CHAPTER

Utilities and Energy Efficient Design

3

KEY LEARNING OBJECTIVES

- How processes are heated and cooled
- The systems used for delivering steam, cooling water, and other site utilities
- Methods used for recovering process waste heat
- How to use the pinch design method to optimize process heat recovery
- How to design a heat-exchanger network
- How energy is managed in batch processes

3.1 INTRODUCTION

Very few chemical processes are carried out entirely at ambient temperature. Most require process streams to be heated or cooled to reach the desired operation temperature, to add or remove heats of reaction, mixing, adsorption, etc., to sterilize feed streams, or to cause vaporization or condensation. Gas and liquid streams are usually heated or cooled by indirect heat exchange with another fluid: either another process stream or a utility stream such as steam, hot oil, cooling water, or refrigerant. The design of heat exchange equipment for fluids is addressed in Chapter 19. Solids are usually heated and cooled by direct heat transfer, as described in Chapter 18. This chapter begins with a discussion of the different utilities that are used for heating, cooling, and supplying other needs such as power, water, and air to a process.

The consumption of energy is a significant cost in many processes. Energy costs can be reduced by recovering waste heat from hot process streams and by making use of the fuel value of waste streams. Section 3.4 discusses how to evaluate waste stream combustion as a source of process heat. Section 3.3 introduces other heat recovery approaches.

When it is economically attractive, heating and cooling are accomplished by heat recovery between process streams. The design of a network of heat exchangers for heat recovery can be a complex task if there are many hot and cold streams in a process. Pinch analysis, introduced in Section 3.5, is a systematic method for simplifying this problem.

The efficient use of energy in batch and cyclic processes is made more complicated by the sequential nature of process operations. Some approaches to energy efficient design of batch and cyclic processes are discussed in Section 3.6.

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3.2 UTILITIES

The word "utilities" is used for the ancillary services needed in the operation of any production process. These services are normally supplied from a central site facility, and include:

1. Electricity
2. Fuel for fired heaters
3. Fluids for process heating
   a. Steam
   b. Hot oil or specialized heat transfer fluids
4. Fluids for process cooling
   a. Cooling water
   b. Chilled water
   c. Refrigeration systems
5. Process water
   a. Water for general use
   b. Demineralized water
6. Compressed air
7. Inert-gas supplies (usually nitrogen)

Most plants are located on sites where the utilities are provided by the site infrastructure. The price charged for a utility is mainly determined by the operating cost of generating and transmitting the utility stream. Some companies also include a capital recovery charge in the utility cost, but if this is done then the offsite (OSBL) capital cost of projects must be reduced to avoid double counting and biasing the project capital-energy trade-off, leading to poor use of capital.

Some smaller plants purchase utilities "over the fence" from a supplier such as a larger site or a utility company, in which case the utility prices are set by contract and are typically pegged to the price of natural gas, fuel oil, or electricity.

The utility consumption of a process cannot be estimated accurately without completing the material and energy balances and carrying out a pinch analysis, as described in Section 3.5.6. The pinch analysis gives targets for process heat recovery and hence for the minimum requirements of hot and cold utilities. More detailed optimization then translates these targets into expected demands for fired heat, steam, electricity, cooling water, and refrigeration. In addition to the utilities required for heating and cooling, the process may also need process water and air for applications such as washing, stripping, and instrument air supply. Good overviews of methods for design and optimization of utility systems are given by Smith (2005) and Kemp (2007).

3.2.1 Electricity

The electricity demand of the process is mainly determined by the work required for pumping, compression, air coolers, and solids-handling operations, but also includes the power needed for instruments, lights, and other small users. The power required may be generated on site, but will more usually be purchased from the local supply company. Some plants generate their own electricity using a gas-turbine cogeneration plant with a heat recovery steam generator (waste-heat boiler) to raise steam (Figure 3.1). The overall thermal efficiency of such systems can be in the range 70% to 80%; compared with the 30% to 40% obtained from a conventional power station, where the

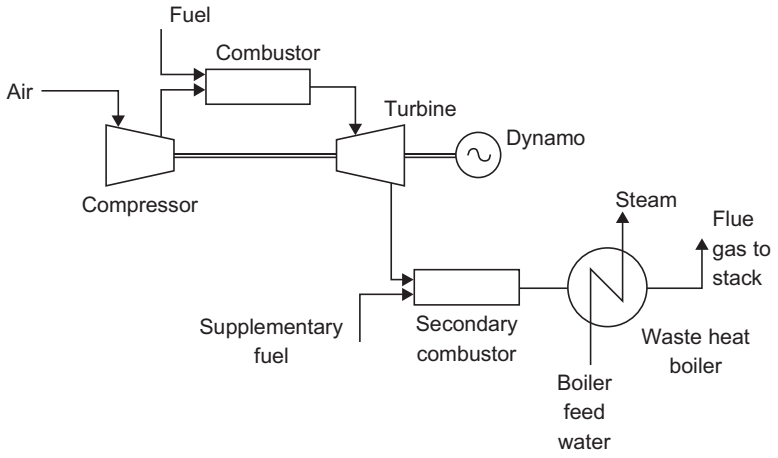


FIGURE 3.1

Gas-turbine-based cogeneration plant.

heat in the exhaust steam is wasted in the condenser. The cogeneration plant can be sized to meet or exceed the plant electricity requirement, depending on whether the export of electricity is an attractive use of capital. This "make or buy" scenario gives chemical producers strong leverage when negotiating electric power contracts and they are usually able to purchase electricity at or close to wholesale prices. Wholesale electricity prices vary regionally (see www.eia.gov for details), but are typically about $0.06/kWh in North America at the time of writing.

The voltage at which the supply is taken or generated will depend on the demand. In the United States, power is usually transmitted over long distances at 135, 220, 550, or 750 kV. Local substations step the power down to 35 to 69 kV for medium voltage transmission and then to 4 to 15 kV local distribution lines. Transformers at the plant are used to step down the power to the supply voltages used on site. Most motors and other process equipment run on 208 V three-phase power, while 120/240 V single-phase power is used for offices, labs, and control rooms.

On any site it is always worth considering driving large compressors and pumps with steam turbines instead of electric motors and using the exhaust steam for local process heating.

Electric power is rarely used for heating in large-scale chemical plants, although it is often used in smaller batch processes that handle nonflammable materials, such as biological processes. The main disadvantages of electrical heating for large-scale processes are:

- Heat from electricity is typically two to three times more expensive than heat from fuels, because of the thermodynamic inefficiency of power generation.
- Electric heating requires very high power draws that would substantially increase the electrical infrastructure costs of the site.
- Electric heating apparatus is expensive, requires high maintenance, and must comply with stringent safety requirements when used in areas where flammable materials may be present.
- Electric heaters are intrinsically less safe than steam systems. The maximum temperature that a steam heater can reach is the temperature of the steam. The maximum temperature of an electric

heater is determined by the temperature controller (which could fail) or by burn-out of the heating element. Electric heaters therefore have a higher probability of overheating.

Electric heating is more likely to be attractive in small-scale batch or cyclic processes, particularly when the cost of heating is a small fraction of overall process costs and when the process calls for rapid on-off heating.

A detailed account of the factors to be considered when designing electrical distribution systems for chemical process plants, and the equipment used (transformers, switch gear, and cables), is given by Silverman (1964). Requirements for electrical equipment used in hazardous (classified) locations are given in the National Electrical Code (NFPA 70), as described in Section 10.3.5.

3.2.2 Fired Heat

Fired heaters are used for process heating duties above the highest temperatures that can be reached using high pressure steam, typically about 250 ºC (482 ºF). Process streams may be heated directly in the furnace tubes, or indirectly using a hot oil circuit or heat transfer fluid, as described in Section 3.2.4. The design of fired heaters is described in Section 19.17. The cost of fired heat can be calculated from the price of the fuel fired. Most fired process heaters use natural gas as fuel, as it is cleaner burning than fuel oil and therefore easier to fit NO_x control systems and obtain permits. Natural gas also requires less maintenance of burners and fuel lines and natural gas burners can often co-fire process waste streams such as hydrogen, light organic compounds, or air saturated with organic compounds.

Natural gas and heating oil are traded as commodities and prices can be found at any online trading site or business news site (e.g., www.cnn.money.com). Historic prices for forecasting can be found in the *Oil and Gas Journal* or from the U.S. Energy Information Adminstration (www.eia.gov).

The fuel consumed in a fired heater can be estimated from the fired heater duty divided by the furnace efficiency. The furnace efficiency will typically be about 0.85 if both the radiant and convective sections are used (see Chapter 19) and about 0.6 if the process heating is in the radiant section only.

Example 3.1

Estimate the annual cost of providing heat to a process from a fired heater using natural gas as fuel if the process duty is 4 MW and the price of natural gas is $3.20 /MMBtu (million Btu).

Solution

If we assume that the fired heater uses both the radiant and convective sections then we can start by assuming a heater efficiency of 0.85, so

$$\text{Fuel required} = \text{Heater duty/heater efficiency} = 4/0.85 = 4.71\ \text{MW}$$

$$1\ \text{Btu/h} = 0.29307\ \text{W, so } 4.71\ \text{MW} = 4.71/0.29307 = 16.07\ \text{MMBtu/h}$$

Assuming 8000 operating hours per year, the total annual fuel consumption would be

$$\text{Annual fuel use} = 16.07 \times 8000 = 128.6 \times 10^3\ \text{MMBtu}$$

$$\text{Annual cost of fired heat} = 128.6 \times 10^3 \times 3.20 = \underline{\underline{\$411,400}}$$

Note that if we had decided to carry out all of the heating in the radiant section only, then the fuel required would have been 4/0.6 = 6.67 MW and the annual cost of heating would increase to $582,600 unless we could find some other process use for the heat available in the convective section of the heater.

3.2.3 Steam

Steam is the most widely-used heat source in most chemical plants. Steam has a number of advantages as a hot utility:

- The heat of condensation of steam is high, giving a high heat output per pound of utility at constant temperature (compared to other utilities such as hot oil and flue gas that release sensible heat over a broad temperature range).
- The temperature at which heat is released can be precisely controlled by controlling the pressure of the steam. This enables tight temperature control, which is important in many processes.
- Condensing steam has very high heat transfer coefficients, leading to cheaper heat exchangers.
- Steam is nontoxic, nonflammable, visible if it leaks externally, and inert to many (but not all) process fluids.

Most sites have a pipe network supplying steam at three or more pressure levels for different process uses. A typical steam system is illustrated in Figure 3.2. Boiler feed water at high pressure is preheated and fed to boilers where high pressure steam is raised and superheated above the dew point to allow for heat losses in the piping. Boiler feed water preheat can be accomplished using process waste heat or convective section heating in the boiler plant. High pressure (HP) steam is

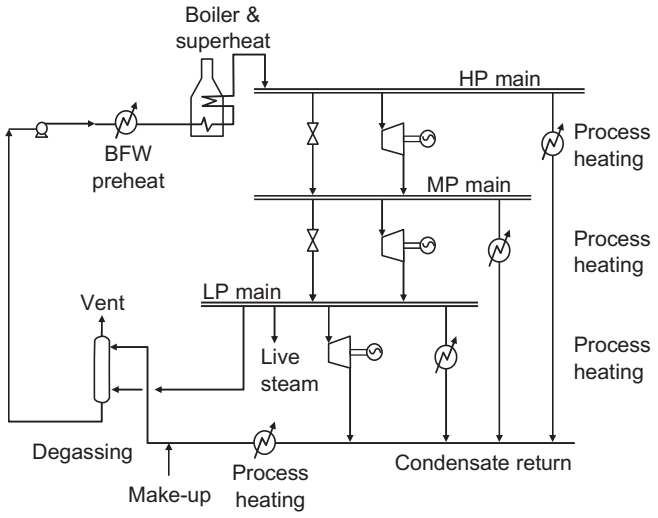


FIGURE 3.2

Steam system.

typically at about 40 bar, corresponding to a condensing temperature of 250 ºC, but every site is different. Some of the HP steam is used for process heating at high temperatures. The remainder of the HP steam is expanded either through let-down valves or steam turbines known as back-pressure turbines to form medium pressure (MP) steam. The pressure of the MP steam mains varies widely from site to site, but is typically about 20 bar, corresponding to a condensing temperature of 212 ºC. Medium pressure steam is used for intermediate temperature heating or expanded to form low pressure (LP) steam, typically at about 3 bar, condensing at 134 ºC. Some of the LP steam may be used for process heating if there are low-temperature heat requirements. Low pressure (or MP or HP) steam can also be expanded in condensing turbines to generate shaft work for process drives or electricity production. A small amount of LP steam is used to strip dissolved noncondensable gases such as air from the condensate and make-up water. Low pressure steam is also often used as "live steam" in the process, for example, as stripping vapor or for cleaning, purging, or sterilizing equipment.

When steam is condensed without coming into contact with process fluids, the hot condensate can be collected and returned to the boiler feed water system. Condensate can also sometimes be used as a low-temperature heat source if the process requires low-temperature heat.

The price of HP steam can be estimated from the cost of boiler feed water treatment, the price of fuel, and the boiler efficiency:

$$P_{HPS} = P_F \times \frac{dH_b}{\eta_B} + P_{BFW} \tag{3.1}$$

where P_{HPS} = price of high pressure steam ($/1000 lb, commonly written $/Mlb)
P_F = price of fuel ($/MMBtu)
dH_b = heating rate (MMBtu/Mlb steam)
η_B = boiler efficiency
P_{BFW} = price or cost of boiler feed water ($/Mlb)

Package boilers typically have efficiencies similar to fired heaters, in the range 0.8 to 0.9.

The heating rate should include boiler feed water preheat, the latent heat of vaporization, and the superheat specified.

The steam for process heating is usually generated in water-tube boilers, using the most economical fuel available.

The cost of boiler feed water includes allowances for water make-up, chemical treatment, and degassing, and is typically about twice the cost of raw water; see Section 3.2.7. If no information on the price of water is available, then 0.50 $/1000 lb can be used as an initial estimate. If the steam is condensed and the condensate is returned to the boiler feed water (which will normally be the case), then the price of steam should include a credit for the condensate. The condensate credit will often be close enough to the boiler feed water cost that the two terms cancel each other out and can be neglected.

The prices of medium and low pressure steam are usually discounted from the high pressure steam price, to allow for the shaft work credit that can be gained by expanding the steam through a turbine, and also to encourage process heat recovery by raising steam at intermediate levels and using low-grade heat when possible. Several methods of discounting are used. The most rational of these is to calculate the shaft work generated by expanding the steam between levels and price this

as equivalent to electricity (which could be generated by attaching the turbine to a dynamo or else would be needed to run a motor to replace the turbine if it is used as a driver). The value of the shaft work then sets the discount between steam at different levels. This is illustrated in the following example.

Example 3.2

A site has steam levels at 40 bar, 20 bar, and 6 bar. The price of fuel is $6/MMBtu and electricity costs $0.05/kWh. If the boiler efficiency is 0.8 and the steam turbine efficiency is 0.85, suggest prices for HP, MP, and LP steam.

Solution

The first step is to look up the steam conditions, enthalpies, and entropies in steam tables:

Steam level	HP	MP	LP
Pressure (bar)	40	20	6
Saturation temperature (°C)	250	212	159

The steam will be superheated above the saturation temperature to allow for heat losses in the pipe network. The following superheat temperatures were set to give an adequate margin above the saturation temperature for HP steam and also to give (roughly) the same specific entropy for each steam level. The actual superheat temperatures of MP and LP steam will be higher, due to the nonisentropic nature of the expansion.

Superheat temperature (°C)	400	300	160
Specific entropy, s_g, (kJ/kg.K)	6.769	6.768	6.761
Specific enthalpy, h_g, (kJ/kg)	3214	3025	2757

We can then calculate the difference in enthalpy between levels for isentropic expansion:

Isentropic delta enthalpy (kJ/kg)	189	268

Multiplying by the turbine efficiency gives the nonisentropic enthalpy of expansion:

Actual delta enthalpy (kJ/kg)	161	228

This can be converted to give the shaft work in customary units:

Shaft work (kWh/Mlb)	20.2	28.7

Multiplying by the price of electricity converts this into a shaft work credit:

Shaft work credit ($/Mlb)	1.01	1.44

The price of high pressure steam can be found from Equation 3.1, assuming that the boiler feed water cost is cancelled out by a condensate credit. The other prices can then be estimated by subtracting the shaft work credits.

Steam price ($/Mlb)	6.48	5.47	4.03

For quick estimates, this example can easily be coded into a spreadsheet and updated with the current prices of fuel and power. A sample steam costing spreadsheet is available in the online material at booksite.elsevier.com/Towler.

3.2.4 Hot Oil and Heat Transfer Fluids

Circulating systems of hot oil or specialized heat transfer fluids are often used as heat sources in situations where fired heat or steam are not suitable. Heat transfer fluids and mineral oils can be used over a temperature range from 50 ºC to 400 ºC. The upper temperature limit on use of hot oils is usually set by thermal decomposition of the oil, fouling, or coking of heat-exchange tubes. Some heat transfer fluids are designed to be vaporized and condensed in a similar manner to the steam system, though at lower pressures; however, vaporization of mineral oils is usually avoided, as less volatile components in the oil could accumulate and decompose, causing accelerated fouling.

The most common situation where a hot oil system is used is in plants that have many relatively small high-temperature heating requirements. Instead of building several small fired heaters, it can be more economical to supply heat to the process from circulating hot oil streams and build a single fired heater that heats the hot oil. Use of hot oil also reduces the risk of process streams being exposed to high tube-wall temperatures that might be experienced in a fired heater. Hot oil systems are often attractive when there is a high pressure differential between the process stream and HP steam and use of steam would entail using thicker tubes. Hot oil systems can sometimes be justified on safety grounds if the possibility of steam leakage into the process is very hazardous.

The most commonly used heat transfer fluids are mineral oils and Dowtherm A. Mineral oil systems usually require large flow rates of circulating liquid oil. When the oil is taken from a process stream, as is common in oil refining processes, then it is sometimes called a pump-around system. Dowtherm A is a mixture of 26.5 wt% diphenyl in diphenyl oxide. Dowtherm A is very thermally stable and is usually operated in a vaporization-condensation cycle similar to the steam system, although condensed liquid Dowtherm is sometimes used for intermediate temperature heating requirements. The design of Dowtherm systems and other proprietary heat transfer fluids are discussed in detail in Singh (1985) and Green and Perry (2007).

The cost of the initial charge of heat transfer fluid usually makes a negligible contribution to the overall cost of running a hot oil system. The main operating cost is the cost of providing heat to the hot oil in the fired heater or vaporizer. If a pumped liquid system is used then the pumping costs should also be evaluated. The costs of providing fired heat are discussed in Section 3.2.2. Hot oil heaters or vaporizers usually use both the radiant and convective sections of the heater and have heater efficiencies in the range 80% to 85%.

3.2.5 Cooling Water

When a process stream requires cooling at high temperature, various heat recovery techniques should be considered. These include transferring heat to a cooler process stream, raising steam, preheating boiler feed water, etc., as discussed in Section 3.3.

When heat must be rejected at lower temperatures, below about 120 ºC (248 ºF) (more strictly, below the pinch temperature), then a cold utility stream is needed. Cooling water is the most commonly used cold utility in the temperature range 120 ºC to 40 ºC, although air cooling is preferred in regions where water is expensive or the ambient humidity is too high for cooling water systems to operate effectively. The selection and design of air coolers are discussed in Section 19.16. If a process stream must be cooled to a temperature below 40 ºC, cooling water or air cooling would be

used down to a temperature in the range 40 ºC to 50 ºC, followed by chilled water or refrigeration down to the target temperature.

Natural and forced-draft cooling towers are generally used to provide the cooling water required on a site, unless water can be drawn from a convenient river or lake in sufficient quantity. Sea water, or brackish water, can be used at coastal sites and for offshore operations, but if used directly will require the use of more expensive materials of construction for heat exchangers (see Chapter 6). The minimum temperature that can be reached with cooling water depends on the local climate. Cooling towers work by evaporating part of the circulating water to ambient air, causing the remaining water to be chilled. If the ambient temperature and humidity are high, then a cooling water system will be less effective and air coolers or refrigeration would be used instead.

A schematic of a cooling water system is shown in Figure 3.3. Cooling water is pumped from the cooling tower to provide coolant for the various process cooling duties. Each process cooler is served in parallel and cooling water almost never flows to two coolers in series. The warmed water is returned to the cooling tower and cooled by partial evaporation. A purge stream known as a blowdown is removed upstream of the cooling tower, to prevent the accumulation of dissolved solids as water evaporates from the system. A make-up stream is added to compensate for evaporative losses, blowdown losses, and any other losses from the system. Small amounts of chemical additives are also usually fed into the cooling water to act as biocides and corrosion and fouling inhibitors.

The cooling tower consists of a means of providing high surface area for heat and mass transfer between the warm water and ambient air, and a means of causing air to flow countercurrent to the water. The surface area for contact is usually provided by flowing the water over wooden slats or high-voidage packing. The cooled water is then collected at the bottom of the tower. In most modern cooling towers the air flow is induced by fans that are placed above the packed section of the tower. For very large cooling loads natural-draft cooling towers are used, in which a large hyperbolic chimney is placed above the packed section to induce air flow. Some older plants use spray ponds instead of cooling towers.

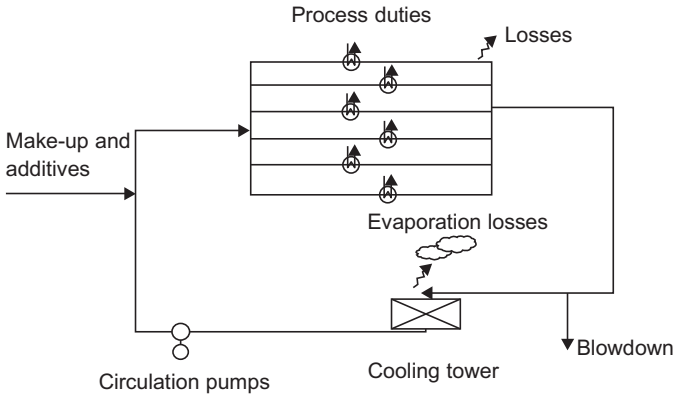


FIGURE 3.3

Schematic of cooling water system.

Cooling water systems can be designed using a psychrometric chart if the ambient conditions are known. A psychrometric chart is given in Figure 3.4. The cooling tower is usually designed so that it will operate effectively except under the hottest (or most humid) conditions that can be expected to occur no more than a few days each year.

The ambient temperature and humidity can be plotted on the psychrometric chart, allowing the inlet air wet bulb temperature to be determined. This is the coldest temperature that the cooling water could theoretically reach; however, in practice most cooling towers operate with a temperature approach to the air wet bulb temperature of at least 2.8 ºC (5 ºF). Adding the approach temperature to the inlet air wet bulb temperature, we can then mark the outlet water condition on the saturation curve. For example, if the hottest ambient condition for design purposes is a dry bulb temperature of 35 ºC (95 ºF) with 80% humidity, then we can mark this point on the psychrometric chart (point A) and read the wet bulb temperature as roughly 32 ºC (89.6 ºF). Adding a 2.8 ºC temperature approach would give a cold water temperature of about 35 ºC (95 ºF), which can then be marked on the saturation line (point B).

The inlet water condition, or cooling water return temperature, is found by optimizing the trade-off between water circulation costs and cooling tower cost. The difference between the cooling water supply (coldest) and return (hottest) temperatures is known as the *range* or *cooling range* of

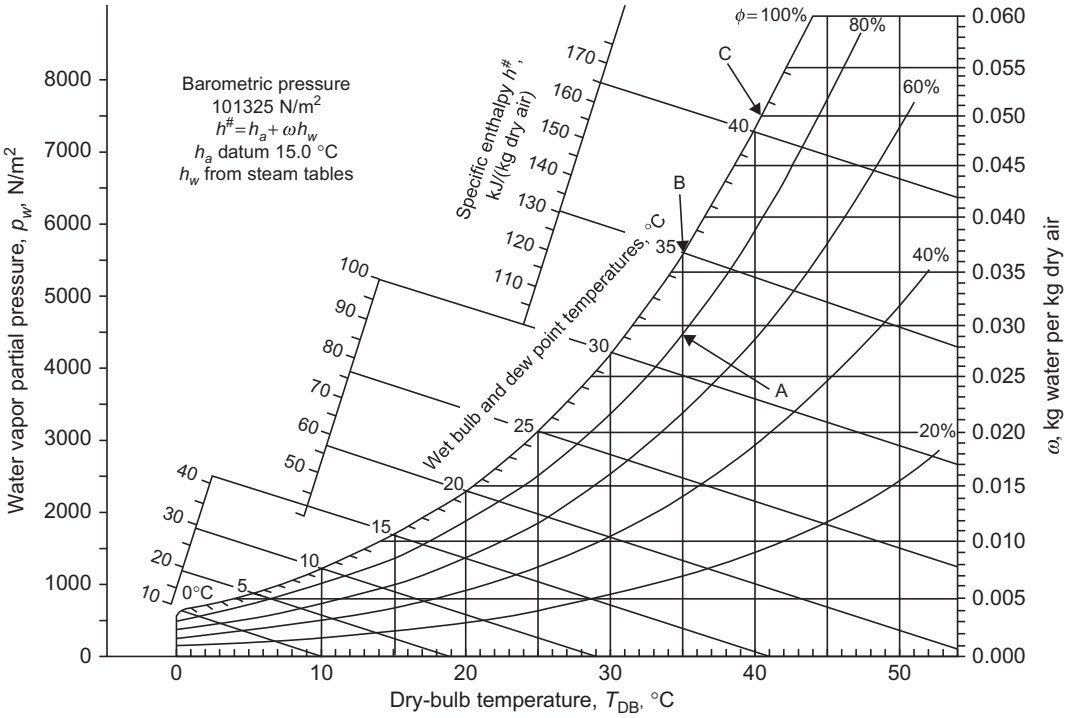


FIGURE 3.4

Psychrometric chart (adapted with permission from Balmer (2010)).

the cooling tower. As the cooling range is increased, the cost of the cooling tower is increased, but the water flow rate that must be circulated decreases, and hence the pumping cost decreases. Note that since most of the cooling is accomplished by evaporation of water rather than transfer of sensible heat to the air, the evaporative losses do not vary much with the cooling range. Most cooling towers are operated with a cooling range between 5 °F and 20 °F (2.8 °C to 11.1 °C). A typical initial design point would be to assume a cooling water return temperature about 10 °F (5.5 °C) hotter than the cold water temperature. In the example above, this would give a cooling water return temperature of 40.5 °C (105 °F), which can also be marked on the psychrometric chart (point C). The difference in air humidity across the cooling tower can now be read from the right-hand axis as the difference in moisture loadings between the inlet air (point A) and the outlet air (point C). The cooling tower design can then be completed by determining the cooling load of the water from an energy balance and hence determining the flow rate of air that is needed based on the change in air humidity between ambient air and the air exit condition. The exit air is assumed to have a dry bulb temperature equal to the cooling water return temperature, and the minimum air flow will be obtained when the air leaves saturated with moisture. Examples of the detailed design of cooling towers are given in Green and Perry (2007).

When carrying out the detailed design of a cooling tower it is important to check that the cooling system has sufficient capacity to meet the site cooling needs over a range of ambient conditions. In practice, multiple cooling water pumps are usually used so that a wide range of cooling water flow rates can be achieved. When new capacity is added to an existing site, the limit on the cooling system is usually the capacity of the cooling tower. If the cooling tower fans cannot be upgraded to meet the increased cooling duty, additional cooling towers must be added. In such cases, it is often cheaper to install air coolers for the new process rather than upgrading the cooling water system.

The cost of providing cooling water is mainly determined by the cost of electric power. Cooling water systems use power for pumping the cooling water through the system and for running fans (if installed) in the cooling towers. They also have costs for water make-up and chemical treatment. The power used in a typical recirculating cooling water system is usually between 1 and 2 kWh/1000 gal of circulating water. The costs of water make-up and chemical treatment usually add about $0.02/1000 gal.

3.2.6 Refrigeration

Refrigeration is needed for processes that require temperatures below those that can be economically obtained with cooling water, i.e., below about 40 °C. For temperatures down to around 10 °C, chilled water can be used. For lower temperatures, down to −30 °C, salt brines (NaCl and $CaCl_2$) are sometimes used to distribute the "refrigeration" around the site from a central refrigeration machine. Large refrigeration duties are usually supplied by a standalone packaged refrigeration system.

Vapor compression refrigeration machines are normally used, as illustrated in Figure 3.5. The working fluid (refrigerant) is compressed as a vapor, and then cooled and condensed at high pressure, allowing heat rejection at high temperature in an exchanger known as a condenser. Heat is usually rejected to a coolant such as cooling water or ambient air. The liquid refrigerant is then expanded across a valve to a lower pressure, where it is vaporized in an exchanger known as an evaporator, taking up heat at low temperature. The vapor is then returned to the compressor, completing the cycle.

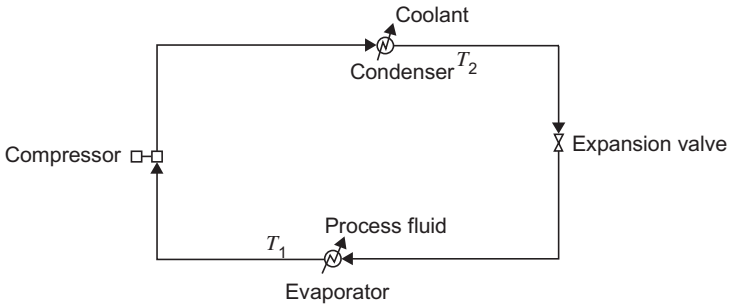


FIGURE 3.5

Simple refrigeration cycle.

The working fluid for a refrigeration system must satisfy a broad range of requirements. It should have a boiling point that is colder than the temperature that must be reached in the process at a pressure that is above atmospheric pressure (to prevent leaks into the system). It should have a high latent heat of evaporation, to reduce refrigerant flow rate. The system should operate well below the critical temperature and pressure of the refrigerant, and the condenser pressure should not be too high or the cost will be excessive. The freezing temperature of the refrigerant must be well below the minimum operating temperature of the system. The refrigerant should also be nontoxic, nonflammable, and have minimal environmental impact.

A wide range of materials have been developed for use as refrigerants, most of which are halogenated hydrocarbons. In some situations ammonia, nitrogen, and carbon dioxide are used. Cryogenic gas separation processes usually use the process fluids as working fluid; for example, ethylene and propylene refrigeration cycles are used in ethylene plants.

Refrigeration systems use power to compress the refrigerant. The power can be estimated using the cooling duty and the refrigerator coefficient of performance (*COP*).

$$COP = \frac{\text{Refrigeration produced (Btu/hr or MW)}}{\text{Shaft work used (Btu/hr or MW)}} \tag{3.2}$$

The *COP* is a strong function of the temperature range over which the refrigeration cycle operates. For an ideal refrigeration cycle (a reverse Carnot cycle), the *COP* is

$$COP = \frac{T_e}{(T_c - T_e)} \tag{3.3}$$

where T_e = evaporator absolute temperature (K)
T_c = condenser absolute temperature (K)

The *COP* of real refrigeration cycles is always less than the Carnot efficiency. It is usually about 0.6 times the Carnot efficiency for a simple refrigeration cycle, but can be as high as 0.9 times the Carnot efficiency if complex cycles are used. If the temperature range is too large then it may be more economical to use a cascaded refrigeration system, in which a low-temperature cycle rejects heat to a higher-temperature cycle that rejects heat to cooling water or ambient air. Good

overviews of refrigeration cycle design are given by Dincer (2003), Stoecker (1998), and Trott and Welch (1999).

The operating cost of a refrigeration system can be determined from the power consumption and the price of power. Refrigeration systems are usually purchased as packaged modular plants and the capital cost can be estimated using commercial cost estimating software as described in Section 7.10. An approximate correlation for the capital cost of packaged refrigeration systems is also given in Table 7.2.

Example 3.3

Estimate the annual operating cost of providing refrigeration to a condenser with duty 1.2 MW operating at −5 °C. The refrigeration cycle rejects heat to cooling water that is available at 40 °C, and has an efficiency of 80% of the Carnot cycle efficiency. The plant operates for 8000 hours per year and electricity costs $0.06/kWh.

Solution

The refrigeration cycle needs to operate with an evaporator temperature below −5 °C, say at −10 °C or 263 K. The condenser must operate above 40 °C, say at 45 °C (318 K).

For this temperature range the Carnot cycle efficiency is

$$COP = \frac{T_e}{(T_c - T_e)} = \frac{263}{318 - 263} = 4.78 \tag{3.3}$$

If the cycle is 80% efficient then the actual coefficient of performance = $4.78 \times 0.8 = 3.83$

The shaft work needed to supply 1.2 MW of cooling is given by

$$\text{Shaft work required} = \frac{\text{Cooling duty}}{COP} = \frac{1.2}{3.83} = 0.313 \text{ MW}$$

The annual operating cost is then = 313 kW × 8000 h/y × 0.06 $/kWh = 150,000 $/y

3.2.7 Water

The water required for general purposes on a site will usually be taken from the local mains supply, unless a cheaper source of suitable quality water is available from a river, lake, or well. Raw water is brought in to make up for losses in the steam and cooling water systems and is also treated to generate demineralized and deionized water for process use. Water is also used for process cleaning operations and to supply fire hydrants.

The price of water varies strongly by location, depending on fresh water availability. Water prices are often set by local government bodies and often include a charge for waste water rejection. This charge is usually applied on the basis of the water consumed by the plant, regardless of whether that water is actually rejected as a liquid (as opposed to being lost as vapor or incorporated into a product by reaction). A very rough estimate of water costs can be made by assuming $2 per 1000 gal ($0.5 per metric ton).

Demineralized Water

Demineralized water, from which all the minerals have been removed by ion-exchange, is used where pure water is needed for process use, and as boiler feed water. Mixed and multiple-bed ion-exchange units are used; one resin converting the cations to hydrogen and the other removing the anions. Water with less than 1 part per million of dissolved solids can be produced. The design of ion exchange units is discussed in Section 16.5.5. Demineralized water typically costs about double the price of raw water, but this obviously varies strongly with the mineral content of the water and the disposal cost of blowdown from the demineralization system. A correlation for the cost of a water ion exchange plant is given in Table 7.2.

3.2.8 Compressed Air

Compressed air is needed for general use, for oxidation reactions, air strippers, aerobic fermentation processes, and for pneumatic control actuators that are used for plant control. Air is normally distributed at a mains pressure of 6 bar (100 psig), but large process air requirements are typically met with standalone air blowers or compressors. Rotary and reciprocating single-stage or two-stage compressors are used to supply utility and instrument air. Instrument air must be dry and clean (free from oil). Air is usually dried by passing it over a packed bed of molecular sieve adsorbent. For most applications, the adsorbent is periodically regenerated using a temperature-swing cycle. Temperature swing adsorption (TSA) is discussed in more detail in Section 16.2.1.

Air at 1 atmosphere pressure is freely available in most chemical plants. Compressed air can be priced based on the power needed for compression (see Section 20.6). Drying the air, for example for instrument air, typically adds about \$0.005 per standard m^3 (\$0.14/1000 scf).

Cooling Air

Ambient air is used as a coolant in many process operations; for example, air cooled heat exchangers, cooling towers, solids coolers, and prilling towers. If the air flow is caused by natural draft then the cooling air is free, but the air velocity will generally be low, leading to high equipment cost. Fans or blowers are commonly used to ensure higher air velocities and reduce equipment costs. The cost of providing cooling air is then the cost of operating the fan, which can be determined from the fan power consumption. Cooling fans typically operate with very high flow rates and very low pressure drop, on the order of a few inches of water. The design of a cooling fan is illustrated in the discussion of air cooled heat exchangers in Section 19.16.

3.2.9 Nitrogen

Where a large quantity of inert gas is required for the inert blanketing of tanks and for purging (see Chapter 10) this will usually be supplied from a central facility. Nitrogen is normally used, and can be manufactured on site in an air liquefaction plant, or purchased as liquid in tankers.

Nitrogen and oxygen are usually purchased from one of the industrial gas companies via pipeline or a small dedicated "over-the-fence" plant. The price varies depending on local power costs, but is typically in the range \$0.01 to \$0.03 per lb for large facilities.

3.3 ENERGY RECOVERY

Process streams at high pressure or temperature contain energy that can be usefully recovered. Whether it is economical to recover the energy content of a particular stream depends on the value of the energy that can be usefully extracted and the cost of recovery. The value of the energy is related to the marginal cost of energy at the site. The cost of recovery will be the capital and operating cost of any additional equipment required. If the savings exceed the total annualized cost, including capital charges, then the energy recovery will usually be worthwhile. Maintenance costs should be included in the annualized cost (see Chapter 9).

Some processes, such as air separation, depend on efficient energy recovery for economic operation, and in all processes the efficient use of energy will reduce product cost.

When setting up process simulation models, the design engineer needs to pay careful attention to operations that have an impact on the energy balance and heat use within the process. Some common problems to watch out for include:

1. Avoid mixing streams at very different temperatures. This usually represents a loss of heat (or cooling) that could be better used in the process.
2. Avoid mixing streams at different pressures. The mixed stream will be at the lowest pressure of the feed streams. The higher pressure streams will undergo cooling as a result of adiabatic expansion. This may lead to increased heating or cooling requirements or lost potential to recover shaft work during the expansion.
3. Segment heat exchangers to avoid internal pinches. This is particularly necessary for exchangers where there is a phase change. When a liquid is heated, boiled, and superheated, the variation of stream temperature with enthalpy added looks like Figure 3.6. Liquid is heated to the boiling point (A–B), then the heat of vaporization is added (B–C) and the vapor is superheated (C–D). This is a different temperature-enthalpy profile than a straight line between the initial and final states (A–D). If the stream in Figure 3.6 were matched against a heat source that had a temperature profile like line E-F in Figure 3.7, then the exchanger would appear feasible based on the inlet and outlet temperatures, but would in fact be infeasible because of the cross-over of

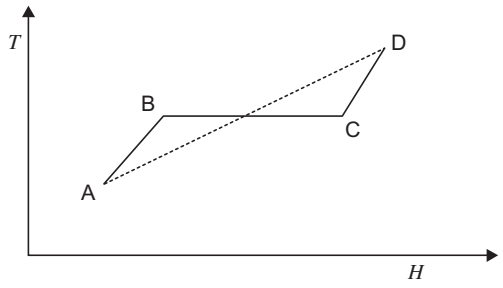


FIGURE 3.6
Temperature-enthalpy profile for a stream that is vaporized and superheated.

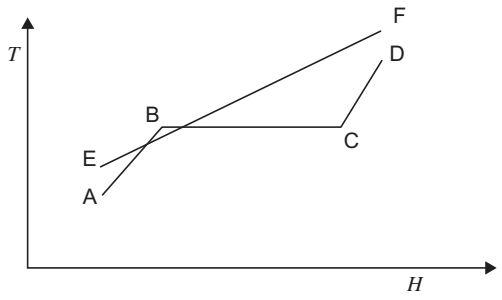


FIGURE 3.7
Heat transfer to a stream that is vaporized and superheated.

the temperature profiles at B. A simple way to avoid this problem is to break up the preheat, boiling, and superheat into three exchangers in the simulation model, even if they will be carried out in a single piece of equipment in the final design. The same problem also occurs with condensers that incorporate desuperheat and subcooling.

4. Check for heat of mixing. This is important whenever acids or bases are mixed with water. If the heat of mixing is large, two or more stages of mixing with intercoolers may be needed. If a large heat of mixing is expected, but is not predicted by the model, then check that the thermodynamic model includes heat of mixing effects.
5. Remember to allow for process inefficiency and design margins. For example, when sizing a fired heater, if process heating is carried out in the radiant section only, the heating duty calculated in the simulation is only 60% of the total furnace duty (see Sections 3.2.2 and 19.17). The operating duty will then be the process duty divided by 0.6. The design duty must be increased further by a suitable design factor, say 10%. The design duty of the fired heater is then 1.1/0.6 = 1.83 times the process duty calculated in the simulation.

Some of the techniques used for energy recovery in chemical process plants are described briefly in the following sections. The references cited give fuller details of each technique. Miller (1968) gives a comprehensive review of process energy systems, including heat exchange and power recovery from high-pressure fluid streams. Kenney (1984) reviews the application of thermodynamic principles to energy recovery in the process industries. Kemp (2007) gives a detailed description of pinch analysis and several other methods for heat recovery.

3.3.1 Heat Exchange

The most common energy-recovery technique is to use the heat in a high-temperature process stream to heat a colder stream. This saves part or all of the cost of heating the cold stream, as well as part or all of the cost of cooling the hot stream. Conventional shell and tube exchangers are normally used. The cost of the heat exchange surface may be increased relative to using a hot utility as heat source, due to the reduced temperature driving forces, or decreased, due to needing fewer exchangers. The cost of recovery will be reduced if the streams are located conveniently close within the plant.

The amount of energy that can be recovered depends on the temperature, flow, heat capacity, and temperature change possible, in each stream. A reasonable temperature driving force must be maintained to keep the exchanger area to a practical size. The most efficient exchanger will be the one in which the shell and tube flows are truly countercurrent. Multiple tube-pass exchangers are usually used for practical reasons. With multiple tube passes the flow is part countercurrent and part cocurrent and temperature crosses can occur, which reduce the efficiency of heat recovery (see Chapter 19). In cryogenic processes, where heat recovery is critical to process efficiency, brazed or welded plate exchangers are used to obtain true countercurrent performance and very low temperature approaches on the order of a few degrees Celsius are common.

The hot process streams leaving a reactor or a distillation column are frequently used to preheat the feed streams ("feed-effluent" or "feed-bottoms" exchangers).

In an industrial process there will be many hot and cold streams and there will be an optimum arrangement of the streams for energy recovery by heat exchange. The problem of synthesizing a

network of heat exchangers has been the subject of much research and is covered in more detail in Section 3.5.

3.3.2 Waste-heat Boilers

If the process streams are at a sufficiently high temperature and there are no attractive options for process-to-process heat transfer, then the heat recovered can be used to generate steam.

Waste-heat boilers are often used to recover heat from furnace flue gases and the process gas streams from high-temperature reactors. The pressure, and superheat temperature, of the steam generated depend on the temperature of the hot stream and the approach temperature permissible at the boiler exit. As with any heat-transfer equipment, the area required increases as the mean temperature driving force (log mean ΔT) is reduced. The permissible exit temperature may also be limited by process considerations. If the gas stream contains water vapor and soluble corrosive gases, such as HCl or SO_2, the exit gas temperature must be kept above the dew point.

Hinchley (1975) discusses the design and operation of waste-heat boilers for chemical plants. Both fire-tube and water-tube boilers are used. A typical arrangement of a water-tube boiler on a reformer furnace is shown in Figure 3.8 and a fire-tube boiler in Figure 3.9.

The application of a waste-heat boiler to recover energy from the reactor exit streams in a nitric acid plant is shown in Figure 3.10. The selection and operation of waste-heat boilers for industrial furnaces is discussed by Dryden (1975).

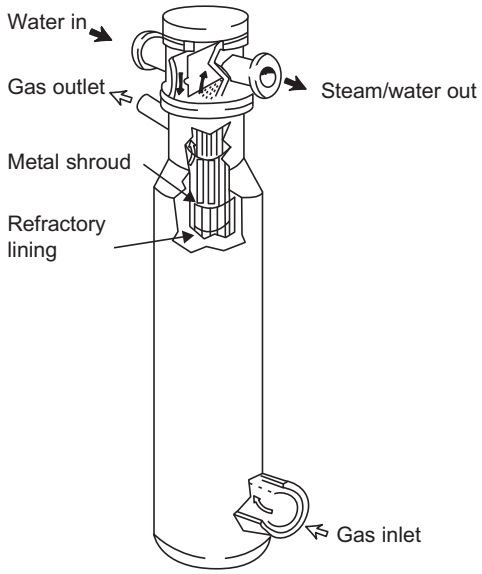


FIGURE 3.8

Reformed gas waste-heat boiler arrangement of vertical U-tube water-tube boiler. (Reprinted by permission of the Council of the Institution of Mechanical Engineers from the Proceedings of the Conference on Energy Recovery in the Process Industries, London, 1975.)

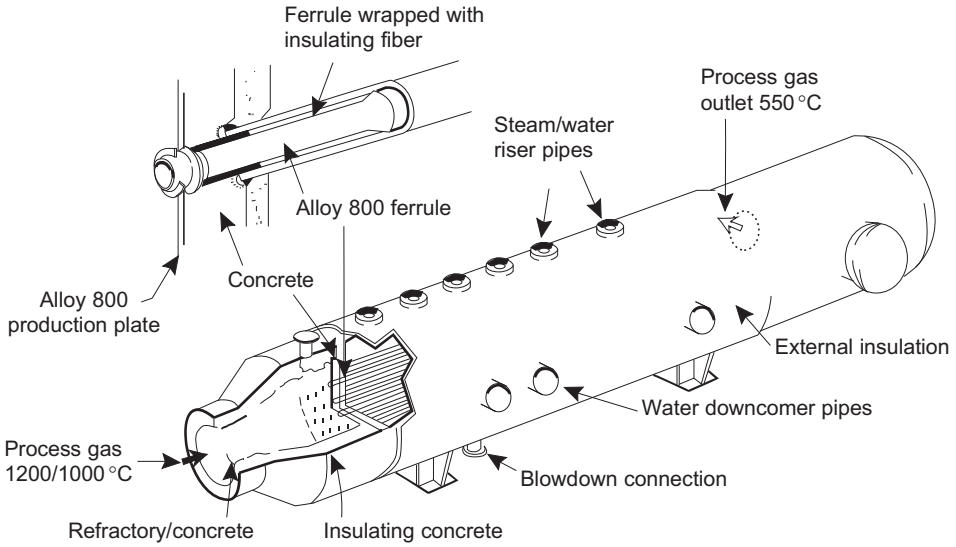
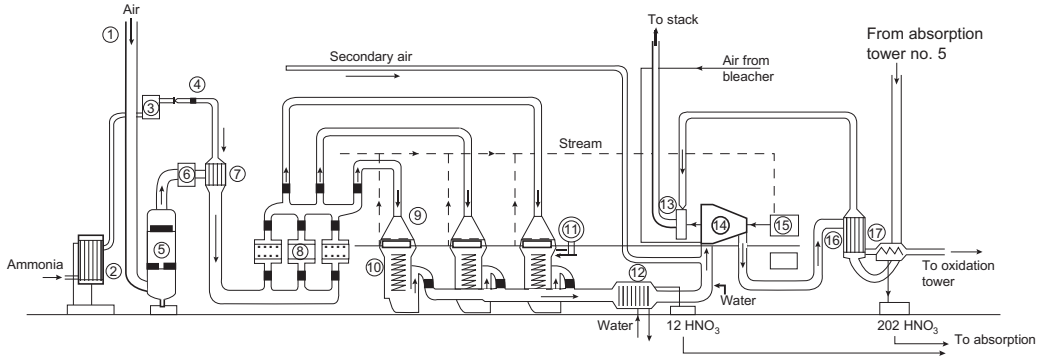


FIGURE 3.9

Reformed gas waste-heat boiler, principal features of typical natural circulation fire-tube boilers. (Reprinted by permission of the Council of the Institution of Mechanical Engineers from the Proceedings of the Conference on Energy Recovery in the Process Industries, London, 1975.)



1. Air entry
2. Ammonia vaporiser
3. Ammonia filter
4. Control valves
5. Air-scrubbing tower
6. Air preheater
7. Gas mixer
8. Gas filters
9. Converters
10. Lamont boilers
11. Steam drum
12. Gas cooler No. 1
13. Exhaust turbine
14. Compressor
15. Steam turbine
16. Heat exchanger
17. Gas cooler No. 2

(From nitric acid manufacture, Miles (1961), with permission)

FIGURE 3.10

Connections of a nitric acid plant, intermediate pressure type.

3.3.3 High-temperature Reactors

If a reaction is highly exothermic, cooling will be needed. If the reactor temperature is high enough, the heat removed can be used to generate steam. The lowest steam pressure normally used in the process industries is about 2.7 bar (25 psig), so any reactor with a temperature above 150 °C is a potential steam generator. Steam is preferentially generated at as high a pressure as possible, as high pressure steam is more valuable than low pressure steam (see Section 3.2.3). If the steam production exceeds the site steam requirements, some steam can be fed to condensing turbines to produce electricity to meet site power needs.

Three systems are used:

1. Figure 3.11(a). An arrangement similar to a conventional water-tube boiler. Steam is generated in cooling pipes within the reactor and separated in a steam drum.
2. Figure 3.11(b). Similar to the first arrangement but with the water kept at high pressure to prevent vaporization. The high-pressure water is flashed to steam at lower pressure in a flash drum. This system would give more responsive control of the reactor temperature.
3. Figure 3.11(c). In this system a heat-transfer fluid, such as Dowtherm A (see Section 3.2.4 and Singh (1985) for details of heat-transfer fluids), is used to avoid the need for high-pressure tubes. The steam is raised in an external boiler.

3.3.4 High-pressure Process Streams

Where high-pressure gas or liquid process streams are throttled to lower pressures, energy can be recovered by carrying out the expansion in a suitable turbine.

Gas Streams

The economic operation of processes that involve the compression and expansion of large quantities of gases, such as ammonia synthesis, nitric acid production, and air separation, depends on the efficient recovery of the energy of compression. The energy recovered by expansion is often used to drive the compressors directly, as shown in Figure 3.10. If the gas contains condensable components, it may be advisable to consider heating the gas by heat exchange with a higher temperature

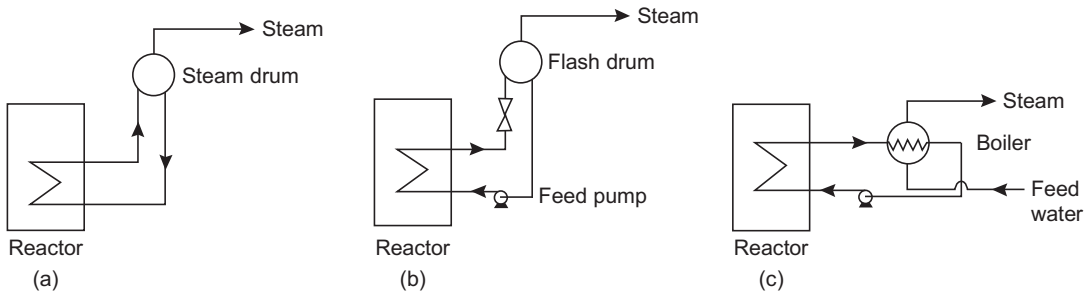


FIGURE 3.11

Steam generation.

process stream before expansion. The gas can then be expanded to a lower pressure without condensation and the power generated increased.

The process gases do not have to be at a particularly high pressure for expansion to be economical if the gas flow rate is high. For example, Luckenbach (1978) in U.S. patent 4,081,508 describes a process for recovering power from the off-gas from a fluid catalytic cracking process by expansion from about 2 to 3 bar (15 to 25 psig) down to just over atmospheric pressure (1.5 to 2 psig).

The energy recoverable from the expansion of a gas can be estimated by assuming polytropic expansion; see Section 20.6.3 and Example 20.4. The design of turboexpanders for the process industries is discussed by Bloch *et al.* (1982).

Liquid Streams

As liquids are essentially incompressible, less energy is stored in a compressed liquid than a gas; however, it is often worth considering power recovery from high-pressure liquid streams (>15 bar), as the equipment required is relatively simple and inexpensive. Centrifugal pumps are used as expanders and are often coupled directly to other pumps. The design, operation, and cost of energy recovery from high-pressure liquid streams is discussed by Jenett (1968), Chada (1984), and Buse (1981).

3.3.5 Heat Pumps

A heat pump is a device for raising low-grade heat to a temperature at which the heat can be used. It pumps the heat from a low temperature source to the higher temperature sink, using a small amount of energy relative to the heat energy recovered. A heat pump is essentially the same as a refrigeration cycle (Section 3.2.6 and Figure 3.5), but the objective is to deliver heat to the process in the condensation step of the cycle, as well as (or instead of) removing heat in the evaporation step.

Heat pumps are increasingly finding applications in the process industries. A typical application is the use of the low-grade heat from the condenser of a distillation column to provide heat for the reboiler; see Barnwell and Morris (1982) and Meili (1990). Heat pumps are also used with dryers; heat is abstracted from the exhaust air and used to preheat the incoming air.

Details of the thermodynamic cycles used for heat pumps can be found in most textbooks on engineering thermodynamics, and in Reay and MacMichael (1988). In the process industries, heat pumps operating on the mechanical vapor compression cycle are normally used. A vapor compression heat pump applied to a distillation column is shown in Figure 3.12(a). The working fluid, usually a commercial refrigerant, is fed to the reboiler as a vapor at high pressure and condenses, giving up heat to vaporize the process fluid. The liquid refrigerant from the reboiler is then expanded over a throttle valve and the resulting wet vapor is fed to the column condenser. In the condenser the wet refrigerant is dried, taking heat from the condensing process vapor. The refrigerant vapor is then compressed and recycled to the reboiler, completing the working cycle.

If the conditions are suitable, the process fluid can be used as the working fluid for the heat pump. This arrangement is shown in Figure 3.12(b). The hot process liquid at high pressure is expanded over the throttle valve and fed to the condenser, to provide cooling to condense the vapor from the column. The vapor from the condenser is compressed and returned to the base of the column. In an alternative arrangement, the process vapor is taken from the top of the column, compressed, and fed to the reboiler to provide heating.

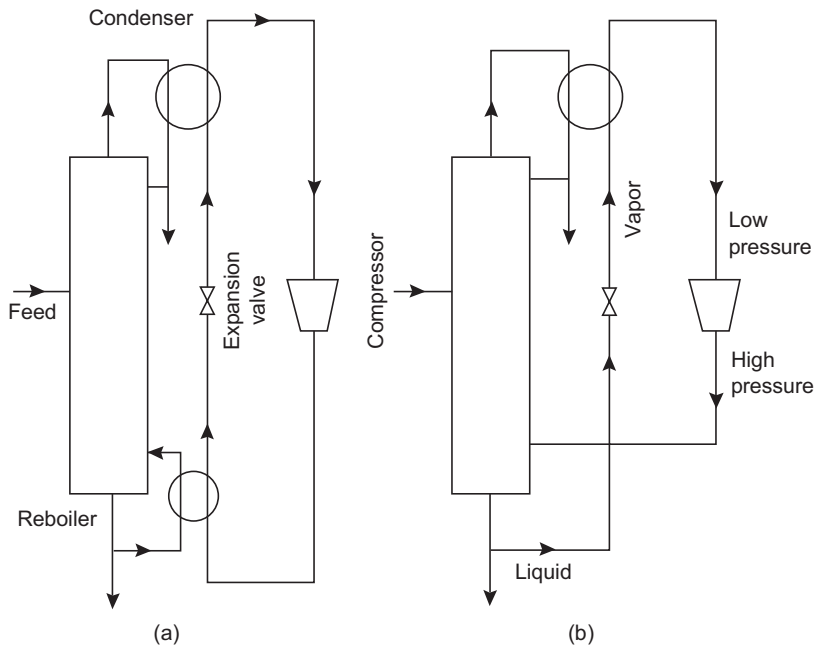


FIGURE 3.12

Distillation column with heat pump: (*a*) separate refrigerant circuit; (*b*) using column fluid as the refrigerant.

The "efficiency" of a heat pump is measured by the heat pump coefficient of performance, COP_h:

$$COP_h = \frac{\text{energy delivered at higher temperature}}{\text{energy input to the compressor}} \tag{3.4}$$

The COP_h depends principally on the working temperatures. Heat pumps are more efficient (higher COP_h) when operated over a narrow temperature range. They are thus most often encountered on distillation columns that separate close-boiling compounds. Note that the COP_h of a heat pump is not the same as the COP of a refrigeration cycle (Section 3.2.6).

The economics of the application of heat pumps in the process industries is discussed by Holland and Devotta (1986). Details of the application of heat pumps in a wide range of industries are given by Moser and Schnitzer (1985).

3.4 WASTE STREAM COMBUSTION

Process waste products that contain significant quantities of combustible material can be used as low-grade fuels, for raising steam or direct process heating. Their use will only be economic if the intrinsic value of the fuel justifies the cost of special burners and other equipment needed to burn the waste. If the combustible content of the waste is too low to support combustion, the waste must be supplemented with higher calorific value primary fuels.

3.4.1 Reactor Off-gases

Reactor off-gases (vent gas) and recycle stream purges are often of high enough calorific value to be used as fuels. Vent gases will typically be saturated with organic compounds such as solvents and high volatility feed compounds. The calorific value of a gas can be calculated from the heats of combustion of its constituents; the method is illustrated in Example 3.4.

Other factors which, together with the calorific value, determine the economic value of an off-gas as a fuel are the quantity available and the continuity of supply. Waste gases are best used for steam raising, rather than for direct process heating, as this decouples the source from the use and gives greater flexibility.

Example 3.4: Calculation of Waste-Gas Calorific Value

The typical vent-gas analysis from the recycle stream in an oxyhydrochlorination process for the production of dichloroethane (DCE) (British patent BP 1,524,449) is given below, percentages on volume basis.

$$O_2\ 7.96,\quad CO_2 + N_2\ 87.6,\quad CO\ 1.79,\quad C_2H_4\ 1.99,\quad C_2H_6\ 0.1,\quad DCE\ 0.54$$

Estimate the vent-gas calorific value.

Solution

Component calorific values, from Perry and Chilton (1973):

$$\begin{aligned} CO\ 67.6\ \text{kcal/mol} &= 283\ \text{kJ/mol} \\ C_2H_4\ 372.8 &= 1560.9 \\ C_2H_6\ 337.2 &= 1411.9 \end{aligned}$$

The value for DCE can be estimated from the heats of formation.

Combustion reaction:

$$C_2H_4Cl_2(g) + 2\tfrac{1}{2}O_2(g) \rightarrow 2CO_2(g) + H_2O(g) + 2HCl(g)$$

The heats of formation ΔH_f° are given in Appendix C, which is available in the online material at booksite.Elsevier.com/Towler.

$$\begin{aligned} CO_2 &= -393.8\ \text{kJ/mol} \\ H_2O &= -242.0 \\ HCl &= -92.4 \\ DCE &= -130.0 \\ \Delta H_c^\circ &= \sum \Delta H_f^\circ\ \text{products} - \sum \Delta H_f^\circ\ \text{reactants} \\ &= [2(-393.8) - 242.0 + 2(-92.4)] - [-130.0] \\ &= -1084.4\ \text{kJ} \end{aligned}$$

Estimation of vent-gas calorific value, basis 100 mol.

Component	mol/100 mol		Calorific Value		Heating Value (kJ/mol)
CO	1.79	×	283.0	=	506.6
C_2H_4	1.99		1560.9		3106.2
C_2H_6	0.1		1411.9		141.2
DCE	0.54		1084.4		585.7
				Total	4339.7

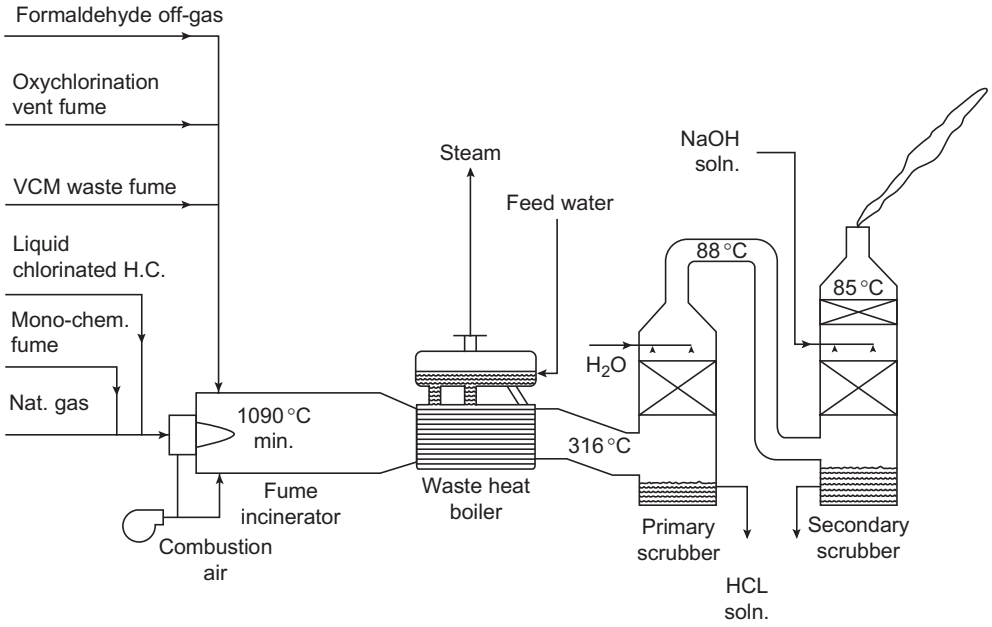


FIGURE 3.13

Typical incinerator-heat recovery-scrubber system for vinyl-chloride-monomer process waste. (Courtesy of John Thurley Ltd.)

$$\text{Calorific value of vent gas} = \frac{4339.7}{100} = 43.4\,\text{kJ/mol}$$

$$= \frac{43.4}{22.4} \times 10^3 = 1938\,\text{kJ/m}^3\ (52\,\text{Btu/ft}^3)\ \text{at 1 bar, 0 °C}$$

This calorific value is very low compared to 37 MJ/m^3 (1000 Btu/ft^3) for natural gas. The vent gas is barely worth recovery, but if the gas has to be burnt to avoid pollution it could be used in an incinerator such as that shown in Figure 3.13, giving a useful steam production to offset the cost of disposal.

3.4.2 Liquid and Solid Wastes

Combustible liquid and solid waste can be disposed of by burning, which is usually preferred to dumping. Incorporating a steam boiler in the incinerator design will enable an otherwise unproductive, but necessary, operation to save energy. If the combustion products are corrosive, corrosion-resistant materials will be needed, and the flue gases must be scrubbed to reduce air pollution. An incinerator designed to handle chlorinated and other liquid and solid wastes is shown in Figure 3.13. This incinerator incorporates a steam boiler and a flue-gas scrubber. The disposal of chlorinated wastes is discussed by Santoleri (1973).

Dunn and Tomkins (1975) discuss the design and operation of incinerators for process wastes. They give particular attention to the need to comply with the current clean-air legislation, and the problem of corrosion and erosion of refractories and heat-exchange surfaces.

3.5 HEAT-EXCHANGER NETWORKS

The design of a heat-exchanger network for a simple process with only one or two streams that need heating and cooling is usually straightforward. When there are multiple hot and cold streams, the design is more complex and there may be many possible heat exchange networks. The design engineer must determine the optimum extent of heat recovery, while ensuring that the design is flexible to changes in process conditions and can be started up and operated easily and safely.

In the 1980s, there was a great deal of research into design methods for heat-exchanger networks; see Gundersen and Naess (1988). One of the most widely applied methods that emerged was a set of techniques termed *pinch technology*, which was developed by Bodo Linnhoff and his collaborators at ICI, Union Carbide, and the University of Manchester. The term derives from the fact that in a plot of the system temperatures versus the heat transferred, a *pinch* usually occurs between the hot stream and cold stream curves, see Figure 3.19. It has been shown that the pinch represents a distinct thermodynamic break in the system and that, for minimum energy requirements, heat should not be transferred across the pinch, Linnhoff *et al.* (1982).

In this section the fundamental principles of the pinch technology method for energy integration will be outlined and illustrated with reference to a simple problem. The method and its applications are described fully in a guide published by the Institution of Chemical Engineers, Kemp (2007); see also Douglas (1988), Smith (2005), and El-Halwagi (2006).

3.5.1 Pinch Technology

The development and application of the method can be illustrated by considering the problem of recovering heat between four process streams: two hot streams that require cooling, and two cold streams that must be heated. The process data for the streams is set out in Table 3.1. Each stream starts from a source temperature T_s, and is to be heated or cooled to a target temperature T_t. The heat capacity flow rate of each stream is shown as CP. For streams where the specific heat capacity can be taken as constant, and there is no phase change, CP will be given by

$$CP = mC_p \tag{3.5}$$

where m = mass flow-rate, kg/s

C_p = average specific heat capacity between T_s and T_t kJ kg^{-1}°C^{-1}

Table 3.1 Data for Heat Integration Problem

Stream Number	Type	Heat Capacity Flow Rate CP, kW/°C	T_s °C	T_t °C	Heat Load, kW
1	hot	3.0	180	60	360
2	hot	1.0	150	30	120
3	cold	2.0	20	135	230
4	cold	4.5	80	140	270

The heat load shown in the table is the total heat required to heat, or cool, the stream from the source to the target temperature.

There is clearly scope for energy integration between these four streams. Two require heating and two cooling, and the stream temperatures are such that heat can be transferred from the hot to the cold streams. The task is to find the best arrangement of heat exchangers to achieve the target temperatures.

Simple Two-stream Problem

Before investigating the energy integration of the four streams shown in Table 3.1, the use of a temperature-enthalpy diagram will be illustrated for a simple problem involving only two streams. The general problem of heating and cooling two streams from source to target temperatures is shown in Figure 3.14. Some heat is exchanged between the streams in the heat exchanger. Additional heat, to raise the cold stream to the target temperature, is provided by the hot utility (usually steam) in the heater; and additional cooling, to bring the hot stream to its target temperature, is provided by the cold utility (usually cooling water) in the cooler.

In Figure 3.15(a) the stream temperatures are plotted on the y-axis and the enthalpy change of each stream on the x-axis. This is known as a temperature-enthalpy (T-H) diagram. For heat to be exchanged, a minimum temperature difference must be maintained between the two streams. This is shown as $\Delta T_{\min}$ on the diagram. The practical minimum temperature difference in a heat exchanger will usually be between 5 °C and 30 °C; see Chapter 19.

The slope of the lines in the T-H plot is proportional to $1/CP$, since $\Delta H = CP \times \Delta T$, so $\mathrm{d}T/\mathrm{d}H = 1/CP$. Streams with low heat capacity flow rate thus have steep slopes in the T-H plot and streams with high heat capacity flow rate have shallow slopes.

The heat transferred between the streams is given by the range of enthalpy over which the two curves overlap each other, and is shown on the diagram as ΔH_{ex}. The heat transferred from the hot utility, ΔH_{hot}, is given by the part of the cold stream that is not overlapped by the hot stream. The heat transferred to the cold utility, ΔH_{cold}, is similarly given by the part of the hot stream that is not overlapped by the cold stream. The heats can also be calculated as

$$\Delta H = CP \times (\textit{temperature change})$$

Since we are only concerned with changes in enthalpy, we can treat the enthalpy axis as a relative scale and slide either the hot stream or the cold stream horizontally. As we do so, we change

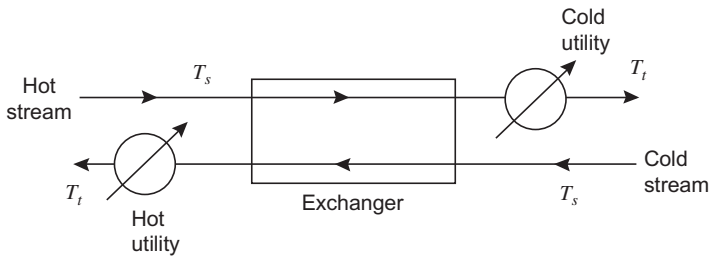


FIGURE 3.14

Two-stream exchanger problem.

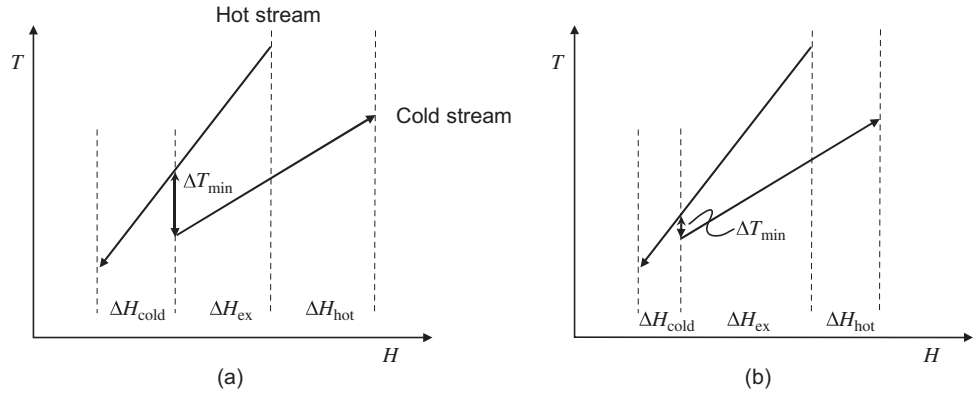


FIGURE 3.15

Temperature-enthalpy (*T-H*) diagram for two-stream example.

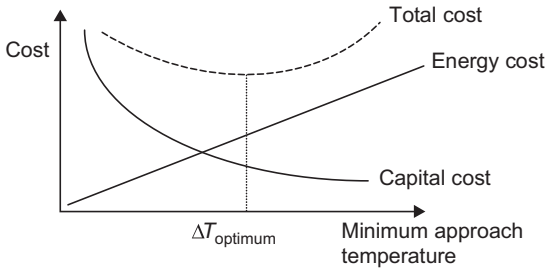


FIGURE 3.16

The capital-energy trade-off in process heat recovery.

the minimum temperature difference between the streams, ΔT_{min}, and also the amount of heat exchanged and the amounts of hot and cold utilities required.

Figure 3.15(b) shows the same streams plotted with a lower value of ΔT_{min}. The amount of heat exchanged is increased and the utility requirements have been reduced. The temperature driving force for heat transfer has also been reduced, so the heat exchanger has both a larger duty and a smaller log-mean temperature difference. This leads to an increase in the heat transfer area required and in the capital cost of the exchanger. The capital cost increase is partially offset by capital cost savings in the heater and cooler, which both become smaller, as well as by savings in the costs of hot and cold utility. In general, there will be an optimum value of ΔT_{min}, as illustrated in Figure 3.16. This optimum is usually rather flat over the range 10 ℃ to 30 ℃.

The maximum feasible heat recovery is reached at the point where the hot and cold curves touch each other on the *T-H* plot, as illustrated in Figure 3.17. At this point, the temperature driving force at one end of the heat exchanger is zero and an infinite heat exchange surface is required, so the design is not practical. The exchanger is said to be *pinched* at the end where the hot and cold curves meet. In Figure 3.17, the heat exchanger is pinched at the cold end.

It is not possible for the hot and cold streams to cross each other, as this would be a violation of the second law of thermodynamics and would give an infeasible design.

Four-stream Problem

In Figure 3.18(a) the hot streams given in Table 3.1 are shown plotted on a temperature-enthalpy diagram.

As the diagram shows changes in the enthalpy of the streams, it does not matter where a particular curve is plotted on the enthalpy axis; as long as the curve runs between the correct temperatures. This means that where more than one stream appears in a temperature interval, the stream heat capacities can be added to form a composite curve, as shown in Figure 3.18(b).

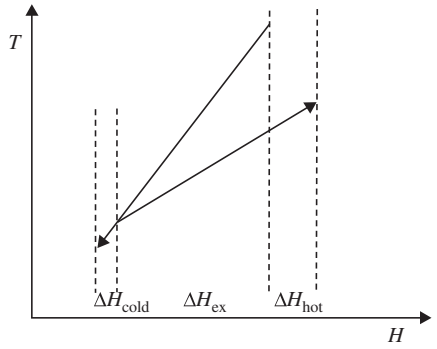


FIGURE 3.17

Maximum feasible heat recovery for two-stream example.

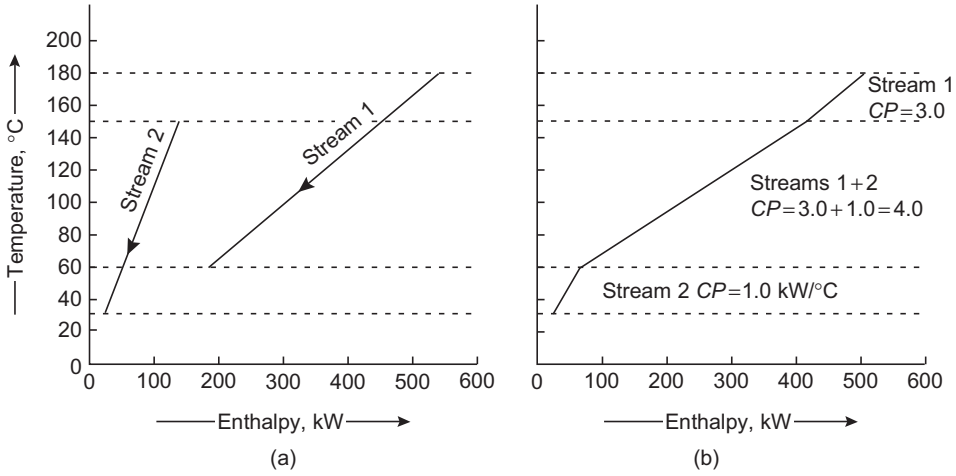


FIGURE 3.18

Hot stream temperature v. enthalpy: (a) separate hot streams; (b) composite hot stream.

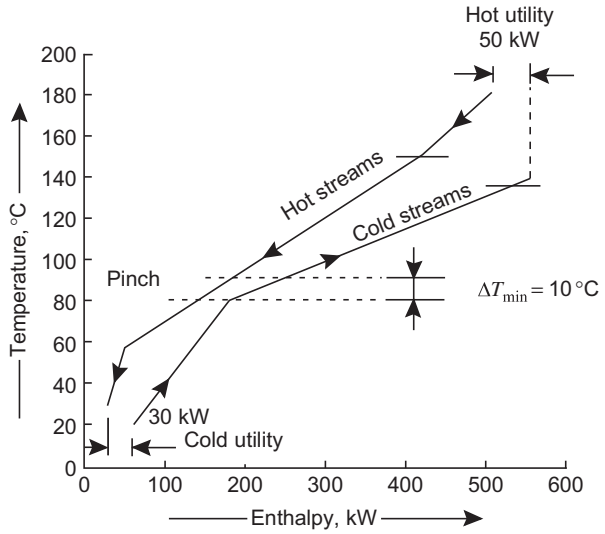


FIGURE 3.19

Hot and cold stream composite curves.

In Figure 3.19, the composite curve for the hot streams and the composite curve for the cold streams are drawn with a minimum temperature difference, the displacement between the curves, of 10 °C. This implies that in any of the exchangers to be used in the network the temperature difference between the streams will not be less than 10 °C.

As for the two-stream problem, the overlap of the composite curves gives a target for heat recovery, and the displacements of the curves at the top and bottom of the diagram give the hot and cold utility requirements. These will be the minimum values needed to satisfy the target temperatures. This is valuable information. It gives the designer target values for the utilities to aim for when designing the exchanger network. Any design can be compared with the minimum utility requirements to check if further improvement is possible.

In most exchanger networks the minimum temperature difference will occur at only one point. This is termed the *pinch*. In the problem being considered, the pinch occurs at between 90 °C on the hot stream curve and 80 °C on the cold stream curve.

For multi-stream problems, the pinch will usually occur somewhere in the middle of the composite curves, as illustrated in Figure 3.19. The case when the pinch occurs at the end of one of the composite curves is termed a threshold problem and is discussed in Section 3.5.5.

Thermodynamic Significance of the Pinch

The pinch divides the system into two distinct thermodynamic regions. The region above the pinch can be considered a heat sink, with heat flowing into it from the hot utility, but no heat flow out of it. Below the pinch the converse is true. Heat flows out of the region to the cold utility. No heat flows across the pinch, as shown in Figure 3.20(a).

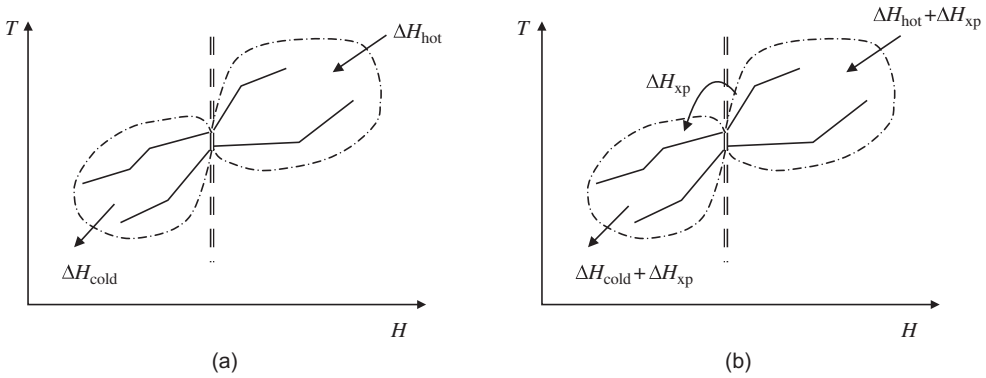


FIGURE 3.20

Pinch decomposition.

If a network is designed in which heat is transferred from any hot stream at a temperature above the pinch (including hot utilities) to any cold stream at a temperature below the pinch (including cold utilities), then heat is transferred across the pinch. If the amount of heat transferred across the pinch is ΔH_{xp}, then in order to maintain energy balance the hot utility and cold utility must both be increased by ΔH_{xp}, as shown in Figure 3.20(b). Cross-pinch heat transfer thus always leads to consumption of both hot and cold utilities that is greater than the minimum values that could be achieved.

The pinch decomposition is very useful in heat-exchanger network design, as it decomposes the problem into two smaller problems. It also indicates the region where heat transfer matches are most constrained, at or near the pinch. When multiple hot or cold utilities are used there may be other pinches, termed utility pinches, that cause further problem decomposition. Problem decomposition can be exploited in algorithms for automatic heat-exchanger network synthesis.

3.5.2 The Problem Table Method

The problem table is a numerical method for determining the pinch temperatures and the minimum utility requirements, introduced by Linnhoff and Flower (1978). It eliminates the sketching of composite curves, which can be useful if the problem is being solved manually. It is not widely used in industrial practice any more, due to the wide availability of computer tools for pinch analysis (see Section 3.5.7).

The procedure is as follows:

1. Convert the actual stream temperatures T_{act} into interval temperatures T_{int} by subtracting half the minimum temperature difference from the hot stream temperatures, and by adding half to the cold stream temperatures:

$$\text{hot streams } T_{int} = T_{act} - \frac{\Delta T_{min}}{2}$$

$$\text{cold streams } T_{int} = T_{act} + \frac{\Delta T_{min}}{2}$$

The use of the interval temperature rather than the actual temperatures allows the minimum temperature difference to be taken into account. $\Delta T_{min} = 10$ °C for the problem being considered; see Table 3.2.

2. Note any duplicated interval temperatures. These are bracketed in Table 3.2.
3. Rank the interval temperatures in order of magnitude, showing the duplicated temperatures only once in the order; see Table 3.3.
4. Carry out a heat balance for the streams falling within each temperature interval.
   For the n^{th} interval:

$$\Delta H_n = (\sum CP_c - \sum CP_h)(\Delta T_n)$$

   where ΔH_n = net heat required in the n^{th} interval
   ΣCP_c = sum of the heat capacities of all the cold streams in the interval
   ΣCP_h = sum of the heat capacities of all the hot streams in the interval
   ΔT_n = interval temperature difference = $(T_{n-1} - T_n)$

   See Table 3.4.
5. "Cascade" the heat surplus from one interval to the next down the column of interval temperatures; see Figure 3.21(a).

   Cascading the heat from one interval to the next implies that the temperature difference is such that the heat can be transferred between the hot and cold streams. The presence of a negative value in the column indicates that the temperature gradient is in the wrong direction and that the exchange is not thermodynamically possible.

   This difficulty can be overcome if heat is introduced into the top of the cascade:

Table 3.2 Interval Temperatures for ΔT_{min} = 10 °C

Stream	Actual Temperature		Interval Temperature	
1	180	60	175	55
2	150	30	145	25
3	20	135	(25)	140
4	80	140	85	(145)

Table 3.3 Ranked Order of Interval Temperatures

Rank	Interval ΔT_n °C	Streams in Interval
175		
145	30	−1
140	5	4 − (2 + 1)
85	55	(3 + 4) − (1 + 2)
55	30	3 − (1 + 2)
25	30	3 − 2

Note: Duplicated temperatures are omitted. The interval ΔT *and streams in the intervals are included as they are needed for Table 3.4.*

Table 3.4 Problem Table

Interval	Interval Temp. °C	ΔT_n °C	$\Sigma CP_c - \Sigma CP_h$* kW/°C	ΔH kW	Surplus or Deficit
	175				
1	145	30	−3.0	−90	s
2	140	5	0.5	2.5	d
3	85	55	2.5	137.5	d
4	55	30	−2.0	−60	s
5	25	30	1.0	30	d

**Note: The streams in each interval are given in Table 3.3.*

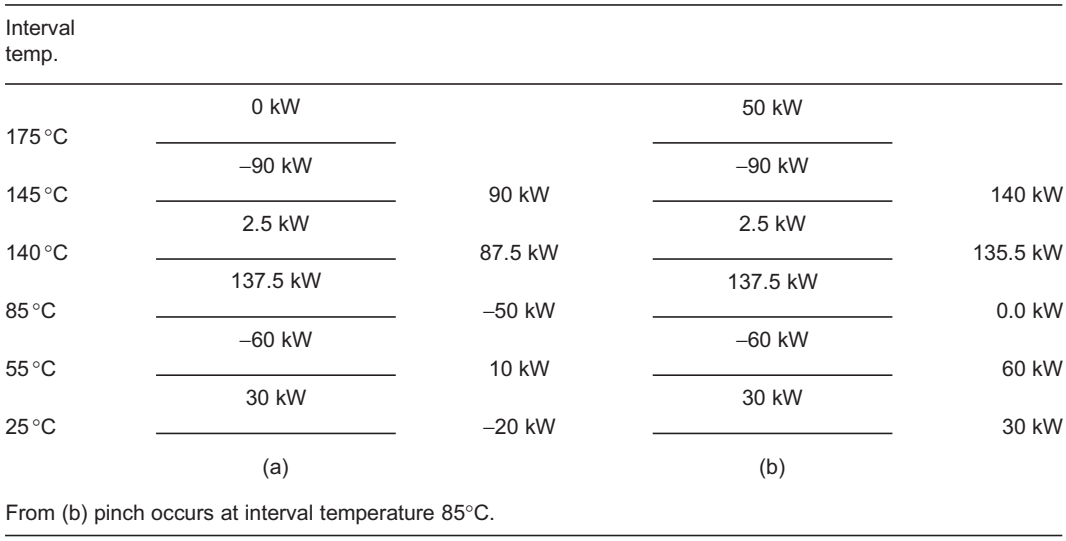


FIGURE 3.21

Heat cascade.

6. Introduce just enough heat to the top of the cascade to eliminate all the negative values; see Figure 3.21(b).

Comparing the composite curve, Figure 3.19, with Figure 3.21(b) shows that the heat introduced to the cascade is the minimum hot utility requirement and the heat removed at the bottom is the minimum cold utility required. The pinch occurs in Figure 3.21(b) where the heat flow in the cascade is zero. This is as would be expected from the rule that for minimum utility requirements no heat flows across the pinch. In Figure 3.21(b) the pinch is at an interval temperature

of 85 ºC, corresponding to a cold stream temperature of 80 ºC and a hot stream temperature of 90 °C, as was found using the composite curves.

It is not necessary to draw up a separate cascade diagram. This was done in Figure 3.21 to illustrate the principle. The cascaded values can be added to the problem table as two additional columns; see Example 3.5.

Summary

For maximum heat recovery and minimum use of utilities:

1. Do not transfer heat across the pinch.
2. Do not use hot utilities below the pinch.
3. Do not use cold utilities above the pinch.

3.5.3 Heat-exchanger Network Design

Grid Representation

It is convenient to represent a heat-exchanger network as a grid; see Figure 3.22. The process streams are drawn as horizontal lines, with the stream numbers shown in square boxes. Hot streams are drawn at the top of the grid, and flow from left to right. The cold streams are drawn at the bottom, and flow from right to left. The stream heat capacities *CP* are shown in a column at the end of the stream lines.

Heat exchangers are drawn as two circles connected by a vertical line. The circles connect the two streams between which heat is being exchanged; that is, the streams that would flow through the actual exchanger. Heaters and coolers can be drawn as a single circle, connected to the appropriate utility. If multiple utilities are used then these can also be shown as streams. Exchanger duties are usually marked under the exchanger and temperatures are also sometimes indicated on the grid diagram.

Network Design for Maximum Energy Recovery

The analysis carried out in Figure 3.19 and Figure 3.21 has shown that the minimum utility requirements for the problem set out in Table 3.1 are 50 kW of the hot and 30 kW of the cold utility, and that the pinch occurs where the cold streams are at 80 ºC and the hot streams are at 90 °C.

The grid representation of the streams is shown in Figure 3.23. The vertical dotted lines represent the pinch and separate the grid into the regions above and below the pinch. Note that the hot and cold streams are offset at the pinch, because of the difference in pinch temperature.

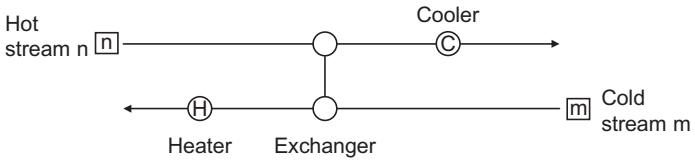


FIGURE 3.22

Grid representation.

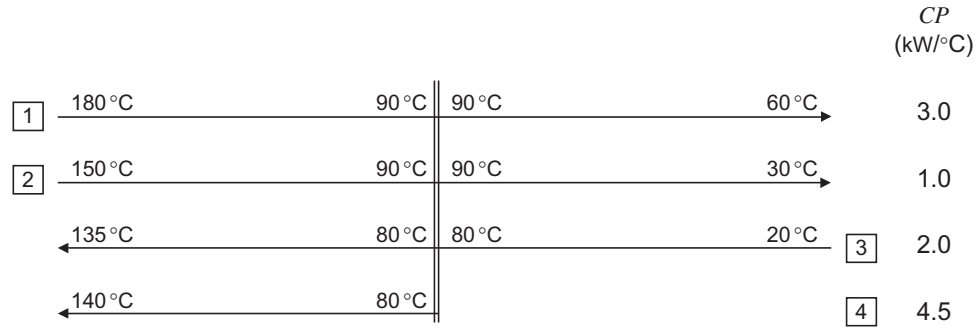


FIGURE 3.23

Grid for four-stream problem.

For maximum energy recovery (minimum utility consumption) the best performance is obtained if no cooling is used above the pinch. This means that the hot streams above the pinch should be brought to the pinch temperature solely by exchange with the cold streams. The network design is therefore started at the pinch, finding feasible matches between streams to fulfill this aim. In making a match adjacent to the pinch the heat capacity CP of the hot stream must be equal to or less than that of the cold stream. This is to ensure that the minimum temperature difference between the curves is maintained. The slope of a line on the temperature-enthalpy diagram is equal to the reciprocal of the heat capacity. So, above the pinch the lines will converge if CP_h exceeds CP_c and as the streams start with a separation at the pinch equal to ΔT_{min}, the minimum temperature condition would be violated. Every hot stream must be matched with a cold stream immediately above the pinch, otherwise it will not be able to reach the pinch temperature.

Below the pinch the procedure is the same; the aim being to bring the cold streams to the pinch temperature by exchange with the hot streams. For streams adjacent to the pinch the criterion for matching streams is that the heat capacity of the cold stream must be equal to or greater than the hot stream, to avoid breaking the minimum temperature difference condition. Every cold stream must be matched with a hot stream immediately below the pinch.

Network Design Above the Pinch

$$CP_h \leq CP_c$$

1. Applying this condition at the pinch, stream 1 can be matched with stream 4, but not with 3.

Matching streams 1 and 4 and transferring the full amount of heat required to bring stream 1 to the pinch temperature gives

$$\Delta H_{ex} = CP(T_s - T_{pinch})$$

$$\Delta H_{ex} = 3.0(180 - 90) = 270 \text{ kW}$$

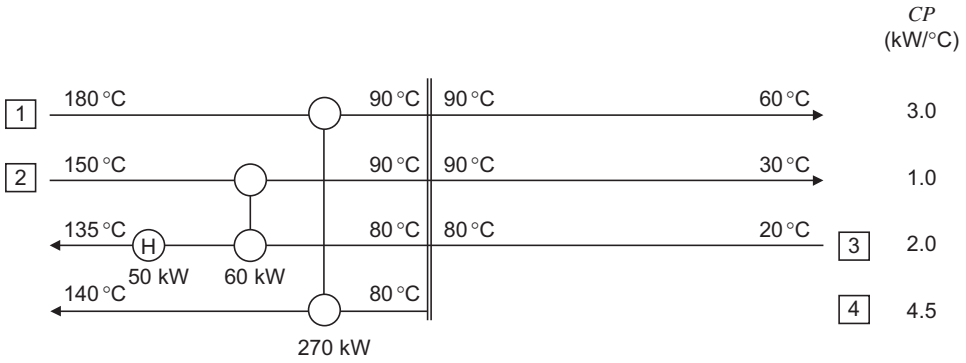


FIGURE 3.24

Network design above the pinch.

This will also satisfy the heat load required to bring stream 4 to its target temperature:

$$\Delta H_{ex} = 4.5(140 - 80) = 270\,\text{kW}$$

2. Stream 2 can be matched with stream 3, while satisfying the heat capacity restriction. Transferring the full amount to bring stream 2 to the pinch temperature:

$$\Delta H_{ex} = 1.0(150 - 90) = 60\,\text{kW}$$

3. The heat required to bring stream 3 to its target temperature, from the pinch temperature, is

$$\Delta H = 2.0(135 - 80) = 110\,\text{kW}$$

So a heater will have to be included to provide the remaining heat load:

$$\Delta H_{hot} = 110 - 60 = 50\,\text{kW}$$

This checks with the value given by the problem table, Figure 3.21(b).

The proposed network design above the pinch is shown in Figure 3.24.

Network Design Below the Pinch

$$CP_h \geq CP_c$$

4. Stream 4 begins at the pinch temperature, $T_s = 80$ °C, and so is not available for any matches below the pinch.

5. A match between streams 1 and 3 adjacent to the pinch will satisfy the heat capacity restriction but not one between streams 2 and 3. So 1 is matched with 3 transferring the full amount to bring stream 1 to its target temperature:

$$\Delta H_{ex} = 3.0(90 - 60) = 90\,\text{kW}$$

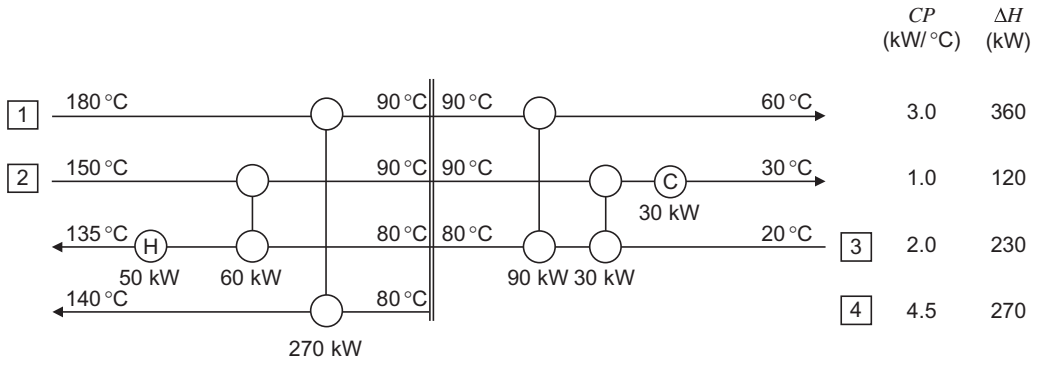


FIGURE 3.25

Proposed heat exchanger network for ΔT_{min} = 10 °C.

6. Stream 3 requires more heat to bring it to the pinch temperature; the amount needed is

$$\Delta H = 2.0(80 - 20) - 90 = 30\,\text{kW}$$

This can be provided from stream 2, as the match is now away from the pinch.

The rise in temperature of stream 3 will be given by

$$\Delta T = \Delta H / CP$$

So transferring 30 kW will raise the temperature from the source temperature to

$$20 + 30/2.0 = 35\,^{\circ}\text{C}$$

and this gives a stream temperature difference on the outlet side of the exchanger of

$$90 - 35 = 55\,^{\circ}\text{C}$$

So the minimum temperature difference condition, 10 °C, will not be violated by this match.

7. Stream 2 needs further cooling to bring it to its target temperature, so a cooler must be included; the cooling required is

$$\Delta H_{\text{cold}} = 1.0(90 - 30) - 30 = 30\,\text{kW}$$

which is the amount of the cold utility predicted by the problem table.

The proposed network for maximum energy recovery is shown in Figure 3.25.

Stream Splitting

If the heat capacities of streams are such that it is not possible to make a match at the pinch without violating the minimum temperature difference condition, then the heat capacity can be altered by splitting a stream. Dividing the stream will reduce the mass flow rates in each leg and hence the heat capacities. This is illustrated in Example 3.5.

Similarly, if there are not enough streams available to make all of the required matches at the pinch then streams with large *CP* can be split to increase the number of streams.

Guide rules for stream matching and splitting are given in the Institution of Chemical Engineers Guide, Kemp (2007) and by Smith (2005).

Summary

The guide rules for devising a network for maximum heat recovery are given below:

1. Divide the problem at the pinch.
2. Design away from the pinch.
3. Above the pinch match streams adjacent to the pinch, meeting the restriction

$$CP_h \leq CP_c$$

4. Below the pinch match streams adjacent to the pinch, meeting the restriction

$$CP_h \geq CP_c$$

5. If the stream matching criteria cannot be satisfied, split a stream.
6. Maximize the exchanger heat loads.
7. Supply external heating only above the pinch and external cooling only below the pinch.

3.5.4 Minimum Number of Exchangers

The network shown in Figure 3.25 was designed to give the maximum heat recovery, and will therefore give the minimum consumption, and cost, of the hot and cold utilities.

This will not necessarily be the optimum design for the network. The optimum design will be that which gives the lowest total annualized cost, taking into account the capital cost of the system, in addition to the utility and other operating costs. The number of exchangers in the network, and their size, will determine the capital cost.

In Figure 3.25 it is clear that there is scope for reducing the number of exchangers. The 30 kW exchanger between streams 2 and 3 can be deleted and the heat loads of the cooler and heater increased to bring streams 2 and 3 to their target temperatures. Heat would cross the pinch and the consumption of the utilities would be increased. Whether the revised network would be better, or more economic, depends on the relative cost of capital and utilities and the operability of each design. For any network, there will be an optimum design that gives the least annual cost: capital charges plus utility and other operating costs. The estimation of capital and operating costs are covered in Chapters 7 and 8.

To find the optimum design it is necessary to cost a number of alternative designs, seeking a compromise between the capital costs, determined by the number and size of the exchangers, and the utility costs, determined by the heat recovery achieved.

For simple networks Holmann (1971) has shown that the minimum number of exchangers is given by

$$Z_{min} = N' - 1 \tag{3.6}$$

where Z_{min} = minimum number of exchangers needed, including heaters and coolers
N' = the number of streams, including the utilities

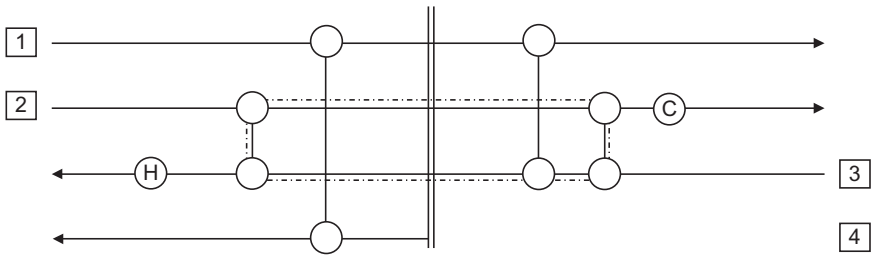


FIGURE 3.26

Loop in network.

For complex networks a more general expression is needed to determine the minimum number of exchangers:

$$Z_{\min} = N' + L' - S \tag{3.7}$$

where L' = the number of internal loops present in the network
S = the number of independent branches (subsets) that exist in the network

A loop exists where a closed path can be traced through the network. There is a loop in the network shown in Figure 3.25. The loop is shown in Figure 3.26. The presence of a loop indicates that there is scope for reducing the number of exchangers.

For a full discussion of Equation 3.7 and its applications see Linnhoff, Mason, Wardle (1979), Smith (2005), or Kemp (2007).

In summary, to seek the optimum design for a network:

1. Start with the design for maximum heat recovery. The number of exchangers needed will be equal to or less than the number for maximum energy recovery.
2. Identify loops that cross the pinch. The design for maximum heat recovery will usually contain loops.
3. Starting with the loop with the least heat load, break the loops by adding or subtracting heat.
4. Check that the specified minimum temperature difference $\Delta T_{\min}$ has not been violated. If the violation is significant, revise the design as necessary to restore $\Delta T_{\min}$. If the violation is small then it may not have much impact on the total annualized cost and can be ignored.
5. Estimate the capital and operating costs, and the total annual cost.
6. Repeat the loop breaking and network revision to find the lowest cost design.
7. Consider the safety, operability, and maintenance aspects of the proposed design.

3.5.5 Threshold Problems

Problems that show the characteristic of requiring only either a hot utility or a cold utility (but not both) over a range of minimum temperature differences, from zero up to a threshold value, are known as threshold problems. A threshold problem is illustrated in Figure 3.27.

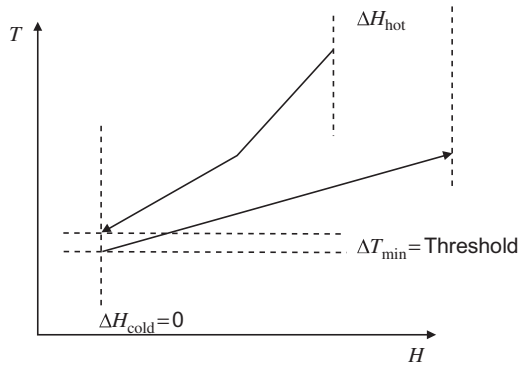


FIGURE 3.27

Threshold problem.

To design the heat-exchanger network for a threshold problem, it is normal to start at the most constrained point. The problem can often be treated as one half of a problem exhibiting a pinch.

Threshold problems are often encountered in the process industries. A pinch can be introduced in such problems if multiple utilities are used, as in the recovery of heat to generate steam, or if the chosen value of ΔT_{min} is greater than the threshold value.

The procedures to follow in the design of threshold problems are discussed by Smith (2005) and Kemp (2007).

3.5.6 Determining Utility Consumption

Pinch analysis can be used to determine targets for process utility consumption. Initial targets for total hot and cold utility use can be calculated directly from the problem table algorithm or read from the composite curves. A more detailed breakdown of the utility needs can be determined from the initial heat-exchanger network.

The following guidelines should be followed when using the pinch method to determine utility consumption targets:

1. Do not use cold utilities above the pinch temperature. This means that no process stream should be cooled from a temperature above the pinch temperature using a cold utility.
2. Do not use hot utilities below the pinch. This means no process stream should be heated from a temperature below the pinch temperature using a hot utility.
3. On either side of the pinch, maximize use of the cheapest utility first. Above the pinch this means use LP steam wherever possible before considering MP steam, then HP steam, hot oil, etc. Below the pinch, maximize use of cooling water before considering refrigeration.
4. If the process pinch is at a high temperature, consider boiler feed water preheat and steam generation as potential cold utility streams.
5. If the process pinch is at a low temperature, consider steam condensate and spent cooling water as hot streams.

6. If the process requires cooling to a very low temperature, consider using cascaded refrigeration cycles to improve the overall *COP*.
7. If the process requires heating to a very high temperature and a fired heater is needed, consider using the convective section heat either for process heating or for steam generation. For process control reasons, it may be necessary to operate the heater with process heating in the radiant section only, but the convective section heat is still available for use. In strict pinch terms, this heat can be used at any temperature above the pinch temperature, but in practice convective section heat recovery is usually limited by the acid-gas dew point of the flue gas or other furnace design considerations (see Section 19.17).
8. If a process condition leads to the use of a more expensive utility, then consider process modifications that would make this unnecessary. For example, if a product must be cooled and sent to storage at 30 ºC, the cooling cannot be carried out using cooling water and refrigeration must be used. The designer should question why 30 ºC was specified for the storage. If it was because a vented tank was selected, then choosing a non-vented (floating roof) tank instead might allow the product to be sent to storage at 40 ºC, in which case the refrigeration system could be eliminated.

Graphical methods and numerical approaches have been developed to assist in the optimal design of utility systems. For simple problems, these methods are not needed, as the heaters and coolers that have been identified in the heat-exchange network can be assigned to the appropriate utility stream using the simple rules above. When a stream requires heating or cooling over a broad temperature range, the designer should consider whether it is cheaper to break the duty into several exchangers, each served by the appropriate utility for a given temperature range, or whether it makes more economic sense to use a single exchanger, served by the hottest or coldest utility. The problem of placing multiple utilities is illustrated in Example 3.6.

3.5.7 Process Integration: Integration of Other Process Operations

The pinch technology method can give many other insights into process synthesis, beyond the design of heat-exchanger networks. The method can also be applied to the integration of other process units, such as separation columns, reactors, compressors and expanders, boilers, and heat pumps. The wider applications of pinch technology are discussed in the Institution of Chemical Engineers Guide, Kemp (2007) and by El-Halwagi (2006) and Smith (2005).

The techniques of process integration have been expanded for use in optimizing mass transfer operations, and have been applied in waste reduction, water conservation, and pollution control; see El-Halwagi (1997) and Dunn and El-Halwagi (2003).

3.5.8 Computer Tools for Heat-exchanger Network Design

Most pinch analysis in industry is carried out using commercial pinch analysis software. Programs such as Aspen HX-Net™ (Aspen Technology Inc.), SUPERTARGET™ (Linnhoff March Ltd.) and UniSim™ ExchangerNet™ (Honeywell International Inc.) allow the design engineer to plot composite curves, optimize $\Delta T_{\min}$, set targets for multiple utilities, and design the heat-exchanger network.

Most of these programs are able to automatically extract stream data from process simulation programs, although great care should be taken to check the extracted data. There are many possible pitfalls in data extraction; for example, not recognizing changes in the *CP* of a stream or partial vaporization or condensation of a stream, any of which could lead to a kink in the stream *T-H* profile. See Smith (2005) or Kemp (2007) for more information on data extraction.

The commercial pinch technology tools also usually include automatic heat-exchanger network synthesis features. The automatic synthesis methods are based on MINLP optimization of superstructures of possible exchanger options (see Chapter 12 for discussion of MINLP methods). These tools can be used to arrive at a candidate network, but the optimization must be properly constrained so that it does not introduce a large number of stream splits and add a lot of small exchangers. Experienced designers seldom use automatic heat-exchanger network synthesis methods, as it usually requires more effort to turn the resulting network into something practical than it would take to design a practical network manually. The NLP optimization capability of the software is widely used though, for fine tuning the network temperatures by exploitation of loops and stream split ratios.

Example 3.5

Determine the pinch temperatures and the minimum utility requirements for the streams set out in the table below, for a minimum temperature difference between the streams of 20 °C. Devise a heat-exchanger network to achieve the maximum energy recovery.

Stream Number	Type	Heat Capacity Flow Rate, kW/°C	Source Temp. °C	Target Temp. °C	Heat Load, kW
1	hot	40.0	180	40	5600
2	hot	30.0	150	60	1800
3	cold	60.0	30	180	9000
4	cold	20.0	80	160	1600

Solution

The problem table to find the minimum utility requirements and the pinch temperature can be built in a spreadsheet. The calculations in each cell are repetitive and the formula can be copied from cell to cell using the cell copy commands. A spreadsheet template for the problem table algorithm is available in MS Excel format in the online material at booksite.Elsevier.com/Towler. The use of the spreadsheet is illustrated in Figure 3.28 and described below.

First calculate the interval temperatures, for $\Delta T_{min} = 20$ °C

$$\text{hot streams } T_{int} = T_{act} - 10\ °C$$

$$\text{cold streams } T_{int} = T_{act} + 10\ °C$$

Company Name Address	Project Name Project Number					Sheet		1 of 1
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PROBLEM TABLE ALGORITHM								
Form XXXXX-YY-ZZ								

1. Minimum temperature approach

T_{min} 20 °C

2. Stream data

	Actual temperature (°C)		Interval temperature (°C)		Heat capacity flow	Heat load
Stream No.	Source	Target	Source	Target	rate *CP* (kW/°C)	(kW)
1	180	40	170	30	40	5600
2	150	60	140	50	30	2700
3	30	180	40	190	60	9000
4	80	160	90	170	20	1600
5						0
6						0
7						0
8						0

3. Problem table

Interval	Interval temp (°C)	Interval *T* (°C)	Sum *CP*c sum CP_h (kW/°C)	d*H* (kW)	Cascade (kW)	(kW)
1	190				0	2900
2	170	20	60	1200	1200	1700
3	170	0	60	0	1200	1700
4	140	30	40	1200	2400	500
5	90	50	10	500	2900	0
6	50	40	10	400	2500	400
7	40	10	20	200	2700	200
8	30	10	40	400	2300	600

FIGURE 3.28

Problem table algorithm spreadsheet.

	Actual Temp. °C		Interval Temp. °C	
Stream	**Source**	**Target**	**Source**	**Target**
1	180	40	170	30
2	150	60	140	50
3	30	180	40	190
4	80	160	90	170

In the spreadsheet this can be done by using an IF function to determine whether the source temperature is lower than the target temperature, in which case the stream is a cold stream and should have $\Delta T_{min}/2$ added.

Next rank the interval temperatures, ignoring any duplicated values. In the spreadsheet this is done using the LARGE function. Determine which streams occur in each interval. For a stream to be present in a given interval the largest stream interval temperature must be greater than the lower end of the interval range and the lowest stream interval temperature must also be greater than or equal to the lower end of the interval range. This can be calculated in the spreadsheet using IF, AND, and OR functions. Once the streams in each interval have been determined it is possible to calculate the combined stream heat capacities. These calculations are not strictly part of the problem table, so they have been hidden in the spreadsheet (in columns to the right of the table).

The sum of CP values for the cold streams minus that for the hot streams can then be multiplied by the interval ΔT to give the interval ΔH, and the interval ΔH values can be cascaded to give the overall heat flow. The amount of heat that must be put in to prevent the heat flow from becoming negative is the lowest value in the column, which can be found using the SMALL function. The final column then gives a cascade showing only positive values, with zero energy cascading at the pinch temperature.

In the last column 2900 kW of heat have been added to eliminate the negative values in the previous column; so the hot utility requirement is 2900 kW and the cold utility requirement, the bottom value in the column, is 600 kW.

The pinch occurs where the heat transferred is zero, that is at interval number 4, interval temperature 90 °C.

So at the pinch hot streams will be at

$$90 + 10 = 100\,^{\circ}\mathrm{C}$$

and the cold streams will be at

$$90 - 10 = 80\,^{\circ}\mathrm{C}$$

Note that in the table both stream 1 and stream 4 had an interval temperature of 170 ºC, which led to a duplicate line in the list of ranked interval temperatures. Strictly, this line could have been eliminated, but since it gave a zero value for the ΔT, it did not affect the calculation. The programming of the spreadsheet is a lot easier if duplicate temperatures are handled in this manner.

To design the network for maximum energy recovery, start at the pinch and match streams, following the rules on stream heat capacities for matches adjacent to the pinch. Where a match is made, transfer the maximum amount of heat.

The proposed network is shown in Figure 3.29.

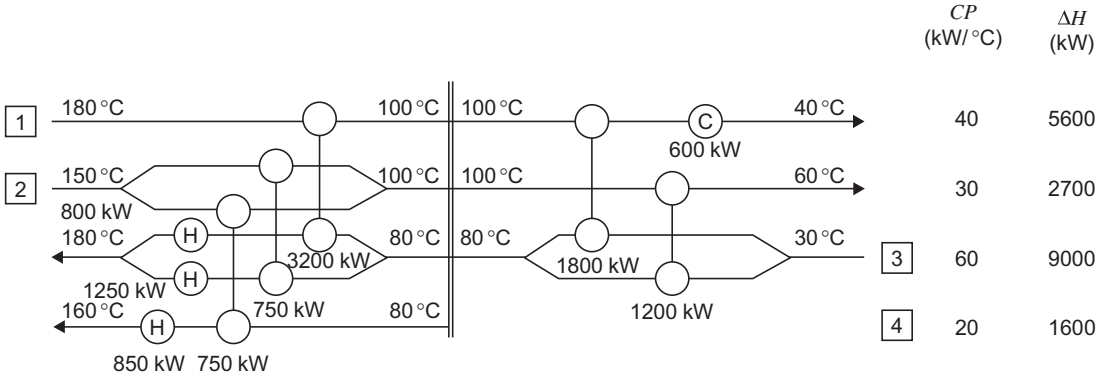

FIGURE 3.29

Proposed heat-exchanger network for Example 3.5.

The methodology followed in devising this network was:

Above Pinch

1. $CP_h \leq CP_c$
2. We can match stream 1 or 2 with stream 3 but neither stream can match with stream 4. This creates a problem, since if we match stream 1 with 3 then stream 2 will not be able to make a match at the pinch. Likewise, if we match stream 2 with 3 then stream 1 will not be able to make a match at the pinch.
3. Check the heat available in bringing the hot streams to the pinch temperature.

$$\text{stream } 1\ \Delta H = 40.0(180 - 100) = 3200\,\text{kW}$$

$$\text{stream } 2\ \Delta H = 30.0(150 - 100) = 1500\,\text{kW}$$

4. Check the heat required to bring the cold streams from the pinch temperature to their target temperatures.

$$\text{stream } 3\ \Delta H = 60.0(180 - 80) = 6000\,\text{kW}$$

$$\text{stream } 4\ \Delta H = 20.0(160 - 80) = 1600\ \text{kW}$$

5. If we split stream 3 into two branches with *CP* of 40.0 and 20.0, then we can match the larger branch with stream 1 and transfer 3200 kW, which satisfies (ticks off) stream 1.
6. We now have two cold streams, both with *CP* of 20.0, and one hot stream (2) with *CP* of 30.0. We need to split stream 2 into two branches. As an initial guess these can both have *CP* of 15.0. We can then match one branch of stream 2 with the smaller branch of 4 and transfer 750 kW, and the other branch with stream 3, also for 750 kW, which then ticks off stream 2.
7. Include a heater on the larger branch of stream 3 to bring it to its target temperature:

$$\Delta H_{\text{hot}} = 40(100) - 3200 = 800\,\text{kW}$$

8. Include a heater on the smaller branch of stream 3 to provide the balance of the heat required:

$$\Delta H_{\text{hot}} = 20(100) - 750 = 1250\,\text{kW}$$

9. Include a heater on stream 4 to provide the balance of the heat required:

$$\Delta H_{\text{hot}} = 1600 - 750 = 850\,\text{kW}$$

Check sum of heater duties = 800 + 1250 + 850 = 2900 kW = hot utility target.

Below Pinch

10. $CP_{\text{h}} \geq CP_{\text{c}}$

11. Note that stream 4 starts at the pinch temperature and so cannot provide any cooling below the pinch.

12. We cannot match stream 1 or 2 with stream 3 at the pinch.

13. Split stream 3 to reduce *CP*. An even split will allow both streams 1 and 2 to be matched with the split streams adjacent to the pinch, so try this initially.

14. Check the heat available from bringing the hot streams from the pinch temperature to their target temperatures:

$$\text{stream 1 } \Delta H = 40.0(100 - 40) = 2400\,\text{kW}$$

$$\text{stream 2 } \Delta H = 30.0(100 - 60) = 1200\,\text{kW}$$

15. Check the heat required to bring the cold streams from their source temperatures to the pinch temperature:

$$\text{stream 3 } \Delta H = 60.0(80 - 30) = 3000\,\text{kW}$$

Stream 4 is at the pinch temperature

16. Note that stream 1 cannot be brought to its target temperature of 40 °C by full interchange with stream 3 as the source temperature of stream 3 is 30 °C, so ΔT_{min} would be violated. So transfer 1800 kW to one leg of the split stream 3.

17. Check temperature at exit of this exchanger:

$$\text{Temp out} = 100 - \frac{1800}{40} = 55\,^{\circ}\text{C, satisfactory}$$

18. Provide cooler on stream 1 to bring it to its target temperature; the cooling needed is

$$\Delta H_{\text{cold}} = 2400 - 1800 = 600\,\text{kW}$$

19. Transfer the full heat load from stream 2 to second leg of stream 3; this satisfies both streams.

Note that the heating and cooling loads, 2900 kW and 600 kW, respectively, match those predicted from the problem table.

Note also that in order to satisfy the pinch decomposition and the stream matching rules we ended up introducing a large number of stream splits. This is quite common in heat-exchanger network design. None of the three split fractions was optimized, so substantial savings as well as simplification of the network could be possible. For example, loops exist between the branches of stream 3 and stream 1 and between the branches of stream 3 and stream 2. With the current split ratios these loops cannot be eliminated, but with other ratios it might be possible to eliminate one or two exchangers.

The introduction of multiple stream splits is often cited as a drawback of the pinch method. Stream splits can be problematic in process operation. For example, when an oil or other multicomponent stream is heated and partially vaporized, then the stream is a two-phase mixture. It is difficult to control the splitting of such streams to give the required flow rate in each branch. Experienced designers usually constrain the network to avoid multiple stream splits whenever possible, even if this leads to designs that have higher than minimum utility consumption.

Example 3.6

Determine the mix of utilities to use for the process introduced in Example 3.5, if the following utility streams are available:

Utility Stream	T_{supply} (°C)	T_{return} (°C)	Cost
MP steam (20 bar)	212	212	\$5.47/1000 lb
LP steam (6 bar)	159	159	\$4.03/1000 lb
Cooling water	30	40	\$0.10/1000 gal
Chilled water	10	20	\$4.50/GJ

Solution

From the solution to Example 3.5, we have the following heating and cooling duties that require utilities:

Cooler on stream 1, duty 600 kW, to cool stream 1 from 55 °C to 40 °C
Heater on large branch of stream 3, duty 800 kW, to heat from 160 °C to 180 °C
Heater on small branch of stream 3, duty 1250 kW, to heat from 117.5 °C to 180 °C
Heater on stream 4, duty 750 kW, to heat from 117.5 °C to 160 °C

It is obvious by inspection that if we are to maintain an approach temperature of 20 °C, then we will need to use MP steam and chilled water in at least some of the utility exchangers.

We can start by converting the utility costs into annual costs to provide a kW of heating or cooling, based on an assumed 8000 hours per year of operation.

For MP steam at 20 bar:

$$\text{Heat of condensation (by interpolation in steam tables)} \approx 1889\ \text{kJ/kg}$$

$$1\ \text{kW} = 3600 \times 8000\ \text{kJ/yr, therefore requires } 3600 \times 8000/1889 = 15.2 \times 10^3\ \text{kg/y}$$

$$\text{Annual cost per kW} = 15.2 \times 10^3 \times 2.205\ (\text{lb/kg}) \times 5.47\ (\$/1000\ \text{lb})/1000 = \$183\ /\text{y}$$

Similarly for LP steam at 6 bar:

$$\text{Heat of condensation (by interpolation in steam tables)} \approx 2085\ \text{kJ/kg}$$

$$1\ \text{kW} = 3600 \times 8000\ \text{kJ/yr, therefore requires } 3600 \times 8000/2085 = 13.8 \times 10^3\ \text{kg/y}$$

$$\text{Annual cost per kW} = 13.8 \times 10^3 \times 2.205\ (\text{lb/kg}) \times 4.03\ (\$/1000\ \text{lb})/1000 = \$123/\text{y}$$

For cooling water with a cooling range of 10 ºC:

$$1\,\text{kW of cooling requires } CP = 1/10 = 0.1\,\text{kW/°C}$$

Heat capacity of water ≈ 4.2 kJ/kgºC, so:

$$\text{Annual flow rate of cooling water per kW} = 0.1 \times 3600 \times 8000/4.2 = 686 \times 10^3\,\text{kg/y}$$

1000 gal of water = 3785 liters and has mass roughly 3785 kg, so:

$$\text{water flow rate} = 686 \times 10^3/3785 = 181.2\,\text{thousand gallons per year,}$$

$$\text{which has annual cost } 0.1 \times 181.2 = \$18.1/\text{y}$$

For chilled water:

$$1\,\text{kW of cooling} = 3600 \times 8000 = 28.8 \times 10^6\,\text{kJ/y} = 28.8\,\text{GJ/y}$$

So, annual cost = 28.8 × 4.50 = $129.6/y.

It is clearly cheaper to use LP steam rather than MP steam and to use cooling water instead of chilled water whenever it is feasible to do so.

Beginning with the design below the pinch, if we are to maintain a minimum temperature difference of 20 ºC, then we cannot use cooling water below 30 + 20 = 50 ºC. The lowest utility cost design would therefore use cooling water to cool stream 1 from 55 ºC to 50 ºC (duty 200 kW). A second cooler would then be needed to cool stream 1 from 50 ºC to 40ºC using chilled water (duty 400 kW). The annual utility cost of this design would be 200(18.1) + 400(129.6) = $55,460.

It might reasonably be argued that the utility savings from using the minimum cost of coolant do not justify the capital cost of an extra exchanger. Two possible alternatives can be considered. If all of the cooling is carried out using chilled water, then the minimum temperature difference constraint is not violated and a single cooler of duty 600 kW can be used. The annual utility cost would be 600(129.6) = $77,760. The use of chilled water gives larger log-mean temperature difference in the cooler, so the total surface area required in this design is less than the sum of the areas needed for the two exchangers proposed above. The incremental operating cost would have to be traded against the capital cost savings. Alternatively, if we jettison the 20 ºC minimum temperature difference and allow a 10 ºC minimum temperature difference in the cooler, then we can cool stream 1 using only cooling water in a single cooler of duty 600 kW. The annual utility cost would be 600(18.1) = $10,860. The savings in operating cost would have to be traded against the increased capital cost that would result from having a lower log-mean temperature difference for this exchanger.

Turning now to the design above the pinch, LP steam cannot be used for heating any stream that is above a temperature of 159 – 20 = 139 ºC. The minimum utility cost design would therefore use the following heaters:

LP steam to heat stream 4 from 117.5 ºC to 139 ºC
LP steam to heat the small branch of stream 3 from 117.5 ºC to 139 ºC
MP steam to heat the small branch of stream 3 from 139 ºC to 180 ºC
MP steam to heat the large branch of stream 3 from 160 ºC to 180 ºC
MP steam to heat stream 4 from 139 ºC to 160 ºC

Again, although this design has the minimum utility cost, other designs may be more optimal when capital costs are also considered. For example, there is no reason why the two branches of stream 3 must be sent to separate MP steam heaters. These two heaters could be combined, even though that violates the rule of thumb about not mixing streams at different temperatures, as we are well away from the pinch and have already ensured maximum use of LP steam. This modification would reduce capital cost with no increase in operating cost, so would almost certainly be adopted. Another modification to consider would be to examine allowing a smaller minimum temperature difference for the heaters that use LP steam. This would increase LP steam use at the expense of more capital (reduced temperature difference in the exchangers) and so would require a trade-off between the additional capital and the energy cost savings.

Note that by introducing the lowest cost utilities into the design we went from needing three heaters and one cooler in Figure 3.29 to using two coolers and five heaters in the lowest utility cost design. The introduction of multiple utilities almost always leads to an increase in the number of heat exchangers needed in a design as well as the surface area requirements, and the energy cost savings must justify the resulting increase in capital cost.

3.6 ENERGY MANAGEMENT IN UNSTEADY PROCESSES

The energy recovery approaches described above have been for steady-state processes, where the rate of energy generation or consumption did not vary with time. Batch and cyclic processes present multiple challenges for energy management. The designer must not only consider the amount of heat that must be added to or removed from the process, but also the dynamics of heat transfer. Limitations on the rate of heat transfer often cause heating and cooling steps to become the rate-limiting steps in determining the overall cycle time. The sequential nature of batch operations can also reduce the possibilities for heat recovery by heat exchange, unless multiple batches are processed in parallel and sequenced such that heat can be transferred from one batch to the next.

3.6.1 Differential Energy Balances

If a batch process is considered, or if the rate of energy generation or removal varies with time, it is necessary to set up a differential energy balance. For batch processes, the total energy requirements can usually be estimated by taking a single batch as the time basis for the calculation; but the maximum rate of heat generation must also be estimated to size any heat-transfer equipment needed.

A generalized differential energy balance can be written as

$$\text{Energy out} = \text{Energy in} + \text{generation} - \text{consumption} - \text{accumulation} \tag{3.8}$$

The energy in and energy out terms should include both heat transfer and convective heat flows, while the generation and consumption terms include heat of mixing, heat of reaction, etc. An unsteady state mass balance must usually be solved simultaneously with the differential energy balance.

Most batch processing operations are carried out in the liquid phase in stirred tanks. In the simplest case, heat is only added or removed when the vessel is full, and the convective heat flows can

be neglected. If there is no heat of reaction or mixing, then Equation 3.8 simplifies to

$$\text{Rate of heat accumulation} = \text{rate of heat transfer into vessel} \tag{3.9}$$

$$MC_p \frac{dT}{dt} = UA\Delta T_m \tag{3.10}$$

where M = the mass contained in the vessel, kg
C_p = the specific heat capacity of the vessel contents, J/kg°C
T = temperature of the vessel contents, °C
t = time, s
U = the overall heat-transfer coefficient, W/m^2°C
A = heat-transfer area, m^2
ΔT_m = the mean temperature difference, the temperature driving force, °C

The mean temperature difference for heat transfer, ΔT_m, will generally be a function of the temperature of the vessel contents, T, as well as depending on the nature of the heating or cooling medium (isothermal or nonisothermal) and the type of heat transfer surface used. Batch tanks are usually heated or cooled using internal coils, jacketed vessels, or external heat exchangers. Heat transfer to vessels is discussed in more detail in Section 19.18.

In more complex cases, it is usually a good idea to set up a dynamic simulation model of the process. The use of dynamic simulation allows the designer to include additional heat source and sink terms such as losses to the environment. The designer can also use the dynamic model to investigate the interaction between the process, the heat transfer equipment, and the process control system, and hence to develop control algorithms that ensure rapid heating or cooling but do not cause excessive overshoot of the target temperature.

The application of differential energy balances to simple problems is illustrated in Examples 3.7 and 15.6.

3.6.2 Energy Recovery in Batch and Cyclic Processes

Most batch processes operate at relatively low temperatures, below 200 °C, where use of steam or hot oil will give high heat transfer rates for process heating. High heat transfer rates allow shorter heating times and enable use of internal coils and jacketed vessels, reducing the number of pieces of equipment in the plant. If the energy cost is a very small fraction of the total cost of production then recovering heat from the process may not be economically attractive, as the resulting increase in capital cost will not be justified.

Many batch processes need cooling to temperatures that require some degree of refrigeration. Fermentation processes are often operated at temperatures below 40 °C, where use of cooling water can be problematic and chilled water or other refrigerants are used instead. Food processes often require refrigeration or freezing of the product. Recovery of "cooling" from chilled streams is not possible when the product must be delivered in chilled form.

Three of the most commonly used methods for recovering heat in batch and cyclic processes are described below. Energy optimization in batch plants has been the subject of much research, and is discussed in more detail in the papers by Vaselenak, Grossman, and Westerberg (1986), Kemp and

Deakin (1989), and Lee and Reklaitis (1995) and the books by Smith (2005), Kemp (2007), and Majozi (2010).

Semi-continuous Operation

The simplest approach to allow some degree of heat recovery in a batch process is to operate part of the plant in a continuous mode. The use of intermediate accumulation tanks can allow sections of the plant to be fed continuously or to accumulate product for batching into other operations.

Semi-continuous operation is often deployed for feed sterilizers and pasteurizers in food processing and fermentation plants. In a pasteurization operation, the feed must be heated to a target temperature, held at that temperature for long enough to kill unwanted species that may be present in the feed, and then cooled to the process temperature. The high temperature residence time is usually obtained by passing the process fluid through a steam-traced or well-insulated pipe coil. The initial heating of the feed can be accomplished by heat exchange with the hot fluid leaving the coil, allowing the use of a smaller steam heater to reach the target temperature, as shown in Figure 3.30. This design is common in food-processing plants, but care must be taken to ensure that there is no leakage across the heat exchanger, which could potentially lead to contamination of the "sterile" feed with components from the raw feed.

Another situation where semi-continuous operation is often adopted is in the separation section of a batch plant. Some energy-intensive separations such as distillation and crystallization are easier to control to high recovery and tight product specifications when operated in continuous mode. In these cases a surge tank can feed the continuous section of the plant and typical heat recovery schemes such as feed-bottoms heat exchange can be considered.

If a batch plant is designed so that batches are transferred from one vessel to another (as opposed to undergoing successive steps in the same vessel), then heat can be transferred between streams as they are pumped from one vessel to the next. During the pumping operation the flow is at a pseudo-steady state, and a heat exchanger between two streams behaves the same as a heat exchanger in a continuous plant. Figure 3.31 shows such an arrangement in which a hot stream flows from vessel *R1* to vessel *R2*, while a cold stream flows from vessel *R3* to vessel *R4*. The flowing streams exchange heat in a heat exchanger that is shown as being countercurrent, but could equally well be cross-flow or cocurrent if the temperatures were suitable. This arrangement is sometimes referred to as a "countercurrent" heat integration, although it should be stressed that the exchanger can be cocurrent or cross-flow.

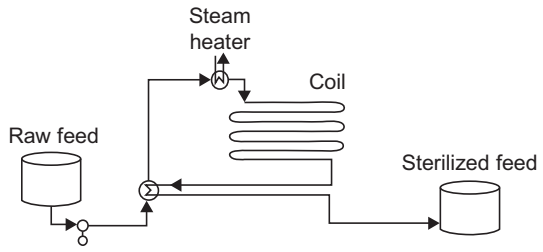

FIGURE 3.30

Heat integration of feed sterilization system.

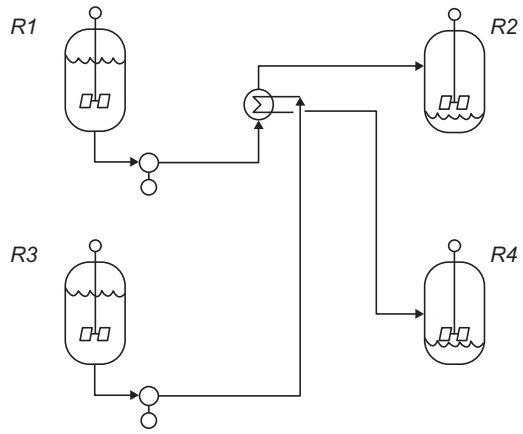

FIGURE 3.31

Stream-to-stream ("countercurrent") heat integration of batch vessels.

When stream-to-stream heat transfer is used, a high degree of heat recovery can be obtained. The exchanger will perform well and maintain roughly constant stream outlet temperatures during the period when the vessels are being pumped out. When the liquid level in the vessels becomes too low for pump operation, the flow rates in the exchanger become too low for the exchanger to function effectively. If batch-to-batch contamination is not important and there are no safety hazards, product quality issues, or fouling concerns, then the exchanger can be isolated ("blocked in") while the remaining tank contents are drained through bypass lines, and the exchanger is then ready to be reused when tanks *R1* and *R3* are again ready to be drained. In the case where batch-to-batch mixing is not desired, or where there are other reasons why the exchanger cannot be left full of process fluid, provision must be made to flush, drain, and clean the exchanger once the upstream tanks are empty.

Sequencing Multiple Batches

If a plant contains several batches that are undergoing different steps of a process at the same time, or if several different batch plants are grouped close to each other, then the batches can sometimes be sequenced so that heat can be transferred from one batch to another.

Suppose a batch process contains the steps of heating reagents, reacting them at a desired temperature and then cooling the products before sending them for further processing. If two reactors are used, a heat exchanger can be employed to exchange heat from the reactor that is being cooled to the reactor that is being heated. For example, in Figure 3.32, hot fluid from vessel *R5* is pumped through an exchanger where it transfers heat to cold fluid that is pumped from vessel *R6*. The fluid from each vessel is returned to the vessel that it came from. The heat exchanger in Figure 3.31 is shown as being countercurrent, but cocurrent or cross-flow heat exchange could be used if the temperatures were appropriate.

The graph on the right of Figure 3.32 is a schematic of the temperature-time profile for both vessels. As time progresses, they become closer in temperature, and would eventually reach thermal equilibrium. In practice, it is usually not economical to run the exchanger for very long times, and heat transfer is

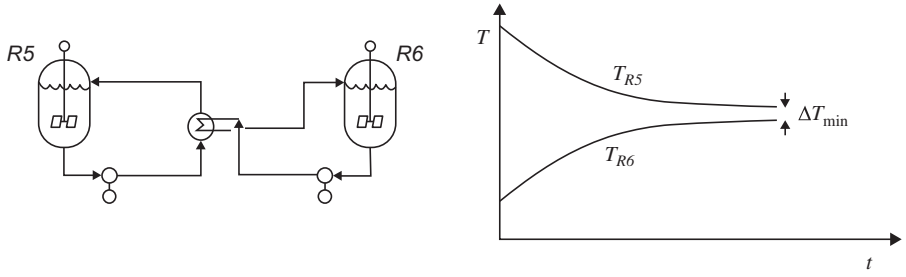

FIGURE 3.32

Tank-to-tank ("cocurrent") heat integration of batch vessels.

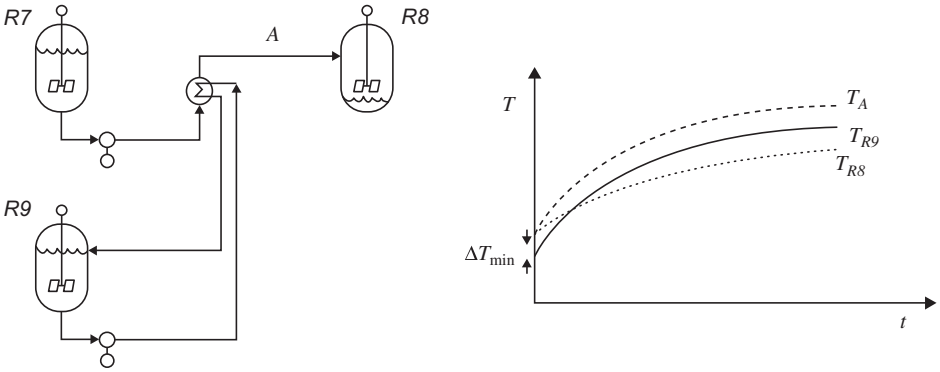

FIGURE 3.33

Stream-to-tank ("cocurrent/countercurrent") heat integration of batch vessels.

stopped when an acceptable minimum temperature difference between the vessels is reached, shown as ΔT_{min} in the figure. Tank-to-tank heat transfer does not allow as efficient heat recovery as stream-to-stream, as the hottest temperatures in the hot tank are matched with the coldest temperatures in the cold tank, as they would be in a cocurrent heat exchanger, hence Vaselenak, *et al.* (1986) named this type of batch heat integration "cocurrent" heat integration. It should again be stressed that the heat exchanger is usually designed to be countercurrent or cross-flow.

An improvement on this scheme is to use stream-to-tank heat transfer, shown in Figure 3.33, in which a stream that is transferred from one vessel to another exchanges heat with a stream that is returned to the tank from which it originated. In Figure 3.33, hot fluid flows from *R7* to *R8* and transfers heat to a cold stream that is pumped from *R9* and returned to *R9*. The graph on the right of Figure 3.33 is a schematic of the temperature behavior of *R9*, *R8*, and the location marked as *A* on the line entering *R8*. The temperature of the cold fluid in *R9* increases over time as heat is transferred to it. The temperature at *A* is the temperature of the hot fluid at the exit of the heat exchanger. The heat exchanger will usually be designed to pinch at the cold end, since the recirculating flow from *R9* can be much greater than the pump-out flow from *R7*. Consequently, the temperature at *A* will be equal to the temperature in *R9* plus the temperature

approach of the heat exchanger, and so the temperature at *A* has a profile offset above the *R9* temperature profile. The temperature in *R8* is the time-averaged integral of the temperature of the feed to the vessel, i.e., the time-averaged integral of the temperature at *A*. Although the fluid entering *R8* becomes hotter with time, it is mixed with an accumulating volume of colder fluid, so the temperature in *R8* does not increase so rapidly as the temperature in *R9*, and *R8* can even be colder than *R9* when the heat transfer is complete. This process is therefore intermediate in thermal efficiency between tank-to-tank heat transfer and stream-to-stream heat transfer. It is sometimes known as "cocurrent/countercurrent" heat integration. The derivation of the equations needed to accurately describe the temperature profiles for this arrangement is given by Vaselenak, *et al.* (1986).

When tank-to-tank or tank-to-stream heat transfer is selected, care must be taken to ensure that the heat exchanger doesn't cause problems when not in use. If the designer anticipates that there could be problems with fouling, corrosion, batch-to-batch contamination, product degradation, safety issues, or any other issue with leaving the exchanger filled, then the design must include means to drain, flush, and clean the exchanger between batches.

When considering the use of stream-to-stream, stream-to-tank, or tank-to-tank heat transfer in a batch process, the designer must ensure that the batch schedules allow both streams to be available at the same time and for a sufficient time to accomplish the desired heat recovery. When draining, flushing, and cleaning of the heat exchanger are necessary, these steps must also be taken into account. For a process that handles multiple batches simultaneously or a site with multiple batch plants, the resulting scheduling problem becomes too large to optimize by hand and numerical methods must be used. See Vaselenak, *et al.* (1986), Kemp and Deakin (1989), and Lee and Reklaitis (1995) for approaches to solving such problems.

Indirect Heat Recovery

An alternative method of heat recovery that can be used in batch processing is to recover heat indirectly through the utility system or using a heat storage system. Although less thermally efficient than process-to-process heat recovery, this method eliminates problems from sequencing of operations.

In indirect heat recovery, heat from a hot process stream is transferred to a utility stream, such as a reservoir of heat-transfer fluid. The heat-transfer fluid can then be used for heating elsewhere in the process. Indirect heat recovery can be used in any of the flow schemes described above, but in all cases the use of an intermediate stream will reduce the thermal efficiency and the amount of heat that can be recovered. Heat storage systems can only be used when there is a large enough temperature difference between the process heat source and process heat sink to allow for the thermal inefficiency of transfer of heat to the storage medium, cooling losses during storage, and transfer of heat to the process heat sink.

Example 3.7: Differential Energy Balance

In the batch preparation of an aqueous solution, the water is first heated to 80 °C in a jacketed, agitated vessel; 1000 Imp. gal. (4545 kg) is heated from 15 °C. If the jacket area is 300 ft^2 (27.9 m^2) and the overall heat-transfer coefficient can be taken as 50 Btu ft^{-2} h^{-1} $°F^{-1}$ (285 W m^{-2} K^{-1}), estimate the heating time. Steam is supplied at 25 psig (2.7 bar).

Solution

The rate of heat transfer from the jacket to the water will be given by Equation 3.10:

$$MC_p \frac{dT}{dt} = UA\Delta T_m \tag{3.10}$$

Since steam is used as the heating medium, the hot side is isothermal and we can write

$$\Delta T_m = T_s - T$$

where T_s = the steam saturation temperature.

Integrating:

$$\int_0^{t_B} dt = \frac{MC_p}{UA} \int_{T_1}^{T_2} \frac{dT}{(T_s - T)}$$

Batch heating time, t_B:

$$t_B = -\frac{MC_p}{UA} \ln \frac{T_s - T_2}{T_s - T_1}$$

For this example,

$$MC_p = 4.18 \times 4545 \times 10^3 \text{ JK}^{-1}$$

$$UA = 285 \times 27 \text{ WK}^{-1}$$

$$T_1 = 15\,^\circ\text{C},\ T_2 = 80\,^\circ\text{C},\ T_s = 130\,^\circ\text{C}$$

$$t_B = -\frac{4.18 \times 4545 \times 10^3}{285 \times 27.9} \ln \frac{130 - 80}{130 - 15}$$

$$= 1990 \text{ s} = 33.2 \text{ min}$$

In this example the heat capacity of the vessel and the heat losses have been neglected for simplicity. They would increase the heating time by 10 to 20 percent.

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NOMENCLATURE

		Dimensions in $MLT\theta$
A	Area	$\mathbf{L}^2$
CP	Stream heat capacity flow rate	$\mathbf{ML}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
CP_c	Stream heat capacity flow rate, cold stream	$\mathbf{ML}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
CP_h	Stream heat capacity flow rate, hot stream	$\mathbf{ML}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
C_p	Specific heat at constant pressure	$\mathbf{L}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
ΣCP_c	Sum of heat capacity flow rates of cold streams	$\mathbf{ML}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
ΣCP_h	Sum of heat capacity flow rates of hot streams	$\mathbf{ML}^2\mathbf{T}^{-2}\boldsymbol{\theta}^{-1}$
COP	Coefficient of performance for a refrigeration cycle	—
COP_h	Coefficient of performance for a heat pump	—
dH_b	Boiler heating rate	$\mathbf{L}^{-2}\mathbf{T}^2$
H	Enthalpy	$\mathbf{ML}^2\mathbf{T}^{-2}$
ΔH	Change in enthalpy	$\mathbf{ML}^2\mathbf{T}^{-2}$
ΔH_{cold}	Heat transfer from cold utility	$\mathbf{ML}^2\mathbf{T}^{-3}$
ΔH_{ex}	Heat transfer in exchanger	$\mathbf{ML}^2\mathbf{T}^{-3}$
ΔH_{hot}	Heat transfer from hot utility	$\mathbf{ML}^2\mathbf{T}^{-3}$
ΔH_n	Net heat required in nth interval	$\mathbf{ML}^2\mathbf{T}^{-3}$
ΔH_{xp}	Cross-pinch heat transfer	$\mathbf{ML}^2\mathbf{T}^{-3}$
$-\Delta H^\circ_c$	Standard heat of combustion	$\mathbf{L}^2\mathbf{T}^{-2}$
ΔH°_f	Standard enthalpy of formation	$\mathbf{L}^2\mathbf{T}^{-2}$
h_g	Specific enthalpy of steam	$\mathbf{L}^2\mathbf{T}^{-2}$
L'	Number of internal loops in network	—
M	Mass	$\mathbf{M}$
m	Mass flow-rate	$\mathbf{MT}^{-1}$
N	Number of cold streams, heat-exchanger networks	—
N'	Number of streams	—
P_{BFW}	Price of boiler feed water	$\$\mathbf{M}^{-1}$

(*Continued*)

		Dimensions in MLT\theta$
P_F	Price of fuel	$\mathbf{\$M^{-1}L^{-2}T^2}$
P_{HPS}	Price of high pressure steam	$\mathbf{\$M^{-1}}$
S	Number of independent branches	—
s_g	Specific entropy	$\mathbf{L^2T^{-2}\theta^{-1}}$
T	Temperature, absolute	$\boldsymbol{\theta}$
T_1	Initial temperature	$\boldsymbol{\theta}$
T_2	Final temperature	$\boldsymbol{\theta}$
T_{act}	Actual stream temperature	$\boldsymbol{\theta}$
T_c	Condenser temperature	$\boldsymbol{\theta}$
T_e	Evaporator temperature	$\boldsymbol{\theta}$
T_{int}	Interval temperature	$\boldsymbol{\theta}$
T_n	Temperature in nth interval	$\boldsymbol{\theta}$
T_{pinch}	Pinch temperature	$\boldsymbol{\theta}$
T_{return}	Return temperature for utility	$\boldsymbol{\theta}$
T_s	Source temperature	$\boldsymbol{\theta}$
T_{supply}	Supply temperature for utility	$\boldsymbol{\theta}$
T_s	Steam saturation temperature	$\boldsymbol{\theta}$
T_t	Target temperature	$\boldsymbol{\theta}$
ΔT_m	Mean temperature difference	$\boldsymbol{\theta}$
ΔT_{min}	Minimum temperature difference (minimum approach) in heat exchanger	$\boldsymbol{\theta}$
ΔT_n	Interval temperature difference	$\boldsymbol{\theta}$
t	Time	$\mathbf{T}$
t_B	Batch heating time	$\mathbf{T}$
U	Overall heat transfer coefficient	$\mathbf{MT^{-3}\theta^{-1}}$
Z_{min}	Minimum number of heat exchangers in network	—
η_B	Boiler efficiency	—

PROBLEMS

3.1. A process heater uses Dowtherm A heat transfer fluid to provide 850 kW of heat. Estimate the annual operating cost of the heater if the Dowtherm evaporator is 80% efficient and the price of natural gas is $4.60/MMBtu. Assume 8000 operating hours per year.

3.2. A site steam system consists of HP steam at 40 bar, MP steam at 18 bar, and LP steam at 3 bar. If natural gas costs $3.50/MMBtu and electricity is worth $0.07/kWh, estimate the cost of steam at each level in $/metric ton.

3.3. Make a rough estimate of the cost of steam per ton, produced from a packaged boiler. 10,000 kg per hour of steam are required at 15 bar. Natural gas will be used as the fuel, calorific value 39 MJ/m^3 (roughly 1 MMBtu/1000 scf). Take the boiler efficiency as 80%. No condensate will be returned to the boiler.

3.4. A crystallization process requires operation at −5° C. The refrigeration system can reject heat to cooling water that is available at 35° C. If a single refrigeration cycle has an efficiency of 60% of Carnot cycle performance then estimate the cost of providing 1 kW of cooling to this process using a single-stage cycle and using a cascaded-two stage cycle (in which the colder cycle rejects heat to the warmer cycle). Electricity costs $0.07/kWh and the cost of cooling water can be neglected.

3.5. A gas produced as a by-product from the carbonization of coal has the following composition, mole %: carbon dioxide 4, carbon monoxide 15, hydrogen 50, methane 12, ethane 2, ethylene 4, benzene 2, balance nitrogen. Using the data given in Appendix C (available online at booksite.Elsevier.com/Towler), calculate the gross and net calorific values of the gas. Give your answer in MJ/m^3, at standard temperature and pressure.

3.6. Determine the pinch temperature and the minimum utility requirements for the process set out below. Take the minimum approach temperature as 15 °C. Devise a heat-exchanger network to achieve maximum energy recovery.

Stream Number	Type	Heat Capacity kW/°C	Source Temp. °C	Target Temp. °C
1	hot	13.5	180	80
2	hot	27.0	135	45
3	cold	53.5	60	100
4	cold	23.5	35	120

3.7. Determine the pinch temperature and the minimum utility requirements for the process set out below. Take the minimum approach temperature as 15 °C. Devise a heat-exchanger network to achieve maximum energy recovery.

Stream Number	Type	Heat Capacity kW/°C	Source Temp. °C	Target Temp. °C
1	hot	10.0	200	80
2	hot	20.0	155	50
3	hot	40.0	90	35
4	cold	30.0	60	100
5	cold	8.0	35	90

3.8. To produce a high purity product two distillation columns are operated in series. The overhead stream from the first column is the feed to the second column. The overhead from the second column is the purified product. Both columns are conventional distillation columns fitted with reboilers and total condensers. The bottom products are passed to other processing units, which do not form part of this problem. The feed to the first column passes through a

preheater. The condensate from the second column is passed through a product cooler. The duty for each stream is summarized below:

No.	Stream	Type	Source Temp. °C.	Target Temp. °C	Duty, kW
1	Feed preheater	cold	20	50	900
2	First condenser	hot	70	60	1350
3	Second condenser	hot	65	55	1100
4	First reboiler	cold	85	87	1400
5	Second reboiler	cold	75	77	900
6	Product cooler	Hot	55	25	30

Find the minimum utility requirements for this process, for a minimum approach temperature of 10 °C.

Note: the stream heat capacity is given by dividing the exchanger duty by the temperature change.

3.9. At what value of the minimum approach temperature does the problem in Example 3.5 become a threshold problem? Design a heat-exchanger network for the resulting threshold problem. What insights does this give into the design proposed in Example 3.5?

CHAPTER

Process Simulation

4

KEY LEARNING OBJECTIVES

- How to use commercial process simulation software to build a process heat and material balance model
- How to select thermodynamic models for prediction of phase equilibrium and stream properties
- How to use user-specified models and components when the simulator does not have what you need
- How to converge flowsheets that include recycles and overcome convergence problems
- How to optimize converged flowsheets

4.1 INTRODUCTION

This chapter addresses the use of process simulation tools in developing an overall process mass and energy balance. A process flow diagram or flowsheet typically includes material balances made over the complete process and each individual unit. Energy balances are also made to determine the energy flows and the utility requirements.

Most flowsheet calculations are carried out using commercial process simulation programs. The process simulation programs contain models for most unit operations as well as thermodynamic and physical property models. All the commercial programs feature some level of custom modeling capability that allows the designer to add models for nonstandard operations.

Many companies developed proprietary flowsheeting programs between 1960 and 1980. The cost of maintaining and updating proprietary software is high; consequently, very few of the proprietary flowsheeting programs are still in use, and most companies now rely entirely on commercially-available software. Each of the commercial process simulation programs has its own unique idiosyncrasies, but they share many common features. The discussion in this chapter addresses general problems of process simulation and flowsheeting rather than software-specific issues. The latter are usually thoroughly documented in the user manuals and online help that come with the software. Examples have been provided in this chapter using both Aspen Plus® (Aspen Technology Inc.) and UniSim™ Design Suite (Honeywell International Inc.). UniSim Design is

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