16 Heat Exchanger Networks II – Energy Targets

The design philosophy started at the heart of the onion with the reactor and moved out to the next layer, the separation and recycle system (Figure 1.7). Acceptance of the major processing steps (reactors, separators and recycles) in the inner two layers of the onion fixes the material and energy balance. Thus, the heating and cooling duties for the next layer of the onion, the heat recovery system, are known. However, completing the design of the heat exchanger network is not necessary in order to assess the completed design. *Targets* can be set for the heat exchanger network to assess the performance of the complete process design without actually having to carry out the network design. These targets allow both energy and capital cost for the heat exchanger network to be assessed. Moreover, the targets allow the designer to suggest process changes for the reactor and separation and recycle systems to improve the targets for energy and capital cost of the heat exchanger network.

Using targets for the heat exchanger network, rather than designs, allows many design options for the overall process to be screened quickly and conveniently. Screening many design options by completed designs is usually simply not practical in terms of the time and effort required. First consider the details of how to set energy targets. Capital cost targets will be considered in the next chapter. In later chapters, energy targets will be used to suggest design improvements to the reaction, separation and recycle systems.

16.1 COMPOSITE CURVES

The analysis of the heat exchanger network first identifies sources of heat (termed *hot streams*) and sinks (termed *cold streams*) from the material and energy balance. Consider first a very simple problem with just one hot stream (heat source) and one cold stream (heat sink). The initial temperature (termed *supply temperature*), final temperature (termed *target temperature*) and enthalpy change of both streams are given in Table 16.1.

Steam is available at 180°C and cooling water at 20°C. Clearly, it is possible to heat the cold stream using steam and cool the hot stream, in Table 16.1, using cooling water. However, this would incur excessive energy cost. It is also incompatible with the goals of sustainable industrial activity, which call for use of the minimum energy consumption. Instead, it is preferable to try to recover the heat between process streams, if this is possible. The scope for heat recovery can be determined by plotting both streams from Table 16.1 on temperature-enthalpy axes. For feasible heat exchange between the two streams, the hot stream must at all points be hotter than the cold stream. Figure 16.1a shows the temperature-enthalpy plot for this problem with a minimum temperature difference (ΔT_{min}) of 10°C. The region of overlap between the two streams in Figure 16.1a determines the amount of heat recovery possible (for $\Delta T_{min} = 10$°C). For this problem, the heat recovery (Q_{REC}) is 11 MW. The part of the cold stream that extends beyond the start of the hot stream in Figure 16.1a cannot be heated by recovery and requires steam. This is the minimum hot utility or *energy target* (Q_{Hmin}), which for this problem is 3 MW. The part of the hot stream that extends beyond the start of the cold stream in Figure 16.1a cannot be cooled by heat recovery and requires cooling water. This is the minimum cold utility (Q_{Cmin}), which for this problem is 1 MW. Also shown at the bottom of Figure 16.1a is the arrangement of heat exchangers corresponding with the temperature-enthalpy plot.

The temperatures or enthalpy change for the streams (and hence their slope) cannot be changed, but the relative position of the two streams can be changed by moving them horizontally relative to each other. This is possible since the reference enthalpy for the hot stream can be changed independently from the reference enthalpy for the cold stream. Figure 16.1b shows the same two streams moved to a different relative position such that ΔT_{min} is now 20°C. The amount of overlap between the streams is reduced (and hence heat recovery is reduced) to 10 MW. A greater amount of the cold stream now extends beyond the start of the hot stream, and hence the amount of steam is increased to 4 MW. Also, more of the hot stream extends beyond the start of the cold stream, increasing the cooling water demand to 2 MW. Thus, the approach of plotting a hot and a cold stream on the same temperature-enthalpy axes can determine hot and cold utility for a given value of ΔT_{min}.

Table 16.1 Two-stream heat recovery problem.

Stream	Type	Supply temperature T_S (°C)	Target temperature T_T (°C)	ΔH (MW)
1	Cold	40	110	14
2	Hot	160	40	−12

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 ISBNs: 0-471-48680-9 (HB); 0-471-48681-7 (PB)

with 2 mm wall thickness. The tube pitch can be assumed to be $1.25d_O$ with a square configuration. The length to shell diameter can be assumed to be 5. The physical property data are given in Table 15.19.

Table 15.19 Physical property data for isopropanol and water.

Property	Isopropanol (83°C)	Water (30°C)
Density ($kg{\cdot}m^{-3}$)	732	996
Liquid heat capacity ($J{\cdot}kg^{-1}{\cdot}K^{-1}$)	3370	4180
Viscosity ($N{\cdot}s{\cdot}m^{-2}$)	0.502×10^{-3}	0.797×10^{-3}
Thermal conductivity ($W{\cdot}m^{-1}{\cdot}K^{-1}$)	0.131	0.618
Heat of vaporization ($J{\cdot}kg^{-1}$)	678,000	–

Assume the fouling coefficients to be 10,000 $W{\cdot}m^{-2}{\cdot}K^{-1}$ and 5000 $W{\cdot}m^{-2}{\cdot}K^{-1}$ for isopropanol and cooling water respectively. For an assumed cooling water velocity of 1 $m{\cdot}s^{-1}$, estimate the heat transfer area for:

a. a horizontal condenser with shell-side condensation.

b. a vertical condenser with tube-side condensation.

12. For the case of vertical tube-side condensation from Exercise 11, the condensate is to be subcooled to 45°C. By dividing the condenser into two zones for condensation and subcooling, estimate the heat transfer area.
13. A reboiler of a distillation column is required to supply 10 $kg{\cdot}s^{-1}$ of toluene vapor. The column operating pressure at the bottom of the column is 1.6 bar. At this pressure, the toluene vaporizes at 127°C and can be assumed to be isothermal. Steam at 160°C is to be used for the vaporization. The latent heat of vaporization of toluene is 344,000 $J{\cdot}kg^{-1}$, the critical pressure is 40.5 bar and critical temperature is 594 K. Steel tubes with 30 mm outside diameter, 2 mm wall thickness and length 3.95 m are to be used. The film coefficient (including fouling) for the condensing steam can be assumed to be 5700 $W{\cdot}m^{-2}{\cdot}K^{-1}$. Estimate the heat transfer area for:
    a. kettle reboiler.
    b. vertical thermosyphon reboiler.
14. The purge gas from a petrochemical process is at 25°C and contains a mole fraction of methane of 0.6, the balance being hydrogen. This purge gas is to be burnt in a furnace to provide heat to a process with a cold stream pinch temperature of 150°C (ΔT_{min} = 50°C). Ambient temperature is 10°C.
    a. Calculate the theoretical flame temperature if 15% excess air is used in the combustion. Standard heats of combustion are given in Table 15.11 and mean molar heat capacities in Table 15.15.
    b. Calculate the furnace efficiency.
    c. Suggest ways in which the furnace efficiency could be improved.

REFERENCES

1. Kern DQ (1950) *Process Heat Transfer*, McGraw-Hill.
2. Hewitt GF (1992) *Handbook of Heat Exchangers Design*, Begell House Inc.
3. Hewitt GF, Shires GL and Bott TR (1994) *Process Heat Transfer*, CRC Press Inc.
4. Sinnott RK (1996) *Chemical Engineering, Volume 6 Chemical Engineering Design*, Butterworth Heinemann.
5. Nie X-R (1998) *Optimisation Strategies for Heat Exchanger Network Design Considering Pressure Drop Aspects*, PhD Thesis, UMIST, UK.
6. Bowman RA, Mueller AC and Nagle WM (1940) Mean Temperature Differences in Design, *Trans ASME*, **62**: 283.
7. Ahmad S, Linnhoff B and Smith R (1988) Design of Multipass Heat Exchangers: an Alternative Approach, *Trans ASME J Heat Transfer*, **110**: 304.
8. Ahmad S, Linnhoff B and Smith R (1990) Cost Optimum Heat Exchanger Networks II Targets and Design for Detailed Capital Cost Models, *Comp Chem Eng*, **7**: 751.
9. Zhu XX, Zanfir M and Klemes J (2000) Heat Transfer Enhancement for Heat Exchanger Network Retrofit, *Heat Transfer Eng*, **21**: 7.
10. Polley GT, Reyes Athie CM and Gough M (1992) Use of Heat Transfer Enhancement in Process Integration, *Heat Recovery Syst CHP*, **12**: 191.
11. Nusselt W (1916) Die Oberflachenkondensation des Wasserdampfes, *Z Ver Deut Ing*, **60**: 541, 569.
12. Rosenow WM (1956) Heat Transfer and Temperature Distribution in Laminar Film Condensation, *Trans ASME*, **78**: 1645.
13. Rosenow WM, Webber JH and Ling AT (1956) Effect of Velocity on Laminar and Turbulent Film Condensation, *Trans ASME*, **78**: 1645.
14. Mueller AC in Hewitt GF (1992) *Handbook of Heat Exchanger Design*, Begell House Inc.
15. Palen JW in Hewitt GF (1992) *Handbook of Heat Exchanger Design*, Begell House Inc.
16. Mostinski IL (1963) Calculation of Boiling Heat Transfer Coefficients, Based on the Law of Corresponding States, *Br Chem Eng*, **8**: 580.
17. Frank O and Prickett RD (1973) Design of Vertical Thermosyphon Reboilers, *Chem Eng*, **3**: 107.
18. Hougen OA, Watson KM and Ragatz RA (1954) *Chemical Process Principles Part I Material and Energy Balances*, 2nd Edition, John Wiley.

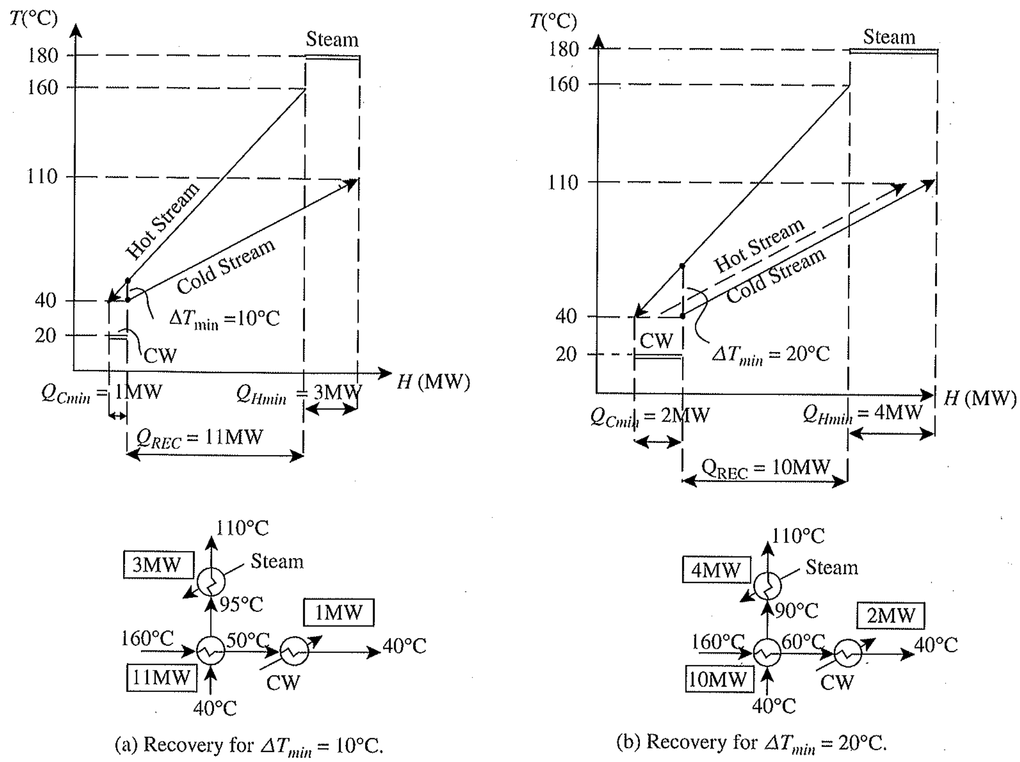


Figure 16.1 A simple heat recovery problem with one hot stream and one cold stream.

The importance of ΔT_{min} is that it sets the relative location of the hot and cold streams in this two-stream problem, and therefore the amount of heat recovery. Setting the value of ΔT_{min} or Q_{Hmin} or Q_{Cmin} sets the relative location and the amount of heat recovery.

Consider now the extension of this approach to several hot streams and cold streams. Figure 16.2 shows a simple flowsheet. Flowrates, temperatures and heat duties for each stream are shown. Two of the streams in Figure 16.2 are sources of heat (hot streams) and two are sinks for heat (cold streams). Assuming that the heat capacities are constant, the data for the hot and cold streams can be extracted as given in Table 16.2. Note that the heat capacities (CP) are total heat capacities, being the product of mass flowrate and specific heat capacity ($CP = m \cdot C_P$). Had the heat capacities varied significantly, the nonlinear temperature-enthalpy behavior could have been represented by a series of linear *segments* (see Chapter 19).

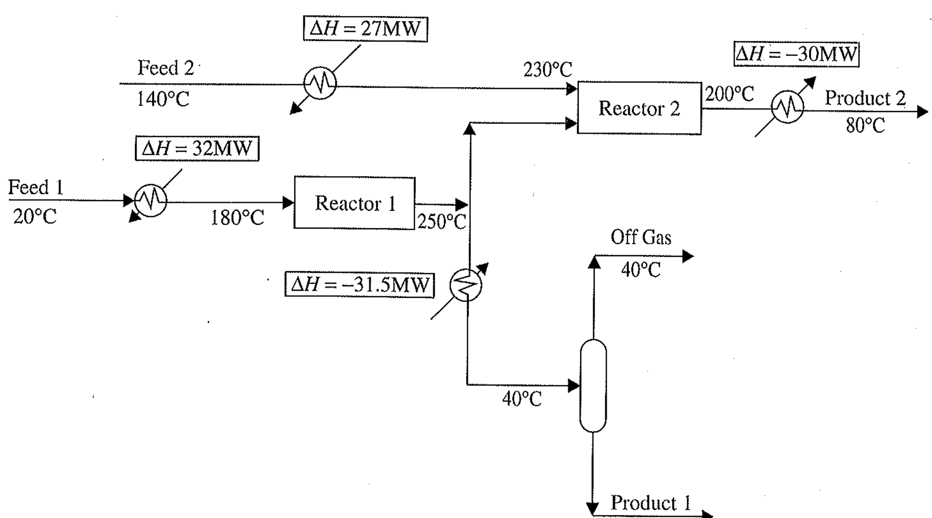


Figure 16.2 A simple flowsheet with two hot streams and two cold streams.

Table 16.2 Heat exchange stream data for the flowsheet Figure 16.2.

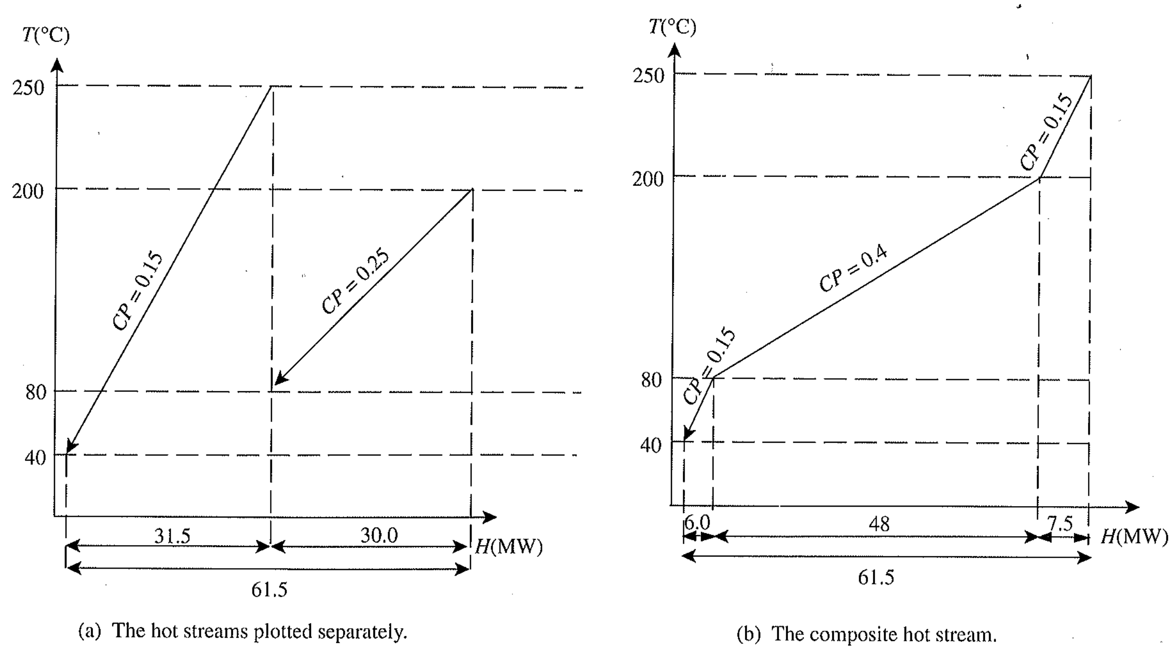
Stream	Type	Supply temperature T_S (°C)	Target temperature T_T(°C)	ΔH (MW)	Heat capacity flowrate CP(MW·K^{-1})
1. Reactor 1 feed	Cold	20	180	32.0	0.2
2. Reactor 1 product	Hot	250	40	−31.5	0.15
3. Reactor 2 feed	Cold	140	230	27.0	0.3
4. Reactor 2 product	Hot	200	80	−30.0	0.25

Instead of dealing with individual streams as given in Table 16.1, an overview of the process is needed. Figure 16.3a shows the two hot streams individually on temperature-enthalpy axes. How these hot streams behave overall can be quantified by combining them in the given temperature ranges[1,2,3]. The temperature ranges in question are defined by changes in the overall rate of change of enthalpy with temperature. If heat capacities are constant, then changes will occur only when streams start or finish. Thus, in Figure 16.3, the temperature axis is divided into ranges defined by the supply and target temperatures of the streams.

Within each temperature range, the streams are combined to produce a composite hot stream. This composite hot stream has a CP in any temperature range that is the sum of the individual streams. Also, in any temperature range, the enthalpy change of the composite stream is the sum of the enthalpy changes of the individual streams. Figure 16.3b shows the *composite curve* of the hot streams[1,2,3]. The composite hot stream is a single stream that is equivalent to the individual hot streams in terms of temperature and enthalpy. Similarly, the composite curve of the cold streams for the problem can be produced, as shown in Figure 16.4. Again, the composite cold stream is a single stream that is equivalent to the individual cold streams in terms of temperature and enthalpy.

The composite hot and cold curves can now be plotted on the same axes, as in Figure 16.5. Plotting the composite hot and cold curves is analogous to plotting the single hot and cold streams in Figure 16.1. The composite curves in Figure 16.5a are set to have a minimum temperature difference (ΔT_{min}) of 10°C. Where the curves overlap in Figure 16.5a, heat can be recovered vertically from the hot streams that comprise the hot composite curve into the cold streams that comprise the cold composite curve. The way in which the composite curves are constructed (i.e. monotonically decreasing hot composite curve and monotonically increasing cold composite curve) allows maximum overlap between the curves and hence maximum heat recovery. Maximizing the energy recovery thereby minimizes the external requirements for heating and cooling duties and minimizes the energy consumption. In this problem, for ΔT_{min} = 10°C, the maximum heat recovery (Q_{REC}) is 51.5 MW.

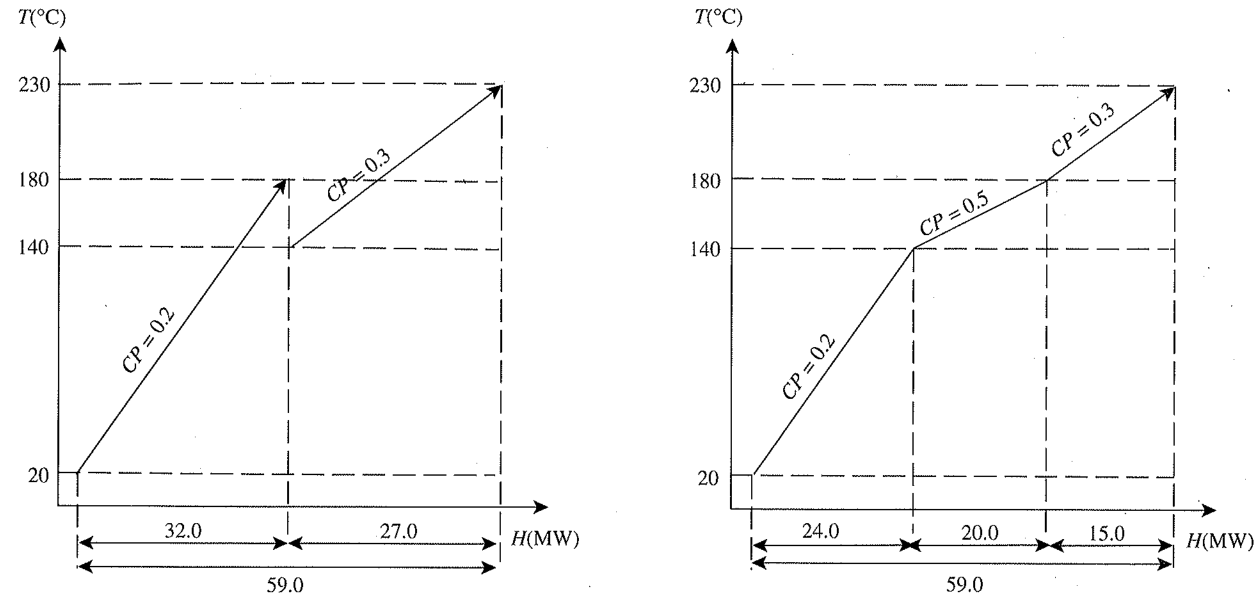
Where the cold composite curve extends beyond the start of the hot composite curve in Figure 16.5a, heat recovery is not possible, and the cold composite must be supplied with external hot utility such as steam. This represents the target for hot utility (Q_{Hmin}). For this problem, with ΔT_{min} = 10°C, Q_{Hmin} = 7.5 MW. Where the hot composite curve



(a) The hot streams plotted separately.

(b) The composite hot stream.

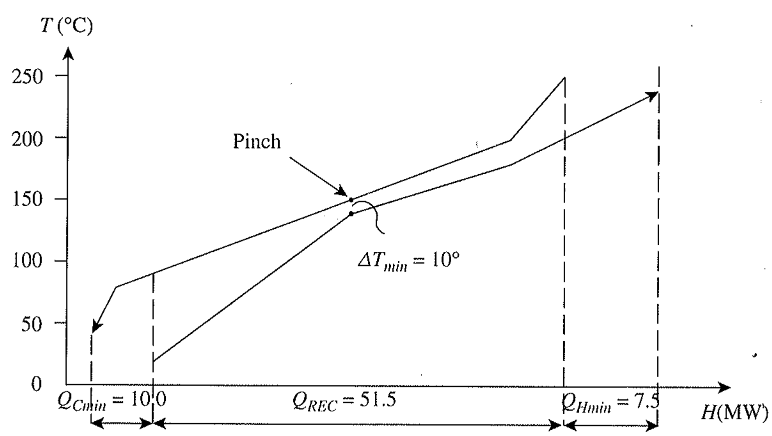
Figure 16.3 The hot streams can be combined to obtain a composite stream.



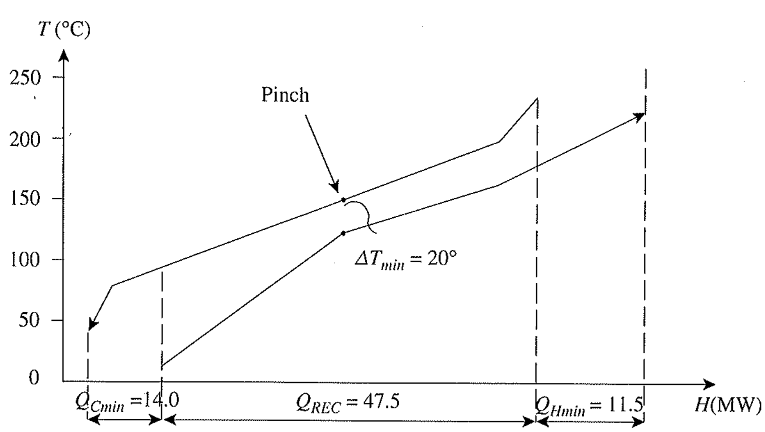
(a) The cold streams plotted separately.

(b) The composite cold stream.

Figure 16.4 The cold streams can be combined to obtain a composite cold stream.



(a) The hot and cold composite curves plotted together at $\Delta T_{min} = 10°C$.



(b) Increasing ΔT_{min} from 10°C to 20°C increases the hot and cold utility targets.

Figure 16.5 Plotting the hot and cold composite curves together allows the targets for hot and cold utility to be obtained.

extends beyond the start of the cold composite curve in Figure 16.5a, heat recovery is again not possible and the hot composite curve must be supplied with external cold utility such as cooling water. This represents the target for cold utility (Q_{Cmin}). For this problem, setting $\Delta T_{min} = 10°C$ gives $Q_{Cmin} = 10.0$ MW.

Specifying the hot utility or cold utility heat duty or ΔT_{min} fixes the relative position of the two curves. As with the simple problem in Figure 16.1, the relative position of the two curves is a degree of freedom[4]. Again, the relative position of the two curves can be changed by moving them horizontally relative to each other. Clearly, to consider heat recovery from hot streams into cold streams, the hot composite curve must be in a position such that it is always above the cold composite curve for feasible heat transfer. Thereafter, the relative position of the curves can be chosen. Figure 16.5b shows the curves with $\Delta T_{min} = 20°C$. The hot and cold utility targets are now increased to 11.5 MW and 14 MW respectively.

Figure 16.6 illustrates what happens to the cost of the system as the relative position of the composite curves is changed over a range of values of ΔT_{min}. When the curves just touch, there is no driving force for heat transfer at one point in the process, which would require infinite heat transfer area and hence infinite capital cost. As the energy target (and hence ΔT_{min} between the curves) is increased, the capital cost decreases. This results from increased temperature differences throughout the process, decreasing the heat transfer area. On the other hand, the energy cost increases as ΔT_{min} increases. There is a trade-off between energy and capital cost and an economic amount of energy recovery. Later, it will be shown how this trade-off can be carried out using energy and capital cost targets.

However, care should be taken not to ignore practical constraints when setting ΔT_{min}. To achieve a small ΔT_{min} in a design requires heat exchangers that exhibit pure countercurrent flow. With shell-and-tube heat exchangers this is not possible, even if single-shell pass and single-tube pass designs are used, because the shell-side stream takes periodic cross-flow. Consequently, operating with a ΔT_{min} less than 10°C should be avoided, unless under special circumstances[5]. A smaller value of 5°C or less can be achieved with plate heat exchangers, and the value can go as low as 1 to 2°C with plate-fin designs[5]. It should be noted, however, that such constraints only apply to the exchangers that operate around the point of closest approach between the composite curves. Additional constraints apply if vaporization or condensation is occurring at the point of closest approach (see Chapter 15).

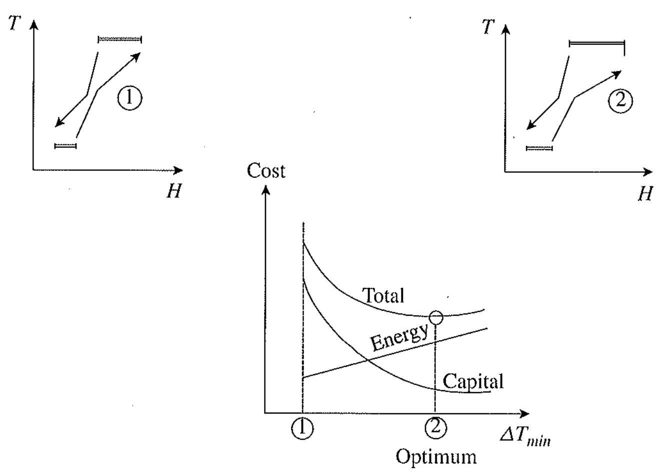


Figure 16.6 The correct setting for ΔT_{min} is fixed economic trade-offs.

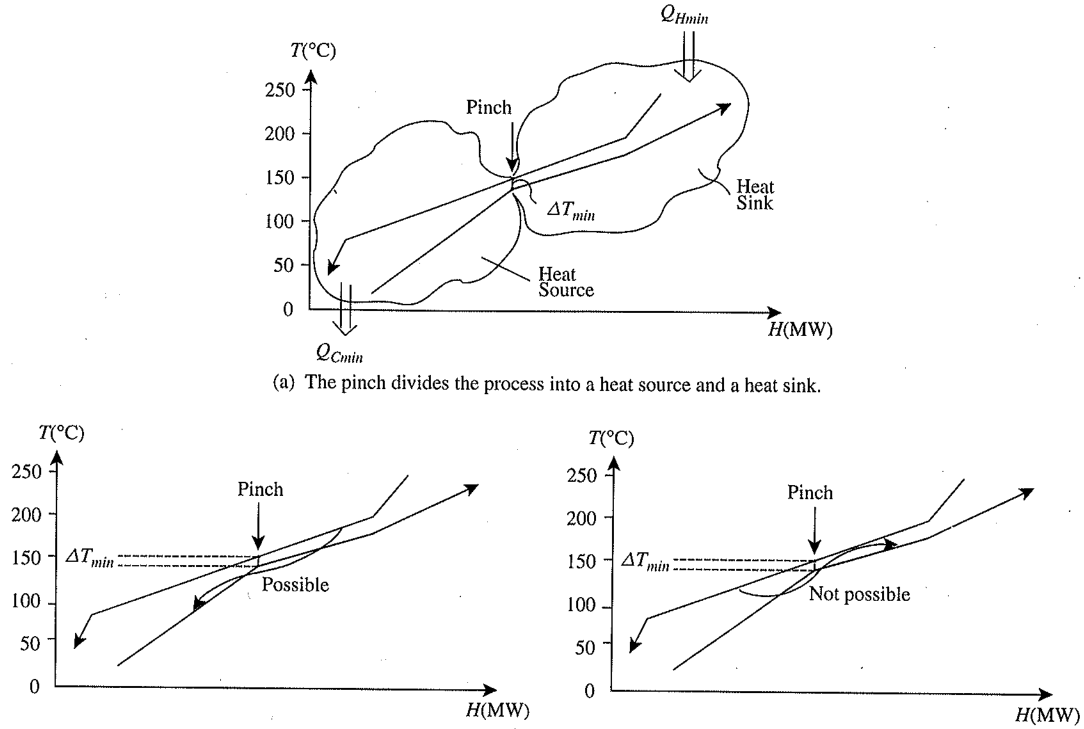
16.2 THE HEAT RECOVERY PINCH

As discussed above, the correct setting for the composite curves is determined by an economic trade-off between energy and capital, corresponding to an economic minimum temperature difference between the curves, ΔT_{min}. Accepting for the moment that the correct economic ΔT_{min} is known, this fixes the relative position of the composite curves and hence the energy target. The value of ΔT_{min} and its location between the composite curves have important implications for design, if the energy target is to be achieved in the design of a heat exchanger network. The ΔT_{min} is normally observed at only one point between the hot and the cold composite curves, called the *heat recovery pinch*[3,6–8]. The pinch point has a special significance.

The trade-off between energy and capital in the composite curves suggests that, "on average", individual exchangers should have a temperature difference no smaller than ΔT_{min}. A good initialization in heat exchanger network design is to assume that no individual heat exchanger has a temperature difference smaller than the ΔT_{min} between the composite curves.

With this rule in mind, divide the process at the pinch as shown in Figure 16.7a. Above the pinch (in temperature terms) the process is in heat balance with the minimum hot utility, Q_{Hmin}. Heat is received from hot utility and no heat is rejected. The process acts as a heat sink. Below the pinch (in temperature terms), the process is in heat balance with the minimum cold utility, Q_{Cmin}. No heat is received but heat is rejected to cold utility. The process acts as a heat source.

Consider now the possibility of transferring heat between these two systems. Figure 16.7b shows that it is possible to transfer heat from hot streams above the pinch to cold streams below it. The pinch temperature for hot streams for the problem is 150°C and for cold streams 140°C. Transfer of heat from above the pinch to below, as shown in Figure 16.7b, means transfer of heat from hot streams with a temperature of 150°C or greater into cold streams with a temperature of 140°C or less. This is clearly possible. By contrast, Figure 16.7c shows that heat transfer from hot streams below the pinch to cold streams



(a) The pinch divides the process into a heat source and a heat sink.

(b) Heat transfer from above the pinch to below the pinch is possible.

(c) Heat transfer from below to above the pinch is not possible without violating ΔT_{min}.

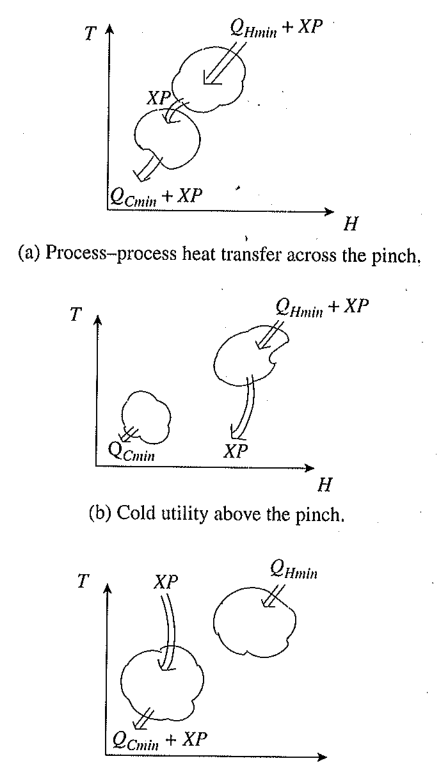
Figure 16.7 The composite curves set the energy target and the location of the pinch.

above is not possible. Such transfer requires heat being transferred from hot streams with a temperature of 150°C or less into cold streams with a temperature of 140°C or greater. This is clearly not possible (without violating the ΔT_{min} constraint).

If an amount of heat XP is transferred from the system above the pinch to the system below the pinch, as in Figure 16.8a, this will create a deficit of heat XP above the pinch and an additional surplus of heat XP below the pinch. The only way this can be corrected is by importing an extra XP amount of heat from hot utility and exporting an extra XP amount of heat to cold utility[3,4].

Analogous effects are caused by the inappropriate use of utilities. Utilities are appropriate if they are necessary to satisfy the enthalpy imbalance in that part of the process. Above the pinch, hot utility (in this case, steam) is needed to satisfy the enthalpy imbalance. Figure 16.8b illustrates what happens if inappropriate use of utilities is made. If cooling to cold utility XP is used to cool hot streams above the pinch, this creates an enthalpy imbalance in the system above the pinch. To satisfy the enthalpy imbalance above the pinch, an import of $(Q_{Hmin} + XP)$ heat from hot utility is required. Overall, $(Q_{Cmin} + XP)$ of cold utility is used[3,4].

Another inappropriate use of utilities involves heating of some of the cold streams below the pinch by hot utility (steam in this case). Below the pinch, cold utility is needed to satisfy the enthalpy imbalance. Figure 16.8c illustrates



(a) Process–process heat transfer across the pinch.

(b) Cold utility above the pinch.

(c) Hot utility below the pinch.

Figure 16.8 Three forms of cross pinch heat transfer.

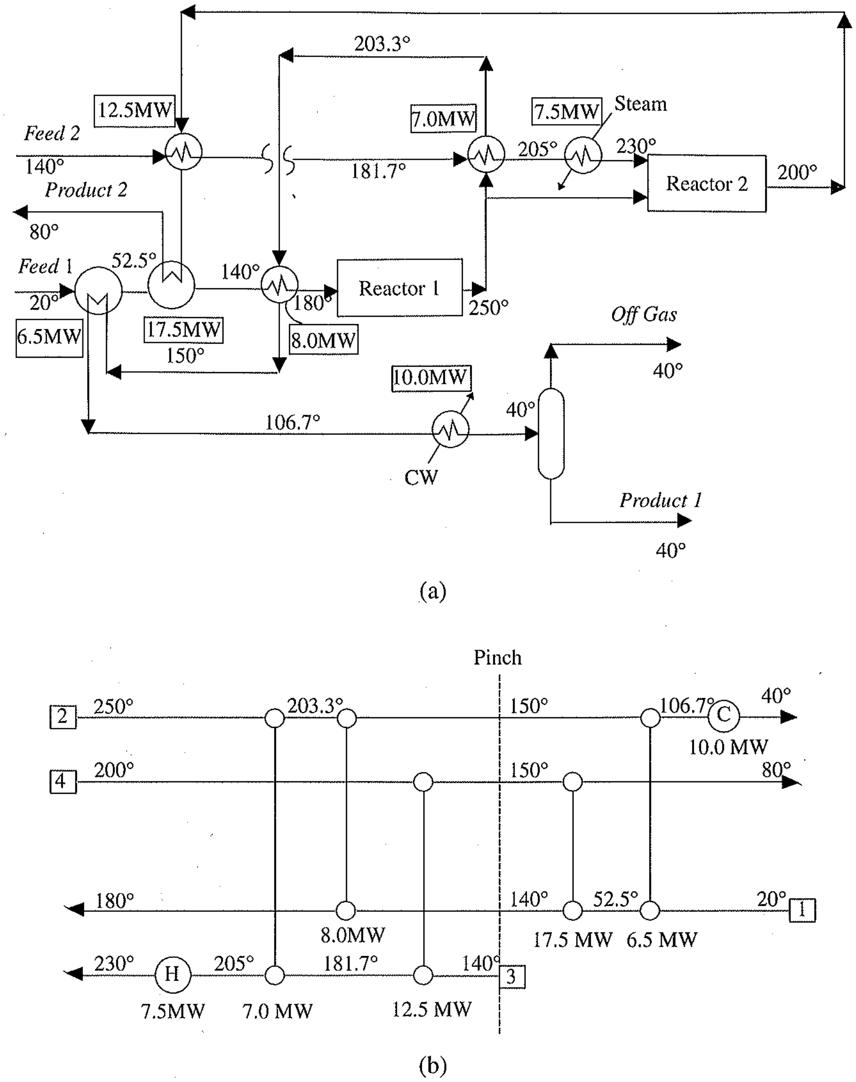


Figure 16.9 A design that achieves the energy target.

what happens if an amount of heat *XP* from hot utility is used below the pinch. Q_{Hmin} must still be supplied above the pinch to satisfy the enthalpy imbalance above the pinch. Overall, $(Q_{Hmin} + XP)$ of steam is used and $(Q_{Hmin} + XP)$ of cooling water[3,4].

In other words, to achieve the energy target set by the composite curves, the designer must not transfer heat across the pinch by[3,4]:

a. Process-to-process heat transfer
b. Inappropriate use of utilities

These rules are both necessary and sufficient to ensure that the target is achieved, providing that the initialization rule is adhered to that no individual heat exchanger should have a temperature difference smaller than ΔT_{min}.

Figure 16.9a shows a design corresponding to the flowsheet in Figure 16.2, which achieves the target of $Q_{Hmin} = 7.5$ MW and $Q_{Cmin} = 10$ MW for $\Delta T_{min} = 10°$C. Figure 16.9b shows an alternative representation of the flowsheet in Figure 16.9a, known as the *grid diagram*[9]. The grid diagram shows only heat transfer operations. Hot streams are at the top running left to right. Cold streams are at the bottom running right to left. A heat exchange match is represented by a vertical line joining two circles on the two streams being matched. An exchanger using hot utility is represented by a circle with an "H". An exchanger using cold utility is represented by a circle with a "C". The importance of the grid diagram is clear in Figure 16.9b, since the pinch, and how it divides the process into two parts, is easily accommodated. Dividing the process into two parts on a conventional diagram such as shown in Figure 16.9a is both difficult and extremely cumbersome.

Details of how the design in Figure 16.9 was developed are explained in Chapter 18. For now, simply take note that the targets set by the composite curves are achievable in design, providing that the pinch is recognized and there is no transfer of heat across the pinch, either process-to-process or through inappropriate use of utilities. However, the insight of the pinch is needed to analyze some of the important decisions still to be made before the network design is addressed.

16.3 THRESHOLD PROBLEMS

Not all problems have a pinch to divide the process into two parts[4]. Consider the composite curves in Figure 16.10a. At this setting, both steam and cooling water are required. As the composite curves are moved closer together, both the steam and cooling water requirements decrease until the setting shown in Figure 16.10b. At this setting, the composite curves are in alignment at the hot end, indicating that there is no longer a demand for hot utility. Moving the curves closer together as shown in Figure 16.10c, decreases the cold utility demand at the cold end but opens up a demand for cold utility at the hot end corresponding with the decrease at the cold end. In other words, as the curves are moved closer together, beyond the setting in Figure 16.10b, the utility demand is constant. The setting shown in Figure 16.10b marks a threshold, and problems that exhibit this feature are known as *threshold problems*[4]. In some threshold problems, the hot utility requirement disappears as in Figure 16.10. In others, the cold utility disappears as shown in Figure 16.11.

Considering the capital-energy trade-off for threshold problems, there are two possible outcomes as shown in Figure 16.12. Below the threshold ΔT_{min}, energy costs are constant, since utility demand is constant. Figure 16.12a shows a situation where the optimum occurs at the threshold ΔT_{min}. Figure 16.12b shows a situation where the optimum occurs above the threshold ΔT_{min}. The flat profile of energy costs below the threshold ΔT_{min} means that the optimum can never occur below the threshold value. It can only be at or above the threshold value.

In a situation, as shown in Figure 16.12a, with the optimum ΔT_{min} at the threshold, there is no pinch. On the other hand, in a situation as shown in Figure 16.12b with the optimum above the threshold value, there is a demand for both utilities and there is a pinch.

It is interesting to note that threshold problems are quite common in practice and although they do not have a process pinch, pinches are introduced into the design when multiple utilities are added. Figure 16.13a shows composite curves similar to the composite curves from Figure 16.10 but with two levels of cold utility used instead of one. In this case, the second cold utility is steam generation. The introduction of this second utility causes a pinch. This is known as a *utility pinch* since it is caused by the introduction of an additional utility[4].

Figure 16.13b shows composite curves similar to those from Figure 16.11, but with two levels of steam used. Again, the introduction of a second steam level causes a utility pinch.

In design, the same rules must be obeyed around a utility pinch as a process pinch. Heat should not be transferred across it by process-to-process transfer and there should

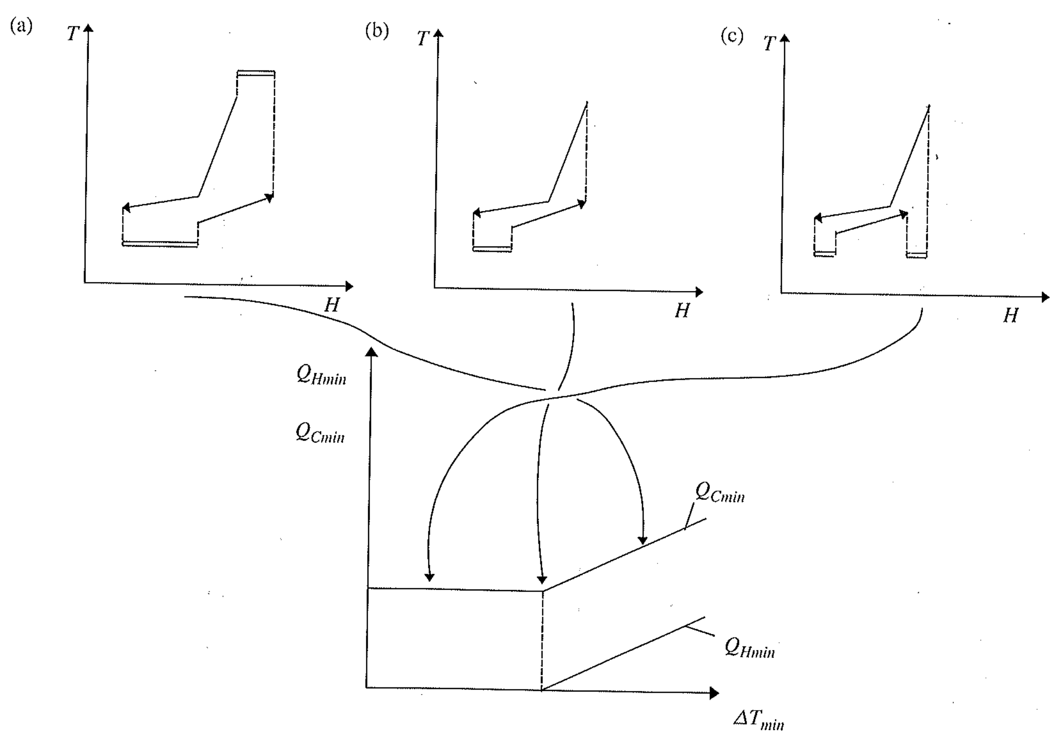


Figure 16.10 As ΔT_{min} is varied, some problems require only cold utility below a threshold value.

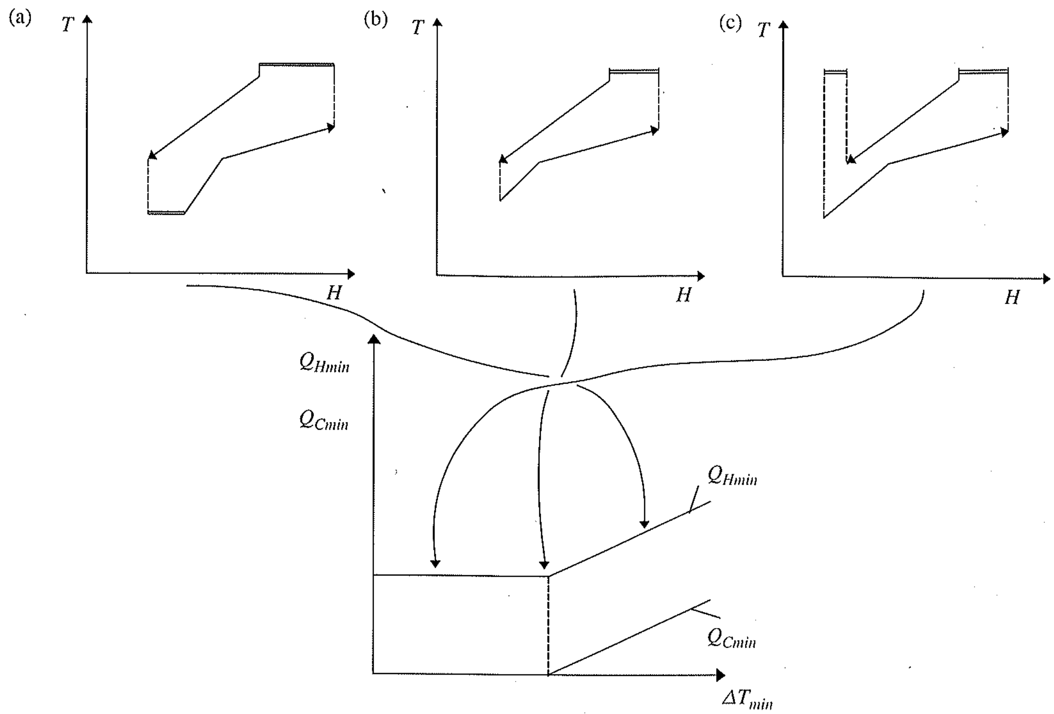


Figure 16.11 In some threshold problems, only hot utility is required below the threshold value of ΔT_{min}.

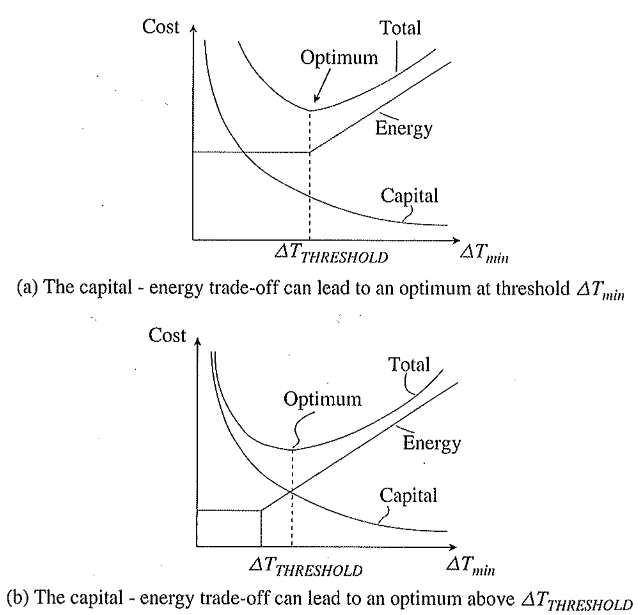


Figure 16.12 The optimum setting of the capital/energy trade off for threshold problems.

be no inappropriate use of utilities. In Figure 16.13a, this means that the only utility to be used above the utility pinch is steam generation and only cooling water below. In Figure 16.13b, this means that the only utility to be used above the utility pinch is high-pressure steam and only low-pressure steam below.

16.4 THE PROBLEM TABLE ALGORITHM

Although composite curves can be used to set energy targets, they are inconvenient since they are based on a graphical construction. A method of calculating energy targets directly without the necessity of graphical construction can be developed[1,9]. The process is first divided into temperature intervals in the same way as was done for construction of the composite curves. Figure 16.14a shows that it is not possible to recover all of the heat in each temperature interval since temperature driving forces are not feasible throughout the interval. Some heat recovery is possible, but all of the heat cannot be recovered. The amount that can be recovered depends on the relative slopes of the two curves in the temperature interval. This problem can be overcome if, purely for the purposes of construction, the hot composite is shifted to be $\Delta T_{min}/2$ colder than it is in practice and that the cold composite is shifted to be $\Delta T_{min}/2$ hotter than it is in practice as shown in Figure 16.14b. The *shifted composite curves* now touch at the pinch. Carrying out a heat balance between the shifted composite curves within a *shifted temperature interval* shows that heat transfer is feasible throughout each shifted temperature interval, since hot streams in practice are actually $\Delta T_{min}/2$ hotter and cold streams $\Delta T_{min}/2$ colder. Within each shifted interval, the hot streams are in reality hotter than the cold streams by ΔT_{min}.

It is important to note that shifting the curves vertically does not alter the horizontal overlap between the curves. It therefore does not alter the amount by which the

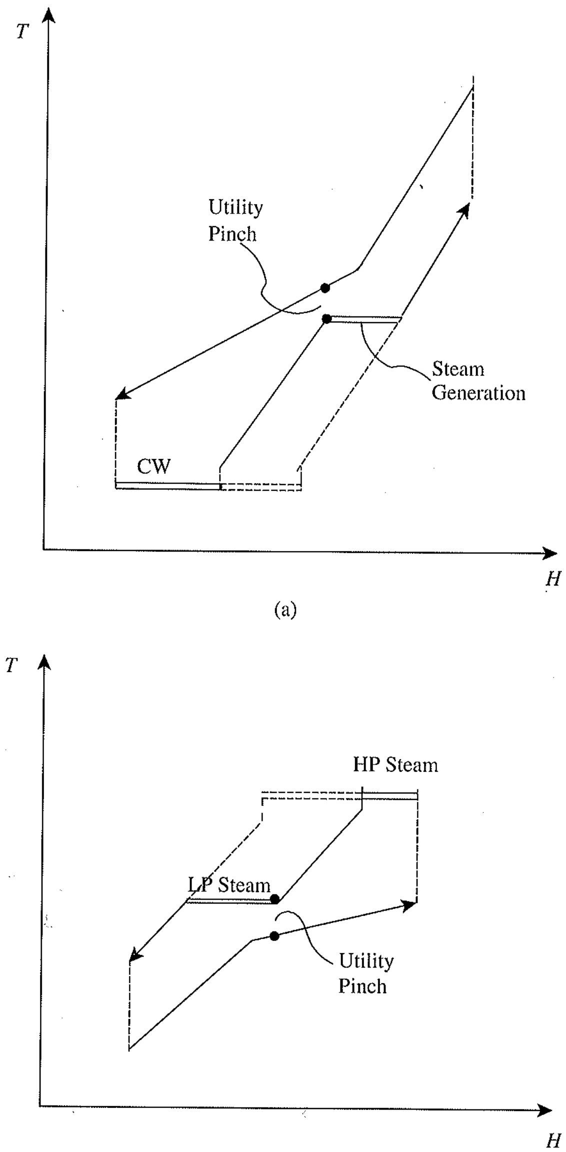
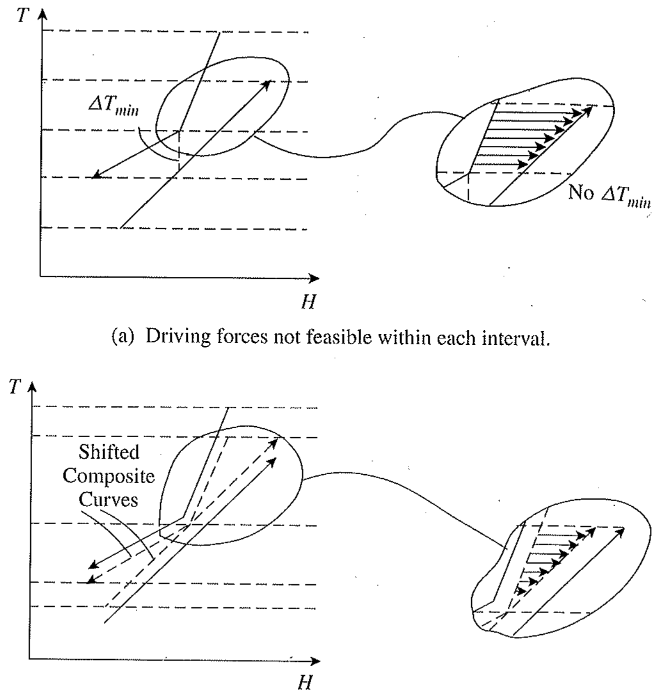


Figure 16.13 Threshold problems are turned into pinched problem when additional utilities are added.



(a) Driving forces not feasible within each interval.

(b) Heat transfer within temperature intervals now feasible.

Figure 16.14 Shifting the composite curves in temperature allows complete heat recovery within temperature intervals.

cold composite curve extends beyond the start of the hot composite curve at the hot end of the problem. Also, it does not alter the amount by which the hot composite curve extends beyond the start of the cold composite curve at the cold end. The shift simply removes the problem of ensuring temperature feasibility within temperature intervals.

This shifting technique can be used to develop a strategy to calculate the energy targets without having to construct composite curves[1,9]:

1. Set up shifted temperature intervals from the stream supply and target temperatures by subtracting $\Delta T_{min}/2$ from the hot streams and adding $\Delta T_{min}/2$ to the cold streams (as in Figure 16.14b).
2. In each shifted temperature interval, calculate a simple energy balance from:

$$\Delta H_i = \left[\underset{Cold\ streams}{\sum CP_C} - \underset{Hot\ streams}{\sum CP_H} \right] \Delta T_i \quad (16.1)$$

   where ΔH_i is the heat balance for shifted temperature interval i and ΔT_i is the temperature difference across it. If the cold streams dominate the hot streams in a temperature interval, then the interval has a net deficit of heat, and ΔH is positive. If hot streams dominate cold streams, the interval has a net surplus of heat, and ΔH is negative. This is consistent with standard thermodynamic convention, for example, for an exothermic reaction, ΔH is negative. If no hot utility is used, this is equivalent to constructing the shifted composite curves shown in Figure 16.15a.
3. The overlap in the shifted curves as shown in Figure 16.15a means that heat transfer is infeasible. At some point, this overlap is a maximum. This maximum overlap is added as hot utility to correct the overlap. The shifted curves now touch at the pinch as shown in Figure 16.15b. Since the shifted curves just touch, the actual curves are separated by ΔT_{min} at this point, Figure 16.15b.

This basic approach can be developed into a formal algorithm known as the *problem table algorithm*[9]. The

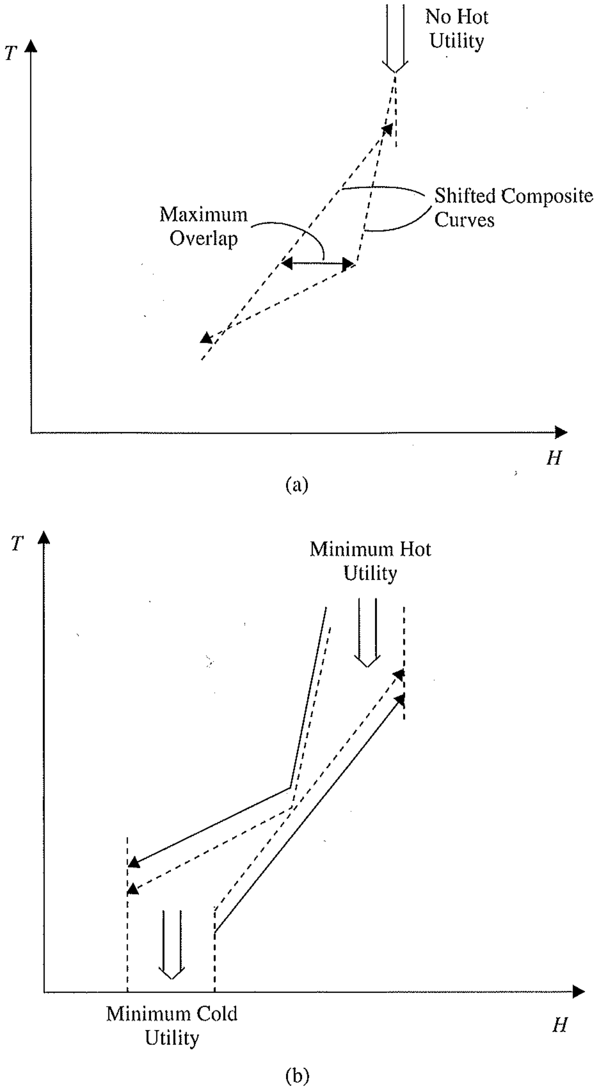


Figure 16.15 The utility target can be determined from the maximum overlap between the shifted composite curves.

algorithm will be explained using the data from Figure 16.2 given in Table 16.2 for $\Delta T_{min} = 10°C$.

First determine the shifted temperature intervals (T^*) from actual supply and target temperatures. Hot streams are shifted down in temperature by $\Delta T_{min}/2$ and cold streams up by $\Delta T_{min}/2$ as detailed in Table 16.3.

The stream population is shown in Figure 16.16 with a vertical temperature scale. The interval temperatures

Table 16.3 Shifted temperatures for the data from Table 16.2.

Stream	Type	T_S	T_T	T_S^*	T_T^*
1	Cold	20	180	25	185
2	Hot	250	40	245	35
3	Cold	140	230	145	235
4	Hot	200	80	195	75

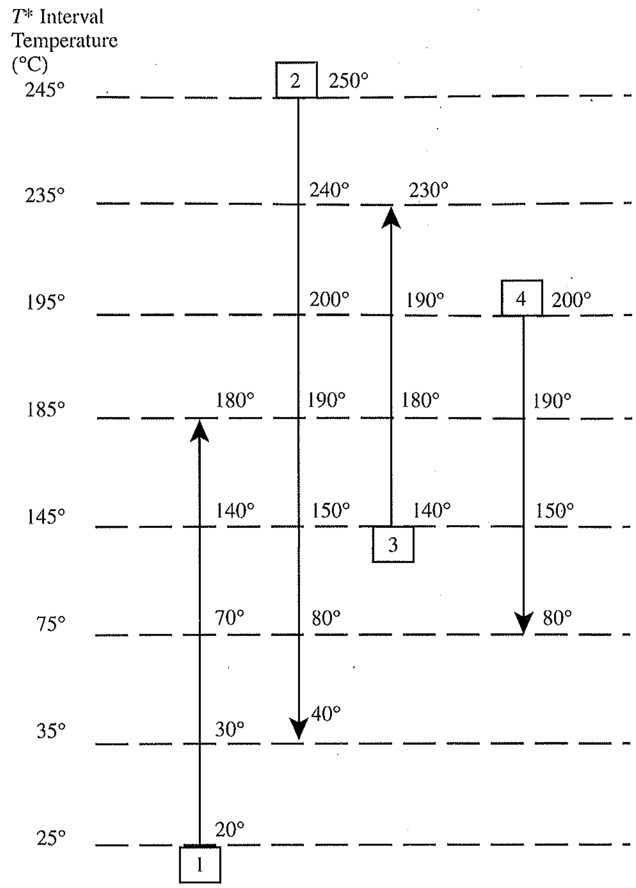
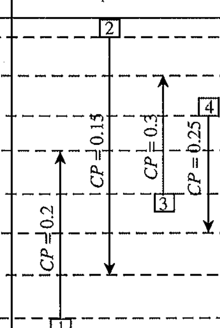


Figure 16.16 The stream population for the data from Figure 16.2.

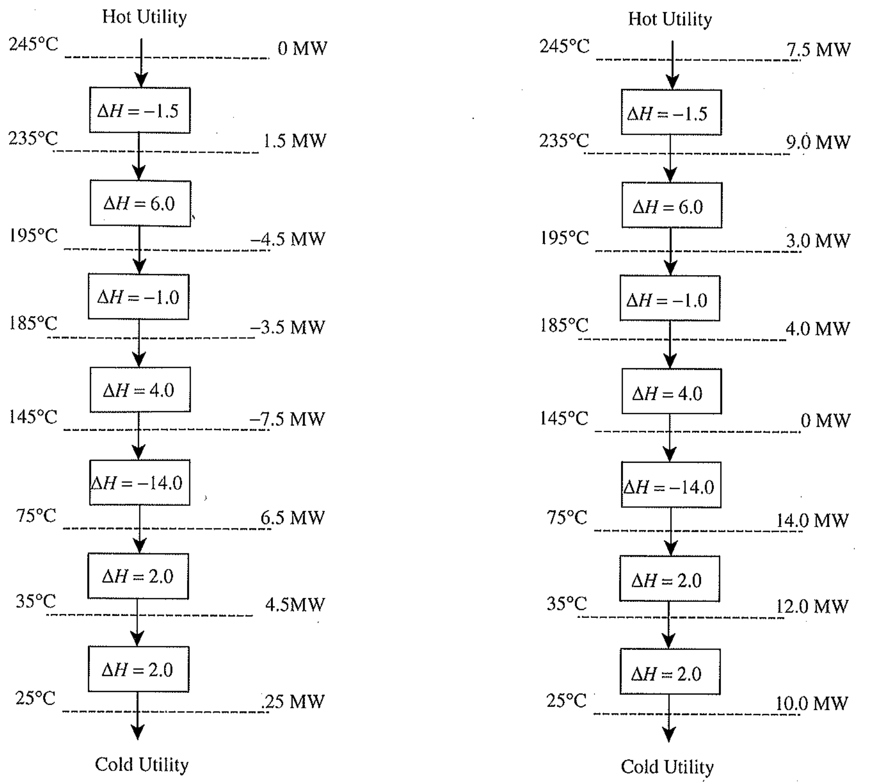


Interval Temperature	$\Delta T_{INTERVAL}$ (°C)	$\Sigma CP_C - \Sigma CP_H$ MW·K^{-1}	$\Delta H_{INTERVAL}$ (MW)	Surplus/Deficit
245°				
	10	−0.15	−1.5	Surplus
235°				
	40	0.15	6.0	Deficit
195°				
	10	−0.1	−1.0	Surplus
185°				
	40	0.1	4.0	Deficit
145°				
	70	−0.2	−14.0	Surplus
75°				
	40	0.05	2.0	Deficit
35°				
	10	0.2	2.0	Deficit
25°				

Figure 16.17 The temperature interval heat balances.

shown in Figure 16.16 are set to $\Delta T_{min}/2$ below hot stream temperatures and $\Delta T_{min}/2$ above cold stream temperatures.

Next, a heat balance is carried out within each shifted temperature interval according to Equation 16.1. The result is given in Figure 16.17, in which some of the shifted intervals are seen to have a surplus of heat and some have a



(a) Cascade surplus heat from high to low temperature.

(b) Add heat from hot utility to make all heat flows zero or positive.

Figure 16.18 The problem table cascade.

deficit. The heat balance within each shifted interval allows maximum heat recovery within each interval. However, recovery must also be allowed between intervals.

Now, *cascade* any surplus heat down the temperature scale from interval to interval. This is possible because any excess heat available from the hot streams in an interval is hot enough to supply a deficit in the cold streams in the next interval down. Figure 16.18 shows the cascade for the problem. First, assume no heat is supplied to the first interval from hot utility, Figure 16.18a. The first interval has a surplus of 1.5 MW, which is cascaded to the next interval. This second interval has a deficit of 6 MW, which leaves the heat cascaded from this interval to be −4.5 MW. In the third interval, the process has a surplus of 1 MW, which leaves −3.5 MW, to be cascaded to the next interval, and so on.

Some of the heat flows in Figure 16.18a are negative, which is infeasible. Heat cannot be transferred up the temperature scale. To make the cascade feasible, sufficient heat must be added from hot utility to make the heat flows to be at least zero. The smallest amount of heat needed from hot utility is the largest negative heat flow from Figure 16.18a, that is 7.5 MW. In Figure 16.18b, 7.5 MW is added from hot utility to the first interval. This does not change the heat balance within each interval, but increases all of the heat flows between intervals by 7.5 MW, giving one heat flow of just zero at an interval temperature of 145°C.

More than 7.5 MW could be added from hot utility to the first interval, but the objective is to find the minimum hot and cold utility, hence only the minimum is added. Thus from Figure 16.18b, $Q_{Hmin} = 7.5$ MW and $Q_{Cmin} = 10$ MW. This corresponds with the values obtained from the composite curves in Figure 16.5. One further important piece of information can be deduced from the cascade in Figure 16.18b. The point where the heat flow goes to zero at $T^* = 145°C$ corresponds to the pinch. Thus, the actual hot and cold stream pinch temperatures are 150°C and 140°C respectively. Again, this agrees with the result from the composite curves in Figure 16.5.

The initial setting for the heat cascade in Figure 16.18a corresponds to the setting for the shifted composite curves in Figure 16.15a where there is an overlap. The setting of the heat cascade for zero or positive heat flows in Figure 16.18b corresponds to the shifted composite curve setting in Figure 16.15b.

The composite curves are useful in providing conceptual understanding of the process but the problem table algorithm is a more convenient calculation tool.

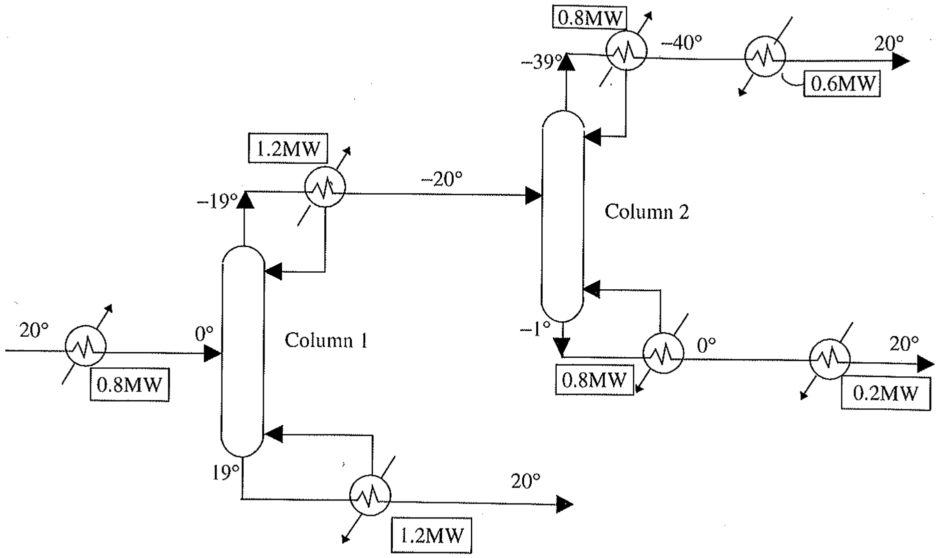


Figure 16.19 A low-temperature distillation process.

Example 16.1 The flowsheet for a low-temperature distillation process is shown in Figure 16.19. Calculate the minimum hot and cold utility requirements and the location of the pinch assuming $\Delta T_{min} = 5°C$.

Solution First extract the stream data from the flowsheet. This is given in Table 16.4 below.

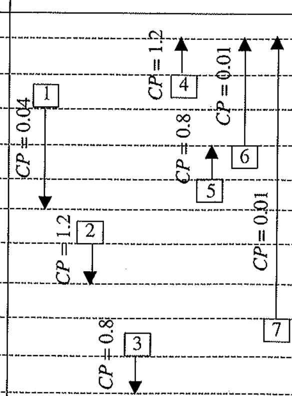
Next, calculate the shifted interval temperatures. Hot stream temperatures are shifted down by 2.5°C and cold stream temperatures shifted up by 2.5°C, as shown in Table 16.5.

Now carry out a heat balance within each shifted temperature interval as shown in Figure 16.20.

Finally, the heat cascade is shown in Figure 16.21. Figure 16.21a shows the cascade with zero hot utility. This leads to negative heat flows, the largest of which is −1.84 MW. Adding 1.84 MW from hot utility as shown in Figure 16.21b gives $Q_{Hmin} = 1.84$ MW, $Q_{Cmin} = 1.84$ MW, hot stream pinch temperature = −19°C and cold stream pinch temperature = −24°C.

Table 16.4 Stream data for low-temperature distillation process.

Stream	Type	Supply temperature T_S(°C)	Target temperature T_T(°C)	ΔH (MW)	CP (MW·K^{-1})
1. Feed to column 1	Hot	20	0	−0.8	0.04
2. Column 1 condenser	Hot	−19	−20	−1.2	1.2
3. Column 2 condenser	Hot	−39	−40	−0.8	0.8
4. Column 1 reboiler	Cold	19	20	1.2	1.2
5. Column 2 reboiler	Cold	−1	0	0.8	0.8
6. Column 2 bottoms	Cold	0	20	0.2	0.01
7. Column 2 overheads	Cold	−40	20	0.6	0.01



Interval Temperature	Stream Population	$\Delta T_{INTERVAL}$	$\Sigma CP_C - \Sigma CP_H$	$\Delta H_{INTERVAL}$	Surplus/ Deficit
22.5°C					
21.5°C		1	1.22	1.22	Deficit
17.5°C		4	0.02	0.08	Deficit
2.5°C		15	−0.02	−0.30	Surplus
1.5°C		1	0.77	0.77	Deficit
−2.5°C		4	−0.03	−0.12	Surplus
−21.5°C		19	0.01	0.19	Deficit
−22.5°C		1	−1.19	−1.19	Surplus
−37.5°C		15	0.01	0.15	Deficit
−41.5°C		4	0	0	---------
−42.5°C		1	−0.8	−0.8	Surplus

Figure 16.20 Temperature interval heat balances for Example 16.1.

Table 16.5 Shifted temperatures for the data in Table 16.4.

Stream	Type	T_S	T_T	T_S^*	T_T^*
1	Hot	20	0	17.5	−2.5
2	Hot	−19	−20	−21.5	−22.5
3	Hot	−39	−40	−41.5	−42.5
4	Cold	19	20	21.5	22.5
5	Cold	−1	0	1.5	2.5
6	Cold	0	20	2.5	22.5
7	Cold	−40	20	−37.5	22.5

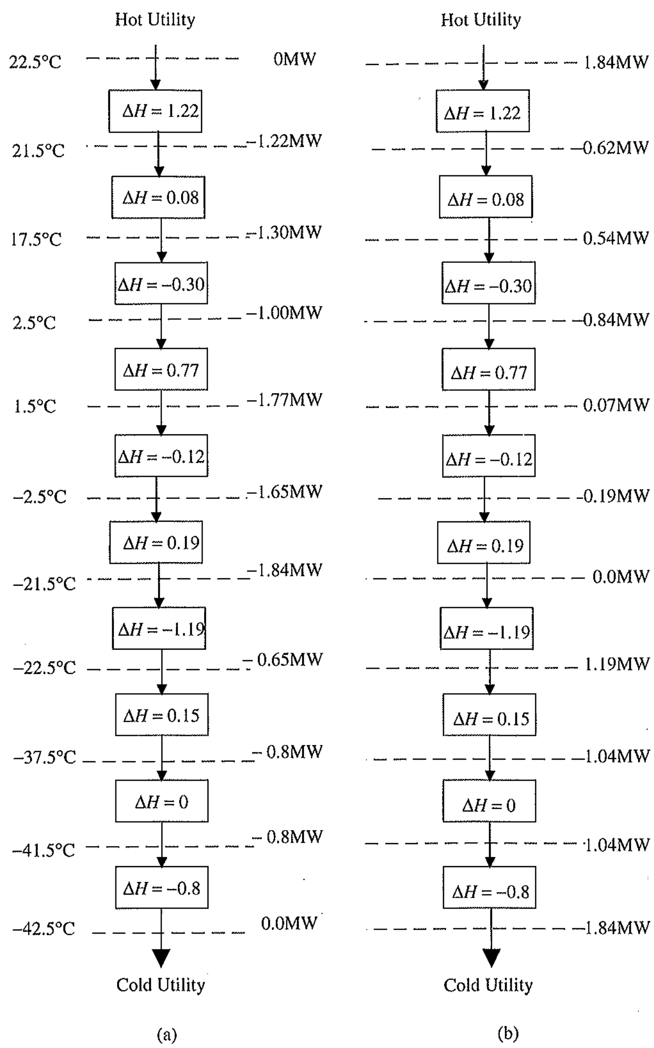


Figure 16.21 The problem table cascade for Example 16.1.

16.5 NONGLOBAL MINIMUM TEMPERATURE DIFFERENCES

So far, it has been assumed that the minimum temperature difference for a heat exchanger network applies globally between all streams in the network. However, there are occasions when nonglobal minimum temperature differences might be required. For example, suppose a heat exchanger network is servicing some streams that are liquid and some that are gaseous. For the liquid–liquid heat transfer matches, a value of perhaps $\Delta T_{min} = 10°C$ is appropriate. But for the gas–gas matches, a larger temperature minimum temperature difference is required, say $\Delta T_{min} = 20°C$. How can this be accommodated in the targeting?

When carrying out the problem table algorithm, the temperatures were shifted according to $\Delta T_{min}/2$ being added to the cold streams and subtracted from the hot streams. This value of $\Delta T_{min}/2$ can be considered to be a *contribution* to the overall ΔT_{min} between the hot and the cold streams. Rather than making the ΔT_{min} contribution equal for all streams, it could be made stream-specific:

$$T^*_{H,i} = T_{H,i} - \Delta T_{min,cont,i}$$

$$T^*_{C,j} = T_{C,j} + \Delta T_{min,cont,j}$$

where $T^*_{H,i}$, $T_{H,i}$ are the shifted and actual temperatures for Hot Stream i, $T^*_{C,j}$, $T_{C,j}$ are the shifted and actual temperatures for Cold Stream j, and $\Delta T_{min,cont,i}$ and $\Delta T_{min,cont,j}$ are the contributions to ΔT_{min} for Hot Stream i and Cold Stream j. Thus, for the above example, if the ΔT_{min} contribution for liquid streams is taken to be 5°C and for gas streams 10°C, then a liquid–liquid match would lead to $\Delta T_{min} = 10°C$, a gas–gas match would lead to $\Delta T_{min} = 20°C$ and a liquid–gas match would lead to $\Delta T_{min} = 15°C$[4]. To include this in the problem table algorithm is straightforward. All that needs to be done is that the appropriate ΔT_{min} contribution is to be allocated to each stream and then that ΔT_{min} contribution is subtracted in the case of hot streams and added in the case of cold streams. This would lead to different interval temperatures compared with a global minimum temperature difference. The remainder of the problem table algorithm would be the same. Once the interval temperatures based on ΔT_{min} contributions have been established, the interval heat balances can be performed and the cascade set up in the same way as for a global ΔT_{min}.

From the point of view of the composite curves, the location of the pinch and the ΔT_{min} at the pinch would depend on which kind of streams were located in the region of the point of closest approach between the composite curves. If only liquid streams were present around the point of closest approach of the composite curves, then in the above example, $\Delta T_{min} = 10°C$ will apply. If there were only gas streams in the region around the point of closest approach, then in the above example, $\Delta T_{min} = 20°C$ would apply. If there was a mixture of liquid and gas streams at the point of closest approach, then $\Delta T_{min} = 15°C$ would apply.

16.6 PROCESS CONSTRAINTS

So far it has been assumed that any hot stream could, in principle, be matched with any cold stream providing there is feasible temperature difference between them. Often, practical constraints prevent this. For example, it might be the case that if two streams are matched in a heat exchanger and a leak develops, such that the two streams come into contact, this might produce an unacceptably hazardous situation. If this were the case, then no doubt a constraint would be imposed to prevent the two streams being matched. Another reason for a constraint might be that two streams are expected to be geographically very distant from each other, leading to unacceptably long pipe runs. Potential control and start-up problems might also call for constraints. There are many reasons why constraints might be imposed.

One common reason for imposing constraints results from *areas of integrity*[10]. A process is often normally designed to have logically identifiable sections or areas. An example might be the "reaction area" and "separation area" of the process. These areas might need to be kept separate for reasons such as start-up, shutdown, operational flexibility, safety, and so on. The areas are often made operationally independent by use of intermediate storage between the areas. Such independent areas are generally described as areas of integrity and impose constraints on the ability to transfer heat. Clearly, to maintain operational independence, two areas cannot be dependent on each other for heating and cooling by recovery.

The question now is, given that there are often constraints to deal with, how to evaluate the effect of these constraints on the system performance? The problem table algorithm cannot be used directly if constraints are imposed. However, often the effect of constraints on the energy performance can be evaluated using the problem table algorithm, together with a little common sense. The following example illustrates how[10].

Example 16.2 A process is to be divided into two operationally independent areas of integrity, Area A and Area B. The stream data for the two areas are given in Table 16.6[10].

Calculate the penalty in utility consumption to maintain the two areas of integrity for $\Delta T_{min} = 20°C$.

Solution To identify the penalty, first calculate the utility consumption of the two areas separate from each other as shown in Figure 16.22a. Next, combine all of the streams from both areas and again calculate the utility consumption, Figure 16.22b. Figure 16.23a shows the problem table cascade for Area A, the cascade for Area B is shown in Figure 16.23b, and that for Areas A and B combined in Figure 16.23c.

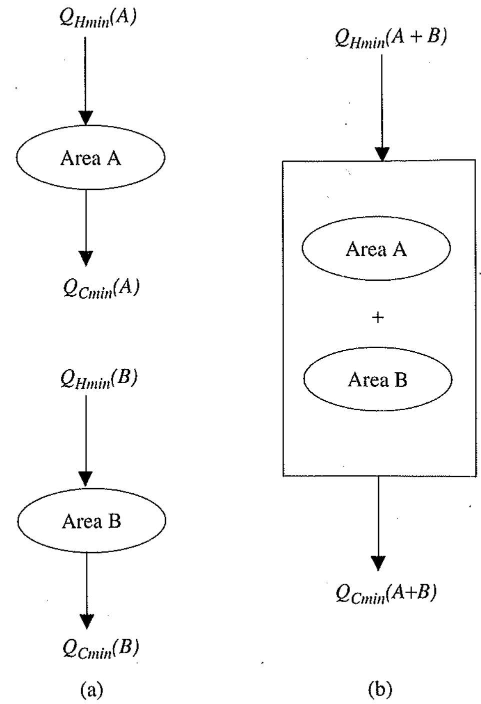


Figure 16.22 The areas of integrity can be targeted separately and then the combination targeted.

With Areas A and B separate, the total hot utility consumption is $(1400 + 0) = 1400$ kW and the total cold utility consumption is $(0 + 1350) = 1350$ kW. With Areas A and B combined, the total utility consumption is 950 kW of hot utility and 900 kW of cold utility.

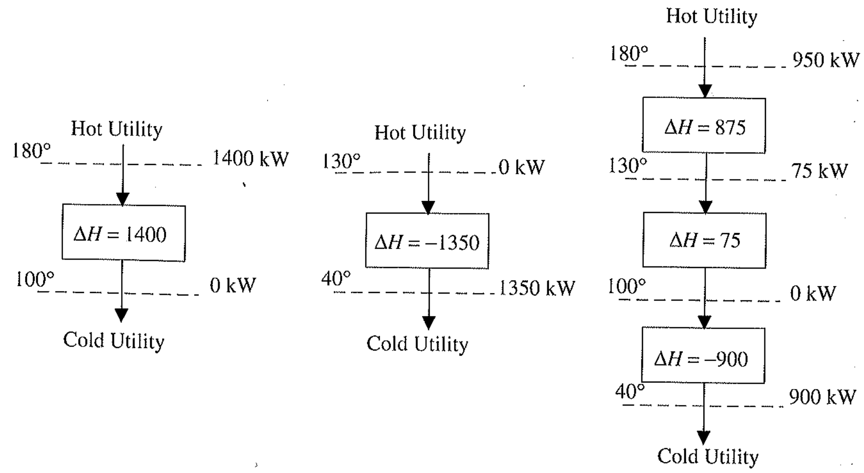
The penalty for maintaining the areas of integrity is thus $(1400 - 950) = 450$ kW of hot utility and $(1350 - 900) = 450$ kW of cold utility.

Having quantified the penalty as a result of imposing constraints, the designer can exercise judgment as to whether it is acceptable or too expensive. If it is too expensive, there is a choice between two options:

a. Reject the constraints and operate the process as a single system.
b. Find a way to overcome the constraint. This is often possible by using a heat transfer fluid. The simplest option is via the existing utility system. For example, rather than have a direct match between two streams,

Table 16.6 Stream data for heat recovery between two areas of integrity.

Area A				Area B			
Stream	T_S (°C)	T_T (°C)	CP (kW·K^{-1})	Stream	T_S (°C)	T_T (°C)	CP (kW·K^{-1})
1	190	110	2.5	3	140	50	20.0
2	90	170	20.0	4	30	120	5.0



(a) Area A (b) Area B (c) Area A + B

Figure 16.23 Problem table cascade for the separate and combined areas of integrity.

it might be possible for the heat source to generate steam to be fed into the steam mains and the heat sink to use steam from the same mains. The utility system then acts as a buffer between the heat sources and sinks. Another possibility might be to use a heat transfer fluid such as hot oil that circulates between the two streams being matched. To maintain operational independence, a standby heater and cooler supplied by utilities can be provided in the hot oil circuit, so that if either the heat source or sink is not operational, utilities could substitute heat recovery for short periods.

Many constraints can be evaluated by scoping out the problem with different boundaries. In Example 16.2, the sets of streams that were constrained to be separate were collected to be within each boundary for targeting. Comparing the targets for the streams within each boundary with that for all the streams put together allows the penalty of the constraint to be evaluated. The approach is more widely applicable than just areas of integrity. Whenever a stream, or set of streams, is to be maintained separate from any other set of streams, the same approach can be used. However, this approach of scoping out the problem with different boundaries has limitations in the evaluation of constraints. More complex constraints require linear programming to obtain the energy target[11,12].

16.7 UTILITY SELECTION

After maximizing heat recovery in the heat exchanger network, those heating and cooling duties not serviced by heat recovery must be provided by external utilities. The most common hot utility is steam. It is usually available at several levels. High-temperature heating duties require furnace flue gas or a hot oil circuit. Cold utilities might be refrigeration, cooling water, air-cooling, furnace air preheating, boiler feedwater preheating or even steam generation if heat needs to be rejected at high temperatures.

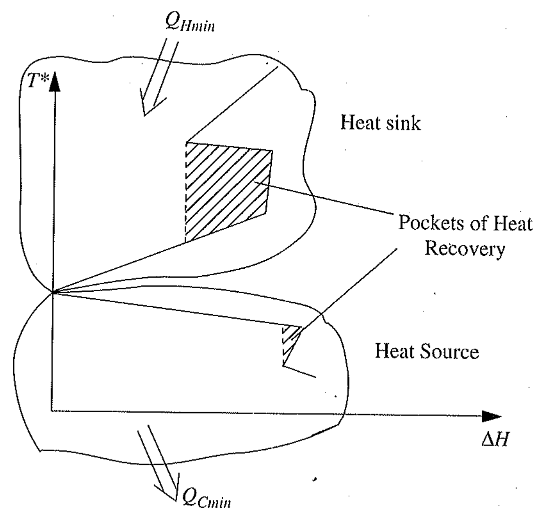


Figure 16.24 The grand composite curve shows the utility requirements both in enthalpy and temperature terms.

Although the composite curves can be used to set energy targets, they are not a suitable tool for the selection of utilities. The *grand composite curve* is a more appropriate tool for understanding the interface between the process and the utility system[4,13,14]. It will also be shown in later chapters to be a useful tool to study the interaction between heat-integrated reactors and separators and the rest of the process.

The grand composite curve is obtained by plotting the problem table cascade. A typical grand composite curve is shown in Figure 16.24. It shows the heat flow through the process against temperature. It should be noted that the temperature plotted here is *shifted temperature* (T^*) and not actual temperature. Hot streams are represented $\Delta T_{min}/2$ colder and cold streams $\Delta T_{min}/2$ hotter than they

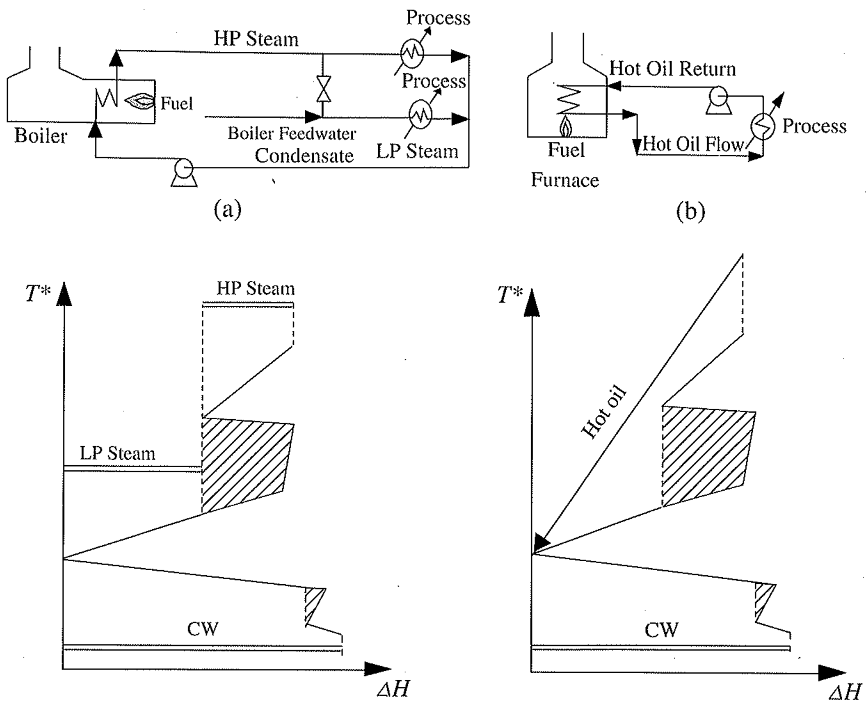


Figure 16.25 The grand composite curve allows alternative utility mixes to be evaluated.

are in practice. Thus, an allowance for ΔT_{min} is built into the construction.

The point of zero heat flow in the grand composite curve in Figure 16.24 is the heat recovery pinch. The open "jaws" at the top and bottom are Q_{Hmin} and Q_{Cmin} respectively. Thus, the heat sink above the pinch and heat source below the pinch can be identified as shown in Figure 16.24. The shaded areas in Figure 16.24, known as *pockets*, represent areas of additional process-to-process heat transfer. Remember that the profile of the grand composite curve represents residual heating and cooling demands after recovering heat within the shifted temperature intervals in the problem table algorithm. In the pockets in Figure 16.24, a local surplus of heat in the process is used at temperature differences in excess of ΔT_{min} to satisfy a local deficit. This reflects the cascading of excess heat from high-temperature intervals to lower temperature intervals in the problem table algorithm.

Figure 16.25a shows the same grand composite curve with two levels of steam used as hot utility. Figure 16.25b shows again the same grand composite curve but with hot oil used as hot utility.

Example 16.3 The problem table cascade for the process in Figure 16.2 is given in Figure 16.18. Using the grand composite curve:

a. For two levels of steam at saturation conditions and temperatures of 240°C and 180°C, determine the loads on the two steam levels that maximizes the use of the lower pressure steam.

b. Instead of using steam, a hot oil circuit is to be used with a supply temperature of 280°C and $C_P = 2.1\ \text{kJ·kg}^{-1}\text{·K}^{-1}$. Calculate the minimum flowrate of hot oil.

Solution

a. For $\Delta T_{min} = 10$°C, the two steam levels are plotted on the grand composite curve at temperatures of 235°C and 175°C. Figure 16.26a shows the loads that maximize the use of the lower pressure steam. Calculate the load on the low-pressure steam by interpolation of the cascade heat flows. At $T^* =$ 175°C:

$$\text{Load of 180°C steam} = \frac{75 - 145}{185 - 145} \times 4$$
$$= 3\ \text{MW}$$
$$\text{Load of 240°C steam} = 7.5 - 3$$
$$= 4.5\ \text{MW}$$

b. Figure 16.26b shows the grand composite curve with hot oil providing the hot utility requirements. If the minimum flowrate is required, then this corresponds to the steepest slope and minimum return temperature. For this problem, the minimum return temperature for the hot oil is pinch temperature ($T^* =$ 145°C, $T = 150$°C for hot streams). Thus,

$$\text{Minimum flowrate} = 7.5 \times 10^3 \times \frac{1}{2.1} \times \frac{1}{(280 - 150)}$$
$$= 27.5\ \text{kg·s}^{-1}$$

In other problems, the shape of the grand composite curve away from the pinch could have limited the flowrate.

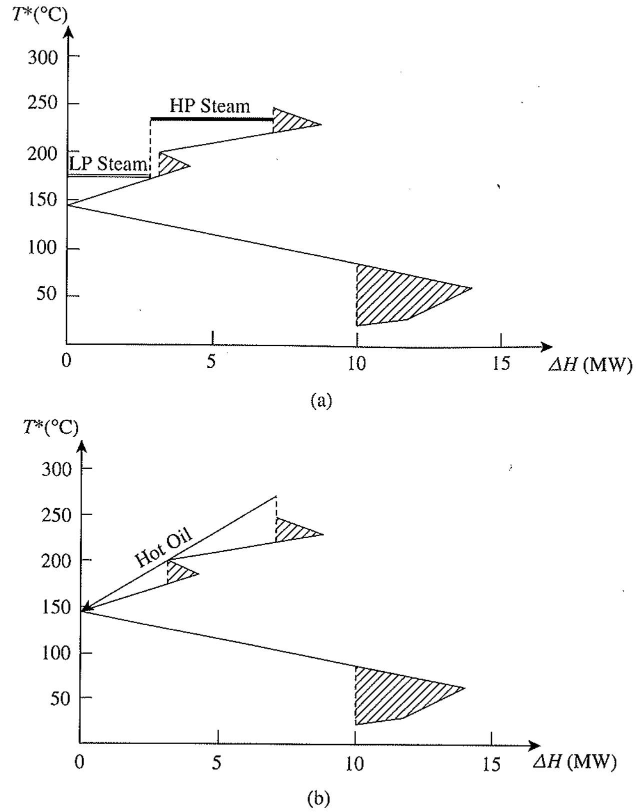


Figure 16.26 Alternative utility mixes for the process in Figure 16.2.

16.8 FURNACES

When hot utility needs to be at a high temperature and/or provide high heat fluxes, radiant heat transfer is used from combustion of fuel in a furnace. Furnace designs vary according to the function of the furnace, heating duty, type of fuel and the method of introducing combustion air (see Chapter 15). Sometimes the function is to purely provide heat, sometimes the furnace is also a reactor and provides heat of reaction. However, process furnaces have a number of features in common. In the chamber in which combustion takes place, heat is transferred mainly by radiation to tubes around the walls of the chamber, through which passes the fluid to be heated. After the flue gas leaves the combustion chamber, most furnace designs extract further heat from the flue gas in a convection section before the flue gas is vented to atmosphere.

Figure 16.27 shows a grand composite curve with a flue gas matched against it to provide hot utility[15]. The flue gas starts at its theoretical flame temperature (see Chapter 15) shifted for ΔT_{min} on the grand composite curve and presents a sloping profile because it is giving up sensible heat. Theoretical flame temperature is the temperature attained when a fuel is burnt in air or oxygen without loss or gain of heat, as explained in Chapter 15.

In Figure 16.27, the flue gas is cooled to pinch temperature before being released to atmosphere. The heat released from the flue gas between pinch temperature and ambient is the stack loss. Thus in Figure 16.27, for a given grand composite curve and theoretical flame temperature, the heat from fuel and stack loss can be determined.

All combustion processes work with an excess of air or oxygen to ensure complete combustion of the fuel. Excess air typically ranges between 5 and 20% depending on the fuel, burner design and furnace design. As excess air is reduced, theoretical flame temperature increases as shown in Figure 16.28. This has the effect of reducing the stack loss and increasing the thermal efficiency of the furnace for a given process heating duty. Alternatively, if

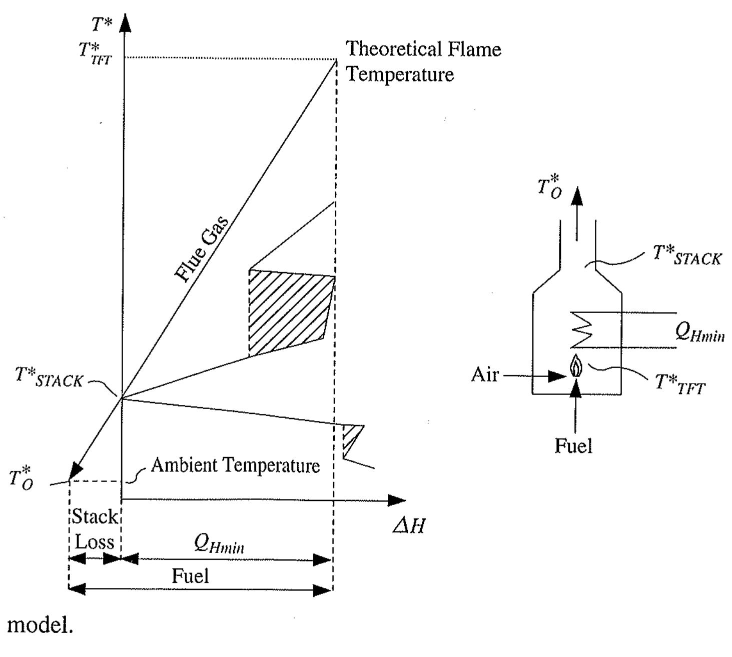


Figure 16.27 Simple furnace model.

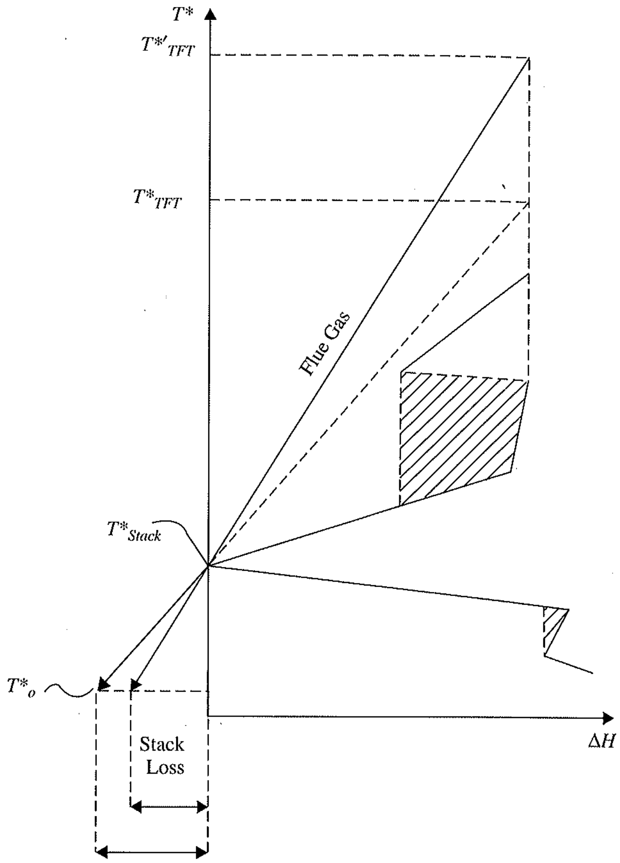


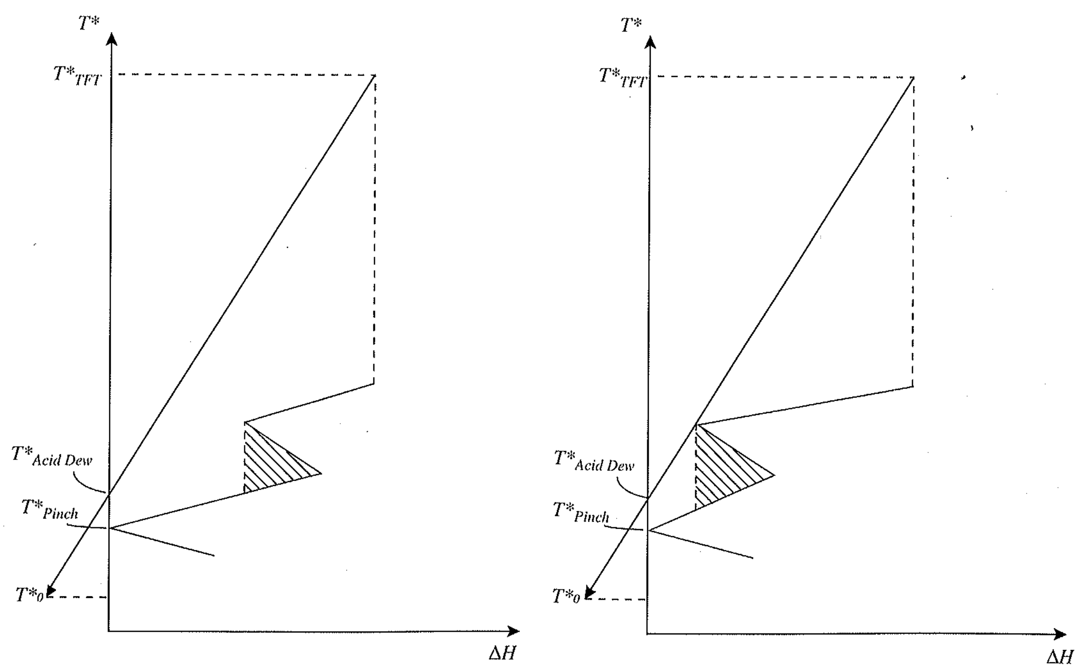
Figure 16.28 Increasing the theoretical flame temperature by reducing excess air or preheating combustion air reduces the stack loss.

the combustion air is preheated (say by heat recovery), then again the theoretical flame temperature increases as shown in Figure 16.28, reducing the stack loss.

Although, the higher flame temperatures in Figure 16.28 reduce the fuel consumption for a given process heating duty, there is one significant disadvantage. Higher flame temperatures increase the formation of oxides of nitrogen, which are environmentally harmful. This point will be returned to in Chapter 25.

In Figures 16.27 and 16.28, the flue gas is capable of being cooled to pinch temperature before being released to atmosphere. This is not always the case. Figure 16.29a shows a situation in which the flue gas is released to atmosphere above pinch temperature for practical reasons. There is a practical minimum, the acid dew point, to which a flue gas can be cooled without condensation causing corrosion in the stack (see Chapter 15). The minimum stack temperature in Figure 16.29a is fixed by acid dew point. Another case is shown in Figure 16.29b where the process away from the pinch limits the slope of the flue gas line and hence the stack loss.

Example 16.4 The process in Figure 16.2 is to have its hot utility supplied by a furnace. The theoretical flame temperature for combustion is 1800°C and the acid dew point for the flue gas is 160°C. Ambient temperature is 10°C. Assume $\Delta T_{min} = 10°C$ for process-to-process heat transfer but $\Delta T_{min} = 30°C$ for flue gas to process heat transfer. A high value for ΔT_{min} for flue gas to process heat transfer has been chosen because of poor heat-transfer coefficients in the convection bank of the furnace. Calculate the fuel required, stack loss and furnace efficiency.



(a) Stack temperature limited by acid dew point.

(b) Stack temperature limited by process away from the pinch.

Figure 16.29 Furnace stack temperature can be limited by other factors than pinch temperature.

Solution The first problem is that a different value of ΔT_{min} is required for different matches. The problem table algorithm is easily adapted to accommodate this. This is achieved by assigning ΔT_{min} contributions to streams. If the process streams are assigned a contribution of 5°C and flue gas a contribution of 25°C, then a process/process match has a ΔT_{min} of $(5+5) = 10$°C and a process/flue gas match has a ΔT_{min} of $(5+25) = 30$°C. When setting up the interval temperatures in the problem table algorithm, the interval boundaries are now set at hot stream temperatures minus their ΔT_{min} contribution, rather than half the global ΔT_{min}. Similarly, boundaries are now set on the basis of cold stream temperatures plus their ΔT_{min} contribution.

Figure 16.30 shows the grand composite curve plotted from the problem table cascade in Figure 16.18b. The starting point for the flue gas is an actual temperature of 1800°C, which corresponds to a shifted temperature of $(1800 - 25) = 1775$°C on the grand composite curve. The flue gas profile is not restricted above the pinch and can be cooled to pinch temperature corresponding with a shifted temperature of 145°C before venting to atmosphere. The actual stack temperature is thus $(145 + 25) = 170$°C. This is just above the acid dew point of 160°C. Now calculate the fuel consumption.

$$Q_{Hmin} = 7.5 \text{ MW}$$

$$CP_{FLUE\ GAS} = \frac{7.5}{1775 - 145} = 0.0046 \text{ MW·K}^{-1}$$

The fuel consumption is now calculated by taking the flue gas from theoretical flame temperature to ambient temperature:

$$\text{Fuel required} = 0.0046(1800 - 10) = 8.23 \text{ MW}$$

$$\text{Stack loss} = 0.0046(170 - 10) = 0.74 \text{ MW}$$

$$\text{Furnace efficiency} = \frac{Q_{H\,min}}{Fuel\ Required} \times 100 = \frac{7.5}{8.23} \times 100 = 91\%$$

16.9 COGENERATION (COMBINED HEAT AND POWER GENERATION)

More complex utility options are encountered when combined heat and power generation (or *cogeneration*) is exploited. Here the heat rejected by a heat engine such as a steam turbine, gas turbine or diesel engine is used as hot utility.

Fundamentally, there are two possible ways to integrate a heat engine exhaust[14]. In Figure 16.31, the process is

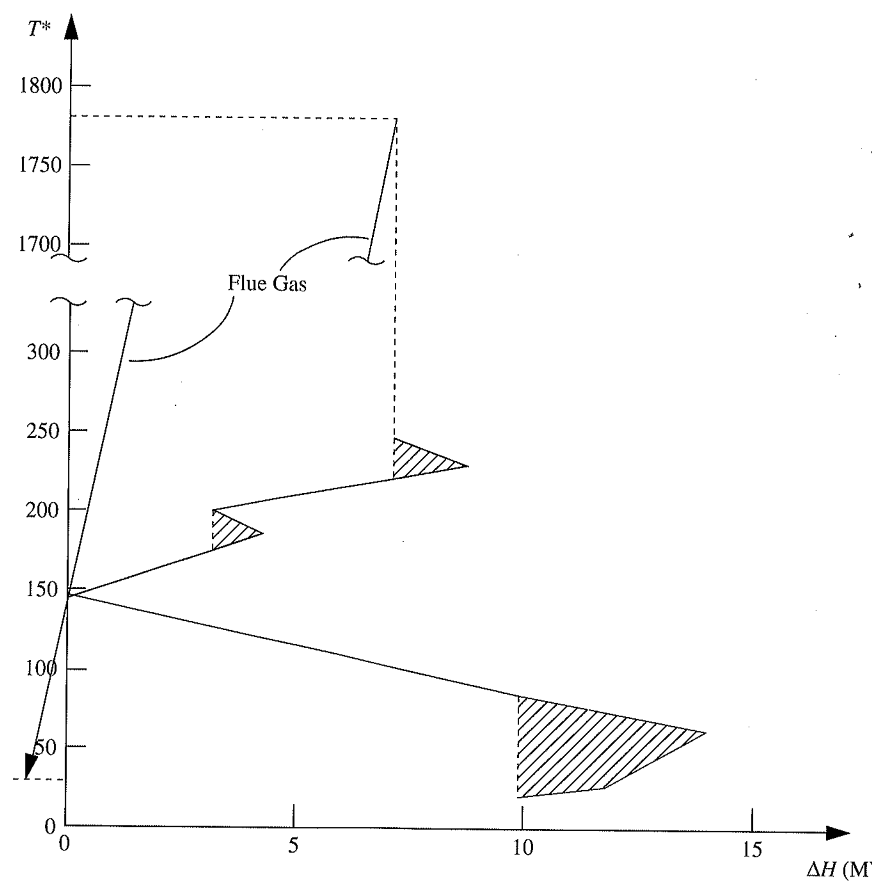


Figure 16.30 Flue gas matched against the grand composite curve of the process in Figure 16.2.

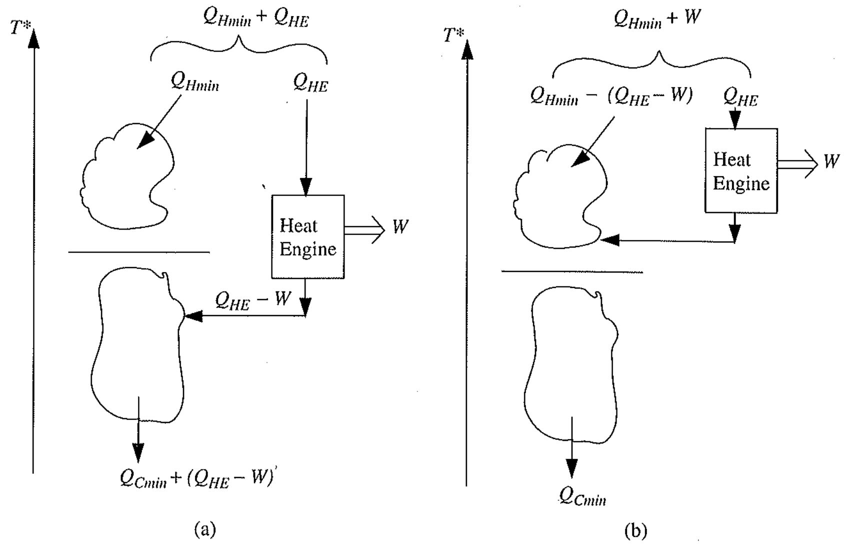


Figure 16.31 Heat engine exhaust can be integrated either across or not across the pinch.

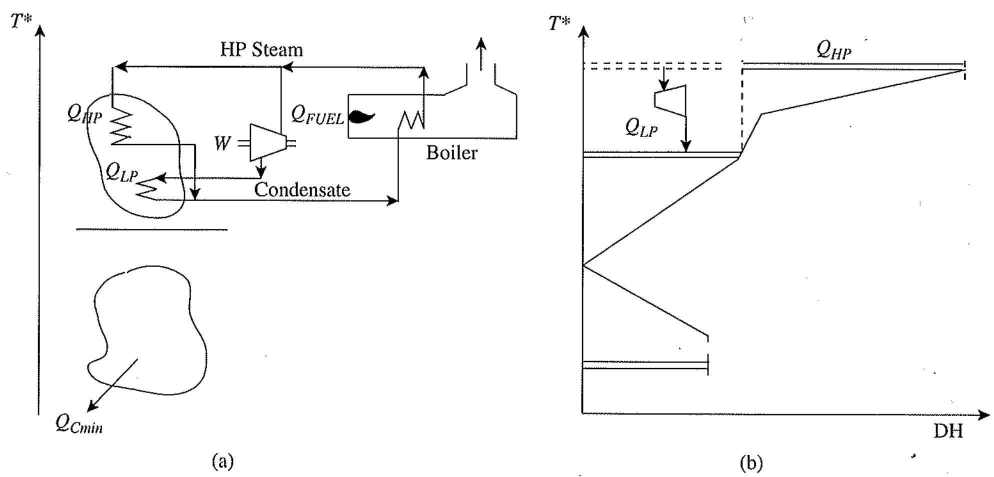


Figure 16.32 Integration of a steam turbine with the process.

represented as a heat sink and heat source separated by the pinch. Integration of the heat engine across the pinch as shown in Figure 16.31a is counterproductive. The process still requires Q_{Hmin} and the heat engine performs no better than operated stand-alone. There is no saving by integrating a heat engine across the pinch[14].

Figure 16.31b shows the heat engine integrated above the pinch. In rejecting heat above the pinch, it is rejecting heat into the part of the process, which is overall a heat sink. In so doing, it is exploiting the temperature difference that exists between the utility source and the process sink, producing power at high efficiency. The net effect in Figure 16.31b is the import of extra energy W from heat sources to produce W power. Owing to the process and heat engine acting as one system, apparently conversion of heat to power at 100% efficiency is achieved[14].

Now consider the two most commonly used heat engines (steam and gas turbines) in more detail to see whether they achieve this in practice. To make a quantitative assessment of any combined heat and power scheme, the grand composite curve should be used and the heat engine exhaust treated like any other utility.

Figure 16.32 shows a steam turbine integrated with the process above the pinch. A steam turbine is conceptually like a centrifugal compressor, but working in reverse. Steam is expanded from high to low pressure in the machine, producing power. In Figure 16.32 heat Q_{HP} is taken into the process from high-pressure steam. The balance of the hot utility demand Q_{LP} is taken from the steam turbine exhaust. In Figure 16.32a, heat Q_{FUEL} is taken into the boiler from fuel. An overall energy balance gives:

$$Q_{FUEL} = Q_{HP} + Q_{LP} + W + Q_{LOSS} \qquad (16.2)$$

The process requires $(Q_{HP} + Q_{LP})$ to satisfy its enthalpy imbalance above the pinch. If there were no losses from

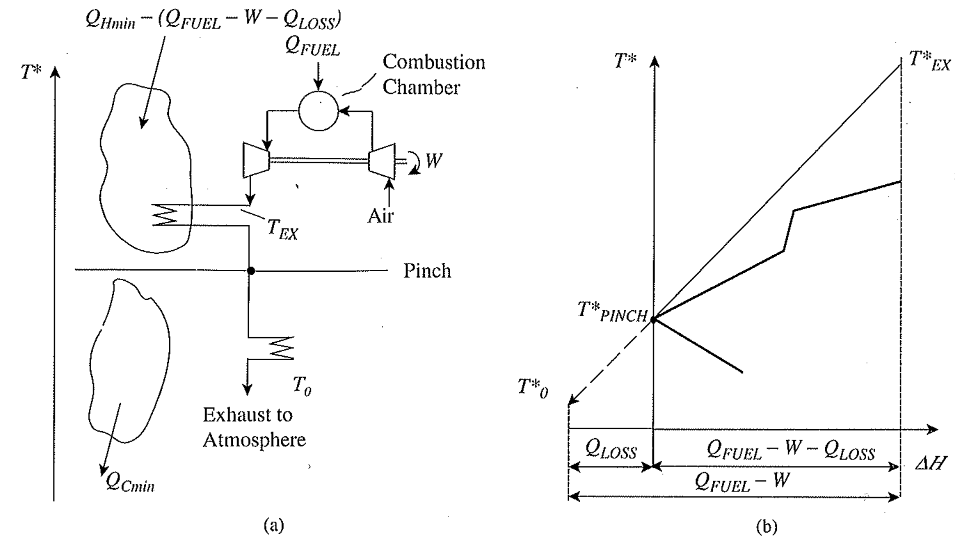


Figure 16.33 A gas turbine exhaust matched against the process (same as a flue gas).

the boiler, then fuel W would be converted to power W at 100% efficiency. However, the boiler losses Q_{LOSS} reduce this to below 100% conversion. In practice, in addition to the boiler losses, there can also be significant losses from the steam distribution system. Figure 16.32b shows how the grand composite curve can be used to size steam turbine cycles[14]. Steam turbines will be dealt with in more detail in Chapter 23.

Figure 16.33 shows a schematic of a simple gas turbine. The machine is essentially a rotary compressor mounted on the same shaft as a turbine. Air enters the compressor where it is compressed before entering a combustion chamber. Here the combustion of fuel increases its temperature. The mixture of air and combustion gases is expanded in the turbine. The input of energy to the combustion chamber allows enough power to be developed in the turbine to both drive the compressor and provide useful power. The performance of the machine is specified in terms of the power output, airflow rate through the machine, efficiency of conversion of heat to power and the temperature of the exhaust. Gas turbines are normally used only for relatively large-scale applications, and will be dealt with in more detail in Chapter 23.

Figure 16.33 shows a gas turbine matched against the grand composite curve[14]. As with the steam turbine, if there was no stack loss to atmosphere (i.e. if Q_{LOSS} was zero), then W heat would be turned into W power. The stack losses in Figure 16.33 reduce the efficiency of conversion of heat to power. The overall efficiency of conversion of heat to power depends on the turbine exhaust profile, the pinch temperature and the shape of the process grand composite.

Example 16.5 The stream data for a heat recovery problem are given in Table 16.7 below.

Table 16.7 Stream data for Example 16.5.

Stream No.	Stream Type	T_S (°C)	T_T (°C)	Heat capacity flowrate (MW·K^{-1})
1	Hot	450	50	0.25
2	Hot	50	40	1.5
3	Cold	30	400	0.22
4	Cold	30	400	0.05
5	Cold	120	121	22.0

A problem table analysis for $\Delta T_{min} = 20°C$ results in the heat cascade given in Table 16.8.

Table 16.8 Problem table cascade for Example 16.5.

T^* (°C)	Cascade heat flow (MW)
440	21.9
410	29.4
131	23.82
130	1.8
40	0
30	15

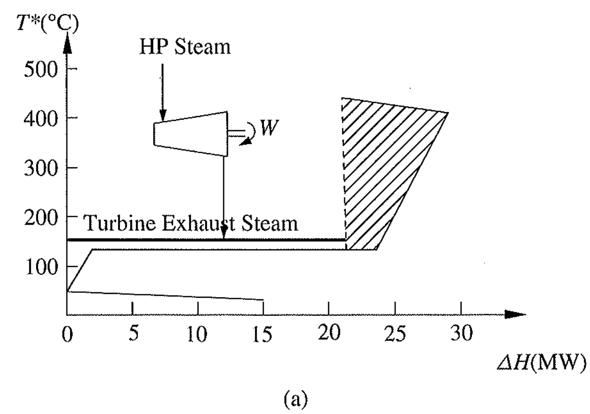
The process also has a requirement for 7 MW of power. Two alternative cogeneration schemes are to be compared economically.

a. A steam turbine with its exhaust saturated at 150°C used for process heating is one of the options to be considered. Superheated steam is generated in the central boiler house at 41 bar with a temperature of 300°C. This superheated steam can be expanded in a single-stage turbine with an isentropic efficiency of 85%. Calculate the maximum generation of power possible by matching the exhaust steam against the process.

b. A second possible scheme uses a gas turbine with a flowrate of air of 97 $kg \cdot s^{-1}$, which has an exhaust temperature of 400°C. Calculate the power generation if the turbine has an efficiency of 30%. Ambient temperature is 10°C.
c. The cost of heat from fuel for the gas turbine is \$4.5 GW^{-1}. The cost of imported electricity is \$19.2 GW^{-1}. Electricity can be exported with a value of \$14.4 GW^{-1}. The cost of fuel for steam generation is \$3.2 GW^{-1}. The overall efficiency of steam generation and distribution is 80%. Which scheme is most cost-effective, the steam turbine or the gas turbine?

Solution

a. This is shown in Figure 16.34a. The steam condensing interval temperature is 140°C.



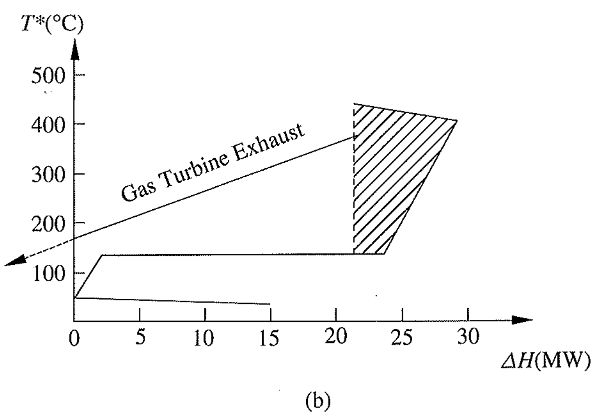


Figure 16.34 Alternative combined heat and power schemes for Example 16.5.

Heat flow required from the turbine exhaust

$$= 21.9 \text{ MW}$$

From steam tables, inlet conditions at $T_1 = 300°C$ and $P_1 =$ 41 bar are:

$$H_1 = 2959 \text{ kJ}\cdot\text{kg}^{-1}$$

$$S_1 = 6.349 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$$

Turbine outlet conditions for isentropic expansion to 150°C from steam tables are:

$$P_2 = 4.77 \text{ bar}$$

$$S_2 = 6.349 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$$

The wetness fraction (X) can be calculated from

$$S_2 = XS_L + (1 - X)S_V$$

where S_L and S_V are the saturated liquid and vapor entropies. Taking saturated liquid and vapor entropies from steam tables at 150°C and 4.77 bar:

$$6.349 = 1.842X + 6.838(1 - X)$$

$$X = 0.098$$

The turbine outlet enthalpy for an isentropic expansion can now be calculated from:

$$H_2 = XH_L + (1 - X)H_V$$

where H_L and H_V are the saturated liquid and vapor enthalpies. Taking saturated liquid and vapor enthalpies from steam tables at 150°C and 4.77 bar:

$$H_2 = 0.098 \times 632 + (1 - 0.098)2747$$

$$= 2540 \text{ kJ}\cdot\text{kg}^{-1}$$

For a single-stage expansion with isentropic efficiency of 85%:

$$H_2' = H_1 - \eta_{IS}(H_1 - H_2)$$

$$= 2959 - 0.85(2959 - 2540)$$

$$= 2603 \text{ kJ}\cdot\text{kg}^{-1}$$

The actual wetness fraction (X') can be calculated from:

$$H_2' = X'H_L + (1 - X')H_V$$

where H_L and H_V are the saturated liquid and vapor enthalpies.

$$H_2' = 2603 = 632X' + 2747(1 - X')$$

$$X' = 0.068$$

Assume in this case that the saturated steam and condensate are separated after the turbine and only the saturated steam used for process heating.

$$\text{Steam flow to process} = \frac{21.9 \times 10^3}{2747 - 632}$$

$$= 10.35 \text{ kg}\cdot\text{s}^{-1}$$

$$\text{Steam flow through turbine} = \frac{10.35}{(1 - 0.068)}$$

$$= 11.11 \text{ kg}\cdot\text{s}^{-1}$$

$$\text{Power generated } W = 11.11(2959 - 2603) \times 10^{-3}$$

$$= 3.96 \text{ MW}$$

b. The exhaust from the gas turbine is primarily air with a small amount of combustion gases. Hence, the CP of the exhaust can be approximated to be that of the airflow. Assuming C_P for air = 1.03 kJ·kg^{-1}·K^{-1}.

$$CP_{EX} = 97 \times 1.03$$
$$= 100 \text{ kW·K}^{-1}$$

The gas turbine option is shown in Figure 16.34b.

$$Q_{EX} = CP_{EX}(T_{EX} - T_0)$$
$$= 0.1 \times (400 - 10)$$
$$= 39 \text{ MW}$$
$$Q_{FUEL} = \frac{Q_{EX}}{(1 - \eta_{GT})}$$
$$= \frac{39}{(1 - 0.3)}$$
$$= 55.71 \text{ MW}$$
$$W = Q_{FUEL} - Q_{EX}$$
$$= 16.71 \text{ MW}$$

c. Steam turbine economics:

$$\text{Cost of fuel} = (21.9 + 3.96) \times \frac{3.2 \times 10^{-3}}{0.8}$$
$$= \$0.10 \text{ s}^{-1}$$
$$\text{Cost of imported electricity} = (7 - 3.96) \times 19.2 \times 10^{-3}$$
$$= \$0.06 \text{ s}^{-1}$$
$$\text{Net cost} = \$0.16 \text{ s}^{-1}$$

Gas turbine economics:

$$\text{Cost of fuel} = 55.71 \times 4.5 \times 10^{-3}$$
$$= \$0.25 \text{ s}^{-1}$$
$$\text{Electricity credit} = (16.71 - 7) \times 14.4 \times 10^{-3}$$
$$= \$0.14 \text{ s}^{-1}$$
$$\text{Net cost} = \$0.11 \text{ s}^{-1}$$

Hence the gas turbine is the most profitable in terms of energy costs. However, this is only part of the story since the capital cost of a gas turbine installation is likely to be significantly higher than that of a steam turbine installation.

Example 16.6 The problem table cascade for a process is given in Table 16.9 below for $\Delta T_{min} = 10°\text{C}$.

It is proposed to provide process cooling by steam generation from boiler feedwater with a temperature of 100°C.

Table 16.9 Problem table cascade.

Interval temperature (°C)	Heat flow (MW)
495	3.6
455	9.2
415	10.8
305	4.2
285	0
215	16.8
195	17.6
185	16.6
125	16.6
95	21.1
85	18.1

a. Determine how much steam can be generated at a saturation temperature of 230°C.
b. Determine how much steam can be generated with a saturation temperature of 230°C and superheated to the maximum temperature possible against the process.
c. Calculate how much power can be generated from the superheated steam from Part *b*, assuming a single-stage condensing steam turbine is to be used with an isentropic efficiency of 85%. Cooling water is available at 20°C and is to be returned to the cooling tower at 30°C.

Solution

a. Heat available for steam generation at 235°C interval temperature

$$= 12.0 \text{ MW}$$

From steam tables, the latent heat of water at a saturated temperature of 230°C is 1812 kJ·kg^{-1}.

$$\text{Steam production} = \frac{12.0 \times 10^3}{1812}$$
$$= 6.62 \text{ kg·s}^{-1}$$

Taking the heat capacity of water to be 4.3 kJ·kg^{-1}·K^{-1}, heat duty on boiler feedwater preheating

$$= 6.62 \times 4.3 \times 10^{-3}(230 - 100)$$
$$= 3.70 \text{ MW}$$

The profile of steam generation is shown against the grand composite curve in Figure 16.35a. The process can support both boiler feedwater preheat and steam generation.

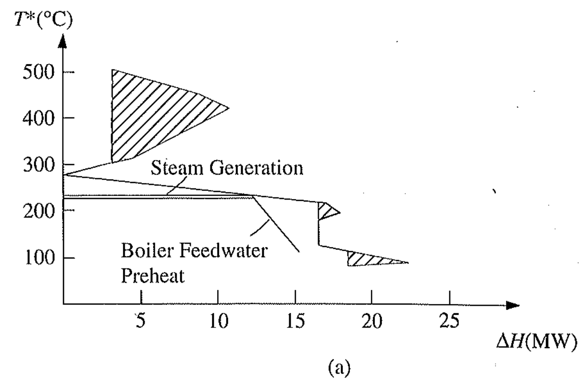
b. Maximum superheat temperature

$$= 285°\text{C interval}$$
$$= 280°\text{C actual}$$

The profile is shown against the grand composite curve in Figure 16.35b.

From steam tables, enthalpy of superheated steam at 280°C and 28 bar

$$= 2947 \text{ kJ·kg}^{-1}$$



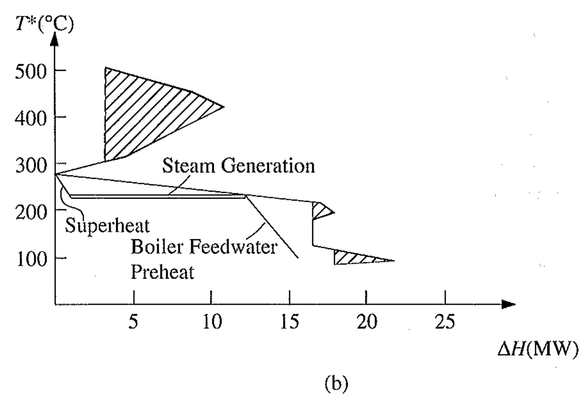


Figure 16.35 Alternative cold utilities for Example 16.6.

and enthalpy of saturated water at 230°C and 28 bar

$$= 991 \text{ kJ·kg}^{-1}$$

$$\text{Steam production} = \frac{12.0 \times 10^3}{(2947 - 991)}$$

$$= 6.13 \text{ kg·s}^{-1}$$

c. In a condensing turbine, the exhaust from the turbine is condensed under vacuum against cooling water. The lower the condensing temperature, the greater the power generation. The lowest condensing temperature for this problem is cooling water temperature plus ΔT_{min}, that is, $30 + 10 = 40$°C.
From steam tables, inlet conditions at $T_1 = 280$°C and $P_1 =$ 28 bar are:

$$H_1 = 2947 \text{ kJ·kg}^{-1}$$

$$S_1 = 6.488 \text{ kJ·kg}^{-1}\text{·K}^{-1}$$

Turbine outlet conditions for isentropic expansion to 40°C from steam tables are:

$$P_2 = 0.074 \text{ bar}$$

For $S_2 = 6.488$ kJ·kg^{-1}·K^{-1}, the wetness fraction (X) and outlet enthalpy H_2 can be calculated as shown in Example 14.4.

$$X = 0.23$$

$$H_2 = 2020 \text{ kJ·kg}^{-1}$$

For a single-stage expansion with isentropic efficiency of 85%:

$$H_2' = 2947 - 0.85(2947 - 2020)$$

$$= 2159 \text{ kJ·kg}^{-1}$$

The power generation (W) is given by:

$$W = 6.13(2947 - 2159) \times 10^{-3}$$

$$= 4.8 \text{ MW}$$

The wetness fraction for the real expansion is given by:

$$H_2' = 2159 = X'H_L + (1 - X')H_V$$

$$= 167.5X' + 2574(1 - X')$$

$$X' = 0.17$$

This wetness fraction is possibly too high, since high levels of wetness can cause damage to the turbine. To allow a lower wetness fraction of say $X = 0.15$, the outlet pressure of the turbine must be raised to 0.2 bar, corresponding to a condensing temperature of 60°C. However, in so doing, the power generation decreases to 4.2 MW.

16.10 INTEGRATION OF HEAT PUMPS

A heat pump is a device that takes in low-temperature heat and upgrades it to a higher temperature to provide process heat. A schematic of a simple vapor compression heat pump is shown in Figure 16.36. In Figure 16.36, the heat pump absorbs heat at a low temperature in the evaporator, consumes power when the working fluid is compressed and rejects heat at a higher temperature in the condenser. The condensed working fluid is expanded and partially vaporizes. The cycle then repeats. The working fluid is usually a pure component, which means that the evaporation and condensation take place isothermally. When considering the integration of a heat pump with the process, there are appropriate and inappropriate ways to integrate heat pumps.

There are two fundamental ways in which a heat pump can be integrated with the process; across and not across the pinch[14]. Integration not across (above) the pinch is illustrated in Figure 16.37a. This arrangement imports W

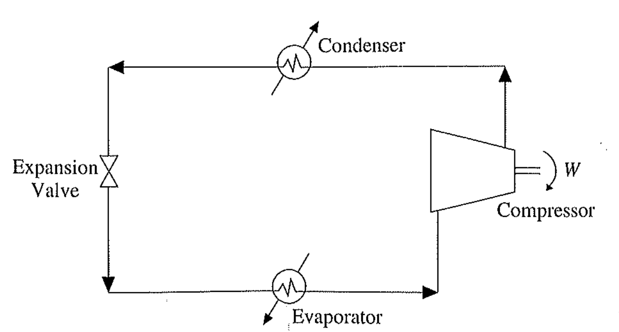
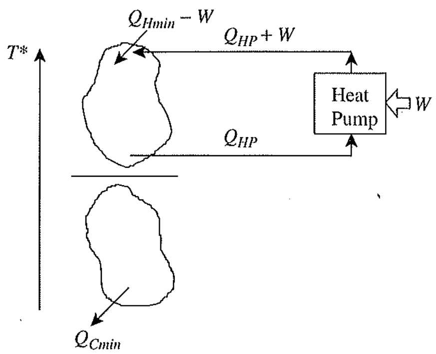
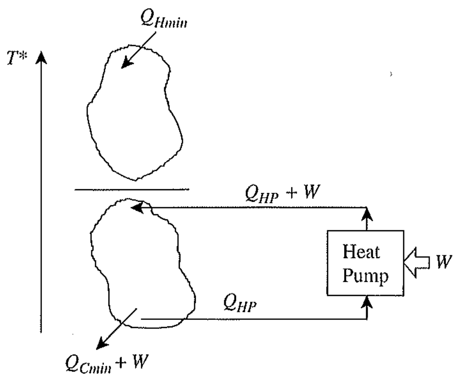


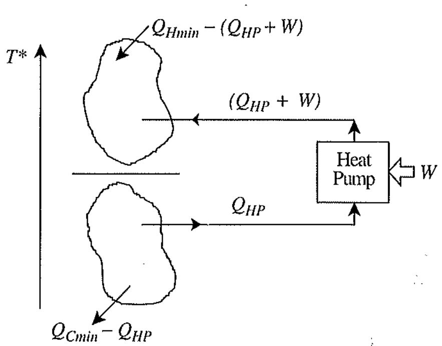
Figure 16.36 Schematic of a simple vapor compression heat pump.



(a) Integration of a heat pump abovethe pinch.



(b) Integration of a heat pump below the pinch.



(c) Integration of a heat pump across the pinch.

Figure 16.37 Integration of heat pump with the process.

power and saves W hot utility. In other words, the system converts power into heat, which is not normally worthwhile economically. Another integration not across (below) the pinch is shown in Figure 16.37b. The result is worse economically. Power is turned into waste heat[14].

Integration across the pinch is illustrated in Figure 16.37c. This arrangement brings a genuine saving. It also makes overall sense since heat is pumped from the part of the process that is overall a heat source to the part that is overall a heat sink.

Figure 16.38 shows a heat pump appropriately integrated against a process. Figure 16.38a shows the overall balance. Figure 16.38b illustrates how the grand composite curve can be used to size the heat pump. How the heat pump performs determines its coefficient of performance. The coefficient of performance for a heat pump can generally be defined as the useful energy delivered to the process divided by the power expended to produce this useful energy. From Figure 16.38a:

$$COP_{HP} = \frac{Q_{HP} + W}{W} \tag{16.3}$$

where COP_{HP} is the heat pump coefficient of performance, Q_{HP} the heat absorbed at low temperature and W the power consumed.

For any given type of heat pump, a higher COP_{HP} leads to better economics. Having a better COP and hence better economics means working across a small temperature lift with the heat pump. The smaller the temperature lift, the better the COP_{HP}. For most applications, a temperature lift greater than 25°C is rarely economic. Attractive heat pump application normally requires a lift much less than 25°C.

Using the grand composite curve, the loads and temperatures of the cooling and heating duties and hence the COP_{HP} of integrated heat pumps can be readily assessed.

Thus, the appropriate placement of heat pumps is that they should be placed across the pinch[14]. Note that the principle needs careful interpretation if there are utility pinches. In such circumstances, heat pump placement above the process pinch or below it can be economic, providing

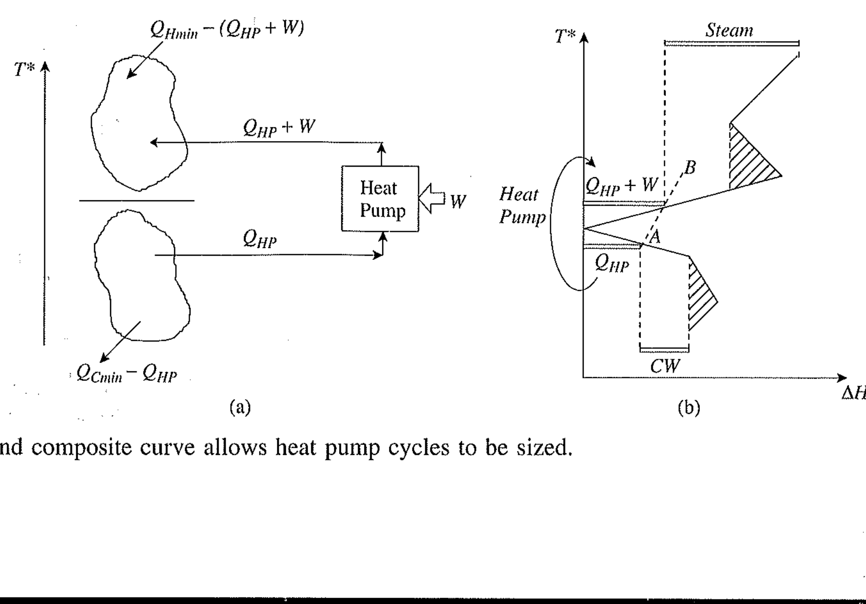
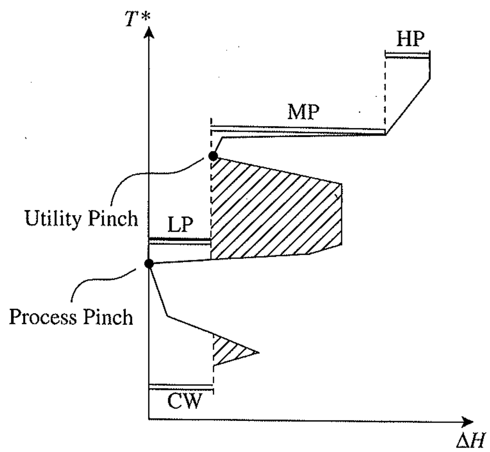
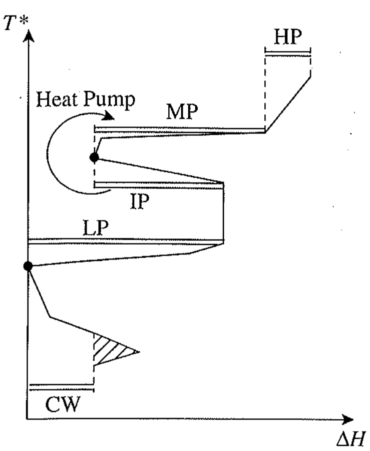


Figure 16.38 The grand composite curve allows heat pump cycles to be sized.



(a) A process that shows little scope for heat pumping across the process pinch.



(b) Heat pump placed across a utility pinch.

Figure 16.39 Heat pumping can be applied across utility pinches as well as the process pinch.

the heat pump is placed across a utility pinch. Figure 16.39a shows a process that does not show a significant potential for heat pumping across the process pinch. The heat source just below the process pinch is small. The heat sink just above the process pinch is also small. Significant heat pumping across the process pinch can only be accomplished across a large temperature difference in this case, which will result in a poor COP_{HP}. However, Figure 16.39b shows another possible heat pump arrangement. A heat source in the pocket of the grand composite curve is pumped through a relatively small temperature difference to replace medium-pressure (MP) steam. To compensate for taking heat from within the pocket of the grand composite curve, additional low-pressure (LP) steam is used for process heating to maintain the heat balance. Thus, the heat pump results in a saving in MP steam for a sacrifice of extra LP steam usage and the power required for the heat pump. This might be economic if there is a large cost difference between MP and LP steam. Although the heat pump does not operate across the process pinch, it does not violate the principle of the appropriate placement of heat pumps. The heat pump in Figure 16.39b operates across a utility pinch.

16.11 HEAT EXCHANGER NETWORK ENERGY TARGETS – SUMMARY

The energy targets for the process can be set without having to design the heat exchanger network and utility system. These energy targets can be calculated directly from the material and energy balance. Thus, energy costs can be established without design for the outer layers of the process onion. Using the grand composite curve, different utility scenarios can be screened quickly and conveniently, including cogeneration and heat pumps.

16.12 EXERCISES

1. A heat recovery problem consists of two streams given in Table 16.10:

Table 16.10 Stream data for Exercise 1.

Stream	Type	Supply temperature (°C)	Target temperature (°C)	Enthalpy change (MW)
1	Hot	100	40	12
2	Cold	10	150	7

Steam is available at 180°C and cooling water at 20°C.

a. For a minimum permissible temperature difference (ΔT_{min}) of 10°C, calculate the minimum hot and cold utility requirements.
b. What are the hot and cold stream pinch temperatures?
c. If the steam is supplied as the exhaust from a steam turbine, is the heat engine appropriately placed?
d. If the (ΔT_{min}) is increased to 20°C, what will happen to the utility requirements?

2. The stream data for a process are given in Table 16.11.

Table 16.11 Stream data for Exercise 2.

Stream No.	Stream Type	T_S (°C)	T_T (°C)	Heat duty (MW)
1	Hot	160	40	3.6
2	Hot	50	50	5.0
3	Hot	140	110	1.5
4	Cold	160	160	5.0
5	Cold	60	150	1.8

a. Sketch the composite curves for $\Delta T_{min} = 10$°C.
b. From the composite curves, determine the target for hot and cold utility for ΔT_{min} 10°C.

3. The problem table cascade for a process is given in the Table 16.12 for $\Delta T_{min} = 10°C$.

Table 16.12 Problem table cascade for Exercise 3.

Interval temperature (°C)	Cascade heat flow (MW)
360	9.0
310	7.6
230	8.0
210	4.8
190	5.8
190	0
170	1.0
150	7.6
60	3.0

The minimum permissible temperature differences for various options are given in Table 16.13.

Table 16.13 ΔT_{min} for different matches.

Type of match	ΔT_{min} (°C)
Process/process	10
Process/steam	10
Process/flue gas (furnace or gas turbine)	50

a. Calculate the amount of fuel required to satisfy the heating requirements for a furnace used to supply hot utility for a theoretical flame temperature of 1800°C and acid dew point of 150°C.

b. If the furnace from Part *a* is used in conjunction with saturated steam at 250°C for heating, such that the heat duty for the steam is maximized, what is the heat duty on the steam and the furnace heat duty?

4. The stream data for a process involving an exothermic chemical reaction are given in the Table 16.14.

Table 16.14 Stream data for Exercise 4.

Stream No.	Stream Type	Enthalpy Change (kW)	T_S (°C)	T_T (°C)
1	Hot	−7000	377	375
2	Hot	−3600	376	180
3	Hot	−2400	180	70
4	Cold	2400	60	160
5	Cold	200	20	130
6	Cold	200	160	260

A problem table analysis of the data indicates that it is a threshold problem requiring only cold utility. The threshold value of ΔT_{min} is 117°C, corresponding with a cold utility duty of 10,100 kW. It is proposed to use steam generation as cold utility for which $\Delta T_{min} = 10°C$.

a. Assume that saturated boiler feedwater is available and that the steam generated is saturated, in order to calculate how much steam can be generated by the process at a pressure of 41 bar. The temperature of saturated steam at this pressure is 252°C and the latent heat is 1706 $kJ{\cdot}kg^{-1}$.

b. If the steam is superheated to a temperature of 350°C, calculate how much steam can be generated at 41 bar. Assume the heat capacity of steam is 4.0 $kJ{\cdot}kg^{-1}{\cdot}K^{-1}$.

c. What would happen if the steam in Part *b* was generated from boiler feedwater at 100°C with a heat capacity of 4.2 $kJ{\cdot}kg^{-1}{\cdot}K^{-1}$. How would the steam generation be calculated under these circumstances?

5. The problem table cascade for a process is given in Table 16.15 for $\Delta T_{min} = 10°C$.

Table 16.15 Problem table cascade for Exercise 5.

Interval temperature (°C)	Heat flow (MW)
495	3.6
455	9.2
415	10.8
305	4.2
285	0
215	16.8
195	17.6
185	16.6
125	16.6
95	21.1
85	18.1

It is proposed to provide process cooling by steam generation from boiler feedwater with a temperature of 80°C.

a. Determine how much saturated steam can be generated at a temperature of 230°C. The latent heat of steam under these conditions is 1812 $kJ{\cdot}kg^{-1}$. The heat capacity of water can be taken as 4.2 $kJ{\cdot}kg^{-1}{\cdot}K^{-1}$.

b. Determine how much steam can be generated with a saturation temperature of 230°C and superheated to the maximum temperature possible against the process. The heat capacity of steam can be taken as 3.45 $kJ{\cdot}kg^{-1}{\cdot}K^{-1}$.

c. Calculate how much power can be generated from the superheated steam from Part *b*, assuming the exhaust steam is saturated at a pressure of 4 bar.

6. The problem table cascade for a process is given in Table 16.16 for $\Delta T_{min} = 20°C$.

Table 16.16 Problem table cascade for Exercise 6.

Interval temperature (°C)	Heat flow (MW)
440	21.9
410	29.4
130	23.82
130	1.8
100	0
95	15

The process also has a requirement for 7 MW of power. Three alternative utility schemes are to be compared economically:

a. A steam turbine with its exhaust saturated at 150°C used for process heating is one of the options to be considered. Steam is raised in the central boiler house at 41 bar with an enthalpy of 3137 $kJ \cdot kg^{-1}$. The enthalpy of saturated steam at 150°C is 2747 $kJ \cdot kg^{-1}$ and its latent heat is 2115 $kJ \cdot kg^{-1}$. Calculate the maximum power generation possible by matching the exhaust steam against the process.
b. A second possible scheme uses a gas turbine with an exhaust temperature of 400°C and heat capacity flowrate of 0.1 $MW \cdot K^{-1}$. Calculate the power generation if the turbine converts heat to power with an efficiency of 30%. Ambient temperature is 10°C.
c. A third possible scheme integrates a heat pump with the process. The power required by the heat pump is given by

$$W = \frac{Q_H}{0.6} \frac{T_H - T_C}{T_H}$$

where Q_H is the heat rejected by the heat pump at T_H (K). Heat is absorbed into the heat pump at T_C (K). Calculate the power required by the heat pump.
d. The cost utilities are given in the Table 16.17.

Table 16.17 Utility costs for Exercise 6.

Utility	Cost ($\$ \cdot MW^{-1}$)
Fuel for gas turbine	0.0042
Fuel for steam generation	0.0030
Imported power	0.018
Credit for exported power	0.014
Cooling water	0.00018

The overall efficiency of steam generation and distribution is 60%. Which scheme is most cost-effective, the steam turbine, the gas turbine or the heat pump?

7. The Table 16.18 represents a problem table cascade (ΔT_{min} = 20°C).

Table 16.18 Problem table cascade for Exercise 7.

Interval temperature (°C)	Heat flow (kW)
160	1000
150	0
130	1100
110	1400
100	900
80	1300
40	1400
10	1800
-10	1900
-30	2200

The following utilities are available:
(i) MP steam at 200°C
(ii) LP steam at 107°C raised from boiler feed water at 60°C
(iii) Cooling water (20 to 40°C)
(iv) Refrigeration at 0°C
(v) Refrigeration at −40°C

For matches between process and refrigeration, ΔT_{min} = 10°C. Draw the process grand composite curve and set the targets for the utilities. Below the pinch use of higher temperature, cold utilities should be maximized. For boiler feedwater, the specific heat capacity is 4.2 $kJ \cdot kg \cdot K^{-1}$ and the latent heat of vaporization is 2238 $kJ \cdot kg^{-1}$.

REFERENCES

1. Hohman EC (1971) *Optimum Networks of Heat Exchange*, PhD Thesis, University of Southern California.
2. Huang F and Elshout RV (1976) Optimizing the Heat Recover of Crude Units, *Chem Eng Prog*, **72**: 68.
3. Linnhoff B, Mason DR and Wardle I (1979) Understanding Heat Exchanger Networks, *Comp Chem Eng*, **3**: 295.
4. Linnhoff B, Townsend DW, Boland D, Hewitt GF, Thomas BEA, Guy AR and Marsland RH (1982) *A User Guide on Process Integration for the Efficient Use of Energy*, IChemE, UK.
5. Polley GT (1993) Heat Exchanger Design and Process Integration, *Chem Eng*, **8**: 16.
6. Umeda T, Itoh J and Shiroko K (1978) Heat Exchange System Synthesis, *Chem Eng Prog*, **74**: 70.
7. Umeda T, Harada T and Shiroko K (1979) A Thermodynamic Approach to the Synthesis of Heat Integration Systems in Chemical Processes, *Comp Chem Eng*, **3**: 273.
8. Umeda T, Niida K and Shiroko K (1979) A Thermodynamic Approach to Heat Integration in Distillation Systems, *AIChE J*, **25**: 423.
9. Linnhoff B and Flower JR (1978) Synthesis of Heat Exchanger Networks, *AIChE J*, **24**: 633.
10. Ahmad S and Hui DCW (1991) Heat Recovery Between Areas of Integrity, *Comp Chem Eng*, **15**: 809.
11. Cerda J, Westerberg AW, Mason D and Linnhoff B (1983) Minimum Utility Usage in Heat Exchanger Network Synthesis – A Transportation Problem, *Chem Eng Sci*, **38**: 373.
12. Papoulias SA and Grossmann IE (1983) A Structural Optimization Approach in Process Synthesis – II Heat Recovery Networks, *Comp Chem Eng*, **7**: 707.
13. Itoh J, Shiroko K and Umeda T (1982) Extensive Application of the T-Q Diagram to Heat Integrated System Synthesis, *International Conference on Proceedings Systems Engineering (PSE-82)*, Kyoto, 92.
14. Townsend DW and Linnhoff B (1983) Heat and Power Networks in Process Design, *AIChE J*, **29**: 742.
15. Linnhoff B and de Leur J (1988) Appropriate Placement of Furnaces in the Integrated Process, *IChemE Symp Ser*, **109**: 259.