COGENERATION AND HEAT RECOVERY SYSTEMS

This material discusses applied heat pump systems, heat recovery systems, and cogeneration systems. Specific details on these subjects can be found in Chapters 7 and 9 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*.

1 Cogeneration

Cogeneration designates on-site electrical generating systems that salvage byproduct or waste heat from the generating process. The magnitude, duration, and coincidence of electrical and thermal loads must be analyzed, and prime movers and waste heat recovery systems must be selected to determine system feasibility and design. The basic components of the cogeneration plant are (1) prime mover, (2) generator, (3) waste heat recovery systems, (4) control systems, and (5) connections to building mechanical and electrical services.

The normal prime movers are reciprocating internal combustion engines, combustion gas turbines, expansion turbines, and steam boiler-turbine combinations. These units convert the heat in the fuel (liquid, solid, gaseous, or nuclear) into rotating shaft energy. Figure 1 is an example of this heat recovery.

Use of the prime mover heat determines overall system efficiency and is one of the critical factors of economic feasibility. Two kinds of energy are available from the prime mover: (1) mechanical energy from the shaft and (2) heat energy remaining after the fuel or steam has acted on the shaft.

Shaft loads (generators, centrifugal chillers, compressors, process equipment) require a given amount of rotating mechanical energy. Once the prime mover has been selected to provide the required shaft output, it has a fixed relationship to system efficiency that is dependent upon the prime mover fuel versus load and the load versus heat balance curves.

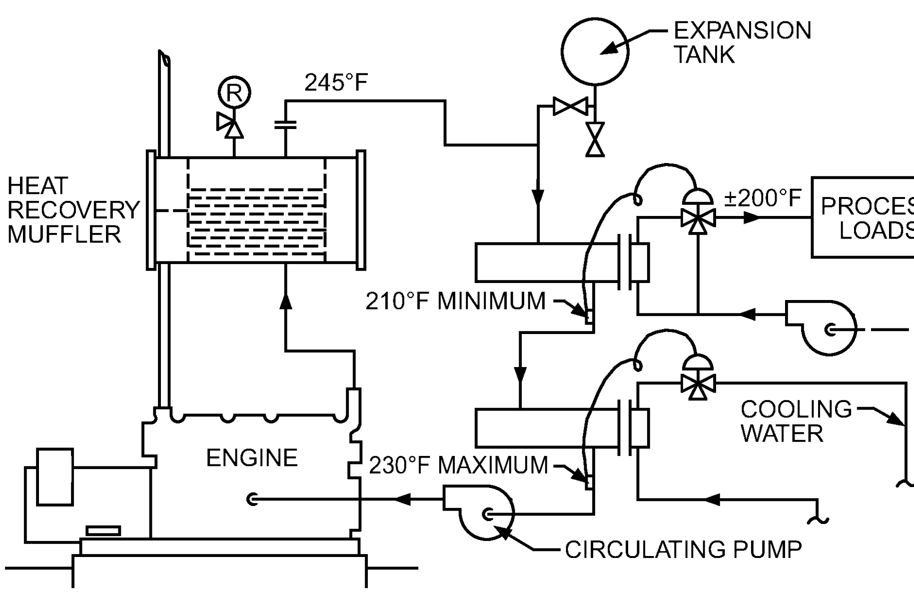

Fig. 1 Hot-Water Heat Recovery
(Figure 46, Chapter 7, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Steam turbine drives can be arranged to extract steam at intermediate turbine stages. The waste heat value of a steam turbine is the enthalpy of the steam at the point it is extracted from the turbine or at the turbine's exhaust outlet. This steam, reduced in pressure and temperature by the amount of shaft work, can be fed to heat exchange equipment, absorption chillers, and steam turbine-driven centrifugal chillers.

The gas turbine cycle has a thermal efficiency of approximately 20%, with the remainder of the fuel energy exhausted or radiated. A minimum exhaust temperature of approximately 300°F (150°C) is required to prevent condensation. The recoverable heat per unit of power is greater for a gas turbine than for a reciprocating engine because the power is less per unit of fuel input. Because gas turbine exhaust contains a large percentage of excess air, afterburners or boost burners may be installed in the exhaust to create a supplementary boiler system. This system can provide additional steam or level the steam production during reduced turbine loads.

In all reciprocating internal combustion engines except small air-cooled units, heat can be reclaimed from the lubricating system, jacket cooling water system, and the exhaust.

Coolant fluids and lubricating oil are generally circulated to remove excess heat conducted into the power train during combustion and heat from friction. Some engines are constructed to convert cooling water to steam within the engine.

The approximate distribution of input fuel energy is as follows:

Useful work	33%
Friction and radiation	7%
Rejected in jacket water	30%
Rejected in exhaust	30%

Of course, neither all of the exhaust heat nor all of the jacket heat can be recovered. Good design could result in the usable portion of the jacket heat and the exhaust heat being about 70% of that shown, or 21% of the input for each.

Depending on engine design, these amounts vary. However, they do indicate that overall cycle thermal efficiency can be greatly improved by waste heat recovery systems if there is a beneficial use for the recovered heat.

Example 1 A 200 kW internal combustion engine power unit produces 2570 lb/h (0.32 kg/s) of exhaust gas at 950°F (510°C). The exhaust gas mixture has a specific heat of 0.252 Btu/lb·°F (1.06 kJ/(kg·K)]. Energy in the exhaust gas is to be used in a waste heat boiler to produce dry saturated steam at 280°F (138°C) from water supplied at 60°F (16°C). The exhaust gas is cooled during the process from 950°F to 400°F (510°C to 204°C). Determine the quantity of steam that can be produced, lb/h.

Solution

From Table 2-1 of Chapter 2 of *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition:

$$h_g \text{ [at 280°F (138°C)]} = 1173.94 \text{ Btu/lb (2730.88 kJ/kg)}$$

$$h_f \text{ [at 60°F (16°C)]} = 28.07 \text{ Btu/lb (67.16 kJ/kg)}$$

Equating the heat transfer rate from the exhaust gas to that for the water/steam, yields

$$m_g c_p\,(t_{g,\text{in}} - t_{g,\text{out}}) = m_s(h_g - h_f)$$

$$2570(0.252)(950 - 400) = m_s(1173.4 - 28.07)$$

$$m_s = 311 \text{ lb/h (0.039 kg/s)}$$

2 Heat Recovery Terminology and Concepts

The following definitions serve as an introduction to heat recovery systems.

Balanced heat recovery. Occurs when internal heat gain equals recovered heat and no external heat is introduced to the conditioned space. Maintaining balance may require raising the temperature of recovered heat.

Break-even temperature. The outdoor temperature at which total heat losses from conditioned spaces equal internally generated heat gains.

Changeover temperature. The outdoor temperature the designer selects as the point of changeover from cooling to heating by the HVAC system.

External heat. Heat generated from sources outside the conditioned area. This heat from gas, oil, steam, electricity, or solar sources supplements internal heat and internal process heat sources. Recovered internal heat can reduce the demand for external heat.

Internal heat. The total passive heat generated within the conditioned space. It includes heat generated by lighting, computers, business machines, occupants, and mechanical and electrical equipment such as fans, pumps, compressors, and transformers.

Internal process heat. Heat from industrial activities and sources such as wastewater, boiler flue gas, coolants, exhaust air, and some waste materials. This heat is normally wasted unless equipment is included to extract it for further use.

Pinch technology. An energy analysis tool that uses vector analysis to evaluate all heating and cooling utilities in a process. Composite curves created by adding the vectors allow identification of a “pinch” point, which is the best thermal location for a heat pump.

Recovered (or reclaimed) heat. Comes from internal heat sources. It is used for space heating, domestic or service water heating, air reheat in air conditioning, process heating in industrial applications, or other similar purposes. Recovered heat may be stored for later use.

Stored heat. Heat from external or recovered heat sources that is held in reserve for later use.

Usable temperature. The temperature or range of temperatures at which heat energy can be absorbed, rejected, or stored for use within the system.

Waste heat. Heat rejected from the building (or process) because its temperature is too low for economical recovery or direct use or storage capacity is not available.

2.1 Definition of Balanced Heat Recovery Systems

In an ideal heat recovery system, all components work year-round to recover all of the internal heat before adding external heat. Any excess heat is either stored or rejected. Such an idealized goal is identified as a balanced heat recovery system.

When the outdoor temperature drops significantly, or when the building is shut down (e.g., on nights and weekends), internal heat gain may be insufficient to meet the space heating requirements. Then, a balanced system provides heat from storage or an external source. When internal heat is again generated, the external heat is automatically reduced to maintain proper temperature in the space. There is a time delay before equilibrium is reached. The size of the equipment and the external heat source can be reduced in a balanced system that includes storage. Regardless of the system, a heat balance analysis establishes the merits of balanced heat recovery at various outdoor temperatures.

Outdoor air less than 55°F to 65°F (13°C to 18°C) may be used to cool building spaces with an air economizer cycle. When considering this method of cooling, the space required by ducts, air shafts, and fans, as well as the increased filtering requirements to remove contaminants and the hazard of possible freeze-up of dampers and coils must be weighted against alternatives such as using deep row coils with antifreeze fluids and efficient heat exchange. Innovative use of heat pump principles may give considerable energy savings and more satisfactory human comfort than an air economizer. In any case, hot and cold air should not be mixed (if avoidable) to control zone temperatures because it wastes energy.

Many buildings, especially those with computers or large interior areas, generate more heat than can be used for most of the year. Operating cost is minimized when the system changes over from net heating to net cooling at the break-even outdoor temperature at which the building heat loss equals the internal heat load. If heat is unnecessarily rejected or added to the space, the changeover temperature varies from the natural break-even temperature, and operating costs increase. Heating costs can be reduced or eliminated if excess heat is stored for later distribution. The concept of ideal heat balance in an overall building project or a single space requires that one of the following takes place on demand:

- Heat must be removed.
- Heat must be added.
- Heat recovered must exactly balance the heat required, in which case heat should be neither added nor removed.

In small air-conditioning projects serving only one space, either cooling or heating satisfies the thermostat demand. If humidity control is not required, operation is simple. Assuming both heating and cooling are available, automatic controls will respond to the thermostat to supply either. A system should not heat and cool the same space simultaneously.

Multiroom buildings commonly require heating in some rooms and cooling in others. Optimum design considers the building as a whole and transfers excess internal heat from one area to another, as required, without introducing external heat that would require waste heat disposal at the same time. The heat balance concept is violated when this occurs.

Humidity control must also be considered. Any system should add or remove only enough heat to maintain the desired temperature and control the humidity. Large percentages of outdoor air with high wet-bulb temperatures, as well as certain types of humidity control, may require reheat, which could upset the desirable balance. Usually, humidity control can be obtained without upsetting the balance. When reheat is unavoidable, internally transferred heat from heat recovery should always be used to the extent it is available before using an external heat source such as a boiler. However, the effect of the added reheat must be analyzed because it affects the heat balance and may have to be treated as a variable internal load.

When a building requires heat and the refrigeration plant is not in use, dehumidification is not usually required and the outdoor air is dry enough to compensate for any internal moisture gains. This should be carefully reviewed for each design.

Heat Balance Studies. The following examples illustrate situations that can occur in nonrecovery and unbalanced heat recovery situation. Figure 2 shows the major components of a building that comprise the total air-conditioning load. Values above the zero line are cooling loads, and values below the zero line are heating loads. On an individual basis, the ventilation and conduction loads cross the zero line, which indicated that these loads can be a heating or a cooling load, depending on outdoor temperature. Solar and internal loads are always a cooling load and are, therefore, above the zero line.

Figure 3 combines all of the loads shown in Figure 2. The graph is obtained by plotting the conduction load of a building at various outdoor temperatures and then adding or subtracting the other loads at each temperature. The project load lines, with and without solar effect, cross the zero line at 16°F (–9°C) and 30°F (–1°C), respectively. These are the outdoor temperatures for the plotted conditions when the naturally created internal load exactly balances the loss.

As plotted, this heat balance diagram includes only the building loads with no allowance for additional external heat from a boiler or other source. If external heat is necessary because of system design, the diagram should include the additional heat.

Figure 4 illustrates what happens when heat recovery is not used. It is assumed that with a temperature of 70°F (21°C), heat from an external source is added to balance conduction through the building's skin in increasing amounts down to the minimum outdoor temperature winter design condition. Figure 4 also adds the heat required for the outdoor air intake. The outdoor air, comprising part or all of the supply air, must be heated from outdoor temperature to room temperature. Only the temperature range above the room temperature is effective for heating to balance the perimeter conduction loss.

These loads are plotted at the minimum outdoor winter design temperature, resulting in a new line passing through points A, D, and E. This line crosses the zero line at –35°F (–37°C), which becomes the artificially created break-even temperature rather than 30°F (–1°C), when not allowing for solar effect. When the sun shines, the added solar heat at the minimum design temperature would further drop the –35°F (–37°C) break-even temperature. Such a design adds more heat than the overall project requires and does not use balanced heat recovery to use the available internal heat. This problem is most evident during mild weather on systems not designed to take full advantage of internally generated heat year-round.

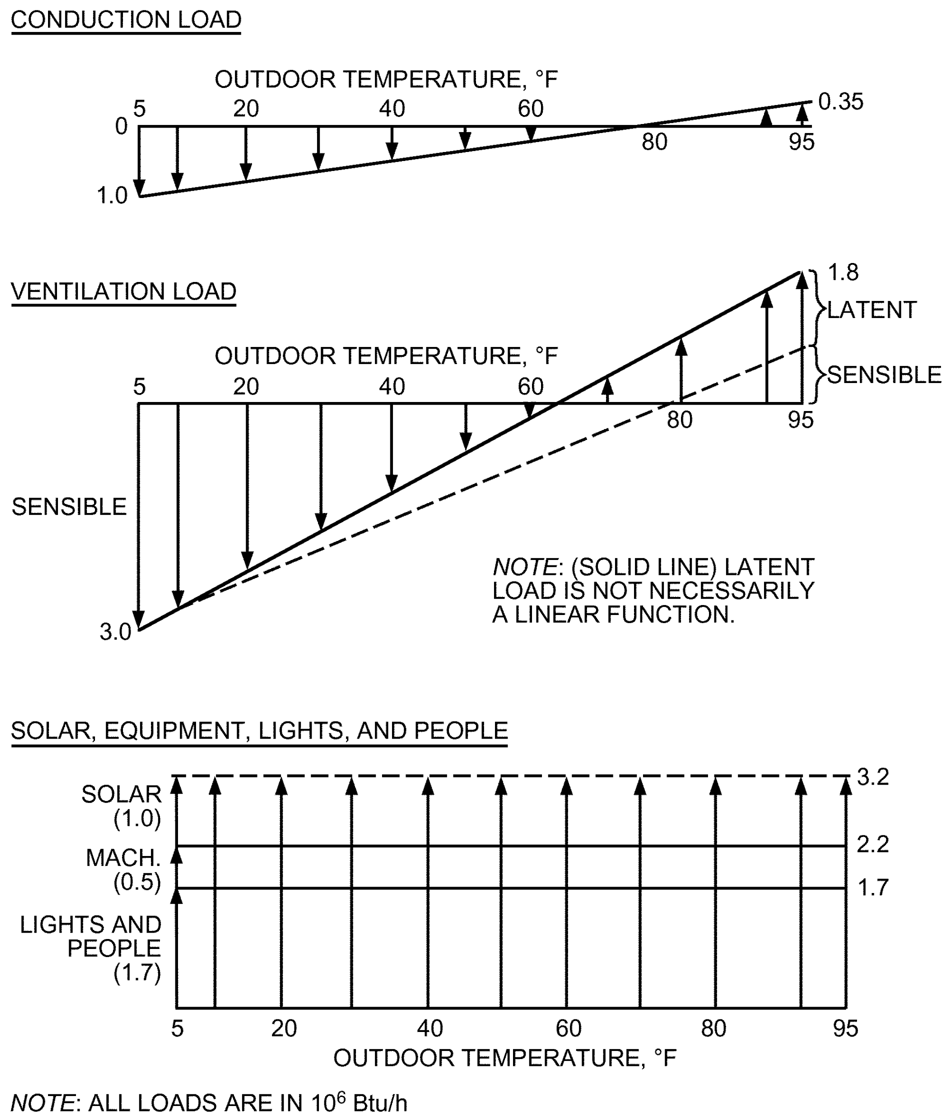

Fig. 2 Major Load Components (I-P)
(Figure 34, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

The following are two examples of situations that can be shown in a heat balance study:

1. As the outdoor air wet-bulb temperature drops, the total heat of the air falls. If a mixture of outdoor and recirculated air is cooled to 55°F (13°C) in summer and the same dry-bulb temperature is supplied by an economizer cycle for interior space cooling in winter, there will be an entirely different result. As the outdoor wet-bulb temperature drops below 55°F (13°C), each unit volume of air introduced does more cooling. To make matters more difficult, this increased cooling is latent cooling, which requires adding latent heat to prevent too low a relative humidity, yet this air is intended to cool. The extent of this added external heat for free cooling is shown to be very large when plotted on a heat balance analysis at 0°F (–18°C) outside temperature.

   Figure 4 is typical for many current non-heat-recovery systems. There may be a need for cooling, even at the minimum design temperature, but the need to add external heat for humidification can be eliminated by using available internal heat. When this asset is thrown away and external heat is added, operation is less efficient.

   Some systems recover heat from exhaust air to heat the incoming air. When a system operates below its natural break-even temperature t_{be}, such as 30°F (–1°C) or 16°F (–9°C) (shown in

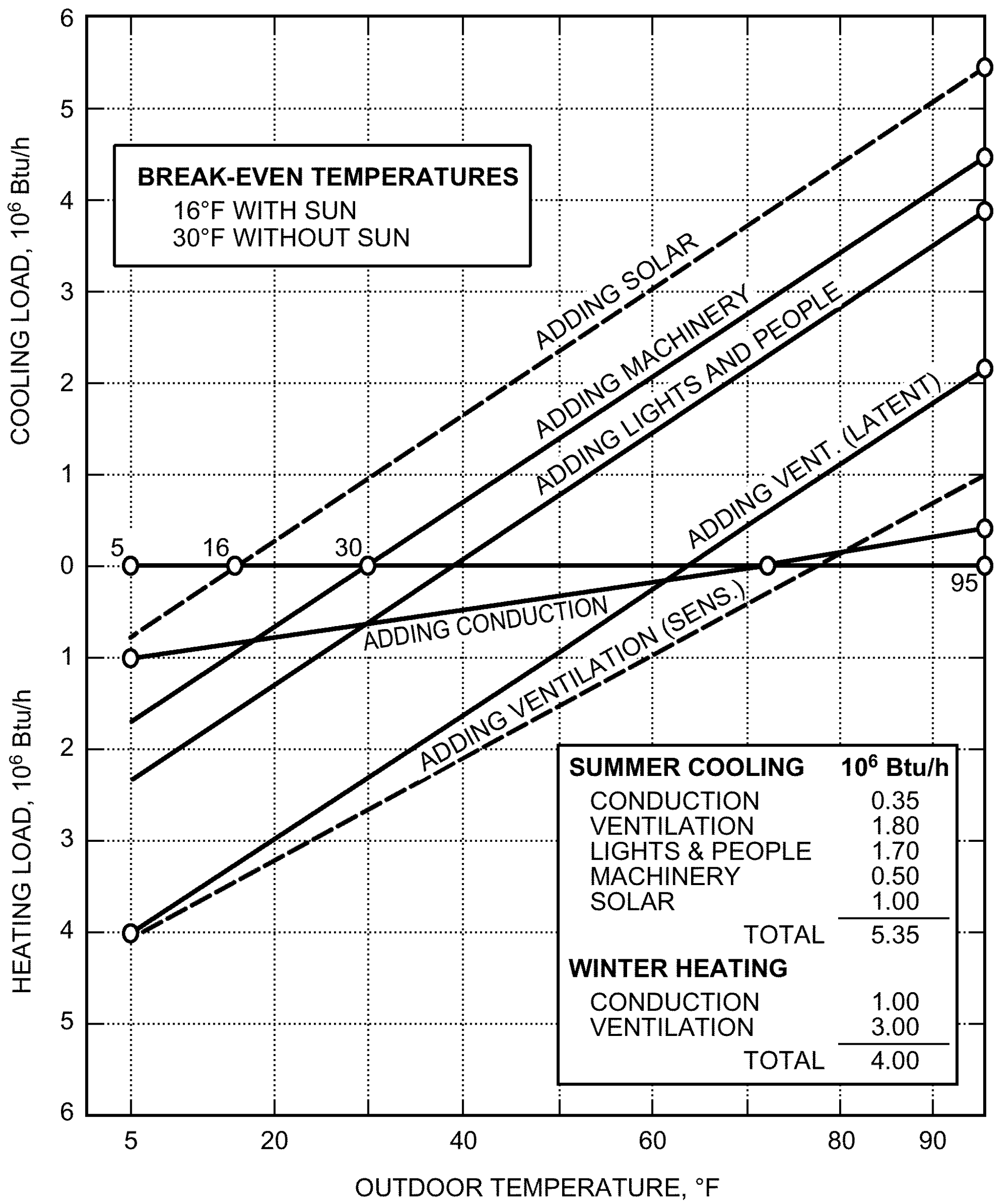

Fig. 3 Composite Plot of Loads in Figure 2 (I-P)
(Figure 35, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Figure 3), the heat recovered from exhaust air is useful and beneficial. This assumes that only the available internal heat is used and that no supplementary heat is added at or above t_{be}. Above t_{be}, the internal heat is sufficient and any recovered heat would become excessive heat to be removed by more outdoor air or refrigeration.

If heat is added to a central system to create an artificial t_{be} of –35°F (–37°C) as in Figure 4, any recovered heat above –35°F (–37°C) requires an equivalent amount of heat removal elsewhere. If the project were in an area with a minimum design temperature of 0°F (–18°C), heat recovery from exhaust air could be a liability at all times for the conditions stipulated in Figure 3. This does not mean that the value of heat recovered from exhaust air should be forgotten. The emphasis should be on recovering heat from exhaust air rather than on adding external heat.

2. A heat balance shows that insulation, double glazing, and so forth can be extremely valuable in some situations. However, these practices may be undesirable in certain regions during the heating season,

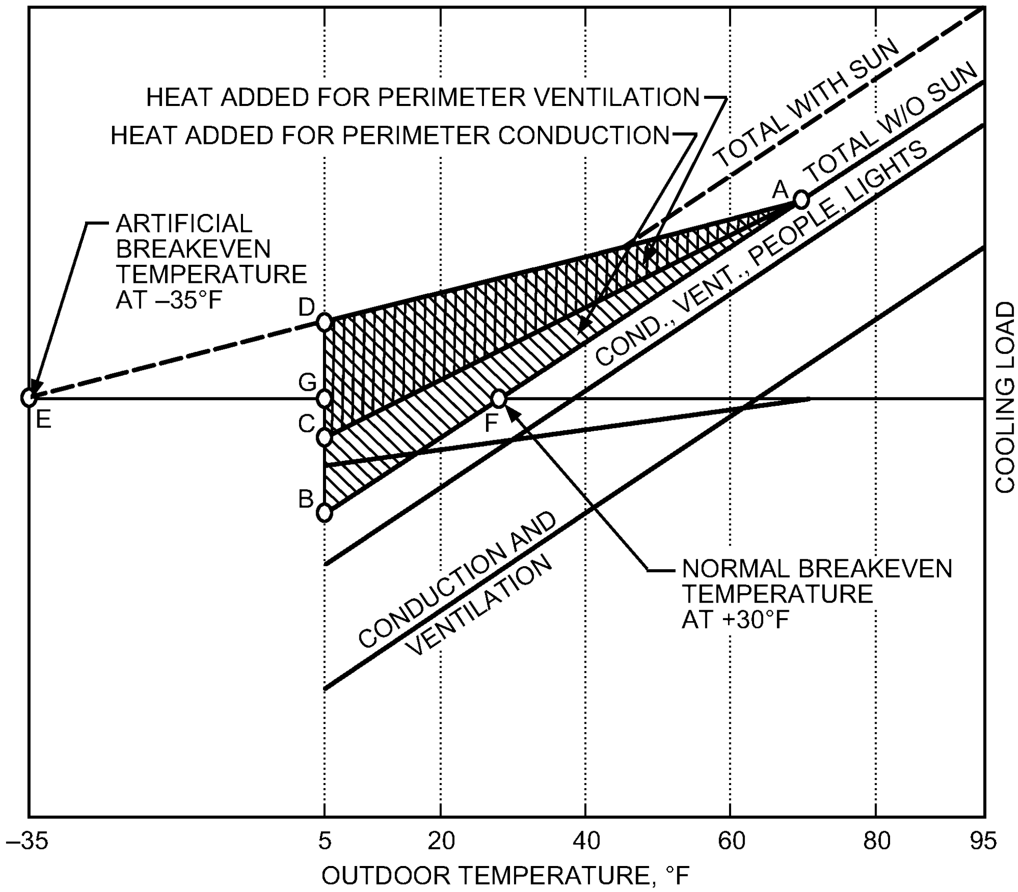

Fig. 4 Non-Heat Recovery System (I-P)
(Figure 36, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

when excess heat must usually be removed from large buildings. For instance, for minimum winter design temperatures of approximately 35°F to 40°F (1.7°C to 4.5°C), it is improbable that the interior core of a large office building will ever reach its break-even temperature. The temperature lag for shutdown periods, such as nights and weekends, at minimum design conditions could never economically justify the added cost of double-pane windows. Therefore, double-pane windows merely require the amount of heat saved to be removed elsewhere. However, in cold climates the double-pane windows may be necessary to provide comfort.

3 Heat Recovery Systems

Figures 5 through 9 show several possible heat recovery/simultaneous heating-cooling systems. Figure 5 illustrates one method of using water as the heat source or sink and as the heating and cooling medium. The compressor, evaporator, condenser, refrigerant piping, and accessories are essentially standard and are available as a factory-packaged water-to-water heat pumps (see Chapter 12 of *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition, and the book's accompanying Panel Heating and Cooling Systems online supplemental material available at www.ashrae.org/PHVAC9th).

The cycle is flexible, and the heating or cooling medium is instantly available at all times. Heating can be provided exclusively to the zone conditioners by closing valves 2 and 3 and opening valves 1 and 4. With the valves in these positions, the water is divided into two separate circuits. The warm water circuit consists of the condenser (where the heat is supplied by the high-temperature refrigerant), valve 1, zone conditioners, and a circulating pump. The cold water circuit consists of the evaporator (where heat is taken from the water by the low-temperature refrigerant), valve 4, a heat exchanger (where heat is taken from the source water),

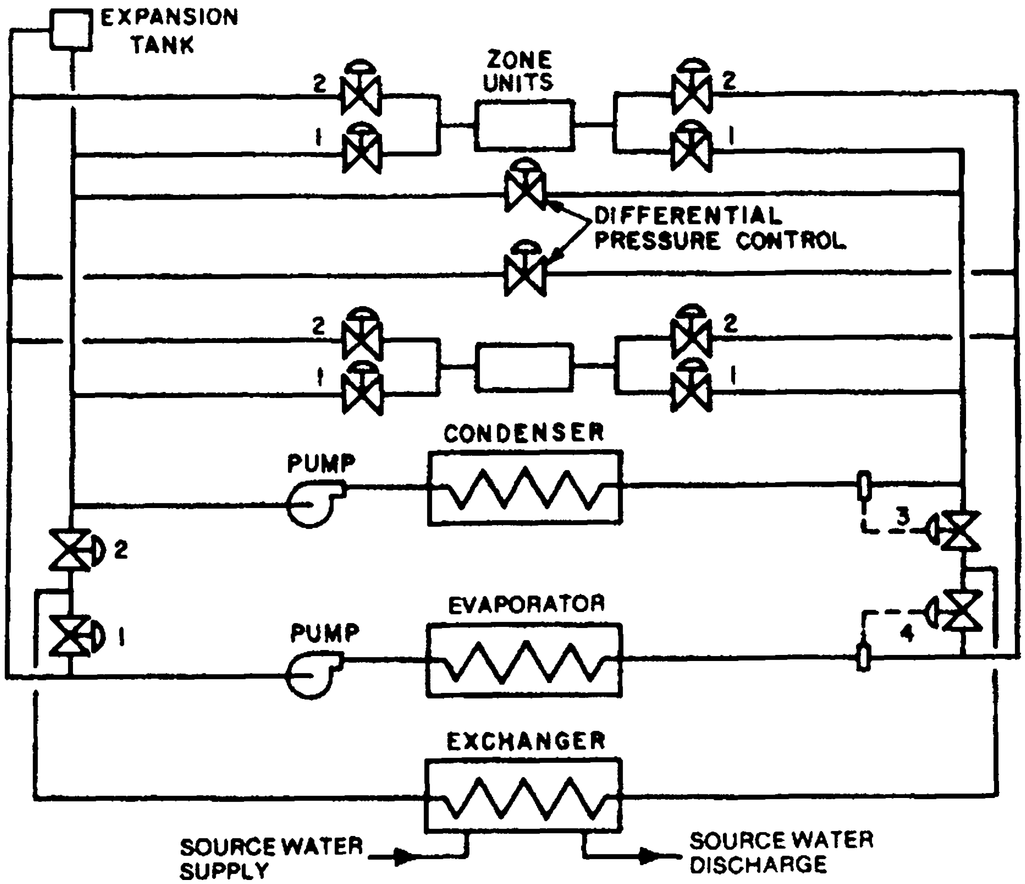

Fig. 5 Water-to-Water Heat Pump Cycle

valve 1, and a circulating pump. The refrigerating compressor operates to maintain the desired leaving water temperature from the condenser.

Similarly, cooling can be exclusively obtained in the cycle of Figure 5 by opening valves 2 and 3 and closing valves 1 and 4. With this arrangement, the cold water circuit consists of the evaporator (where heat is removed from the water by the low-temperature refrigerant), valve 2, zone conditioners, and a circulating pump. The warm water circuit consists of the condenser (which receives the heat from the refrigerant), valve 3, a heat exchanger (where heat is rejected to the source water), valve 2, and a circulating pump. The refrigerating compressor operates to maintain the desired water temperature leaving the evaporator.

During the intermediate season, simultaneous heating and cooling can be provided by the cycle shown in Figure 5. Valves 3 and 4 are modulated when valves 1 and 2 are open. Valve 3 is adjusted to maintain 85°F to 140°F (29°C to 60°C) water in the condenser circuit and valve 4 to maintain 45°F to 50°F (7°C to 10°C) in the evaporator circuit. The excess heating or cooling effect is discharged to the exchanger, which passes it on to the source water.

The source or sink water, if of suitable quality, can be supplied directly to the condenser and evaporator instead of using an exchanger (Figure 5). This eliminates one heat transfer surface and its performance penalty.

A water loop heat pump (WLHP) cycle that combines load transfer characteristics with water-to-air heat pump units is illustrated in Figure 6. Each module or space has one or more water-to-air heat pumps. The units in both the building core and perimeter areas are connected hydronically with a common two-pipe system. Each unit cools conventionally, supplying air to the individual module and rejecting the heat removed to the two-pipe system through its integral condenser. The total heat gathered by the two-pipe system is expelled through a common heat rejection device. This device often includes a closed circuit evaporative cooler with an integral spray pump. If and when some of the modules, particularly on the northern side, require heat, the individual units switch (by means of four-way refrigerant valves) into the heating cycle. The units derive their heat source from the two-pipe water loop, basically obtaining heat from a relatively high temperature source,

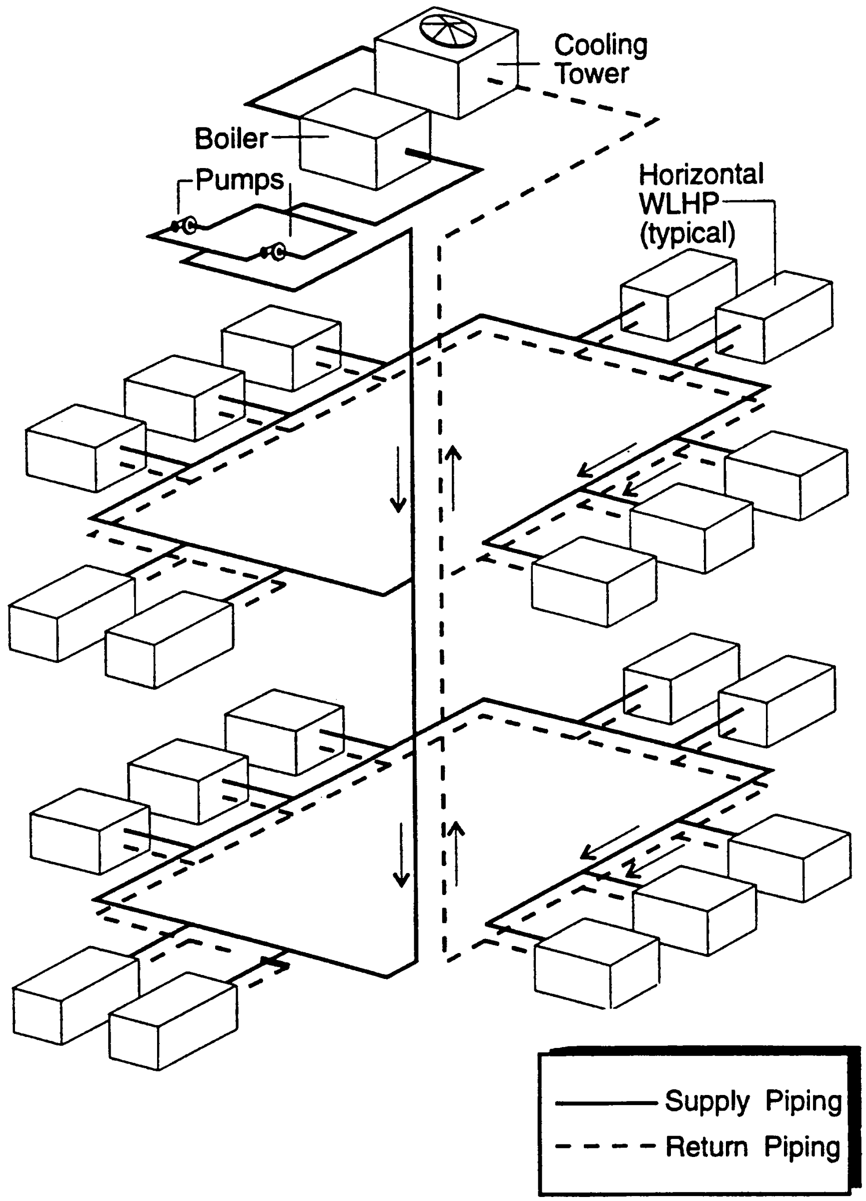

Fig. 6 Heat Recovery System Using Water-to-Air Heat Pump in Closed Loop
(Figure 30, Chapter 9, 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

that is, the condenser water of the other units. When only heating is required, all units are in the heating cycle and, consequently, an external heat input source is needed to provide heating capability. The heat of compression contributes to the heat source. The water loop is usually 60°F to 90°F (15°C to 32°C) and, therefore, seldom requires piping insulation.

A water-to-water heat pump can be added in the closed water loop before the heat rejection device for further heat reclaim. This heat pump reuses the heat and can provide domestic hot water or elevate water temperatures in a storage tank to be bled back into the loop.

In many large buildings, internal heat gains require year-round chiller operation. This internal heat is often discharged through a cooling tower. Prudent design may dictate cascade systems with chillers in parallel or series. Manufacturers can assist with custom components to meet a wide range of load and temperature requirements. The double-bundle condenser working with a reciprocating or centrifugal compressor is most often used in this application. Figure 7 shows the basic configuration of this system, which makes heat avail-

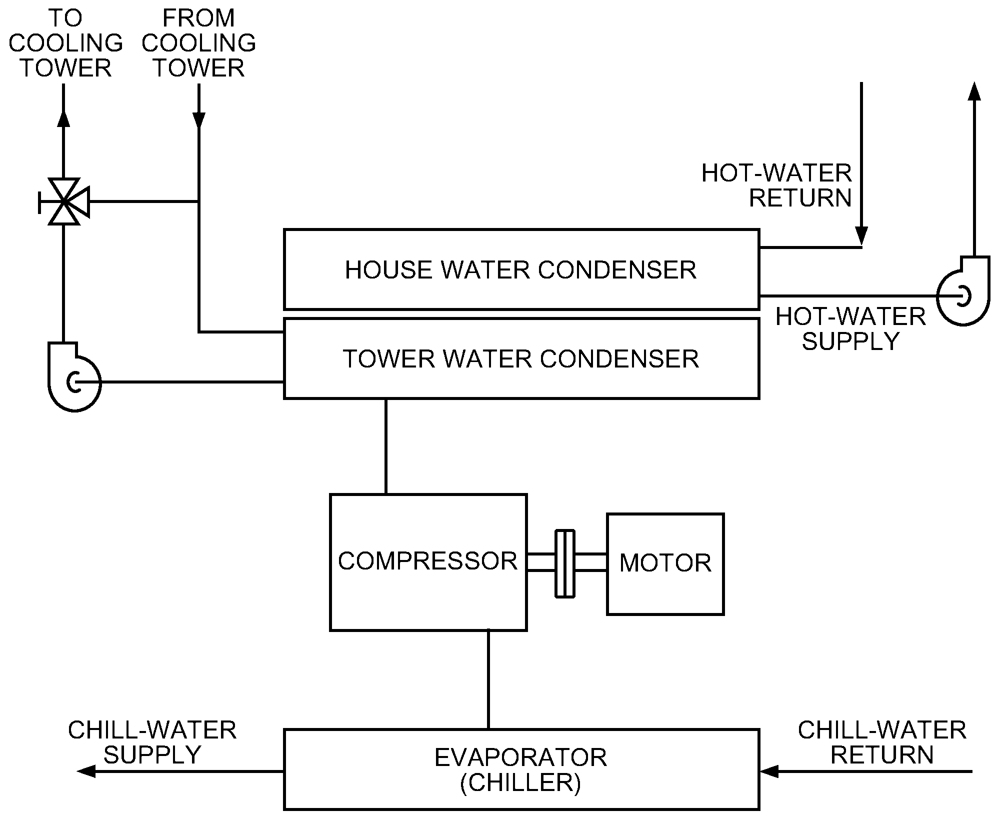

Fig. 7 Heat Transfer Heat Pump with Double-Bundle Condenser
(Figure 23, Chapter 9, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*)

able in the range of 100°F to 130°F (38°C to 54°C). The warm water is supplied as a secondary function of the heat pump and represents recovered heat.

Figure 8 shows a similar cycle, except that a storage tank has been added, enabling the system to store heat during occupied hours by raising the temperature of the water in the tank. During unoccupied hours, water from the tank is gradually fed to the evaporator providing load for the compressor and condenser that heats the building during off hours.

Figure 9 is another transfer system capable of generating 130°F to 140°F (54°C to 60°C) or warmer water whenever there is a cooling load by cascading two compressors hydronically. In this configuration, one chiller can be considered as a chiller only and the second unit as a heating-only heat pump.

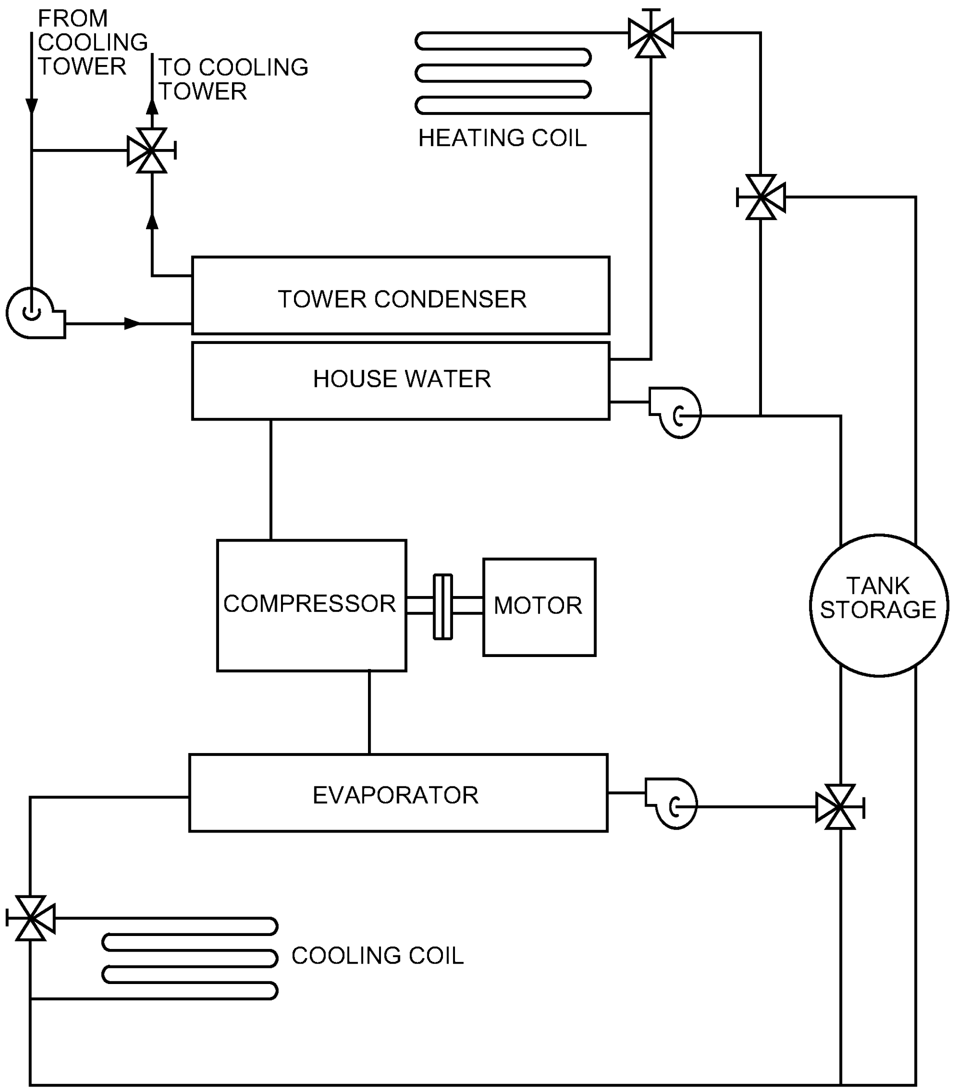

Fig. 8 Heat Transfer System with Storage Tank
(Figure 24, Chapter 9, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*)

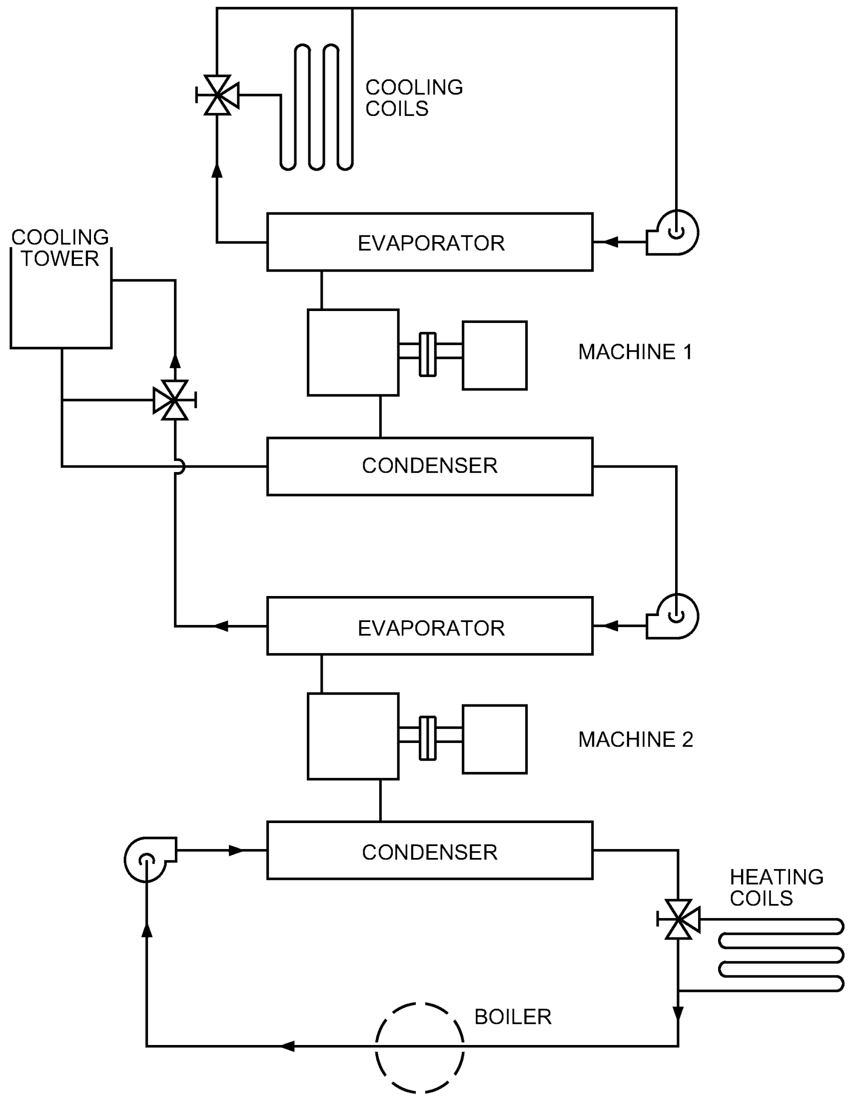

Fig. 9 Multistage (Cascade) Heat Transfer System
(Figure 25, Chapter 9, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*)

4 References

Amende, K.L., J.A. Keen, L.E. Catlin, M. Tosh, A.M. Sneed, and R.H. Howell. 2021. *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition.

ASHRAE. 2012. *ASHRAE Handbook—HVAC Systems and Equipment*.

ASHRAE. 2015. *Combined Heat and Power Design Guide*. Peachtree Corners, GA: ASHRAE.

ASHRAE. 2016. *ASHRAE Handbook—HVAC Systems and Equipment*.

ASHRAE. 2020. *ASHRAE Handbook—HVAC Systems and Equipment*.

Glazer, J. 2019. *ASHRAE Design Guide for Cool Thermal Storage*, 2nd ed. Peachtree Corners, GA: ASHRAE.

Scofield, M., N. DesChamps, and T.S. Weaver. 2016. Variable air volume system heat recovery economizer. *ASHRAE Journal*, May: pp. 34.

Taylor, S.T. 2014. How to design and control waterside economizers. *ASHRAE Journal,* May: pp. 30.

5 Problem

Problem 1 A 100,000 ft^2 building design has a design electrical load of 5 W/ft^2. A reciprocating natural gas engine cogeneration plant is to serve the building. The engine-generator is sized for the electrical load, with salvaged heat being used for heating and for driving a single-effect absorption chiller. The design heating load is 3,000,000 Btu/h. The design cooling load is 250 tons; the absorber requires 20,000 Btu/ton·h input.

Calculate hourly design operating costs for heating and cooling. Any shortfall in heating from recovered heat must be made up by a boiler. Any shortfall in cooling by the absorber with recovered heat must be made up by the boiler as input to the absorber.

Compare design operating costs with hourly design operating costs using conventional equipment (purchased electricity for the building and for cooling with an electric chiller at 1.0 kW/ton, purchased gas for a boiler for heating). Use $1.00 per therm, boiler efficiency of 80% for fuel cost, $0.10/kWh for purchased electricity cost.

SI Resources

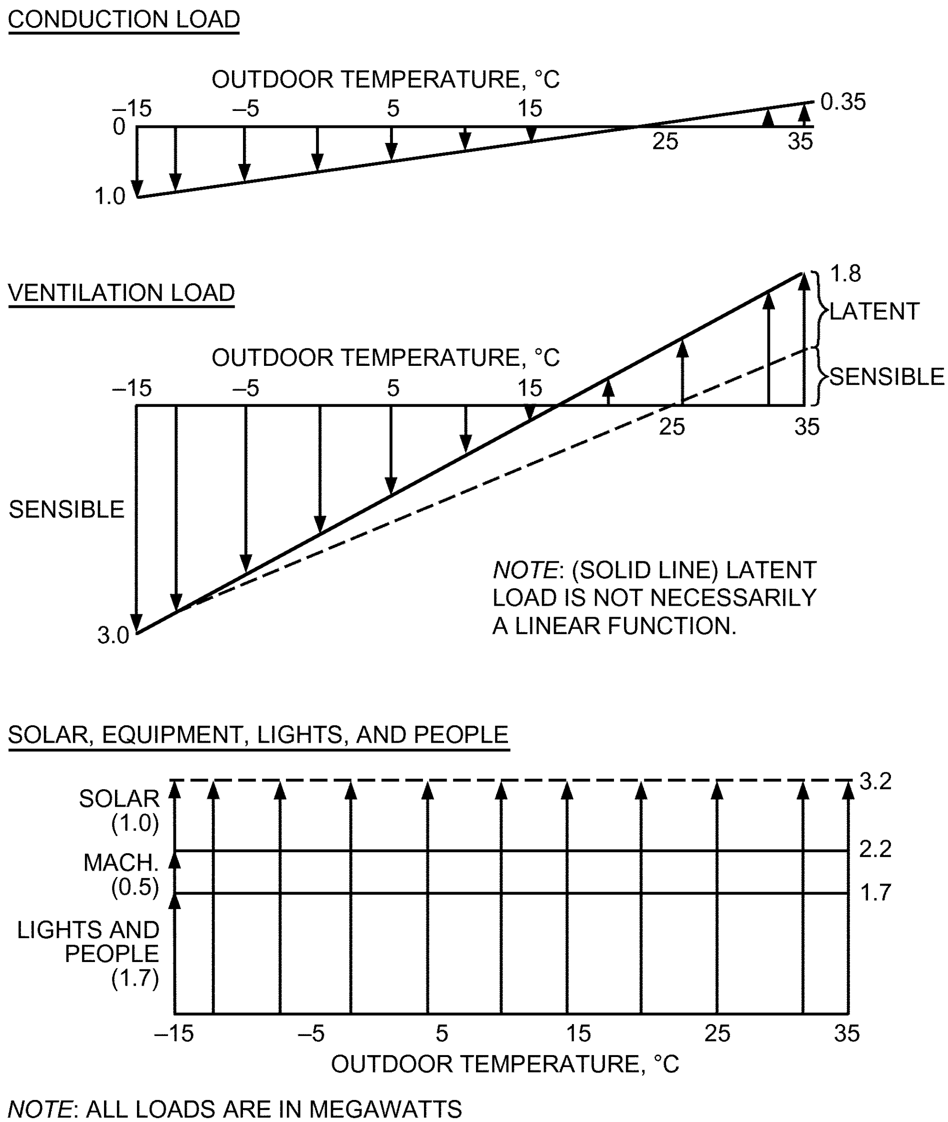

Fig. 2 Major Load Components (SI)
(Figure 34, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

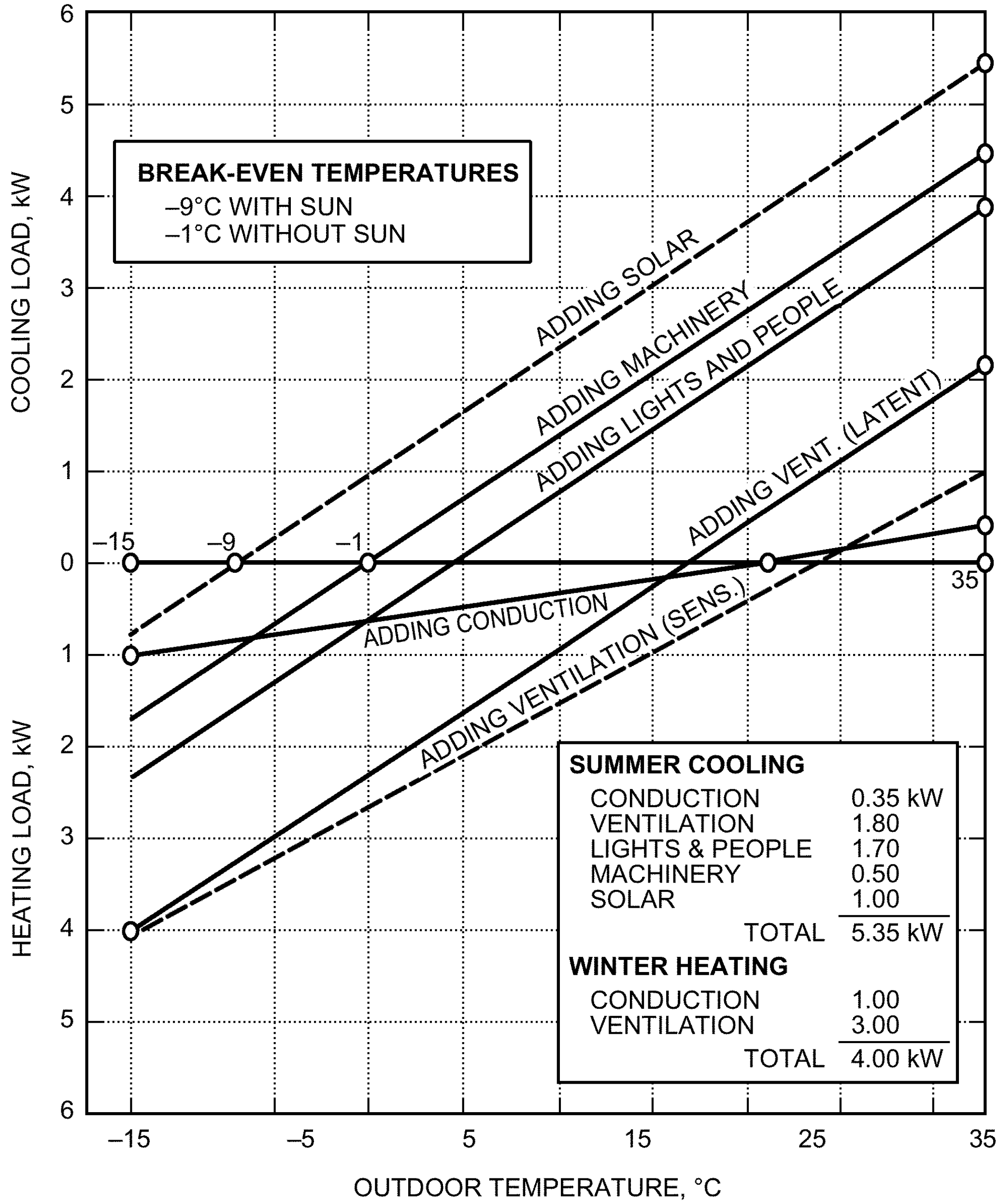

Fig. 3 Composite Plot of Loads in Figure 2 (SI)
(Figure 32, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

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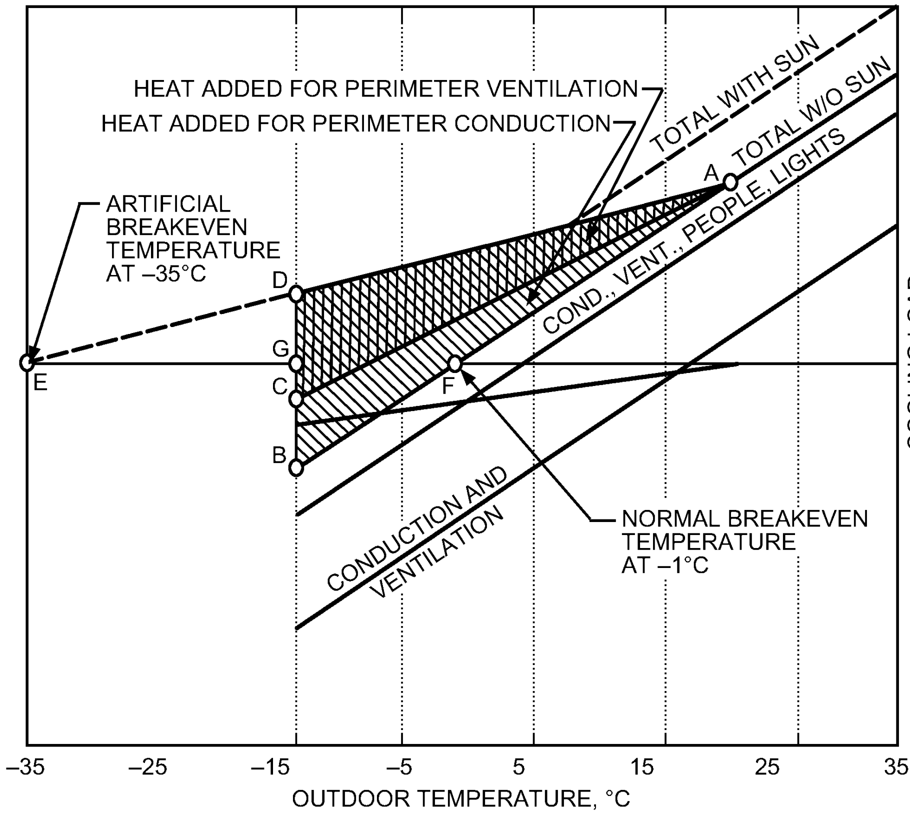

Fig. 4 Non-Heat Recovery System (SI)
(Figure 33, Chapter 9, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

SI Resources