent load.

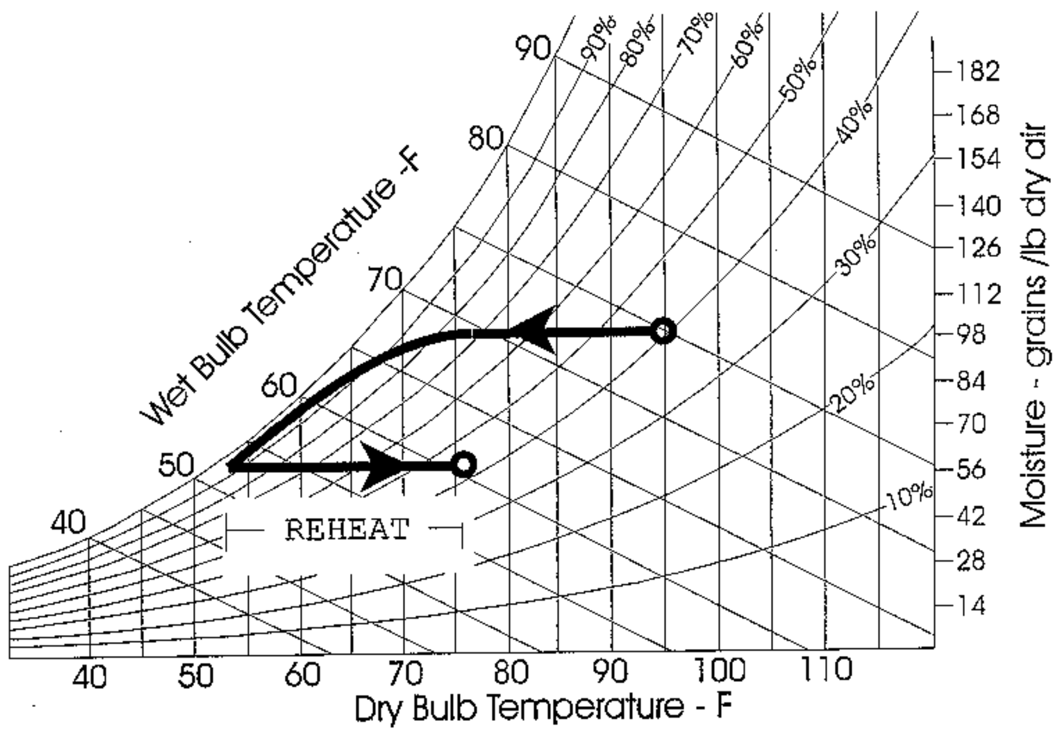


Figure 8.1: Sub-cooling Dehumidification Process (Chamra, Parsons, James, Hodge, and Steele, 2000)

In the conventional system, the same equipment is used for both sensible cooling and dehumidification. If independent humidity and temperature control are required, a provision for reheat of the cooled air must be included. In the example above, the net cooling load is 10.8 BTU/lbm$_{da}$, but the total load on the cooling coil is 16.1 BTU/lbm$_{da}$ with the difference (5.3 BTU/lbm$_{da}$) being added back in during the reheat process. Thus, energy is used both for the extra cooling and for the reheat.

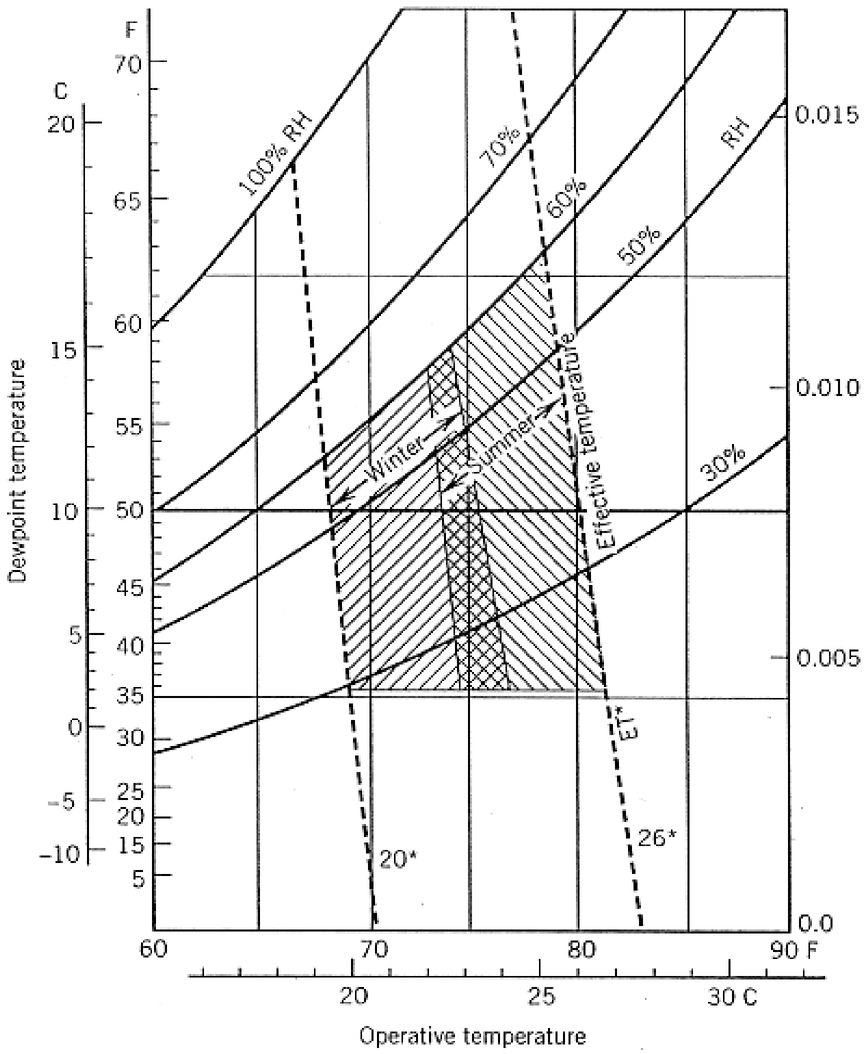


Figure 8.2: ASHRE Comfort Zones (W. R. Grace & Co., 1996)

Another disadvantage of the conventional approach is that the air leaving the evaporator coil is nearly saturated with relative humidity typically above 90 %. This moist air travels through duct work until the air is either mixed with dryer air or reaches the reheat unit. The damp ducts, along with wet evaporator coils and standing water in a condensate collection pan (Figure 8.3), foster problems with microbial growth and the associated health and odor problems.

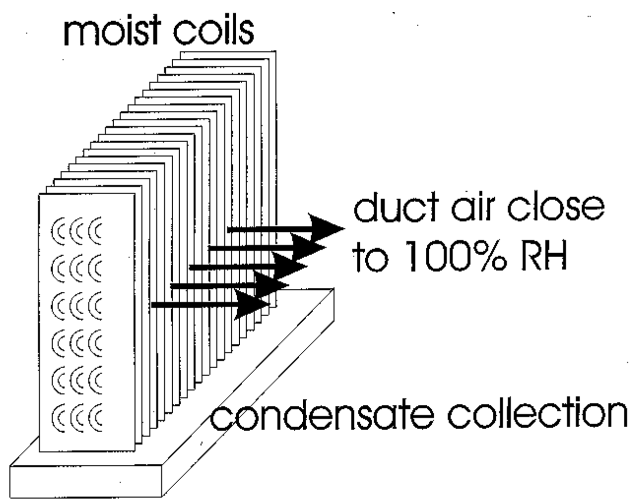


Figure 8.3: Damp Duct Symptoms (Chamra, Parsons, James, Hodge, and Steele, 2000)

Summary of Principles of Desiccant Systems

Desiccant dehumidification systems remove moisture from the air by forcing the water vapor directly into a desiccant material. The moisture from the air is attracted to desiccants since an area of low vapor pressure is created at the surface of the desiccant. The pressure exerted by the water in the air is higher, so the water molecules move from the air to the desiccant and the air is dehumidified.

The functioning of desiccant material might be compared to the action of a sponge in collecting a liquid. When the sponge is dry, it soaks up the liquid effectively. Once it becomes saturated, the sponge is taken to a different spot, the liquid is expelled by squeezing the sponge, and the dry sponge is ready to absorb more liquid. In a desiccant system, if the desiccant material is cool and dry, its surface vapor pressure is low, and moisture is attracted and absorbed from the air, which has a higher vapor pressure. After the desiccant material becomes wet and hot, it is moved to another air stream and the water vapor is expelled by raising the temperature (this step is called "regeneration"). After regeneration, the desiccant material is ready to be brought back to absorb more water vapor. The entire process involves only water vapor -- no liquid is ever condensed.

Desiccants can be either solids or liquids. The difference between solid and liquid desiccants is their reaction to moisture. Some simply collect moisture like a sponge collects water. These desiccants are called adsorbents and are mostly solid materials. Silica gel is an example of a solid adsorbent. Other desiccants undergo a chemical or physiological change as they collect moisture. These are called absorbents and are usually liquids or solids, which become liquid as they absorb moisture. Lithium chloride collects water vapor by absorption. Sodium chloride, common table salt, is another example of an absorbent.

Types of Desiccant Systems

General Classifications

Most commercial desiccant dehumidification systems use as their working material either a solid adsorbent or a liquid absorbent. Briefly, absorption is a process in which the nature of the absorbent is changed, either physically, chemically, or both. The change may include formation of a hydrate or phase change. An adsorbent, on the other hand, does not change either physically or chemically during the sorption process.

A variety of factors dictate whether an adsorbent will be commercially useful. These include cost, long-term stability, moisture removal characteristics (rate, capacity, saturation conditions, suitable temperatures), regeneration requirements (rate of moisture surrender as a function of temperature and humidity), availability, and manufacturing considerations.

Solid Adsorbents

Silica gels and zeolites are used in commercial desiccant equipment. Other solid desiccant materials include activated aluminas and activated bauxites. The desiccant material choice for a particular application depends on factors such as the regeneration temperature, the level of dehumidification, and the operating temperature.

Solid desiccant materials are arranged in a variety of ways in desiccant dehumidification systems. A large desiccant surface area in contact with the air stream is desirable, and a way to bring regeneration air to the desiccant material is necessary.

The most common configuration for commercial space conditioning is the desiccant wheel shown in Figure 8.4a. The desiccant wheel rotates continuously between the process and regeneration air streams. The wheel is constructed by placing a thin layer of desiccant material on a plastic or metal support structure. The support structure, or core, is formed so that the wheel consists of many small parallel channels coated with desiccant. Both "corrugated" and hexagonal (Figure 8.4b) channel shapes are currently in use. The channels are small enough to ensure laminar flow through the wheel. Some kind of sliding seal must be used on the face of the wheel to separate the two streams. Typical rotation speeds are between 6 and 20 rotations per hour. Wheel diameters vary from one foot to over twelve feet. Air filters are an important component of solid desiccant systems. Dust or other contaminants can interfere with the adsorption of water vapor and quickly degrade the system performance. All commercial systems include filters and maintenance directions for keeping the filters functioning properly.

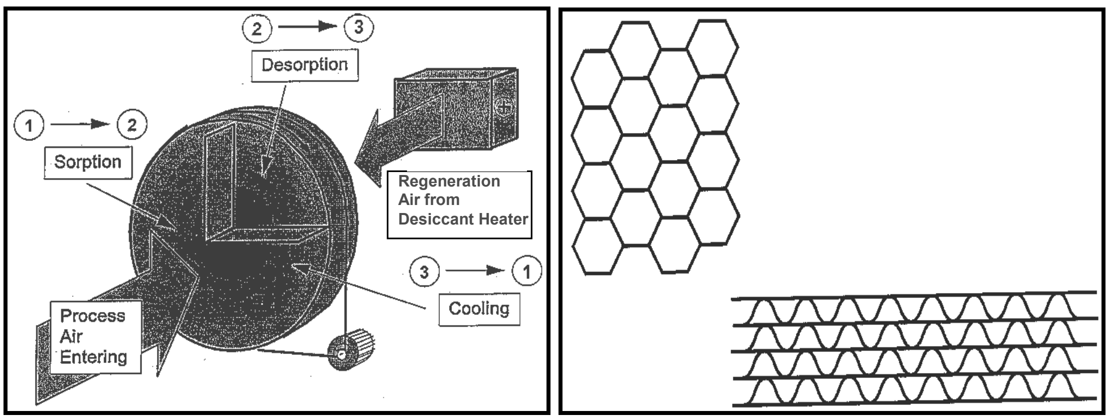


Figure 8.4: (a) Desiccant Wheel (Meckler, et al., 1995) (b) Corrugated and Hexagonal Channel Shapes (Chamra, Parsons, James, Hodge, and Steele, 2000)

Liquid Absorbents

Some materials that function as liquid absorbents are ethylene glycols, sulfuric acid, and solutions of the halogen group such as lithium chloride, calcium chloride, and lithium bromide (ASHRAE Fundamentals Handbook, 1997). A generic configuration for a liquid desiccant system is illustrated in Figure 8.5. The process air is exposed to a concentrated desiccant solution in an absorber, usually by spraying the solution through the air stream. As the solution absorbs water from the air stream, the concentration drops, and the weak solution is taken to a regenerator where heat is used to drive off the water (which is carried away by a regeneration air stream) and the concentrated solution is returned to the absorber.

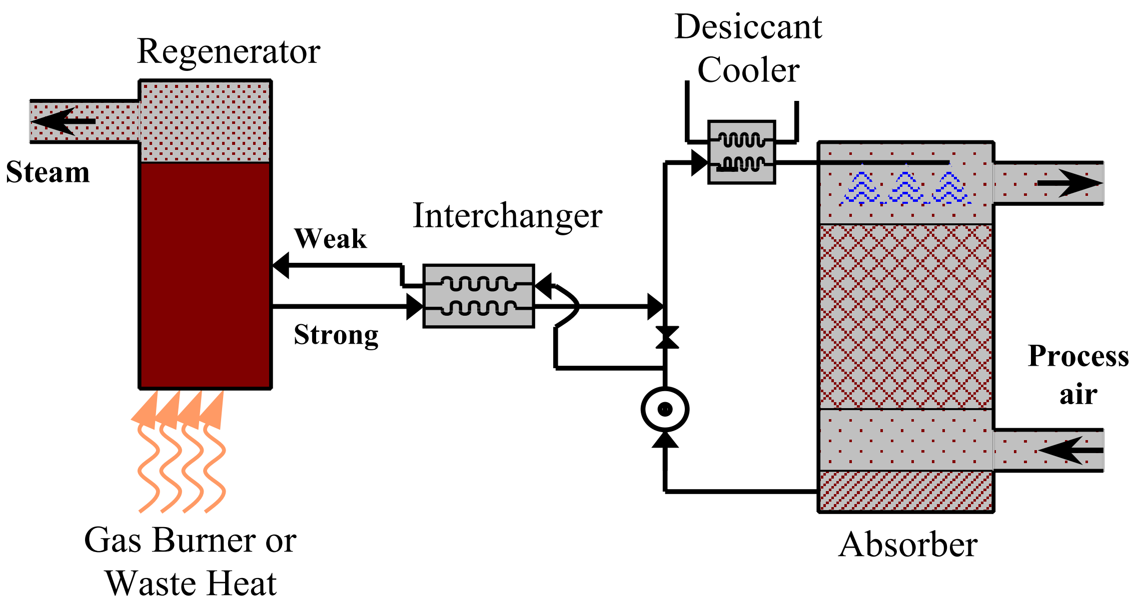


Figure 8.5: Liquid Desiccant System

Liquid desiccant systems provide the added advantage of removing many particulates from the air stream. Some liquid desiccants kill bacteria as well. Furthermore, liquid desiccant systems can be configured to operate with very low regeneration temperatures.

Regeneration

For solid or liquid systems, regeneration energy can be drawn from a variety of sources. In a CHP-B system, regeneration energy is drawn from the waste heat of a power-generation component. Due to the relatively low temperature requirements of regeneration (< 250ºF), waste heat provided by combustion turbines, IC engines, and any of the fuel cell technologies is capable of supplying heat at regeneration temperatures. The thermal energy produced in many CHP-B systems is sufficient to meet the input requirements for absorption refrigeration as well as desiccant regeneration.

Solid Desiccant system

Figure 8.6 illustrates the components of a generic solid desiccant dehumidification system. At a minimum, the system will include separated process and regeneration air-streams for the desiccant device and some kind of heater to raise the temperature of the regeneration air.

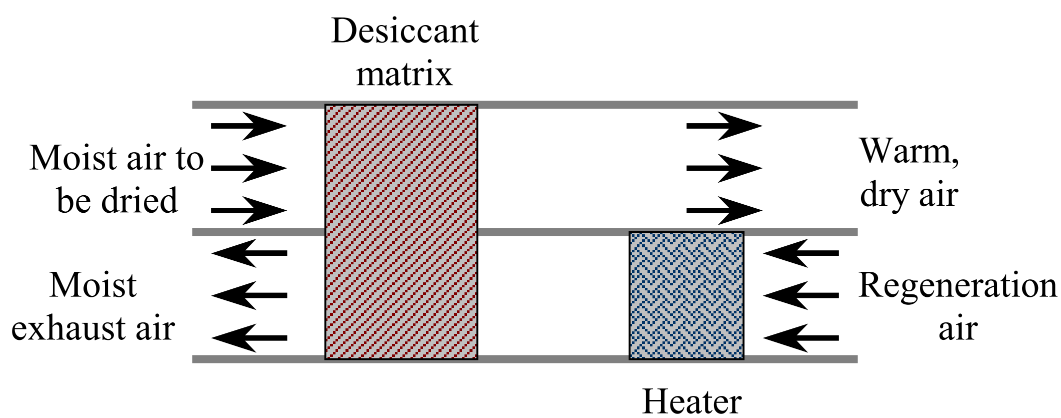


Figure 8.6: Solid Desiccant Dehumidification System

The approximate path of the process air through a desiccant device is shown in Figure 8.7 for the same inlet and outlet conditions as were shown for the sub-cooling system (Figure 8.1). Note that, as implied by the path from point 1 to point 2 in Figure 8.7, the desiccant process increases the dry bulb temperature of the process air. For solid desiccant materials, this increase is a result of the "heat of adsorption" which consists of the latent heat of vaporization of the adsorbed liquid plus an additional "heat of wetting." Heat of wetting is the energy released during dehumidification, in excess of the latent heat of vaporization. The path from point 1 to point 2 is close to a line of constant enthalpy. After the dehumidification process, the process air must undergo a sensible cooling process to reach the end point.

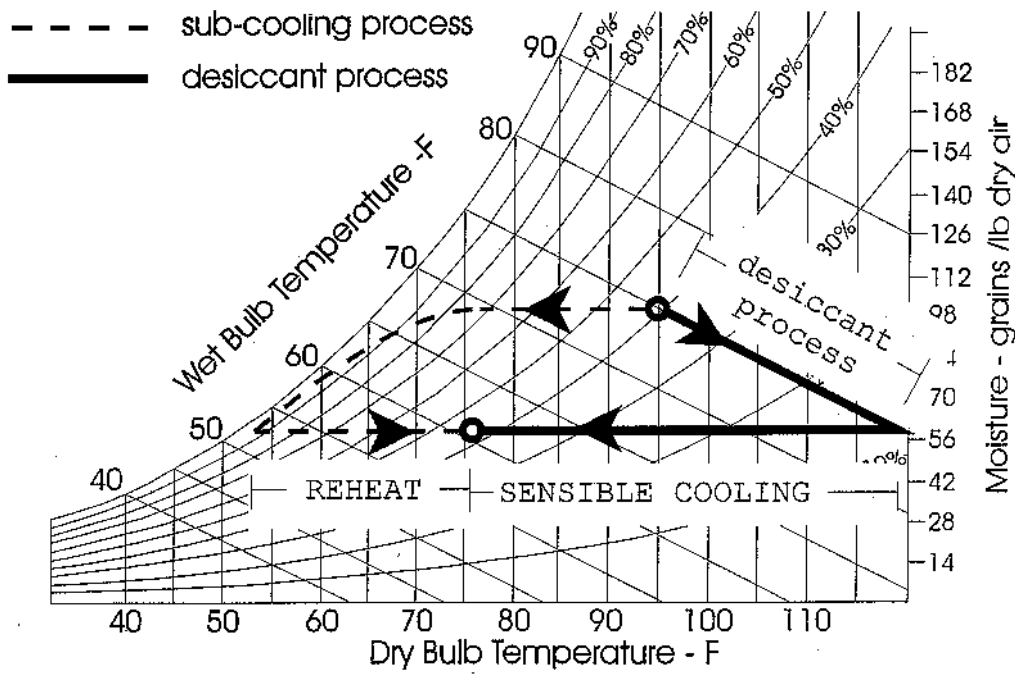


Figure 8.7: Dry Desiccant Dehumidification Process (Chamra, Parsons, James, Hodge, and Steele, 2000)

There are a wide variety of desiccant dehumidification system configurations available. The process and regeneration air inlet conditions and outlet requirements call for different configurations that are suited to individual situations. For illustration purposes, two simple examples are shown in Figures 8.8 and 8.9.

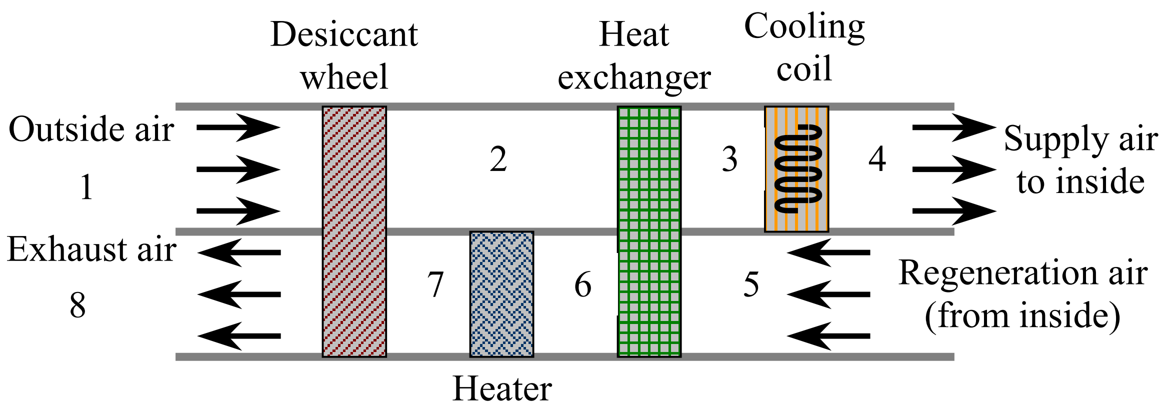


Figure 8.8: Ventilated Desiccant Dehumidification System Configuration

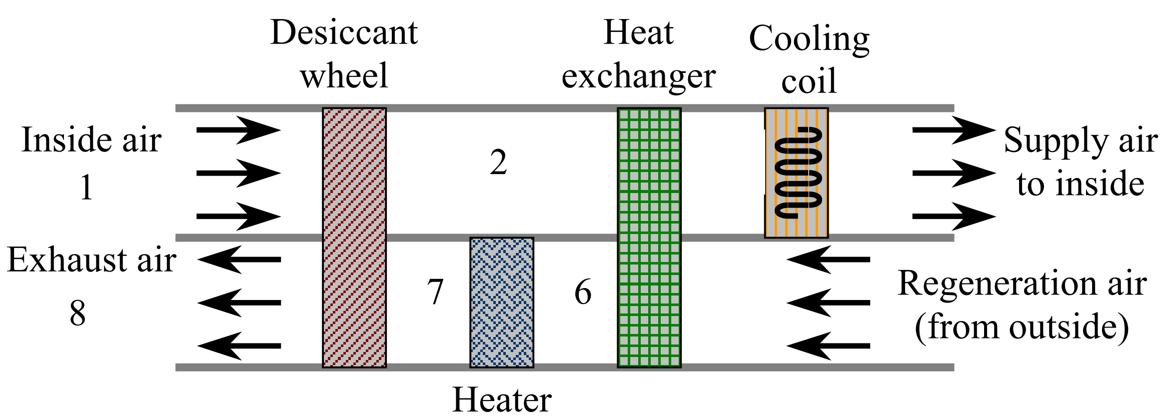


Figure 8.9: Re-circulated Desiccant Dehumidification System Configuration

Figure 8.8 illustrates a possible configuration for a desiccant system that is designed for operation in a ventilation situation; that is, 100 % outside air used to supply the conditioned space. The regeneration air for this illustration is taken completely from within the conditioned space and exhausted outside. The outside air starts at state 1 at approximately 95ºF db, 75ºF wb. From state 1 to state 2, the sensible temperature increases and the moisture content decreases as the outside air passes through a desiccant wheel. From state 2 to state 3, the hot air rejects some heat to the regeneration air stream via a heat exchanger. Finally, the air stream is cooled to the design supply condition by passing through a conventional cooling coil (state 3 to state 4).

The regeneration air stream starts at approximately 75ºF db, 40 % rh. From state 5 to state 6, this air stream acquires heat from the process air stream through a heat exchanger, and from state 6 to state 7; the regeneration stream is further heated to bring it to an appropriate temperature for desiccant regeneration. Finally, the regeneration air is cooled and humidified as it extracts moisture from the desiccant wheel (state 7 to state 8). At state 8, the regeneration stream is exhausted to the outside.

Figure 8.9 illustrates a re-circulating configuration; 100 % of the process air is drawn from the conditioned space (and all is returned to the conditioned space). The regeneration air is 100 % outside air. The equipment arrangement is identical to that of the ventilation illustration in Figure 8.8; only the air stream conditions are different.

Cost Considerations

The humidity-control capability of desiccant technology in a CHP-B system offers many potential cost savings when compared with conventional sub-cooling systems. The capital cost of a desiccant system is often more expensive than an equivalent sub-cooling system due to extra equipment costs. Installed capital cost for active solid desiccant systems (www.bchp.org) range from $4 to $9 per CFM capacity for air handling, depending upon the total capacity and equipment enclosure requirement. The higher-end of the cost range applies to systems with < 5,000 CFM. Some desiccant systems, depending on the specific installation, may result in lower capital cost; however, the reduction of latent heating load will usually result in lower operating costs for desiccant systems. Harriman (1996) discusses the installation of a desiccant dehumidification system in a medical research building where both capital costs and estimated operating costs were lower for the desiccant system.

Because the humidity and temperature can be controlled independently with a desiccant dehumidification and cooling system, the system performance is often more effective than that obtainable with conventional systems. Analysis of the sensible heat ratio (SHR) suggests the energy cost savings potential that a desiccant system may have. The SHR is the ratio of sensible cooling load to the total cooling load (sensible load plus latent load). A sensible heat ratio close to unity implies that very little moisture is removed from the air, while a sensible heat ratio close to zero indicates that most of the load is latent cooling. Air-conditioned environments often have SHR values well below unity, which results in greater energy consumption for a sub-cooling system than that of a desiccant dehumidification and cooling system that meets the same zone temperature requirements.

Some potential cost savings vary with installation and are recognized based on the individual CHP-B application. These cost savings are related to dehumidification and are implied by the process/product benefits outlined in Table 8.1. Other potential costs associated with poor humidity control should not be overlooked. Examples include retail establishments where customer discomfort or dissatisfaction due to unpleasant odors would be detrimental to business. For grocery stores, frost buildup in refrigerated display cases is unsightly, and its elimination requires expensive defrost cycles. Hospitals and nursing homes require careful attention to minimizing conditions favorable to microbial growth and propagation. Additionally, microbial growth can be hazardous to the health of children in school buildings as well as to the occupants of office and other commercial buildings. Ice rinks can reduce "fogging," condensation in the building, and can improve ice quality with lower humidity levels.

Table 8.1: Process/Product Benefits due to dehumidification (Chamra, Parsons, James, Hodge, and Steele, 2000)

Process/Product Benefits	
Process	**Product Benefits**
Lithium battery production	Prevent corrosion and improve production
Computer and electronic equipment production	Prevent condensation and corrosion on metal surfaces
Plastic molding	Improve product finish by preventing condensation on metal surfaces
Archives and museums	Increase longevity of books, artwork, and artifacts
Seeds and grain storage	Optimize seed moisture level and minimize microbial deterioration
Confectionary and pharmaceutical packaging	Deep products from deteriorating
Confectionary Manufacturing	Improve product and appearance production

Manufacturers

A list of desiccant dehumidifier manufacturers, along with brief product descriptions, is provided below.

- Bry-Air, Inc. is headquartered in Sunbury, Ohio and produces desiccant dehumidifiers in packaged units that provide flow rate capacities ranging from 50-cfm to 30,000-cfm. The Bry-Air Dry^3 ™ Compact Series, in Figure 8.10, is available in 300-, 150-, and 75-scfm.

Figure 8.10: Bry-Air Dry^3 ™ Compact desiccant dehumidifiers (www.bry-air.thomasregister.com)

- Kathabar, Inc. in Somerset, New Jersey manufactures liquid desiccant dehumidifier systems that can handle moisture removal at flow rates of 750- to 84,000-cfm. The Kathapac system by Kathabar is presented in Figure 8.11.

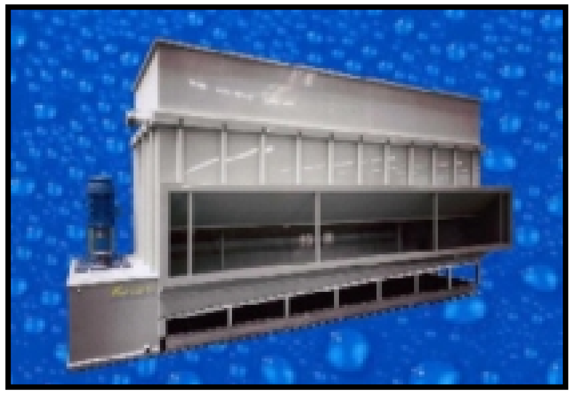


Figure 8.11: Kathapac System by Kathabar dehumidifies large air volumes (www.kathabar.imtech.nl)

- SG America in Frederick, Maryland produces desiccant wheels, enthalpy rotors, integrated desiccant dehumidifier and vapor compression systems, and desiccant dehumidification systems. Product air flow capacities range from 75- to 25,000-cfm. SG America manufactures the E-Save (Figure 8.12), the first desiccant dehumidifier product line designed for use with a microturbine power generator.

Figure 8.12: E-Save Desiccant Dehumidifier by SG America (www.sgamerica.com)

- Stulz-Air Technology Systems in Frederick, Maryland builds desiccant dehumidification systems for applications as low as 125-cfm and as high as 25,000-cfm. Figure 8.13 shows a dehumidification rotor by Stulz-Air.

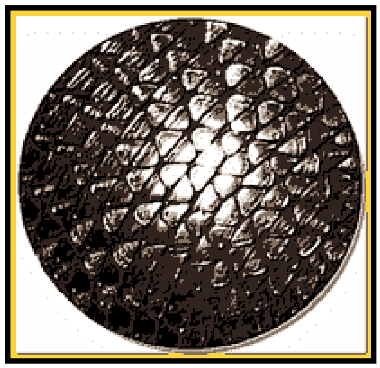


Figure 8.13: DESICAiR Dehumidification Rotor by Stulz-ATS (www.stulz-ats.com)

- Cargocaire Division of Munters USA is located in Amesbury, Massachusetts. Cargocaire traditionally manufactures integrated and packaged desiccant dehumidifier systems for industrial and military applications; however, Cargocaire now includes systems with smaller capacities in an overall production line for air-flows of 30-cfm to 80,000-cfm. A 160-cfm unit is pictured in Figure 8.14.

Figure 8.14: MH-240 by Cargocaire Operates at Flow Rates up to 160-cfm (www.gascooling.com)

CHAPTER IX

CASE STUDY: MISSISSIPPI BAPTIST MEDICAL CENTER

The Mississippi Baptist Medical Center (MBMC) in Jackson, Mississippi, represents an excellent example of a CHP-B system with a long and economically-successful record. The MBMC CHP system was brought on line in 1991, so it has a long operational history. This case study examines details of the system as well as details of the operating experiences.

Figure 9.1 is a site photograph of the MBMC. The full-service hospital has 694 beds, a 24-hour emergency room, a medical staff of 500 and more than 3000 total employees.

Figure 9.1: Site View of the MBMC

The history of CHP at MBMC started in 1990 when hospital officials were interested in exploring options to reduce energy costs. In 1990, the hospital has a large electricity requirement, a large steam requirement, a significant price differential per Btu between electricity and gas, a centralized physical plant, and small daily variations in energy requirements. Taken together, the 1990 energy profile indicated CHP, or cogeneration as it was then called, to be a viable economic option. A CHP system was proposed. A detailed economic analysis projected that the proposed CHP system should have savings of $800,000/year with an initial system cost of $4.2 million for a calculated simple payback period of 5.25 years. The system was expected to provide 70 percent of the electricity requirement, 95 percent of the steam required, and 75 percent of the cooling load. Based on the favorable outcome of the economic study, hospital officials decided to install the BCH system.

The system contains the following components:

1. Gas Turbine Generator Set: Solar Centaur H Model
2. Diverter Valve
3. ABCO Waste Heat Recovery Boiler
4. Economizer
5. Steam Absorption Chiller
6. Primary Switch Gear

The performance specifications as well as details of each of the system components are presented in the next few paragraphs. System operation can be best understood by examining Figure 9.2, which presents a schematic of the system.

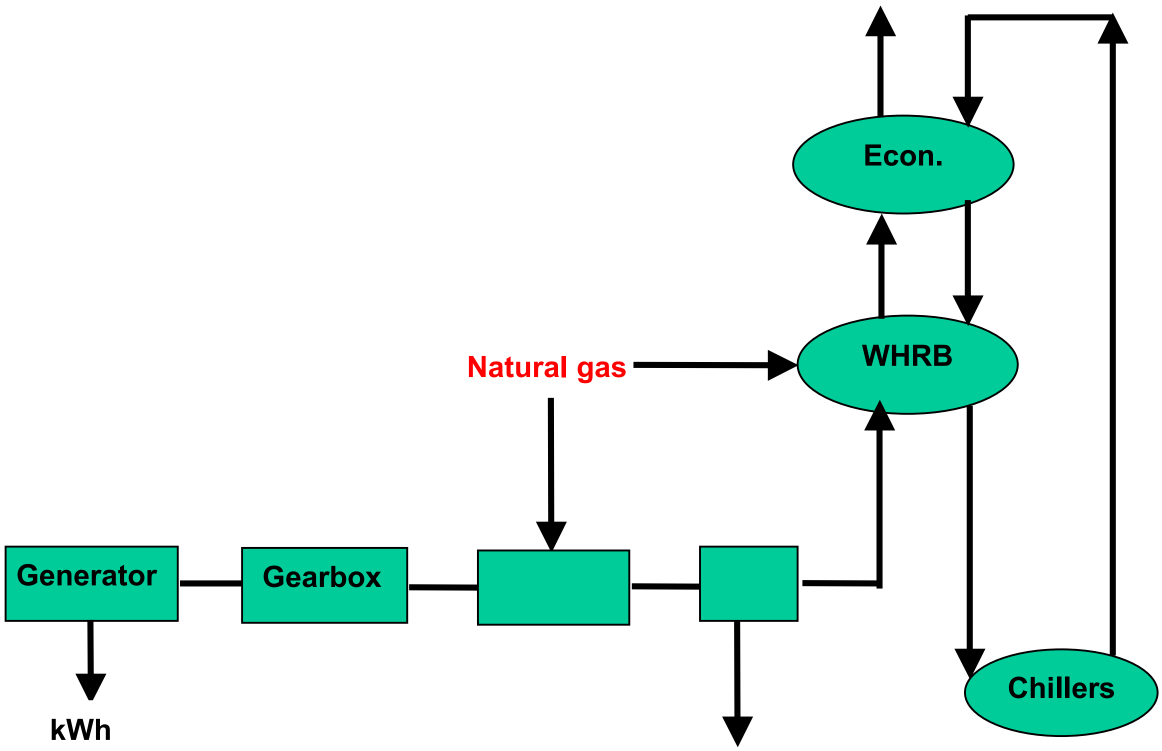


Figure 9.2: MBMC CHP System Schematic

The Solar Centaur Turbine is fired by natural gas; shaft power is extracted from the turbine and, via a gearbox, used to drive the generator. The Solar Centaur H Turbine is rated at 5600 hp with an electrical output of 4.0 MW on an ISO standard day. The turbine is controlled by an Allen Bradley PLC 5/20 microprocessor with starting and synchronizing controls, a fire detection/protection system, and vibration analyzer capability. The nominal generator output is 13, 800 volts. Figure 9.3 shows the turbine installation arrangement.

Figure 9.3: Centaur H Turbine Installation

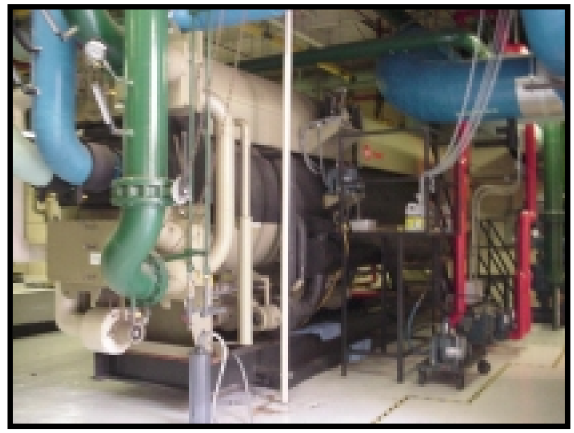
The diverter valve operation is controlled by the waste heat recovery boiler (WHRB) and directs the exhaust gas to the WHRB boiler or out of the bypass stack to maintain the required steam pressure. The ABCO WHRB is rated at 30,000 lb/hr and has two firing modes. In the turbine-firing mode, a 5.8 MMBtu duct burner is available to supplement the turbine exhaust stream. In the direct-fire mode, used when the turbine is off line, direct fresh-air fire at 41.5 MMBtu is available. The diverter valve is pictured in Figure 9.4.

The economizer utilizes the remaining waste heat to preheat boiler feed water; water treatment chemicals are added to the feed water prior to the economizer. Two absorption chillers are used by the system: a 1250-ton York Paraflow double effect chiller utilizes 115 psi steam at 11.8 lb/hr-ton to produce chilled water at 42 F and a 750-

ton Trane double effect chiller utilizes 115 psi steam at 9.6 lb/hr-ton to produce chilled water at 42 F. The chillers supply approximately 60 percent of the MBMC's 2002 total chilled water requirements. Additions to the facility since 1991 have added cooling load and resulted in 60 percent rather than 70 percent of the chilled water load being supplied by the absorption chillers. Figure 9.5 shows the two chillers and their installation arrangements.

Figure. 9.4: Diverter Valve Arrangement

(a) York Chiller



(b) Trane Chiller

Figure 9.5: The York and Trane Absorption Chillers

The primary switchgear is Powell metal-clad 4-bay switchgear that uses vacuum breakers for generator output and primary utility feed and contains generator and utility protective relays as well as synchronizing controls for generator/utility grid interconnect. The turbine-generator does not supply 100 percent of the electrical load, so the MBMC remains connected to the grid in normal operation. If a CHP problem is sensed by the switchgear, the full hospital load is instantaneously shifted to the grid. If the primary utility grid connection fails, the switchgear will shift the electrical load to a secondary utility feed in two seconds. The combination of turbine-generator, primary grid, and secondary grid provides the MBMC with triple redundancy for emergency situations. The turbine control panel and the switchgear panel are pictured in Figure 9.6.

(a) Turbine Control Panel (b) Primary Switchgear

Figure 9.6: Turbine Control Panel and Switchgear Panel

Important dates in the history of the MBMC CHP operations history are listed below.

- Construction started May 1990
- System online March 1991
- Scheduled* overhaul September 1994
- Scheduled* overhaul January 1998
- Scheduled* overhaul November 2001

  *Nominal 30,000 hours

The nominal 30,000 hours (3.42 years) between major overhauls is indicative of the reliability of ground-based turbines, and the actual overhaul dates confirm that reliability! Preventive maintenance requirements for the turbine-generator, as established by Solar, are presented as the following:

- Turbine gearbox overhaul → 3.5 years (See dates above.)
- Oil change* → As needed

  *Last oil change in 1993!

- Filter replacement → 3 months
- Routine inspections every 8 weeks
- Intermediate inspection every 6 months
- Annual inspection every year
- 200 off-line hours/year for inspections

The oil change requirement is based on a spectrographic analysis of the oil for metallic particles. The nearly ten years since the last oil change are a graphic indication of the robustness of the Solar turbine. The maintenance contract with Solar cost $14,000 per month, but includes all parts and labor. Attention to detail and a rigorous preventive maintenance policy are some of the reasons for the successful long-term operation of this CHP system.

Schmidt and Hodge (1998) examined in detail the economics of the MBMC CHP system and concluded that the actual yearly cost avoidance was close to the original estimate of $800,000/year. Their economic study results are presented in Table 9.1.

Table 9.1: Actual Cost Avoidance

Year	Electricity Savings ($)	Natural Gas ($)	Maintenance ($)	Savings ($)
1994	1,250,000	402,000	159,000	686,000
1995	1,240,000	432,000	159,000	648,000
1996	1,400,000	468,000	163,000	770,000
Average cost avoidance				**701,000**

The actual average yearly cost avoidance for 1994-1996 was $701,000 which compares favorably with the original estimate. The actual payback period was about six years, which again compares favorably with the estimated payback period of the original economic study of 1990.

One indication of the overall system performance is the percent of time the CHP system was online. Table 9.2 presents the online percentage for the years for which data were available as well as the total electrical generation as a percentage of the electrical requirement for the entire hospital.

Table 9.2: MBMC Online and Generation Percentages

Year	Percent Generation	Online Percentage
1994	74	94
1995	74	98
1996	78	98
1998	64	84
1999	73	98
2001	61	82
2002	70	93

As illustrated in Table 5-2, except for 1998 and 2001, the system provided in excess of 79 percent of the electricity for the hospital and online percentages mirrored the percent generation. In 1998, generator problems caused a multi-day outage, and in 2001 the high spike in natural gas costs caused the MBMC to rely solely on grid electricity during the peak-cost period. Taken over the 11 years of operation, the system was online and producing electricity for all but two out of 132 months!

Performance data for the last complete calendar year of operation are provided in Table 9.3. These data illustrate the effectiveness of the system in the MBMC.

Table 9.3: Calendar-Year 2001 Performance Data

Total Electricity Used	34,870,334 kWh
Electricity Generated	21,181,009 kWh
Electricity Purchased	13,689,325 kWh
Turbine Gas	334,221 MMBtu
Duct Burner Gas	18,286 MMBtu
Average Steam Prod.	19,130 lb/h

The average steam requirements for the absorption chillers and the ancillary hospital usage was virtually satisfied by the turbine exhaust as the duct burner used only 5 percent of the total gas usage for the system (turbine plus duct burner). Moreover, 2001 was the year in which gas costs caused a curtailment in the turbine usage; none-the-less, the turbine-generator accounted for more than 60 percent of the required electricity.

Many factors have contributed to the successful long-term operation of the MBMC CHP system. The most important of these factors are delineated as follows:

1. A high reliability for all system components.
2. A comprehensive maintenance program.
3. An enthusiastic and competent power-house staff.
4. An accurate and consistent monitoring procedure.
5. A no-penalty switchover-to-grid electrical rate structure.
6. A continuous assessment and improvement process.

Among the new operational policies are real-time monitoring with current utility prices for optimum economic operation and a 36-months firm gas price for economic stability ($3.80/MMBtu).

By virtually any metric, the CHP system at the Mississippi Baptist Medical Center must be rated a success. In the United States, situations where CHP make sense exist by the thousands. This case study illustrates how effective CHP can be as a solution to energy costs.

CHAPTER X

CONCLUSION

CHP-B is an emerging paradigm in energy systems. The driving potential behind CHP-B systems is the thermal efficiency that these systems can achieve. Projected CHP-B system efficiencies of 80% are well above the overall thermal efficiencies experienced by standard energy systems. Reliability, enhanced power quality, energy security, fuel versatility, improved indoor air quality, and lower emissions are additional benefits that make CHP-B systems attractive.

A variety of distributed power generation (DPG) technologies may be selected for a CHP-B system. Table 10.1 illustrates how these technologies compare in efficiency, cost/technology status, emissions, noise, and small-scale capacity. The technologies are ranked from those having the most positive characteristics to those having the most negative characteristics. For example, Fuel Cells have the lowest emissions and therefore the best characteristics in this category, and IC Engines have the highest emissions and therefore the most negative characteristics in this category. The rankings in Table 10.1 are based on the technologies as a whole and may vary in some cases.

The thermally-activated technologies discussed in this module are non-competing. A specific system-configuration may require that only an Absorption Chiller or a Desiccant Dehumidifier be utilized. However, if both cooling and dehumidification are desired, both components may be included in a CHP-B system. Thus, these thermally-activated components are not mutually exclusive and are selected based on product rather than performance.

Table 10.1: Rankings for Distributed Power Generation Technologies

Rankings for DPG Technologies	
Category	**Positive ⇒⇒⇒ Worst**
Efficiency	Fuel Cells ⇒ IC Engines ⇒ Industrial Turbines ⇒ Microtrubines
Cost/Technology Status	IC Engines ⇒ Industrial Turbines ⇒ Microturbines ⇒ Fuel Cells
Emissions	Fuel Cells ⇒ Microturbines ⇒ Industrial Turbines ⇒ IC Engines
Noise	Fuel Cells ⇒ Microturbines ⇒ Industrial Turbines ⇒ IC Engines
Small Scale Capacity	Microturbines, Fuel Cells, IC Engines ⇒ Industrial Turbines

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