CHAPTER 18. REFRIGERATION PLANT AND MINE AIR CONDITIONING SYSTEMS

18.1. INTRODUCTION ........................................................................................................ 2

18.2. THE VAPOUR COMPRESSION CYCLE ....................................................................... 2
  18.2.1. Basic principles ........................................................................................................ 2
  18.2.2. Refrigerant fluids ..................................................................................................... 4
  18.2.3. Basic components of the vapour compression cycle ................................................. 5
  18.2.4. Performance of a refrigeration cycle ......................................................................... 6
     18.2.4.1 The Carnot cycle .................................................................................................... 6
     18.2.4.2. The actual cycle .................................................................................................... 8
  18.2.5. Coefficient of Performance, COP ............................................................................ 8

18.3. COMPONENTS AND DESIGN OF MINE COOLING SYSTEMS ................................. 15
  18.3.1. Overview of mine cooling systems .......................................................................... 15
  18.3.2. Heat exchangers ........................................................................................................ 20
     18.3.2.1. Indirect heat exchangers ...................................................................................... 22
     18.3.2.2. Performance calculations for indirect heat exchangers ........................................ 22
     18.3.2.3. Direct heat exchangers ........................................................................................ 26
     Chilled water spray chambers ......................................................................................... 28
     18.3.2.4. Performance calculations for direct heat exchangers ........................................... 31
     18.3.2.5. Heat exchange across the walls of pipes and ducts ............................................ 41
        Air inside a duct ............................................................................................................. 42
        Water inside a pipe ........................................................................................................ 42
        Air outside a pipe .......................................................................................................... 43
        Radiative heat transfer coefficient ................................................................................ 43
  18.3.3. Water distribution systems ..................................................................................... 45
     18.3.3.1. Pipe sizing .......................................................................................................... 46
     18.3.3.2. Pipe insulation ..................................................................................................... 47
        Table 18.3. Thermal conductivities of pipe materials .................................................. 48
     18.3.3.3. Energy and temperature changes within water systems .................................... 48
  18.3.4. Energy recovery devices ......................................................................................... 53
     18.3.4.1. Water turbines .................................................................................................... 54
     18.3.4.2. Hydrolift systems ............................................................................................... 55
  18.3.5. Design of mine cooling systems .............................................................................. 56
     18.3.5.1. Location of main plant ........................................................................................ 56
        Surface plant .................................................................................................................. 57
        Underground plant ........................................................................................................ 57
        Combinations of surface and underground plant ......................................................... 59
     18.3.5.2. Service water cooling .......................................................................................... 61
     18.3.5.3. Hydropower ........................................................................................................ 62
     18.3.5.4. Ice systems ........................................................................................................... 62
        Manufacture of ice .......................................................................................................... 64
        Transportation of ice ...................................................................................................... 65
        Incorporation into the mine cooling system .................................................................. 66
        Economics of ice systems ............................................................................................... 67
  18.3.6. Summary of design process .................................................................................... 67

18.4. AIR HEATING ............................................................................................................. 68
  18.4.1. Utilization of waste heat .......................................................................................... 69
  18.4.2. Heat recovery from exhaust air ................................................................................ 69
  18.4.3. Direct heating .......................................................................................................... 71
     Table 18.4. Gross calorific values of gases .................................................................... 71
  18.4.4. Indirect heating ........................................................................................................ 72
  18.4.5. Ice stopes ................................................................................................................ 73
  18.4.6. Geothermal and cycled storage heat ...................................................................... 74

References .......................................................................................................................... 76
18.1. INTRODUCTION

One of the earliest methods of temperature control in underground mines was the importation of naturally produced ice from the surface. Blocks of ice were transported in ore cars to cool miners in the Comstock Lode under Virginia City in Nevada, USA, during the 1860's. The vapour compression refrigeration cycle, currently the most widespread method of artificial cooling, appears to have first been used in mining during the 1920's. Examples included the famous Morro Velho Mine in Brazil (1923) and experimental work in British coal mines (Hancock, 1926). Air cooling techniques in mining gained further recognition in the 1930's including their utilization in the gold mines of South Africa and in the Kolar Goldfields of India. However, it was the 1960's that saw the start of a real escalation of installed mine cooling capacity.

Large centralized refrigeration plant, located underground, became popular in the South African gold mines. Limitations on the heat rejection capacity of return air, combined with the development of energy recovery devices for water pipelines in shafts and improved "coolth" distribution systems led to a renewed preference for surface plant. (The term "coolth" is a decidedly unscientific but descriptive word sometimes used in association with chilled fluid distribution systems.)

Although the traditional role of mine cooling has been to combat geothermal heat and the effects of autocompression in deep metal mines, an additional influence has been the escalating amount of mechanized power employed underground, particularly in longwall coal mines. This resulted in smaller scale and more localized use of air cooling units in such mines at depths where, prior to intensive mechanization, heat had not been a limiting environmental problem. Examples of these installations appeared in the United Kingdom and Germany.

In this chapter, we shall examine the essential theory of the vapour compression refrigeration cycle in addition to discussing the design of mine cooling systems and some of the methods of distributing "coolth" to the working areas of a subsurface facility. The chapter concludes with a section on the opposite problem, that of increasing the temperature of the intake air for mines in cold climates.

18.2. THE VAPOUR COMPRESSION CYCLE

There are many transient phenomena that are known to produce a cooling effect, varying from endothermic chemical reactions to the sublimation of solid carbon dioxide (dry ice). Where a continuous cooling effect is required, then a means must be employed by which a supply of mechanical, electrical or thermal energy is utilized to remove heat from some source, and to transport it to a thermal sink where it can be rejected. If the primary objective is to cool the source, then the device is known as a refrigerator. If, however, the desired effect is to heat the sink, then it is called a heat pump. In fact, both effects occur simultaneously. Hence, a domestic refrigerator cools the interior of the container but heats the air in the kitchen.

Of the several devices that have been developed to achieve a continuous refrigeration or heat pump effect, the most common is based on the vapour compression cycle. This may be used on small units such as air conditioning equipment fitted to automobiles or for very large scale cooling of mine workings where many megawatts of heat require to be transferred. In this section, we shall examine the basic principles of the vapour compression cycle, how rates of heat exchange may be calculated, and the essential components of a refrigeration plant.

18.2.1. Basic principles

When a liquid boils, it does so at constant temperature provided that the applied pressure remains fixed. The heat added is utilized in increasing the internal kinetic energy of the molecules until they
can no longer remain in the liquid phase but burst free to form a vapour or gas (Section 2.1.1). If, however, the applied pressure is raised to a higher value, then additional heat is required to vaporize the liquid. The boiling temperature will increase. The relationship between pressure and boiling point for any given liquid may be defined as the **vapour pressure line** on a pressure-temperature diagram such as that shown on Figure 18.1. The fluid is a liquid on the left of the curve and a vapour (gas) on the right of the curve. The liquid may be vaporized either by increasing the temperature or decreasing the pressure. Similarly, condensation from vapour to liquid may occur either by decreasing the temperature or increasing the pressure.

The change in physical appearance of a fluid on crossing the vapour pressure line is quite distinct, liquid to gas or vice versa. However, there is a critical pressure-temperature coordinate beyond which the change of phase is gradual rather than sudden and there is no clearly defined moment of evaporation or condensation. This is known as the **critical point**.

Although the vapour pressure curve appears as a single line on the pressure-temperature diagram, it takes a finite amount of time and energy exchange to cross that line. During this time, part of the fluid will be liquid and the remainder will be vapour. This is the situation that exists inside a boiling kettle. The region within which the two phases coexist is more clearly shown on the PV (pressure against specific volume) diagram of Figure 18.2(a). The corresponding temperature-entropy (Ts) and pressure-enthalpy (PH) diagrams are shown on Figures 18.2(b) and (c) respectively. The latter two diagrams are particularly useful in analyzing and quantifying both the work and heat transfer processes in the vapour compression cycle.

![Figure 18.1 A pressure-temperature diagram showing limits of phase change.](image-url)
In all three diagrams of Figure 18.2, a horizontal line within the two phase region indicates that the pressure and temperature both remain constant during any given isobaric phase change. However, the volume, entropy and enthalpy all increase significantly during vaporization. Different horizontal lines on anyone of these diagrams will indicate different values of pressure and corresponding boiling (or condensation) temperature (Figure 18.1). When a liquid boils at a given value of applied pressure it will extract heat from the surroundings or other available medium. If the vapour thus produced is then transported to a new location and compressed to a higher pressure, then it can be condensed at a correspondingly higher temperature, yielding up its heat of condensation to the new surroundings or any cooling medium that may be supplied. This is the basic principle underlying the vapour compression refrigeration cycle.

18.2.2. Refrigerant fluids

The differing pressure-temperature relationships of various fluids allow each of those fluids to act as a refrigerant over specified temperature ranges. Carbon dioxide and even water have been used as refrigerants. For the ranges of pressures and temperatures acceptable in refrigeration plant for air conditioning, ammonia is a particularly efficient refrigerant although its toxicity limits its use. The ideal refrigerant for mining use is one that gives a high efficiency of heat transfer, is non-corrosive to metal, non-toxic and has a boiling temperature close to 0°C at a positive pressure with respect to the ambient atmosphere. Fluorinated hydrocarbons have been widely employed in many types of industrial plant as well as domestic refrigeration equipment. Although non-toxic, fluorinated hydrocarbons react adversely with atmospheric ozone and are being replaced by alternative refrigerants.

Refrigerant fluids are commonly referred to, not by their chemical names, but by R (refrigerant) number. Table 18.1 indicates some of those designations.

<table>
<thead>
<tr>
<th>Refrigerant No.</th>
<th>Chemical name</th>
<th>Refrigerant No</th>
<th>Chemical Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>R11</td>
<td>trichlorofluoromethane</td>
<td>R122</td>
<td>trichlorodifluoroethane</td>
</tr>
<tr>
<td>R12</td>
<td>dichlorodifluoromethane</td>
<td>R170</td>
<td>ethane</td>
</tr>
<tr>
<td>R22</td>
<td>chlorodifluoromethane</td>
<td>R290</td>
<td>propane</td>
</tr>
<tr>
<td>R23</td>
<td>trifluoromethane</td>
<td>R600</td>
<td>butane</td>
</tr>
<tr>
<td>R30</td>
<td>dichloromethane</td>
<td>R717</td>
<td>ammonia</td>
</tr>
<tr>
<td>R50</td>
<td>methane</td>
<td>R718</td>
<td>water</td>
</tr>
<tr>
<td>R110</td>
<td>hexachloroethane</td>
<td>R744</td>
<td>carbon dioxide</td>
</tr>
<tr>
<td>R113</td>
<td>trichlorotrifluoroethane</td>
<td>R1150</td>
<td>ethylene</td>
</tr>
<tr>
<td>R120</td>
<td>pentachloroethane</td>
<td>R1270</td>
<td>propylene</td>
</tr>
</tbody>
</table>

Table 18.1 Refrigerant numbers of selected fluids.
18.2.3. Basic components of the vapour compression cycle

Figure 18.3 shows that there are four essential components of hardware in a vapour compression refrigeration unit. The evaporator is a heat exchanger, typically of the shell-and-tube configuration in mining refrigeration plant. In the larger units, the refrigerant liquid is on the outside of the tubes while the medium to be cooled (for example, water, brine or glycol) passes through the tubes. Smaller units employed for direct cooling of an airstream are sometimes called "direct evaporators" and contain the refrigerant within the tubes while the air passes over their outer surface.

Within the evaporator, the refrigerant pressure is maintained at a relatively low level and boils at a correspondingly low temperature. For example, refrigerant R12 will boil at 4°C if the pressure is 351 kPa (approximately 3.5 atmospheres). The heat required to maintain the boiling is extracted from the gas or liquid passing on the other side of the tube walls. Hence, that gas or liquid is cooled. The refrigerant, now vaporized, collects at the top of the evaporator and is allowed to gain a few degrees of superheat to ensure full vaporization before it passes on the compressor. Except for direct evaporators, it may be necessary to insulate the external surface of the evaporator in order to prevent excessive heat gain from the ambient atmosphere.

Figure 18.3 Major components of a vapour compression cycle.
The compressor is the device where mechanical work is input to the system. Reciprocating, screw or centrifugal compressors are all employed. The latter are favoured for the larger units and where the required pressure ratio remains constant. However, the development of large diameter screw compressors allows a good efficiency to be maintained under conditions of variable cooling load. (Baker-Duly, 1989). Multi-stage compressors are employed to give high differentials of pressure and, hence, large temperature differences between the evaporator and condenser. Electric motors are normally employed to drive the compressors on mine refrigeration units although internal combustion engines may be used on surface or as standby units. The duty of a refrigeration plant can be modified by changing the speed of the compressor. The flow of vapour through the compressor and, hence, rate of heat transfer can also be controlled by inlet guide vanes.

The refrigerant vapour leaves the compressor and passes into the condenser at a relatively high pressure and temperature. The condenser itself may be of similar construction to the evaporator, that is, a shell-and-tube heat exchanger. Heat is removed from the refrigerant by air, water or some other fluid medium to the extent that the refrigerant cools and condenses back to a liquid. As the pressure is high, this occurs at a relatively high temperature. At a pressure of 1217 kPa, refrigerant R12 will condense at 50°C. The latent heat of condensation is removed by the cooling fluid for subsequent rejection in a cooling tower or other type of separate heat exchanger. As the vapour compression cycle is a closed system (ignoring heat losses or gains from the surrounding atmosphere), the rate at which heat is removed from the refrigerant in the condenser must equal the combined rates of heat addition in the evaporator and work provided by the compressor.

The condensed refrigerant passes from the condenser to the fourth and final component of the cycle. This is the expansion valve whose purpose is simply to reduce the pressure of the refrigerant back to evaporator conditions. An expansion valve may be a simple orifice plate or can be controlled by a float valve. At the exit from the expansion valve, the liquid is at low pressure and has a correspondingly low boiling temperature. Provided that the pipework is insulated, the latent heat for boiling can come only from the liquid refrigerant itself. Hence, the temperature of the refrigerant drops rapidly as it passes from the expansion valve to the evaporator where it enters as a mixture of liquid and vapour, thus closing the cycle.

18.2.4. Performance of a refrigeration cycle

18.2.4.1 The Carnot cycle

Electrical motors or heat engines are devices that convert one form of energy into another. Their efficiency may, therefore, be defined as an energy output/input ratio. However, in the case of refrigerators or heat pumps, the purpose is to remove heat from a given source and to reject it at a higher temperature to a receiving sink. A different measure of performance is required.

Figure 18.4 shows the temperature-entropy diagram for the ideal (frictionless) Carnot cycle to which an actual vapour compression cycle can aspire but never attain. We may follow the ideal cycle by commencing at position 1, the entry of refrigerant vapour to the compressor.

The ideal compression process is isentropic (Section 10.6). The compressed refrigerant vapour enters the condenser at position 2 and, in the ideal condenser, passes through to position 3 with neither a pressure drop nor a fall in temperature. This assumes frictionless flow and perfect heat transfer. The ideal expansion valve allows an isentropic fall in pressure and temperature to station 4, the entrance to the evaporator. The cycle closes via frictionless flow and perfect heat transfer in the evaporator.
In order to quantify the ideal Carnot cycle, we apply the steady-flow energy equation (3.25). The kinetic and potential energy terms are small and, in any event, cancel out in a closed cycle. Using the numbered station points from Figure 18.4, the steady-flow energy equation gives:

compressor: \[ W_{12} = H_2 - H_1 \quad \text{J/kg} \quad (18.1) \]

condenser: \[ q_{23} = H_3 - H_2 \quad \text{J/kg} \quad (18.2) \]

evaporation valve: \[ q_{41} = H_4 - H_3 \quad \text{J/kg} \quad (18.3) \]

evaporator: \[ q_{41} = H_1 - H_4 \quad \text{J/kg} \quad (18.4) \]

cycle summation: \[ W_{12} + q_{23} + q_{41} = 0 \quad \text{J/kg} \quad (18.5) \]

where:

- \( W_{12} \) = mechanical energy added by the compressor
- \( q_{23} \) = heat exchange in the condenser (this is numerically negative as heat is leaving the refrigerant)
- \( q_{41} \) = heat added to the refrigerant in the evaporator
- \( H \) = enthalpy

We can rewrite equation (18.5) as

\[ W_{12} + q_{41} = - q_{23} \quad \text{J/kg} \quad (18.6) \]

thus confirming our earlier statement that the heat rejected in the condenser is numerically equal to the sum of the compressor work and the heat added to the refrigerant in the evaporator.
The measure of performance of a refrigeration cycle is known as the **Coefficient of Performance**, \(\text{COP}\), and is defined as

\[
\text{COP} = \frac{\text{Useful cooling effect (evaporator heat transfer)}}{\text{Work input from the compressor}} = \frac{q_{41}}{W_{12}} \quad (18.7)
\]

Using equation (18.5) once again, this may be rewritten as

\[
\text{COP} = \frac{q_{41}}{-q_{23} - q_{41}} \quad (18.8)
\]

Recalling that the area under a process line on a Ts diagram represents heat (Section 3.5), the terms in equation (18.8) may be related to Figure 18.4 for the Carnot cycle:

\[
q_{41} = \text{Area 41ab} \quad -q_{23} = \text{Area 23ba}
\]

and

\[
-q_{23} - q_{41} = \text{Area 1234} \quad (\text{remember that } q_{23} \text{ is numerically negative})
\]

Hence, \(W_{12} = \text{Area 1234}\) and equation (18.8) becomes

\[
\text{Carnot COP} = \frac{\text{Area 41ab}}{\text{Area 1234}} = \frac{T_1 (s_a - s_b)}{(T_2 - T_1)(s_a - s_b)}
\]

\[
= \frac{T_1}{T_2 - T_1} \quad \text{or} \quad \frac{T_4}{T_3 - T_4} \quad (18.9)
\]

as \(T_1 = T_4\) and \(T_2 = T_3\)

Hence, the ideal or Carnot coefficient of performance is given as the ratio:

\[
\text{Carnot COP} = \frac{\text{Evaporator temperature (absolute)}}{\text{Condenser temperature} - \text{Evaporator temperature}} \quad (18.10)
\]

**Example**
The evaporator and condenser of a refrigeration unit have temperatures of 4 and 50°C respectively. Determine the maximum possible coefficient of performance of this unit.

**Solution**
The Carnot or ideal coefficient of performance is given by equation (18.10)

\[
\text{Carnot COP} = \frac{(273.15 + 4)}{(50 - 4)} = 6.025
\]

18.2.4.2. The actual cycle

A real vapour compression cycle has a coefficient of performance that is necessarily lower than the corresponding Carnot COP. There are two reasons for this. First, in actual compressors and expansion valves, there are, inevitably, increases in entropy. Furthermore, as the refrigerant passes through real condensers and evaporators there will be slight changes in pressure and temperature. In
a well designed unit, the latter are small compared with the differences in pressure or temperature between the condenser and evaporator.

Secondly, it would be impractical to design a refrigeration unit that attempted to follow the Carnot cycle. Referring again to Figure 18.4, it can be seen that station 1 (entry to the compressor) lies within the two-phase region. The presence of liquid droplets would cause severe erosion of the compressor impeller. It is for this reason that a few degrees of superheat are imparted to the vapour before it leaves a real evaporator.

Figure 18.5 illustrates the temperature-entropy and pressure-enthalpy diagrams for a practical vapour compression cycle. The major difference between Figure 18.5(a) and the Carnot cycle shown on Figure 18.4 is that the compression commences slightly beyond the saturated vapour line (station 1) and involves an increase in entropy to station 2. On entering the condenser, the superheated vapour undergoes a near isobaric cooling process (stations 2 to 2'), losing sensible heat to the cooling medium until condensing temperature is reached at station 2'.

Although the process through the expansion valve is no longer isentropic, the fluid remains at constant enthalpy (no heat exchange takes place) and, hence, appears on the pressure-enthalpy diagram as a vertical line 3 to 4.

Equations (18.1 to 18.4) arose from those parts of the steady-flow energy equation that do not involve the friction term. They apply equally well to both the real and the ideal cases. Furthermore, the definition of coefficient of performance given in equation (18.7) remains applicable. Substituting for $W_{12}$ and $q_{41}$ from equations (18.1) and (18.4) respectively gives

$$\text{Actual COP} = \frac{q_{41}}{W_{12}} = \frac{H_1 - H_4}{H_2 - H_1} \quad (18.11)$$

The reason for introducing the pressure-enthalpy diagram now emerges. The coordinate points for real vapour compression cycles can be plotted on pressure-enthalpy-temperature charts that have been derived through tests on many refrigerant fluids. An example is given for refrigerant R12 on Figure 18.6. The values of enthalpy may be read from the relevant chart enabling the actual coefficient of performance and other parameters to be determined. For more precise work, tables of the thermodynamic behaviour of refrigerants are available. The values of enthalpy and entropy on the charts and tables are based on a specified datum temperature. This is usually either absolute zero (-273.15°C) or -40°C. The datum employed has little import as it is differences in values that are used in practical calculations.

A measure of efficiency of the refrigeration unit may now be defined by comparing the actual COP with the ideal Carnot COP.

$$\text{Cycle efficiency} = \frac{\text{Actual COP}}{\text{Carnot COP}} \quad (18.12)$$
Figure 18.5 Temperature-entropy and pressure-enthalpy diagrams for an actual vapour compression cycle.
Figure 18.6  Pressure-enthalpy diagram for refrigerant R12 (based on zero degrees Kelvin).  
Example
A plant employing refrigerant R12 is used to chill water for a mine distribution system. Water is also used to remove heat from the condenser. The following measurements are made:

**Evaporator:**
- Water flowrate: 50 litres/s
- Water inlet temperature: 20°C
- Water outlet temperature: 10°C
- Refrigerant pressure: 363 kPa

**Condenser:**
- Water flowrate: 140 litres/s
- Refrigerant pressure: 1083 kPa

**Compressor:**
- Refrigerant inlet temperature: 7°C
- Refrigerant outlet temperature: 65°C

Using the pressure-enthalpy chart for R12, analyze the performance characteristics of the unit.

Solution
Step 1: On the pressure-enthalpy diagram, draw horizontal lines to represent the pressures in the evaporator (363 kPa) and condenser (1083 kPa). For clarity, the relevant area of the R12 pressure-enthalpy chart has been enlarged and reproduced as Figure 18.7. The corresponding temperatures may be estimated directly from the chart.

\[ t_{\text{evap}} = t_4 = 5°C \]
\[ t_{\text{cond}} = t_3 = 45°C \]

These temperatures can, of course, be verified by direct measurement.

Step 2: Establish the nodal points on the pressure-enthalpy chart and read the corresponding values of enthalpy.

The temperature of the refrigerant entering the compressor (station 1) is given as 7°C. This represents \( 7 - 5 = 2° \) of superheat. Station 1 is, therefore, established on the evaporator pressure line at a temperature of 7°C. Note that the lines of constant temperature change from horizontal to near vertical on passing the saturated vapour line. We have ignored the small pressure drop between the evaporator and the compressor inlet.

Station 2 is established in a similar manner, i.e. at the measured compressor outlet temperature of 65°C and lying on the condenser pressure line.

Station 3, representing the refrigerant leaving the condenser, lies on the junction of the evaporator pressure line and the saturated liquid curve.

The expansion process is represented by a vertical line dropped from station 3 until it intersects the evaporator pressure line at station 4.

The four critical enthalpies may now be estimated from the chart:

<table>
<thead>
<tr>
<th>Component</th>
<th>Enthalpy (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor inlet</td>
<td>( H_1 = 356 )</td>
</tr>
<tr>
<td>Compressor outlet/condenser inlet</td>
<td>( H_2 = 386 )</td>
</tr>
<tr>
<td>Condenser outlet</td>
<td>( H_3 = 244 )</td>
</tr>
<tr>
<td>Evaporator inlet</td>
<td>( H_4 = 244 )</td>
</tr>
</tbody>
</table>
Step 3: Establish the coefficients of performance and cycle efficiency:

Carnot COP = \( \frac{T_4}{(t_3 - t_4)} = \frac{(273.15 + 5)}{(45 - 5)} = 6.954 \) equations (18.9 or 18.10)

Actual COP = \( \frac{(H_1 - H_4)}{(H_2 - H_1)} = \frac{356 - 244}{386 - 356} = 3.733 \) equation (18.11)

Cycle efficiency = \( \frac{Actual COP}{Carnot COP} = \frac{3.733}{6.954} \times 1000 = 53.7 \) per cent equation (18.12)

Step 4: Determine the useful cooling effect, or evaporator duty, \( q_{evap} \). This can be established from the known flowrate and temperature drop of the water being cooled as it passes through the evaporator:

\[
q_{evap} = m_{w, evap} \cdot C_w \cdot \Delta T_{w, evap} \quad kW
\]  

where: water mass flowrate \( m_{w, evap} = 50 \) litres/s = 50 kg/s  
specific heat of water \( C_w = 4.187 \) kJ/kg°C  
temperature drop of water \( \Delta T_{w, evap} = 20 - 10 = 10°C \)

giving \( q_{evap} = 50 \times 4.187 \times 10 \) kJ/s or kW = 2093.5 kW
Step 5: Determine the mass flowrate of refrigerant, \(m_r\), around the system. The evaporator duty, already found to be 2093.5 kW, can also be expressed as

\[
q_{\text{evap}} = m_r (H_1 - H_4) \quad \text{kW}
\]

Hence,

\[
m_r = \frac{2093.5}{356 - 244} = 18.69 \quad \text{kg/s}
\]

Step 6: Determine the compressor duty. The work input from the compressor to the refrigerant, \(W_{12}\), can be established in two ways. First,

\[
W_{12} = m_r (H_2 - H_1) = 18.69(386 - 356) = 560.7 \quad \text{kW}
\]

and, secondly, from the definition of coefficient of performance given in equation (18.7),

\[
W_{12} = \frac{q_{\text{evap}}}{\text{Actual COP}} = \frac{2093.5}{3.733} = 560.8 \quad \text{kW}
\]

The slight difference arises from rounding errors.

Step 7: Evaluate the condenser duty. The heat transferred from the refrigerant in the condenser is simply

\[-q_{\text{cond}} = q_{\text{evap}} + W_{12} = 2093.5 + 560.7 = 2654.2 \quad \text{kW} \quad (\text{equation (18.6)})\]

As the cooling water flowrate through the condenser is known to be \(m_{w,\text{cond}} = 140 \quad \text{kg/s}\), the temperature rise of this water, \(\Delta T_{w,\text{cond}}\), may be determined from

\[-q_{\text{cond}} = m_{w,\text{cond}} C_w \Delta T_{w,\text{cond}}\]

\[
giving \quad \Delta T_{w,\text{cond}} = \frac{2654.2}{140 \times 4.187} = 4.53 \quad \text{°C}
\]

The performance of a refrigeration unit may be enhanced beyond that illustrated in the example. For large units, it may be necessary to employ two or more stages of compression in order to achieve the desired difference in temperature between evaporator and condenser. In that case, it is usual to employ the same number of expansion valves as stages of compression. A fraction of the refrigerant is vaporized on passing through an expansion valve. That fraction can be read from the corresponding point on the pressure-enthalpy diagram or calculated from the isobaric enthalpies at the saturated liquid and saturated vapour curves. This "flash gas" may be separated out and piped to the corresponding intermediate point between compressor stages. This is known as interstage cooling or, simply, intercooling and reduces the total mass of refrigerant that passes through all stages of compression. The result is to reduce the required compressor power and/or increase the useful refrigerating effect.

In some units, the hot liquid refrigerant leaving the condenser is partially cooled by a secondary cooling circuit. This subcooling reduces the fraction of flash gas produced at the downstream side of expansion valves and gives a small increase in the performance of the plant.
The coefficient of performance as defined by equation (18.7) takes into account the isentropic efficiency of the compressor. However, it does not cater for the efficiency of the motor or other device that drives the compressor. Hence, the $W_{12}$ in equation (18.7) may be replaced by the power fed to the compressor drive unit to give a more practical coefficient of performance. Furthermore, the compressor is not the only device that consumes energy in a refrigeration unit. There will also be water pumps or fans to promote the flow of the cooled medium at the evaporator and the cooling medium at the condenser. An overall coefficient of performance may be defined as:

$$\text{Overall } COP = \frac{\text{Heat transfer in the evaporator}}{\text{Total energy consumption of the refrigeration unit}}$$

(18.16)

Care should, therefore, be taken that any quoted values of $COP$ are interpreted in the correct manner.

18.3. COMPONENTS AND DESIGN OF MINE COOLING SYSTEMS

18.3.1. Overview of mine cooling systems

In the majority of large scale mine cooling systems, there are usually three sets of heat transfer involved:

- transfer of heat from the work areas to the evaporators of the refrigeration units
- transfer of heat from the evaporators to the condensers in the refrigeration units (the vapour compression cycle)
- transfer of heat from the condensers to the free atmosphere on surface.

There exist tremendous variations in the duty, complexity and efficiency of the hardware involved in these three phases, dependent upon the severity and dispersion of the heat problem in the mine. In this section we shall discuss a range of systems varying from a simple spot cooler for very localized applications to large integrated systems that may be required for deep and hot mines. This will serve both as an overview and an introduction to a more detailed examination of individual components and overall system design.

Figure 18.8 shows a simple application of a direct evaporator or "spot cooler" to a specific working area in a mine. The evaporator of the refrigeration unit takes the form of copper or cupronickel tubular coils located within an air duct. The refrigerant passes through the inside of the evaporator tubes and cools the air flowing along the duct. The heat from the condenser is rejected directly to the return air. This system has the advantage that the cooling effect produced by the refrigeration unit is utilized directly and immediately at the place where cooling is required. There is no loss of efficiency introduced by an intervening water reticulation system between the evaporator and working area. Similarly, heat rejection from the condenser is direct and immediate. The spot cooler is merely an industrial version of a domestic air conditioning unit where heat is rejected directly to the outside atmosphere.

Unfortunately, the spot cooler has a major disadvantage that limits its application in subsurface ventilation systems. Glancing, again, at the blind heading illustrated in Figure 18.8, it is clear that the air emerging from the end of the duct will be at a reduced temperature when the refrigeration unit is operating. Although this is the desired effect, a consequence will be that the flow of strata heat into the heading will increase due to the lowered air temperature, a phenomenon that is examined in Chapter 15. Yet more heat is added as the air returns over the condenser. The net effect is that the return air leaving the area has a greater enthalpy (and, usually, temperature and humidity) than would
be the case if the refrigeration unit were not operating. The increase in enthalpy is the sum of the additional strata heat and the energy taken by the compressor. The use of spot coolers is restricted by the availability of return air and the debilitating effect on psychrometric conditions within local return airways.

There are three ways of alleviating the situation. First, the condenser could take the form of a shell and tube heat exchanger and be cooled by a water circuit (Figure 18.3). The hot water can then be piped away and recirculated through a heat exchanger in a main return airway. Secondly, the refrigeration unit itself may be located away from the working area and water that has been chilled by the evaporator piped to a heat exchanger in the workings. Thirdly, the first two arrangements may be combined, resulting in the system illustrated in Figure 18.9. Here, the refrigeration plant of, perhaps, 2 MW cooling duty, is sited in a stable location and provides chilled water for a number of work areas. The chilled water flows through thermally insulated pipes to heat exchangers in the faces, stopes or headings and returns to the plant via uninsulated pipes. Some of the chilled water may be used for

Figure 18.8 The simplest application of a direct evaporator or spot cooler with local heat rejection in a mine heading.
dust suppression purposes in which case additional make-up water must be supplied to the plant. Hot water from the condenser is recirculated through heat exchangers that are located in a main return airway. If that return airway is to remain open for persons to travel, either for routine purposes or as an emergency escapeway, then the psychrometric conditions must remain physiologically acceptable (Chapter 17). Again, this provides a limit on the degree of heat rejection and, hence, size of refrigeration plant that can be utilized as a district cooler. If, however, a return route can be dedicated fully to heat rejection, then physiological acceptability limits may be exceeded. In this case, the dedicated return or “dirty pipe”, as it is sometimes called, must be made inaccessible to inadvertent entry by personnel. Inspection or maintenance can be carried out either when the plant is shut down or by persons wearing protective clothing (Section 17.5.6). It may be necessary to seek exemption from national or state legislation in order to utilize a “dirty pipe” arrangement.

For mines that have a widespread heat problem, the economies of scale and the need for flexibility indicate a requirement for large centralized cooling facilities. Banks of individual refrigeration units, each producing a typical 3.5 MW of cooling capacity, may be assembled to give a total duty which might exceed 100 MW for a large and deep metal mine. The “coolth” is normally distributed via chilled water lines to provide any required combination of bulk air cooling, face or stope air cooling and chilled service water.
Until the mid-nineteen seventies, centralized plant tended to be located underground in excavated refrigeration chambers close to shaft bottoms. All of the mine return air could then be utilized for heat rejection. In South Africa, it was common for large cooling towers to be situated in, or adjacent to, the upcast shaft bottoms. Chilled water from the centralized underground plant could be transmitted to other levels. However, at elevations greater than some 500 m below the plant, water pressures in the pipe ranges become excessive. This can be counteracted either by pressure reducing valves (adjustable orifices) or by employing water to water heat exchangers and secondary, low pressure, cold water circuits. Unfortunately, there are difficulties associated with both of these types of device. Pressure reducing valves necessitate the employment of high duty pumps to raise the heated water back to the plant elevation. Maintaining the high pressure water in a closed circuit by means of water to water heat exchangers balances the pressure heads in the supply and return shaft ranges. However, the intermediate water to water heat exchangers produce an additional loss of heat transfer efficiency between the work areas and the refrigeration plant. Furthermore, they are a further source of potential corrosion and fouling of internal tubes.

In the mid 'seventies a number of factors coincided to promote a trend towards the location of centralized refrigeration plant at the surface of deep and/or hot mines. First, there was a significant drop in the wet bulb temperatures at which mining workforces would be expected to work. Cut-off wet bulb temperatures have been reduced from 32°C to 28°C with an expectation of further improvements to 27°C or less. Secondly, the combination of greater depths of workings and more intensive mechanization produced higher heat loads to be handled by the mine environmental control system. These factors combined to give very significant increases in the required cooling capacities of mine refrigeration plant, to the extent that untenable and uneconomic quantities of return air would be required for heat rejection. There was no choice but to locate the larger new plant on surface where heat rejection is relatively straightforward. Thirdly, the problem of high pressures developing in the shaft pipes was combated by the development of energy recovery devices including water turbines (Section 18.3.4).

A simplified schematic of one arrangement involving surface refrigeration plant is indicated on Figure 18.10. The figure illustrates water being cooled by a surface refrigeration plant and collected temporarily in a cold water dam. The hot water from the condensers is cycled around the condenser cooling tower for heat rejection to the atmosphere. If a natural supply of sufficiently cool water is available from a stream or river, then there may be no need for the capital and operating expense of refrigeration plant. Alternatively, in dry climates, a modest required degree of cooling may be achievable simply by spraying the water supply through a surface cooling tower. Furthermore, it may be necessary to operate the refrigeration plant during the summer months only.

The water passes from the surface cold water dam to an insulated shaft pipe through which it falls to the working levels. The water may then be passed through a turbine at one or more subsurface levels before being stored temporarily in underground cold water dams. The turbine(s) achieve three beneficial results. First, the pressure of the water is reduced to that of the mine atmosphere at the corresponding level. Hence, no extensive high pressure water systems need exist in the workings. Secondly, the mechanical output power produced by the turbines may be employed directly to assist in driving the pumps that raise the return hot water to surface. However, because the demand for cold water may be out of phase with the availability of hot return water, it is preferable to use the turbines to drive generators for the production of electrical power. Third, the removal of energy from the water by the turbines results in a reduced temperature rise of that water, improving the cooling efficiency of the system.

From the underground cold water dams, the chilled water may be utilized for a variety of purposes including water to air heat exchangers for cooling the air at the entrance to a stope or face, or for recooling at intermediate points along the stope or face. Water to water heat exchangers may be employed to cool secondary water circuits such as a supply of potable drinking water or dust suppression service water. However, the modern trend is to supply chilled service water directly from the cold water dam.
Figure 18.10 Schematic of centralized plant on surface supplying chilled water for underground heat exchangers and mine service water.
Warm water returning from the heat exchangers or via drainage channels is directed into hot water dams adjacent to a shaft. The pumps that return this water to surface may be powered partly by the energy recovered by the turbines. Where surface refrigeration plant is in use, the hot water from the mine may be sprayed through pre-cooling towers prior to its return to the plant evaporators. This gives an additional low cost supplement to the cooling capacity of the system. Any required make-up water and anti-corrosion chemicals are added at this stage.

The purpose of the dams on both the cold and hot water sides of the layout is to provide capacitance to the system. This permits short term fluctuations in demand for chilled water while using smaller refrigeration plant than would otherwise be necessary. At times of low demand, the temperature of the water in the surface dams can be further reduced by recycling that stored water through the plant. This smoothes the variations in cooling load required of the refrigeration units.

Having introduced the broad concepts of subsurface cooling systems, we are now in a better position to examine the component parts in greater detail.

18.3.2. Heat exchangers

In general, a heat exchanger may be defined as a device that facilitates the transfer of thermal energy from one solid or fluid system to another. In subsurface air conditioning systems, there are two classifications of heat exchanger in common use, both involving heat transfer between fluids.

An indirect heat exchanger promotes heat transfer across a solid medium that separates the two fluids. There is no direct contact between the fluids. Examples include:

- the shell and tube evaporators and condensers of refrigeration units (Figure 18.3)
- shell and tube water to water heat exchangers employed to transfer heat between water systems of differing pressure and/or water quality (Figure 18.11)
- tubular coil heat exchangers to transfer heat from air to water (Figure 18.12).

As the term implies, direct heat exchangers involve direct contact between the two fluids. Cooling towers and other types of spray chambers fall into this category. The objective may be to reject heat from the water to the air as in a conventional cooling tower. Conversely, a spray chamber supplied with chilled water provides a means of cooling an airstream.

The higher efficiency of heat transfer associated with direct heat exchangers caused a distinct trend away from tube coil air coolers during the nineteen eighties. Good performance of large horizontal spray chambers for bulk air cooling promoted the further development of compact and enclosed spray chambers for more localized use.

In this section we shall examine the essential structure and operating principles of both indirect and direct forms of heat exchanger.
Figure 18.11  A water-to-water heat exchanger

Figure 18.12  An air-to-water heat exchanger showing one layer of tubes.
18.3.2.1. Indirect heat exchangers

A shell and tube heat exchanger may contain over 200 tubes. In a counter-flow heat exchanger, the fluids inside and outside the tubes move in opposite directions, maximizing the total heat transfer. However, the water to water heat exchanger illustrated in Figure 18.11 shows that this may be sacrificed in the interest of compactness.

In an air to water indirect heat exchanger, a bank of tubes carrying chilled water is located within an air duct. Figure 18.12 shows one layer in a bank of tubes. The tubes are often inclined to the direction of airflow to improve the efficiency of the unit and to control the movement of condensate.

The two important features of a heat exchanger are that a good efficiency of heat transfer is obtained and that the pressure drops suffered by both fluid streams should remain within acceptable limits. Heat transfer is facilitated by choosing a tube material that has a high thermal conductivity and is also resistant to corrosion and the build-up of scale deposits within the tubes. Chemical additives can be used to control such fouling of the tubes. Copper tubes are commonly employed for air to water heat exchangers. For water to water heat exchangers and in the evaporator and condensers of refrigeration units, cupronickel and stainless steel are more resistant to corrosion. Even titanium may be used within evaporators and condensers.

If tubes are not maintained in a clean condition then the efficiency of the unit can fall dramatically. Periodic manual brushing of the tubes is time-consuming and may be replaced by a variety of automatic tube cleaning devices including reverse flushing with brush inserts or ultrasonic vibration. Air to water tube coils, as illustrated on Figure 18.12, are subject to caking by dust deposits in mine atmospheres. This may be handled by periodic cleaning with a high pressure water jet. Alternatively, sprays may be located permanently upstream from the coils. In humid atmospheres, the coils may be made self-cleaning by arranging them into horizontal banks with air passing upwards over the tubes. The droplets of condensate fall back through the coils giving a continuous and oscillating cleaning action on the outer surfaces of the tubes.

A second major factor that controls the duty of a heat exchanger is the effective area available for thermal transfer. Spiral fins welded to the tubes, as illustrated on Figure 18.12, may be used. Metal plates welded between consecutive and partially flattened tubes have also been used (Mücke, 1984; Weuthen, 1975). It is important that such means of area enlargement should have good thermal contact with the tubes and that they are oriented such that they present minimum resistance to flow over the tubes. In evaporators and condensers, the tube surfaces may be knurled or sintered. In addition to enhancing the surface area, this assists in the promotion of boiling or condensation.

18.3.2.2. Performance calculations for indirect heat exchangers

At equilibrium, the heat gained by one fluid in a heat exchanger must equal the heat lost by the other fluid. Hence, taking the example of an air to water heat exchanger, we can express the rate of heat transfer to be

\[ q = m_w C_w \Delta t_w = m_a \Delta S \quad \text{W} \quad (18.17) \]

where

- \( m_w \) = mass flow of water (kg/s)
- \( C_w \) = specific heat of water (4187 J/(kg °C))
- \( \Delta t_w \) = rise in temperature of the water (°C)
- \( m_a \) = mass flow of air (kg/s)
- \( \Delta S \) = fall in sigma heat of the air (J/kg) (ref. section 14.6).
As each of the factors in equation (18.17) can be calculated easily from measured temperatures and flowrates, either the water or the air may be used to determine the rate of heat transfer, $q$. As measurements on the water circuit can normally be made more accurately than those on the airflow, the former are preferred for a determination of heat transfer. If condensation occurs on the outside of the coils then the third part of equation 18.17 is slightly in error as it does not take into account the heat removed from the system by the condensate. However, this is usually small and may be neglected.

Another way of expressing the heat transfer is in terms of an overall heat transfer coefficient, $U$, (W/m$^2$°C) for the coils and adjacent fluid boundary layers, and the difference between the mean temperatures of the air and the water, $(t_{ma} - t_{mw})$, giving

$$q = UA(t_{ma} - t_{mw}) \quad \text{W}$$  \hspace{1cm} (18.18)

where $A$ = area available for heat transfer. (This is analogous to equation (15.16) for a rock surface.)

As the temperatures of both the air and water streams are likely to vary in a logarithmic rather than a linear fashion through the heat exchanger, it is more accurate to employ logarithmic mean temperature difference in equation (18.18), giving

$$q = \frac{UA(\Delta t_1 - \Delta t_2)}{\ln(\Delta t_1 / \Delta t_2)} \quad \text{W}$$ \hspace{1cm} (18.19)

where ln indicates natural logarithm and $\Delta t_1$ and $\Delta t_2$ are the temperature differences between the fluids at each end of the heat exchanger.

The $UA$ product is usually quoted as a measure of the effectiveness of an indirect heat exchanger.

A performance check should be carried out at monthly or three monthly intervals in order to determine any deterioration in the $UA$ caused by corrosion, scaling, or other forms of deposition on the tube surfaces.

Example

Measurements made on a counterflow air to water cooling coil give the following results.

<table>
<thead>
<tr>
<th>Air</th>
<th>Inlet wet bulb/dry bulb temperature = 28/34°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet wet bulb/dry bulb temperature</td>
<td>22.7/22.7°C</td>
</tr>
<tr>
<td>Mass flowrate</td>
<td>5.1 kg/s</td>
</tr>
<tr>
<td>Barometric pressure</td>
<td>105 kPa</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Water</th>
<th>Inlet temperature = 17 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet temperature</td>
<td>23 °C</td>
</tr>
<tr>
<td>Mass flowrate</td>
<td>4 kg/s</td>
</tr>
</tbody>
</table>

Determine the operating duty of the coil and the $UA$ value.

Solution

From the psychrometric equations given in section 14.6, the sigma heats of the air at inlet and outlet conditions of 28/34 and 22.7/22.7°C, respectively, are found to be

$$S_{in} = 84,696 \text{ J/kg}$$

and

$$S_{out} = 63,893 \text{ J/kg}$$
From equation (18.17) the heat lost from the air is:

\[ q_a = m_a (S_{in} - S_{out}) = 5.1 (84,696 - 63,893) = 106.1 \times 10^3 \text{ W or 106.1 kW}. \]

Also, from equation (18.17), the heat gained by the water is:

\[ q_w = m_w C_w \Delta t_w = 4 \times 4187 \times (23-17) = 100.5 \times 10^3 \text{ W or 100.5 kW}. \]

At equilibrium, \( q_a \) and \( q_w \) must be equal, showing that the errors in measurements have caused a deviation of some 5.5 percent. Much larger discrepancies often occur due mainly to difficulties in making measurements in highly turbulent and, often, saturated airflows. We shall continue the calculation using the rate of heat transfer given by the water circuit, 100.5 kW.

In order to calculate the \( UA \) value from equation (18.19), we must first evaluate the temperature differences between the water and air at the air inlet and air outlet ends (subscripts 1 and 2, respectively).

\[ \Delta t_1 = t_{a,in} - t_{w,out} = 34 - 23 = 11 \text{ °C} \]
\[ \Delta t_2 = t_{a,out} - t_{w,in} = 22.7 - 17 = 5.7 \text{ °C} \]

Notice that we use the dry bulb temperature at the air inlet and before saturation conditions are attained for the determination of \( \Delta t_1 \). This is equivalent to taking the latent heat of condensation as a factor contributing towards the \( UA \) value of the heat exchanger.

Equation (18.19) now gives

\[ UA = q_w \frac{\ln(\Delta t_1 / \Delta t_2)}{\Delta t_1 - \Delta t_2} = 100.5 \frac{\ln(11/5.7)}{11 - 5.7} = 12.47 \text{ kW/°C} \]

The \( UA \) of a clean coil may lie between 10 and 25 kW/°C depending upon the design of the heat exchanger and the configuration of fluid flows. Records should be kept of the periodic performance tests on each heat exchanger. Significant reductions in \( UA \) values indicate the need for cleaning or replacement of the tubes.

Although \( UA \) values are normally determined by measurement as illustrated in the example, they may also be defined by the following equation:

\[ \frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_{ii} A_{ii}} + \frac{x}{k_t A_m} + \frac{1}{h_o A_o} + \frac{1}{h_{io} A_{io}} \text{ °C} \quad (18.20) \]

where \( h \) = heat transfer coefficients (W/(m²°C))
\( A \) = area available for heat transfer (m²)
\( x \) = thickness of tube walls (m)
\( k_t \) = thermal conductivity of tube material (W/(m°C))

and subscripts
\( i \) = inside surface of tubes
\( o \) = outside surface of tubes
\( m \) = mean of inner and outer surfaces.
18-25

\( h_f \) and \( h_{fo} \) are the heat transfer coefficients associated with fouling (deposits) on the inside and outside surfaces of the tubes respectively. On a clean tube there are no such deposits; \( h_f \) and \( h_{fo} \) are both then infinite.

Radiation terms have been left out of equation (18.20) as these are normally small in a heat exchanger. They may, however, become significant in situations such as an uninsulated pipe suspended in an airway.

Equation (18.20) can be further simplified for practical application. First, the term \( x / (k_A m) \) is very small compared with the others. Secondly, the terms involving fouling of the tubes are often combined and attributed to the inside surface only, giving

\[
\frac{1}{UA} = \frac{1}{h_f A_i} + \frac{1}{h_{fo} A_o} + \frac{1}{h_r A_i} \quad {^\circ C}/W \quad (18.21)
\]

As fouling of the tubes occurs, \( h_r \) will decrease causing the \( UA \) value also to fall. Whillier (1982) quotes a typical value of some 3000 W/(m\(^2\)\(^\circ\)C) for \( h_f \) in a mine refrigeration plant. For turbulent flow inside smooth tubes, the heat transfer coefficient can be determined from equation (15.23) and the Colburn equation (A15.15)

\[
h_f = 0.023 \frac{k}{d} \text{Re}^{0.8} \text{Pr}^{0.4} \quad W/(m^2 \circ C) \quad (18.22)
\]

where
- \( k \) = thermal conductivity of fluid (W/(m\(^\circ\)C))
- \( d \) = internal diameter of tube (m)
- \( \text{Re} \) = Reynolds number = \( \rho u d/\mu \) (dimensionless)
- \( \rho \) = fluid density (kg/m\(^3\))
- \( u \) = fluid velocity (m/s)
- \( \mu \) = dynamic viscosity (Ns/m\(^2\))
- \( \text{Pr} \) = Prandtl number = \( C_p \mu/k \) (dimensionless, may be taken as 0.7 for air)

and \( C_p \) = Specific heat at constant pressure (J/(kg\(^\circ\)C))

For air and water, the values of viscosity and thermal conductivity within the temperature range 0 to 60\(^\circ\)C are given in sections 2.3.3 and 15.2.4 respectively. Other expressions for the heat transfer coefficient at rough surfaces are given in Table A15.1.

The values of \( h_o \) for the outer surfaces of the tubes vary widely according to geometry and the configuration of tubes. For turbulent cross flow of air over tubular surfaces, McAdams (1954) gives:

for a single tube:

\[
h_o = 0.24 \frac{k_o}{d} \text{Re}^{0.6} \quad W/m^2 \circ C \quad (18.23)
\]

and for a bank of staggered tubes:

\[
h_o = 0.29 \frac{k_o}{d} \text{Re}^{0.6} \quad W/m^2 \circ C \quad (18.24)
\]

In these relationships, \( d \) is the outer diameter of the tubes and Reynolds' Numbers are determined on the basis of the maximum velocity of the air as it flows between the tubes. The selection of cooling coils is facilitated greatly by tables and graphs provided by manufacturers.
18.3.2.3. Direct heat exchangers

In direct heat exchangers, air is brought into contact with water surfaces. The purpose may be to cool water from the condensers of a refrigeration plant. In this case the hot water is sprayed into a cooling tower and descends as a shower of droplets through an ascending airstream. Heat is transferred from the water droplets to the air by a combination of convection (sensible heat) and evaporation (latent heat). The cooled water that collects at the base of the cooling tower is then returned to the condenser (see the condenser cooling tower on Figure 18.10).

Alternatively, the objective may be to cool the air. In this case, chilled water is sprayed into a vertical or horizontal spray chamber. Provided that the airflow enters with a wet bulb temperature that is higher than the temperature of the water, then heat will be transferred from the air to the water by a combination of convection and condensation.

Although there are significant differences between the designs of cooling towers and spray coolers, there are several common factors that influence the amount and efficiency of heat exchange:

- water mass flowrate
- supply temperature of water
- air mass flowrate
- psychrometric condition of the air at inlet
- duration and intimacy of contact between the air and the water droplets.

This latter factor depends upon the physical design of the heat exchanger. In particular,

(i) the relative velocity between the air and water droplets

and

(ii) the size and concentration of water droplets - governed by the flow and pressure of the supply water, the type and configuration of spray nozzles, and the presence of packing within the heat exchanger.

The traditional use of direct heat exchangers in mine air conditioning systems has been for cooling towers either underground or on surface. Tube coil heat exchangers with closed circuit water systems have been used extensively for air cooling in or close to mine workings. However, the greater efficiency of direct heat exchangers led to the development of large permanent spray chambers for bulk air cooling. Furthermore, through the nineteen eighties, smaller portable spray chambers began to be employed for local cooling.

Cooling Towers

First, let us examine the essential features of cooling towers. Figure 18.13 illustrates a vertical forced draught cooling tower of the type that may be used on the surface of a mine. Hot water from refrigeration plant condensers is sprayed into the cooling tower and moves downward in counterflow to the rising air current. The purpose of the packing is to distribute the water and airflow uniformly over the cross section and to maximize both the time and total area of contact between the air and water surfaces. The packing may take the form of simple splash bars or ruffles arranged in staggered rows, egg-crate geometries or wavy (film-type) surfaces located in vertical configurations. The materials used for packing may be treated fir or redwood timber, galvanized steel, metals with plastic coatings and injection molded PVC or polypropylene. Concrete is used primarily for casings, structural reinforcements and water sumps or dams. Air velocities through counterflow packed cooling towers lie typically in the range 1.5 to 3.6 m/s.
In underground cooling towers, the packing is often eliminated completely or takes the form of one or two screens arranged horizontally across the tower. Such heat exchangers are essentially vertical spray chambers. Other means of maximizing contact time are then employed. These include designs that introduce a swirl into the air at entry, and directing the sprays upward rather than downward. Again, in underground installations, airflow is induced through the cooling towers by the mine ventilating pressure or by booster fans in return airways rather than by fans connected directly to the tower. The pressure drop through underground cooling towers may be further reduced by replacing the mist eliminator screens with an enlarged cross-sectional area; the lower air velocity in this zone decreases the carry-over of water droplets. The optimum air velocity in open spray towers lies in the range 4 to 6 m/s with a maximum of some 8 m/s. Water loadings should not be greater than 16 litres/s per square metre of cross-sectional area (Stroh, 1982).

Cooling towers of the type used for mine air conditioning are typically 10 to 20 m in height and some 3 to 8 m in diameter, depending upon the rate at which heat is to be exchanged. Heat loads may be as high as 30 MW.

Natural draught cooling towers do not have fans but rely upon air flow induction by the action of the sprays, or by density difference between the outside atmosphere and the hot moist air within the tower. The hyperbola shaped cooling towers employed commonly for the high heat loads of power stations are of this type and may be over 150 m high in order to accentuate the chimney effect.

The air leaving a cooling tower is usually saturated. This often results in the formation of a fog plume in the surface atmosphere. The environmental effects of such a plume should be considered carefully and may influence the siting of a surface cooling tower. In large installations with unacceptable fog plumes, part of the hot condenser water may be cooled within a finned tube indirect heat exchanger. This involves sensible heat exchange only and reduces the humidity of the air leaving the cooling tower. However, such an arrangement detracts from the overall efficiency of heat transfer.
In a direct exchange cooling tower, some water is lost continuously from the circuit both by evaporation and by drift (or carry-over) of small droplets. The evaporation loss approximates some one percent for each 7°C of water cooling and drift loss is usually less than 0.2 percent of the water circulation rate (ASHRAE, 1988). However, evaporation can result in a rapid escalation in the concentration of dissolved solids and other impurities leading to scaling, corrosion and sedimentation within the system. In order to limit the buildup of such impurities, some water is continuously removed from the system (bleedoff or blowdown). The bleedoff rate is controlled by monitoring the quality of the water but may be of the order of one percent of the circulation rate. The combined losses from evaporation, drift and bleedoff are made up by adding clean water to the circuit.

Further protection of metal components and, particularly the tubes of condensers is obtained by administering anticorrosion compounds. These generally take the form of chromates, phosphates or polyphosphonates of zinc, and promote the formation of a protective film on metal surfaces. Some of these compounds are toxic and precautions may be necessary against accidental release into natural drainage systems. Biocides such as chlorine are also added on a periodic basis to control the growth of algae and other organic slimes.

**Chilled water spray chambers**

If the water supplied to a direct contact heat exchanger is at a temperature below that of the wet bulb temperature of the air, then cooling and dehumidification of the air is achieved. Chilled water spray chambers fall into two categories with respect to size. First, the larger installations are constructed at fixed sites for bulk cooling of main airflows. Secondly, portable spray chambers for localized use within working areas have many advantages over the more traditional tube coil (indirect) stope or face coolers. We shall consider each of these two applications in turn.

For bulk air cooling, spray chambers may be designed in vertical or horizontal configuration. Indeed, if the cooling tower shown on Figure 18.13 were supplied with chilled water, then it would act equally well as a vertical spray air cooler. Such designs may be employed either on surface or underground for bulk cooling of intake air and may have heat transfer duties up to 20 MW.

**Horizontal spray chambers** have more limited maximum capacities of some 3.5 MW. They are, however, more convenient for underground use in that existing airways may be utilized without additional excavation. Figure 18.14 illustrates a single stage horizontal spray chamber. The sprays may be directed against or across the airflow. The nozzles can be distributed over the cross section as shown in the sketch or, alternatively, located at the sides or near the base of the chamber. Although the position of the nozzles appears not to be critical, it is important that both the sprays and the airflow are distributed uniformly over the cross section. The spray density should lie within the range 2 to 5 litres/second for each square metre of cross sectional area (Bluhm, 1983).

The area of liquid/air interface and efficiency of heat exchange increase for smaller sizes of droplets. However, very small droplets result in excessive carry-over and necessitate highly constrictive mist eliminators. Furthermore, higher water pressures and, therefore, pumping costs are required to produce fine sprays. In practice, droplet diameters of some 0.5 mm and water pressures in the range 150 to 300 kPa give satisfactory results in horizontal spray chambers (Reuther, 1988).

At positions fairly close to the nozzles, the relative directions of the air and water droplets may be counterflow or crossflow, dependent upon the orientation of the nozzles. However, aerodynamic drag rapidly converts the spray to parallel flow, particularly for the smaller droplets. In order to regain the higher efficiency of counterflow heat exchange, the spray chamber may be divided into two or three stages so that the air leaves the chamber at the position of the coldest sprays. Chilled water should be supplied at as low a temperature as practicable but, in any event, not higher than 12°C.
Figure 18.15 illustrates a two stage spray chamber. An additional sump at each end assists in balancing the pumping duties.

If the surrounding strata are unfractured and unaffected by water, then the spray chamber may utilize the full cross section of a bypass airway. If, however, the rock must be protected against the effects of water, then concrete lining or prefabricated sections may be employed to contain the spray chamber. In cases where the strata is very sensitive to water (such as evaporite strata) then it is advisable to protect the rock surfaces for 50 to 100 m downstream from the spray chamber.

The cross-sectional area of the spray chamber should be chosen to give a preferred air velocity of some 4 to 6 m/s, but not more than 7 m/s. Higher air velocities will reduce the efficiency of heat exchange and can result in excessive pressure drops in the airflow.

In addition to cooling and dehumidifying the air, spray coolers can also reduce dust concentrations. However, the buildup of dust particles in the recirculating water system may cause fouling of the pipes and other heat exchangers. This may require filters or sedimentation zones to be included in the design.

Internal packing can also be employed to improve the efficiency of horizontal spray chambers (Stroh, 1980). This can increase the air frictional pressure drop across the cooler. However, the water supply pressure (gauge) may be as low as 30 kPa since the spray nozzles can be replaced by low resistance dribbler bars.

In order to extend the applicability of direct spray coolers, enclosed and portable units have been developed (Thimons, 1980; Ramsden, 1984; Reuther, 1988). These may be mounted on wheels or sleds and are often referred to as spray mesh coolers. Figure 18.16 illustrates the principle of operation. In order to maintain the design to acceptable dimensions for portable application, the water loading may be much higher than open spray chambers and it becomes even more important to maximize the area and time of contact between the air and water. This occurs in three stages. First, the air passes through the lower mesh of plastic or knitted stainless steel. Secondly, the airflow is directed through the upward pointing and finely divided sprays. The final stage of cooling occurs within the upper mesh which also acts as a demister. The distribution of droplets of water falling from the upper to the lower mesh may be assisted by installing drip fins. These also help to prevent water running down the walls (Heller, 1982). Between the two meshes, heat exchange occurs by a combination of counterflow, crossflow and parallel flow. Another type of portable spray cooler employs film packing and crossflow heat exchange (Reuther, 1988).
Figure 18.15 A two-stage horizontal spray chamber

Figure 18.16 A section through a portable mesh cooler.
The dry bulb temperature of air can be reduced by passing it through a spray chamber supplied with unchilled water. The device is then known as an evaporative cooler. No heat is removed from the air. Hence, if the water is supplied at wet bulb temperature, then the exhaust air may attain that same wet bulb temperature while the moisture content will have increased. Evaporative coolers have an application for surface structures in hot dry climates but are seldom used in underground mines. They have been employed for very localized effects in hot evaporite mines. However, the reduced dry bulb temperature encourages enhanced heat flow from the strata (Section 15.2.2) which, when combined with the raised moisture content, reduces the cooling power of the air downstream from the cooled area.

18.3.2.4. Performance calculations for direct heat exchangers.

A common theoretical analysis may be applied to direct heat exchangers irrespective of the direction of heat transfer. The results of such analyses apply equally well to cooling towers and chilled water spray chambers. In order to avoid unnecessary repetition, we will conduct the following analysis on the assumption of a cooling tower (Figure 18.13). Water recycles continuously through the condenser of a refrigeration plant (where it gains heat) and the cooling tower where it rejects that heat to the atmosphere.

The first observation is that if we ignore the small heat losses from interconnecting pipes and the equally small effects of make-up water, then, at steady state, the heat rejected in the cooling tower must be equal to the heat gained in the condenser. This leads to the initially surprising statement that the rate of heat rejection in the cooling tower depends only upon the heat load imposed by the condenser and not at all upon the design of the cooling tower. However, if the cooling tower is inefficient in transferring heat from the water to the air, then the temperature of the water throughout the complete circuit will rise until balance is attained between heat gain in the condenser and heat rejection in the cooling tower. This would be unfortunate as the coefficient of performance of the refrigeration plant deteriorates as the condenser temperature increases (Section 18.2.4).

Figure 18.17 illustrates the decrease in temperature of the water as it falls through the cooling tower and the corresponding increase in the wet bulb temperature of the ascending air.

Figure 18.17 also defines the meaning of two terms that are commonly employed:

The range is the change in temperature of the water

\[
\text{range} = t_{w,\text{in}} - t_{w,\text{out}} \degree C
\]

while the approach is the difference between the temperatures of the water outflow and the wet bulb temperature of the air inflow:

\[
\text{approach} = t_{w,\text{out}} - t_{a,\text{in}} \degree C
\]

where \( t \) = temperature, \( w \) = liquid water, \( a \) = air (wet bulb), \( in \) = inflow and \( out \) = outflow.

Manufacturers will usually accept requests for approach values down to 2\degree C.
The quantitative analysis of the cooling tower commences by writing down the balance that must exist between rates of heat gained by the air and heat lost from the water:

$$m_a (S_{out} - S_{in}) = m_w C_w (t_{w,in} - t_{w,out}) \quad W$$

(18.27)

where \( m \) = mass flow, kg/s

\( S \) = sigma heat of air, J/kg

and \( C_w \) = specific heat of water (4187 J/kg °C)

This is, in fact, an approximation as evaporation changes the value of \( m_w \) within the tower. However, the error does not normally exceed 4 percent and may be neglected for practical purposes.

Figure 18.17 Variation of water temperature, \( t_w \), and wet bulb temperature of the air, \( t_a \), through a cooling tower.
As in so many cases of assessing performance, it is useful to imagine the unattainable perfect cooling tower. In such a device, the two curves on Figure 18.17 would coincide and, in particular, the water would leave at inlet air wet bulb temperature while the air would leave at the temperature of the incoming water, that is:

\[ t_{w,\text{out}} = t_{w,\text{in}} \]  \hspace{1cm} (18.28)

and

\[ t_{a,\text{out}} = t_{w,\text{in}} \]  \hspace{1cm} (18.29)

The concept of a perfect cooling tower allows us to devise efficiencies of heat transfer for both the water and the air streams:

**Water efficiency**

\[ \eta_w = \frac{\text{actual heat lost from the water}}{\text{theoretical maximum heat that could be lost from the water}} \]

\[ = \frac{m_w C_w (t_{w,\text{in}} - t_{w,\text{out}})}{m_w C_w (t_{w,\text{in}} - t_{a,\text{in}})} = \frac{(t_{w,\text{in}} - t_{w,\text{out}})}{(t_{w,\text{in}} - t_{a,\text{in}})} = \frac{\text{range}}{\text{range + approach}} \]  \hspace{1cm} (18.30) (See Figure 18.17)

**Air efficiency**

\[ \eta_a = \frac{\text{actual heat gained by the air}}{\text{theoretical maximum heat that could be gained by the air}} \]

\[ = \frac{m_a (S_{\text{out}} - S_{\text{in}})}{m_a (S_{w,\text{in}} - S_{\text{in}})} = \frac{(S_{\text{out}} - S_{\text{in}})}{(S_{w,\text{in}} - S_{\text{in}})} \]  \hspace{1cm} (18.31)

where \( S_{w,\text{in}} = \) sigma heat of saturated air at a temperature equal to that of the inlet water, J/kg.

[The term "thermal capacity" is sometimes employed where we have used "theoretical maximum heat".]

Although the water and air efficiencies are useful indicators of the efficacy of heat transfer for each of the two fluids, neither gives any clue to the overall quality of design for the complete cooling tower. For example, if the water flow were low giving relatively few small droplets falling through a large airflow, then the water efficiency would be very high. However, the air would be used to only a small fraction of its thermal capacity and, hence, the air efficiency and overall heat transfer would be low.

To examine this problem let us consider, again, perfect heat transfer in a cooling tower. Then,

**maximum thermal capacity of water**  \[ = m_w C_w (t_{w,\text{in}} - t_{a,\text{in}}) \]  J

and  \[ \text{maximum thermal capacity of airstream} = m_a (S_{w,\text{in}} - S_{\text{in}}) \]  J

The ratio of these maximum thermal capacities defines the tower capacity factor, \( R \)

\[ R = \frac{m_w C_w (t_{w,\text{in}} - t_{a,\text{in}})}{m_a (S_{w,\text{in}} - S_{\text{in}})} \]  \hspace{1cm} (dimensionless)  \hspace{1cm} (18.32)

The liquid to gas ratio, \( m_w/m_a \) normally lies within the range 0.4 to 2.0.

The theoretical concept of tower capacity factor ignores the practical reality that in a real cooling tower the rate of heat loss from the water must equal the rate of heat gain by the air. It may be defined in words as follows:
'If both of the given fluid streams were used to their maximum thermal capacity, then $R$ would be the number of Watts lost by the water for each Watt gained by the air.'

An important observation here is that $R$ depends only upon the flowrates and the inlet conditions of the water and air. It is completely independent of the construction of the cooling tower. Notice also that we have been quite arbitrary in choosing to define $R$ as the ratio of water thermal capacity divided by air thermal capacity, rather than the other way round. The values of tower capacity factor vary from 0.5 to 2.

The tower capacity factor is also related to the ratio of air and water efficiencies. From equations (18.30 and 18.31)

$$\frac{\eta_a}{\eta_w} = \frac{(S_{out} - S_{in})}{(S_{w,in} - S_{in})} \frac{(t_{w,in} - t_{a,in})}{(t_{w,in} - t_{w,out})}$$

and substituting for

$$(S_{out} - S_{in}) = \frac{m_w}{m_a} C_w (t_{w,in} - t_{w,out})$$

from equation (18.27) gives

$$\frac{\eta_a}{\eta_w} = \frac{m_w}{m_a} C_w \frac{(t_{w,in} - t_{a,in})}{(S_{w,in} - S_{in})}$$

(18.33)

As this is identical to equation (18.32), then tower capacity factor

$$R = \frac{\eta_a}{\eta_w}$$

(18.34)

indicating that the ratio of air and water efficiencies is also independent of the design of the cooling tower or spray chamber.

The overall effectiveness of heat transfer within the cooling tower must be dependent on the fluid having the smaller thermal capacity, i.e. the fluid having the larger efficiency. To quantify this concept, we can say that the Effectiveness, $E$, is the larger of the values of $\eta_a$ and $\eta_w$. Another way of expressing this, using equation (18.34), is:

$$E = \eta_a \text{ if } R \geq 1$$

and

$$E = \eta_w \text{ if } R < 1$$

(18.35)

$\eta_a$ and $\eta_w$ are, of course, equal when $R = 1$.

Prior to 1977, there seems to have been no one single parameter that could be used to describe the quality of design for any given cooling tower. The rate of heat transfer, air and water efficiencies, and the tower capacity factor are all dependent on inlet temperatures. It is, therefore, somewhat misleading to quote a single value of cooling duty to characterize a heat exchanger.

In the mid 1970's, Whillier of the South African Chamber of Mines conducted an analysis on results published earlier by Hemp (1967 and 1972) relating to a series of tests during which a spray tower was successively supplied with a range of inlet water temperatures. Whillier (1977) noticed that two distinct curves were produced when (a) water efficiency and (b) air efficiency were plotted against tower capacity factor. Furthermore, these two curves appeared to be mirror images of each other. In a moment of inspiration he recalled that the order of the ratio that we choose to define $R$ is quite arbitrary (equation (18.32). Whillier took the values of tower effectiveness, $E$, as defined by equations (18.35) and plotted them against $R^*$, where
\[ R^* = R \text{ when } R < 1 \]
\[ R^* = \frac{1}{R} \text{ when } R > 1 \]

(i.e. inverting the ratio when R exceeds unity).

To his delight, all the points for the spray tower then lay on a single curve within an acceptable experimental scatter, and no matter whether the tower was used to cool water or air. The shape of that curve is illustrated on Figure 18.18. The fact that all test results lay on a single curve held promise that an equation or characteristic number for that curve could be used as a means of quantifying the overall merit of the design and construction of the cooling tower or spray chamber, and independent of the inlet temperatures or flowrates of the air and water. Whillier attempted several curve fitting exercises on his results and found that a reasonable correlation was obtained by the exponential relationship

\[ E = F^{R^*} \]

where \( F \) was a constant for that particular spray tower. Different values of \( F \) in the range 0 to 1 would be obtained for other cooling towers or spray chambers. Whillier termed this parameter the \textbf{Factor of Merit} for the cooling tower. A means had been found by which the performance characteristic of a direct heat exchanger could be quantified in a single number. A factor of merit of 1.0 would indicate a perfect counterflow direct-contact heat exchanger. Practical values lie within the range 0.3 to 0.8.

Further work by Bluhm (1980, 1984) investigated the factor of merit for multi-stage spray chambers, and produced an improved algorithm for spray coolers:
\[ \eta_w = \frac{1 - \exp(-N(1 - R))}{1 - R \exp(-N(1 - R))} \]  

(18.38)

where \( N = \frac{F}{R^{0.4}(1 - F)} \)  

(18.39)

It is recommended that this algorithm is used for factors of merit below 0.4 or above 0.7.

Typical factors of merit suggested by Whillier (1977) and Bluhm (1980) are shown on Table 18.2. Once the factor of merit has been established for any direct-contact heat exchanger, either by testing or by estimation from Table 18.2, then the performance of the unit can be established for any given flow rates and inlet conditions of the air and water streams.

<table>
<thead>
<tr>
<th>Type of unit</th>
<th>Typical Values of Factor of Merit, ( F )</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Vertical spray filled towers</strong></td>
<td></td>
</tr>
<tr>
<td>No packing:</td>
<td></td>
</tr>
<tr>
<td>high water loading</td>
<td>0.5 to 0.6</td>
</tr>
<tr>
<td>low water loading</td>
<td>0.6 to 0.7</td>
</tr>
<tr>
<td>With packing:</td>
<td></td>
</tr>
<tr>
<td>high water loading</td>
<td>0.55 to 0.65</td>
</tr>
<tr>
<td>low water loading</td>
<td>0.65 to 0.75</td>
</tr>
<tr>
<td>Industrial packed tower</td>
<td>0.7 to 0.8</td>
</tr>
<tr>
<td><strong>Horizontal spray chambers</strong></td>
<td></td>
</tr>
<tr>
<td>single-stage</td>
<td>0.4 to 0.5</td>
</tr>
<tr>
<td>two-stage</td>
<td>0.58 to 0.68</td>
</tr>
<tr>
<td>three-stage</td>
<td>0.65 to 0.75</td>
</tr>
<tr>
<td>Packed air coolers</td>
<td></td>
</tr>
<tr>
<td>crossflow</td>
<td>0.55 to 0.65</td>
</tr>
<tr>
<td>counterflow</td>
<td>0.68 to 0.78</td>
</tr>
<tr>
<td>Mesh coolers</td>
<td>0.55 to 0.65</td>
</tr>
</tbody>
</table>

Table 18.2. Typical factors of merit for direct contact heat exchangers.

**Example 1**
A cooling tower operates at a barometric pressure of 95 kPa. The following temperature measurements were obtained:

- air inlet wet bulb temperature, \( t_{a,in} = 18.0 \, ^\circ C \)
- air outlet wet bulb temperature, \( t_{a,out} = 28.0 \, ^\circ C \)
- water inlet, \( t_{w,in} = 32.0 \, ^\circ C \)
- water outlet, \( t_{w,out} = 22.5 \, ^\circ C \)

Determine the factor of merit for the cooling tower.
Solution

Using the psychrometric equations given in section 14.6 (or from a 95 kPa psychrometric chart), sigma heats may be determined to be:

- at air inlet (18.0 °C), \( S_{in} = 52.05 \text{ kJ/kg} \)
- at air outlet (28.0 °C), \( S_{out} = 90.90 \text{ kJ/kg} \)
- at water inlet temperature (32.0 °C), \( S_{w,in} = 111.64 \text{ kJ/kg} \)

**Air efficiency** (from equation (18.31))

\[
\eta_a = \frac{(S_{out} - S_{in})}{(S_{w,in} - S_{in})} = \frac{(90.90 - 52.05)}{(111.64 - 52.05)} = 0.652
\]

**Water efficiency** (from equation (18.30))

\[
\eta_w = \frac{(t_{w,in} - t_{w,out})}{(t_{w,in} - t_{a,in})} = \frac{(32.0 - 22.5)}{(32.0 - 18.0)} = 0.679
\]

**Tower capacity factor** (from equation (18.34))

\[
R = \frac{\eta_a}{\eta_w} = \frac{0.652}{0.679} = 0.961
\]

**Tower effectiveness** (from equations (18.35))

As \( R < 1 \) then \( E = \eta_w = 0.679 \).

Also, from equations (18.36)

\[
R^* = R = 0.961
\]

**Factor of merit** (from equation (18.37))

\[
E = F^{R^*}.
\]

\[
\ln(E) = R^* \ln(F)
\]

or

\[
F = \exp \left[ \frac{\ln(E)}{R^*} \right] = \exp \left[ \frac{\ln(0.679)}{0.961} \right] = 0.668
\]

Note that we have been able to calculate the factor of merit using measured temperatures and the barometric pressure only. We have not required any fluid flowrates.

Bluhm's equations (18.38 and 18.39) give the factor of merit to be 0.659, a difference of some 1.3 percent in this example.

**Example 2**

Suppose the cooling tower of the previous example were to be converted into a vertical spray cooler by supplying it with chilled water. Calculate the operating characteristics of the cooler given that:

- water inlet temperature, \( t_{w,in} = 5^\circ \text{C} \)
- air inlet wet bulb temperature, \( t_{a,in} = 18^\circ \text{C} \)
- water flowrate, \( m_w = 100 \text{ litres/s} \)
- air flowrate, \( m_a = 120 \text{ m}^3/\text{s} \) at an air density of 1.1 kg/m\(^3\)
- barometric pressure, \( P = 95 \text{ kPa} \)
Solution
The wet bulb temperature of the inlet air is the same as in the previous example. Hence, the corresponding sigma heat remains at

\[ S_{in} = 52.05 \text{ kJ/kg} \]

However, at the new inlet water temperature of 5°C, the psychrometric equations of Section 14.6 give

\[ S_{w,in} = 19.38 \text{ kJ/kg}. \]

Tower capacity factor (from equation (18.32))

\[
R = \frac{m_w C_w}{m_a} \left( \frac{t_{w,in} - t_{a,in}}{(S_{w,in} - S_{in})} \right) = \frac{100 \times 4187}{132} \frac{(5 - 18)}{(19.38 - 52.05)1000} = 1.262
\]

Tower effectiveness

As \( R > 1 \) then \( R^* = \frac{1}{R} \) (equation (18.36))

\[
R^* = \frac{1}{1.262} = 0.7923
\]

Now the factor of merit for the tower, \( F \), was calculated in the previous example as 0.668. This remains the same despite the new inlet conditions. Equation (18.37) gives

\[
E = F R^* = 0.668 \times 0.7923 = 0.7264
\]

Fluid efficiencies

As \( R > 1 \), \( E = \eta_a \) (equation (18.35))

Hence \( \eta_a = 0.7264 \) and from equation (18.34)

\[
\eta_w = \frac{\eta_a}{R} = \frac{0.7264}{1.262} = 0.5755
\]

Now, equation (18.30) gives

\[
\eta_w = \frac{(t_{w,in} - t_{w,out})}{(t_{w,in} - t_{a,in})}
\]

or

\[
t_{w,out} = t_{w,in} - \eta_w (t_{w,in} - t_{a,in}) = 5 - 0.5755(5 - 18) = 12.48 \text{ °C}
\]

and equation (18.31) gives

\[
\eta_a = \frac{(S_{out} - S_{in})}{(S_{w,in} - S_{in})}
\]

from which

\[
S_{out} = \eta_a (S_{w,in} - S_{in}) + S_{in}
\]

\[= 0.7264(19380 - 52050) + 52050 = 28319 \text{ J/kg or 28.3 kJ/kg}\]
The air wet bulb temperature giving this sigma heat may be read from the 95 kPa psychrometric chart or calculated by iterating equations (14.44 to 14.47), giving $t_{a,\text{out}} = 9.18^\circ\text{C}$.

**Rate of heat transfer, $q$**

Using the water circuit,

$$q = m_w C_w \left( t_{w,\text{out}} - t_{w,\text{in}} \right) = 100 \times 4187 \left( 12.48 - 5 \right)$$

$$= 3.132 \times 10^6 \text{ or } 3.132 \text{ MW}$$

As a cross-check, we may use the air circuit to give

$$q = m_a \left( S_{\text{out}} - S_{\text{in}} \right) = 132 \left( 28319 - 52050 \right)$$

$$= -3.132 \times 10^6 \text{ W or } -3.132 \text{ MW}$$

The heat gained by the water is shown to balance the heat lost by the air.\footnote{Note that because of measurement uncertainties, good agreement may not be obtained in actual operating conditions.}

**Example 3**

A mesh spray cooler is to be supplied with water at a temperature of 10°C. The unit is required to cool 12 kg/s of air from a wet bulb temperature of 29°C to 23°C and at a barometric pressure of 100 kPa. Determine the rate of water flow that must be sprayed through the cooler, the temperature of the return water and the cooling duty of the unit.

**Solution**

In the absence of further data concerning the device, we take a mean value of 0.6 as the factor of merit for a mesh cooler (from Table 18.2).

The data may then be summarized as follows:

- $t_{a,\text{in}} = 29^\circ\text{C}$
- $t_{a,\text{out}} = 23^\circ\text{C}$
- $m_a = 12 \text{ kg/s}$
- $t_{w,\text{in}} = 10^\circ\text{C}$
- $t_{w,\text{out}} = \text{unknown}$
- $m_w = \text{unknown}$
- $F = 0.6$

The sigma heats can be calculated from the psychrometric equations given in section 14.6 or approximated from the 100 kPa psychrometric chart:

- $S_{\text{in}}$ (at 29°C wet bulb temperature) = 92.28 kJ/kg
- $S_{\text{out}}$ (at 23°C) = 67.11 kJ/kg
- $S_{w,\text{in}}$ (at 10°C) = 29.21 kJ/kg

**Air efficiency (from equation (18.31))**

$$\eta_a = \frac{S_{\text{out}} - S_{\text{in}}}{(S_{w,\text{in}} - S_{\text{in}})} = \frac{67.11 - 92.28}{29.21 - 92.28} = 0.3991$$

**Water efficiency (from equation (18.30))**

$$\eta_w = \frac{t_{w,\text{in}} - t_{w,\text{out}}}{(t_{w,\text{in}} - t_{a,\text{in}})} = \frac{10 - t_{w,\text{out}}}{(10 - 29)} = \frac{t_{w,\text{out}} - 10}{19} \quad (18.40)$$
Capacity factor \( \text{from equation (18.34)} \)

\[
R = \eta_a \frac{\eta_w}{\eta_w} = \frac{0.3991 \times 19}{(t_{w,\text{out}} - 10)} = \frac{7.583}{(t_{w,\text{out}} - 10)} \quad (18.41)
\]

We now have a decision to make - whether to assume that \( R \) is greater or less than unity, allowing us to ascribe values for \( E \) and \( R^* \). Let us assume that \( R > 1 \) (we shall see in a moment that this was the wrong decision). Using equations (18.35 and 18.36), if \( R > 1 \), then \( E = \eta_a \) and \( R^* = 1/R \).

Equation (18.37) gives

\[
E = F^{R^*}
\]
or

\[
0.3991 = 0.6 \left( \frac{t_{w,\text{out}} - 10}{7.583} \right)
\]
giving

\[
(t_{w,\text{out}} - 10) = 7.583 \frac{\ln(0.3991)}{\ln(0.6)} = 13.64
\]
or

\[
t_{w,\text{out}} = 23.64^\circ C.
\]

In order to check the consistency of the value of \( R \), we can return to equation (18.41):

\[
R = \frac{7.583}{(23.64 - 10)} = 0.56
\]

showing that our assumption of \( R > 1 \) was incorrect.

Let us, therefore, repeat the analysis assuming that \( R < 1 \). Then equations (18.35 and 18.36) give \( E = \eta_w \) and \( R^* = R \).

Equation (18.37) gives

\[
E = F^{R^*}
\]
or, using \( \eta_w = (t_{w,\text{out}} - 10)/19 \) from equation (18.40) and \( R = 7.583/(t_{w,\text{out}} - 10) \) from equation (18.41)

\[
\frac{(t_{w,\text{out}} - 10)}{19} = 0.6 \left( \frac{7.583}{(t_{w,\text{out}} - 10)} \right)
\]

To simplify the algebra, we may substitute

\[
(t_{w,\text{out}} - 10) = a
\]
giving

\[
a = 19 \times 0.6 \left( \frac{7.583}{a} \right)
\]

This can easily be solved iteratively on a pocket calculator to give

\[
a = 14.56
\]
and

\[
t_{w,\text{out}} = 24.56^\circ C
\]
Using equation (18.41), again, to check the value of $R$ gives

$$ R = \frac{7.583}{14.56} = 0.521 $$

In this case, our assumption that $R < 1$ is shown to be consistent.

**Water flowrate (from equation (18.32))**

$$ m_w = R \frac{m_a (S_{w,\text{in}} - S_{\text{in}})}{C_w (t_{w,\text{in}} - t_{\text{a,in}})} = 0.521 \times \frac{12(29210 - 92.280)}{4187(10 - 29)} = 4.955 \text{ kg/s} $$

**Cooling duty**

This can be calculated directly from the air data:

$$ q = m_a (S_{\text{out}} - S_{\text{in}}) = 12(67.11 - 92.28) = -302 \text{ kW} $$

To check the heat balance, the heat gain by the water is given as

$$ q = m_w C_w (t_{w,\text{out}} - t_{w,\text{in}}) = 4.955 \times 4.187 \times (24.56 - 10) = 302 \text{ kW} $$

18.3.2.5. Heat exchange across the walls of pipes and ducts

When a fluid flowing through a pipe or duct is at a temperature different to that of the ambient air, heat exchange will occur across the walls of the pipe and any insulating material that surrounds it. The analysis of indirect heat exchangers given in Section 18.3.2.2 applies equally well for pipes and ducts.

Consider the insulated pipe shown on Figure 18.19 carrying a fluid of temperature $t_{\text{in}}$ when the temperature of the outside air is $t_{\text{out}}$. 

![Figure 18.19 Heat transfer across an insulated pipe or duct.](image)
The rate of heat transfer from a short length of pipe over which neither \( t_{in} \) nor \( t_{out} \) change significantly is given as:

\[
q = UA(t_{in} - t_{out}) \quad \text{W (see equation (18.18)}
\]

The \( UA \) value is given by

\[
\frac{1}{UA} = \frac{1}{h_{in} A_{in}} + \frac{r_2 - r_1}{k_p A_{12}} + \frac{r_3 - r_2}{k_{ins} A_{23}} + \frac{1}{(h_{out} + 0.95 h_{r})A_{out}} \quad ^{\circ}C/W \quad (18.42)
\]

where
- \( h_{in} \) = heat transfer coefficient at the inner surface (W/(m\(^2\)°C))
- \( h_{out} \) = heat transfer coefficient at the outer surface (W/(m\(^2\)°C))
- \( h_{r} \) = radiative heat transfer coefficient at the outer surface (W/(m\(^2\)°C))
- \( r \) = radii (m)
- \( k_p \) = thermal conductivity of pipe (W/(m °C))
- \( k_{ins} \) = thermal conductivity of insulation (W/(m °C))
- \( A \) = area (or mean area) across which heat transfer occurs (m)
- 0.95 = assumed value for the product of emissivity and view factor, (Whillier, 1982)

Values of thermal conductivity for pipe materials can be obtained from Table 18.3.

The heat transfer coefficients may be estimated from the following relationships which take into account the variations of fluid viscosity and thermal conductivity with respect to temperature.

**Air inside a duct**

The Colburn equation (18.22) gives the heat transfer coefficient for turbulent flow inside smooth tubes. For air, the Prandtl Number, \( Pr \), remains essentially constant at 0.7 over a wide range of temperatures and pressures (Appendix A15.3). Further analysis of the Colburn equation gives the approximation

\[
h_{in} (\text{air}) = 3.169 \left( \frac{\rho_a u_a}{d^{0.2}} \right)^{0.8} \left( 1 + 9.4 \times 10^{-4} t_{in} - 6.49 \times 10^{-6} t_{in}^2 \right) \quad \text{W/(m}^2\text{°C)} \quad (18.43)
\]

where
- \( \rho_a \) = air density (kg/m\(^3\))
- \( u_a \) = air velocity in duct (m/s)
- \( d \) = internal diameter of duct (m)

The effect of temperature is fairly small over the range of 0 to 60°C. Hence, taking a value of 20°C gives the simpler approximation

\[
h_{in} (\text{air}) = 3.22 \left( \frac{\rho_a u_a}{d^{0.2}} \right)^{0.8} \quad \text{W/(m}^2\text{°C)} \quad (18.44)
\]

The incorporation of air density allows this relationship to be employed for compressed air pipes as well as ventilation ducts.

**Water inside a pipe**

A similar analysis for water inside a pipe leads to the approximation

\[
h_{in} (\text{water}) = \left( 1430 + 20.9 t_{in} \right) \left( \frac{u_w^{0.8}}{d^{0.2}} \right) \quad \text{W/(m}^2\text{°C)} \quad (18.45)
\]

where \( u_w \) = water velocity (m/s)
In this case, the near constant value of water density allows it to be eliminated as a variable. However, the viscosity of water varies significantly in the range 0 to 60°C. Hence, the effect of temperature should not be ignored in equation (18.45).

**Air outside a pipe**

This time, the relationship given by McAdams for the outside of a single tube (equation (18.23)) provides the basis for further analysis. This leads to an approximation that may be used for both longitudinal flow and cross flow in subsurface airways.

\[ h_{out} (\text{air}) = 4.24 \left( \frac{\rho_a U_a}{d} \right)^{0.4} (1 + 0.0015 t_{out}) \quad \text{W/(m}^2 \cdot \text{°C}) \]  

(18.46)

**Radiative heat transfer coefficient**

This can be calculated from the Stefan-Boltzman equation (15.27) using the dry bulb temperature of the surrounding air, giving

\[ h_r = 22.68 \times 10^{-8} \left( 273.15 + t_{out} \right)^3 \quad \text{W/(m}^2 \cdot \text{°C}) \]  

(18.47)

However, an adequate accuracy for most purposes is obtained simply by assuming a value of \( h_r = 6 \) \text{ W/(m}^2 \cdot \text{°C})

Over long lengths of pipe or ducting the inner and outer temperatures, \( t_{in} \) and \( t_{out} \), are each likely to change in a nonlinear manner. In this case the log. mean temperature difference should be employed giving

\[ q = UA \left( \frac{\Delta t_1 - \Delta t_2}{\ln(\Delta t_1 / \Delta t_2)} \right) \quad \text{W} \]  

(18.48)

where \( \Delta t_1 \) and \( \Delta t_2 \) are the values of the temperature difference \( (t_{in} - t_{out}) \) at the two ends of the pipe and \( \ln = \) natural logarithm

Example

A chilled water pipeline is 1000 m long and comprises a 200 mm internal diameter steel pipe of thickness 4 mm, inside a 30 mm sheath of phenol formaldehyde insulation. The water has a velocity of 2 m/s and a mean temperature of 8°C while the surrounding air has a velocity of 4 m/s, a mean dry bulb temperature of 25.5°C and a density of 1.12 kg/m³.

Determine

(a) the UA value per metre length of pipeline,

(b) the actual heat gain and temperature rise of the water

and

(c) the temperature of the outer surface of the insulation.

Solution

(a) Let us first collate the data required by equation (18.42)

**Thermal conductivities** (from Table 18.3)

- steel: \( k_p = 45 \) \text{ W/(m} \cdot \text{°C})
- phenol formaldehyde: \( k_{ns} = 0.032 \) \text{ W/(m} \cdot \text{°C})

**Radii** (with reference to Figure 18.19)

\( r_1 = 0.1 \text{m}; \quad r_2 = 0.104 \text{m}; \quad r_3 = 0.134 \text{m} \)
Areas of heat transfer (per metre length)

\[ A_{in} = 2\pi r_1 = 2\pi \times 0.1 = 0.6283 \text{ m}^2 \]
\[ A_{out} = 2\pi r_3 = 2\pi \times 0.134 = 0.8419 \text{ m}^2 \]

\[ A_{12} = 2\pi \frac{(r_1 + r_2)}{2} = 2\pi \frac{(0.1 + 0.104)}{2} = 0.6409 \text{ m}^2 \]

\[ A_{23} = 2\pi \frac{(r_2 + r_3)}{2} = 2\pi \frac{(0.104 + 0.134)}{2} = 0.7477 \text{ m}^2 \]

Heat transfer coefficients

For water in the pipe (from equation (18.45)):

\[ h_{in} \text{ (water)} = (1430 + 20.9 t_{in}) \frac{u_w^{0.8}}{d_1^{0.2}} \text{ W/(m}^2 \text{ °C)} \]

\[ = (1430 + 20.9 \times 8) \frac{2^{0.8}}{(0.2)^{0.2}} = 3837 \text{ W/(m}^2 \text{ °C)} \]

For air outside the pipe (from equation (18.46)):

\[ h_{out} \text{ (air)} = 4.24 \left(\frac{\rho_g u_g}{d_3^{0.4}} \right)^{0.6} \left(1 + 0.0015 t_{out}\right) \]

\[ = 4.24 \left(\frac{(1.12 \times 4)^{0.6}}{0.268^{0.4}} \right) \left(1 + 0.0015 \times 25.5\right) = 18.33 \text{ W/m}^2 \text{ °C} \]

For radiative heat transfer at outside of pipe (from equation (18.47))

\[ h_r = 22.68 \times 10^{-8} \left(273.15 + 25.5\right)^3 = 6.04 \text{ W/(m}^2 \text{ °C)} \]

The large value of \( h_{in} \) (water) shows the relative ease with which heat is transferred from the water to the steel pipe.

The \( UA \) value per metre length can now be calculated from equation (18.42). Inserting the numerical value gives:

\[ \frac{1}{UA} = \frac{1}{3837 \times 0.6283} + \frac{0.004}{45 \times 0.6409} + \frac{0.03}{0.032 \times 0.7477} + \frac{1}{(18.33 + 0.95 \times 6.04) \times 0.8419} \]

\[ \text{water inside pipe} \quad \text{steel} \quad \text{insulation} \quad \text{air outside insulation} \]

\[ = 0.00041 + 0.00014 + 1.2538 + 0.0493 \]

\[ = 1.3037 \text{ °C/W} \]

or \( UA = 1/1.3037 = 0.7670 \text{ W/°C} \)

The dominant effect of the insulation is shown here. Indeed, the terms involving the water inside the pipe and the steel pipe itself may be neglected without significant loss of accuracy.

(b) The average heat loss per metre length of pipe is then simply

\[ q = UA \left(t_{in} - t_{out}\right) = 0.7670 \left(8 - 25.2\right) = -13.42 \text{ W per metre} \]

(negative as heat is transferred from the air to the water).
Over the 1000 m pipeline, the total heat transferred to the water is then
\[ q_{\text{tot}} = 13.42 \times 1000 = 13420 \text{ W} \]

In order to determine the actual increase in temperature of the water, \( \Delta t_w \), it is first necessary to calculate the mass flow of water

\[ m_w = \rho_w \times u_w \times \text{area of flow} = 1000 \times 2 \times \frac{\pi \times 0.2^2}{4} = 62.83 \text{ kg/s} \]

where \( \rho_w = \text{density of water (1000 kg/m}^3\) \)

Then \( \Delta t_w = \frac{q_{\text{tot}}}{C_w \times m_w} \)

where \( C_w = \text{specific heat of water (4187 J/(kg°C)) giving} \)

\[ \Delta t_w = \frac{13420}{4187 \times 62.83} = 0.051 \text{ °C} \]

This small rise in temperature justifies the assumption of a "constant" water temperature at a mean value. In practice, the actual temperature rise may be significantly greater because of damaged or imperfectly applied insulation, particularly at flange points.

(c) The heat flux remains the same at each surface boundary of the pipe and insulation. This allows the temperatures at those boundaries to be determined. In particular, at the outer surface of the insulation,

\[ q = h_{\text{out}} \times A_{\text{out}} \times (t_{\text{surf}} - t_{\text{out}}) \quad \text{where } t_{\text{surf}} = \text{surface temperature of the insulation.} \]

Inserting the known values gives

\[ t_{\text{surf}} = \frac{-13.42}{18.33 \times 0.8419} + 25.5 = -0.87 + 25.5 = 24.63°C \]

No condensation will occur on the outside of the insulation provided that the wet bulb temperature of the air remains below 24.63°C. However, any imperfections in the insulation may allow the ingress of water vapour and condensation on the steel tube. This can result in problems of corrosion.

The techniques described in this section can be incorporated into mine climate simulation programs (Chapter 16) in order to take into account heat transfers between pipes or ducts and the surrounding airflow. If the fluid inside the pipe moves in the same direction as the ventilating airstream, then the heat transfers and corresponding temperature changes can be computed sequentially along each incremental length of simulated airway (Section 16.2.2.). In the case of the piped fluid and the ventilating airstream moving in opposite directions, then an iterative procedure becomes necessary to compute the equilibrium condition.

18.3.3. Water distribution systems

A significant item in the design and economic analysis of mine air conditioning systems is the choice of piping that carries water between the refrigeration plant and heat exchangers or mining equipment.
There are, essentially, two matters of importance. First, the size of the pipe should be selected carefully. If it is too small for the given flowrate, then the frictional pressure drops and pumping costs will be high. On the other hand, if too large a pipe is chosen the costs of purchase and insulation will be high while the larger surface area and low velocity will exacerbate unwanted heat transfer with the surroundings.

This brings us to the second consideration, the thermal properties of the pipe material and the need for additional insulation. We shall discuss each of these matters in turn.

18.3.3.1. Pipe sizing

The first step in determining the diameters of the various pipes that comprise a water network is to assess the flowrates that are to be carried in each branch. The cooling duty, \( q \), of a heat exchanger is given as

\[
q = m_w C_w (t_{w,out} - t_{w,in}) \quad \text{W}
\]

where

- \( m_w \) = water flowrate through the heat exchanger (litres/s or kg/s)
- \( C_w \) = specific heat of liquid water (4187 J/kg °C)
- \( t_{w,in} \) = inlet water temperature (°C)
- \( t_{w,out} \) = outlet water temperature (°C)

The heat transferred is dependent equally upon the water flowrate and the temperature rise of the water through the unit. The latter, in turn, is governed by the construction of the heat exchanger and the supply temperature of the water. Calculations of the type included in Section 18.3.2 allow flowrates to be assessed. For service water lines, the flowrates must also satisfy the machine requirements for dust suppression or hydraulic power.

Having established the water flowrate, the actual sizing of a pipe becomes a combination of simple analysis and practical considerations. Taking into account capital, operating and installation costs, van Vuuren (1975) suggested optimum water velocities of 1.8 to 2.0 m/s for 4000 kPa pipes and 2.0 to 2.2 m/s for 7000 kPa pipes. A common rule of thumb is to employ 2 m/s for an initial exercise in pipe sizing. The size of pipe is then given as

\[
\frac{\pi d^2}{4} = A = \frac{Q}{u} \quad \text{m}^2
\]

or

\[
d = \frac{4Q}{\pi u} = 1.13 \sqrt{\frac{Q}{u}} \quad \text{m}
\]

where

- \( d \) = pipe diameter (m)
- \( A \) = pipe cross sectional area (m²)
- \( Q \) = flowrate (m³/s)
- \( u \) = velocity (m/s)

The frictional pressure drop, \( p \), along the pipe may then be determined from the Chezy Darcy relationship (equation (2.45)):

\[
p = \frac{4fL \rho_w u^2}{2d} \quad \text{Pa}
\]
For practical calculations, water density, $\rho_w$, may be taken as constant at 1000 kg/m$^3$. The length $L$ is often set at unity to give the frictional pressure loss in Pa per metre.

New steel pipes will have a coefficient of friction, $f$, approximating 0.0039. However, the aging processes of corrosion and scaling may cause this to increase by 20 to 50 percent. Plastic pipes have a coefficient of friction averaging about 0.0032. Manufacturers usually provide charts for their pipes relating frictional pressure losses to pipe diameter and flowrate. The pressure loss that can be tolerated depends upon the pumping cost function which is proportional to the product $pQ$ (ref. equation (9.17)), and the pressure required at the end of the pipeline. If the pumping costs are shown to be high, then consideration should be given to increasing the pipe diameter and reducing the velocity.

Pressure losses are also incurred at bends or pipe fittings. Again, manufacturer-specific data can be employed to determine such losses. However, for mine water systems, it is common to cater for bends and fittings by simply adding ten percent to the computed pressure drop.

Pipe diameters determined through this procedure should be rounded up to the next highest standard size. Consideration should be given to any increases in water flow that may be required in the future. Furthermore, pipe ranges that might be used for fire fighting should be sized to give adequate flow and pressure at the hydrants. The sizes of pipes that are to be installed within existing airways or shafts may also be constrained by the space available.

18.3.3.2. Pipe insulation

The performance of heat exchangers is dependent upon the temperature of the supply water. It is, therefore, important that heat gain along chilled water pipelines is minimized. This is normally accomplished by providing thermal insulation on those pipes. One exception is the special case where the pipeline is located within an intake airway which, itself, requires air cooling. In such circumstances, the insulation may be partially, or totally, omitted.

In addition to having a low thermal conductivity, the insulating materials for chilled water pipes in mines should conform to several other criteria:

- The insulation should provide a barrier against the ingress of water that condenses on the outer surfaces. A porous and permeable insulation that becomes waterlogged will be much more conductive to heat. Furthermore, water that reaches the surface of metal pipes will cause problems of corrosion. The required vapour barrier may be achieved either by covering the insulation with an impermeable sheath or by using an insulating material that is, itself, non-permeable. An example is cellular glass in which the small air pockets are isolated from each other.

- The insulation or outside sheath should have sufficient mechanical strength to withstand the rigours of transport and utilization in mine environments. Insulation on shaft pipelines must be able to withstand impact from falling objects.

- The flammability of insulating materials and the toxicity of gases that they may produce when heated in a mine fire is of great importance. Polyurethane foams have been employed widely in the past. However, they emit carbon monoxide and, perhaps, cyanide vapours at temperatures that may be reached in mine fires. Phenolic foams in which phenols are bound into a resin base are safer in that they have a higher ignition temperature, and a much lower rate of flame spread (Rose, 1989).

- The cost of purchase and ease of installation of insulating materials or pre-insulated piping should also be taken into account.
A common practice has been to apply insulating materials to pipes on the mine site - either on the surface or after the pipework has been installed in its underground location. The application of foam insulators may be carried out by injecting the foam within a PVC vapour barrier sheath that has been pulled over spacers on the outside of the pipe. Pre-cast semicircular sections of cellular glass or other rigid insulators may be bound on to the pipes with adhesives or an outer sealing material. Expanded polystyrene granules and fibreglass have also been used widely (Arkle, 1985; Ramsden, 1985). The thickness of applied insulators varies from 25 to 40 mm.

The continued development of thermoplastics has allowed the manufacture of pipes that combine mechanical strength with good thermal insulating properties. These may consist of concentric and impermeable cylinders with the annulus containing a cellular infill. Typical values of thermal conductivity are given in Table 18.3.

<table>
<thead>
<tr>
<th>Material</th>
<th>Thermal conductivity W/(m °C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Still air</td>
<td>0.028</td>
</tr>
<tr>
<td>Expanded polystyrene</td>
<td>0.038</td>
</tr>
<tr>
<td>Polyurethane and polystyrene</td>
<td>0.034</td>
</tr>
<tr>
<td>Polyisocyanurate</td>
<td>0.023</td>
</tr>
<tr>
<td>Phenol formaldehyde</td>
<td>0.032</td>
</tr>
<tr>
<td>Glass fibre</td>
<td>0.029</td>
</tr>
<tr>
<td>Unplasticised polyvinylchloride (UPVC)</td>
<td>0.14</td>
</tr>
<tr>
<td>Still water</td>
<td>0.62</td>
</tr>
<tr>
<td>Wood</td>
<td>0.17</td>
</tr>
<tr>
<td>Steel</td>
<td>45</td>
</tr>
</tbody>
</table>

Table 18.3. Thermal conductivities of pipe materials.

18.3.3.3. Energy and temperature changes within water systems

In Section 3.4.1 we saw that fluid frictional effects have no influence on temperature changes within a steady-state airflow. This is not the case for water or other near incompressible fluids. Furthermore, in Section 15.3.1, the autocompression of air in a mine shaft was shown to produce an increase in air temperature that depended upon the depth of the shaft. Again, this is not true for water in a shaft pipe.

In order to reassess the situation for water, let us return to the steady flow energy equation (3.25).

$$\frac{u_1^2 - u_2^2}{2} + (Z_1 - Z_2)g + W_{12} = \int_{1}^{2} V \, dP + F_{12} = (H_2 - H_1) - q_{12} \frac{J}{kg}$$ (18.52)

where

- $u$ = velocity (m/s)
- $Z$ = height above datum (m)
- $g$ = gravitational acceleration (m/s$^2$)
- $W$ = work input (J/kg)
- $V$ = specific volume of fluid (m$^3$/kg) = 1/$\rho$ where $\rho$ = fluid density (kg/m$^3$)
- $F$ = work done against friction (J/kg)
- $H$ = enthalpy (J/kg).

and $q$ = heat added from surroundings (J/kg).
In applying this equation to water, we recall the basic definition of enthalpy from equation (3.19):

\[ H = PV + U \quad \text{J/kg} \quad (18.53) \]

where \( U \) = internal energy and by assuming the water to be incompressible, \( V \) = constant.

The right hand sides of equation (18.52) then give

\[ V(P_2 - P_1) + F_{12} = P_2V - P_1V + U_2 - U_1 - q_{12} \]

or

\[ F_{12} = U_2 - U_1 - q_{12} \quad \text{J/kg} \quad (18.54) \]

However, we may take a single value for the specific heat of water, \( C_w = 4187 \text{ J/(kg °C)} \).

Equation (3.28) then gives

\[ \left( U_2 - U_1 \right) = C_w (T_2 - T_1) \quad \text{J/kg} \quad (18.55) \]

Substituting in equation (18.54),

\[ F_{12} = C_w (T_2 - T_1) - q_{12} \quad \text{J/kg} \quad (18.56) \]

Furthermore, the frictional pressure drop, \( p \), is given by equation (2.46)

\[ p = \rho_w F_{12} = \rho_w C_w (T_2 - T_1) - \rho_w q_{12} \quad \text{Pa} \quad (18.57) \]

where \( \rho_w \) = density of water (kg/m\(^3\))

In the particular case of adiabatic flow \( (q_{12} = 0) \), the increase in water temperature along a pipeline becomes

\[ T_2 - T_1 = \frac{p}{\rho_w C_w} \quad ^\circ \text{C} \quad (18.58) \]

The term \( 1/\rho_w C_w \) is sometimes known as the Joule-Thompson coefficient and, for water, has a standard value of \( 1/(1000 \times 4187) = 2.39 \times 10^{-7} \text{ °C/Pa} \).

The adiabatic rise in temperature along a water pipe is shown by equation (18.58) to be directly proportional to the frictional pressure drop and is independent of changes in elevation. The frictional pressure drop, \( p \), may occur by viscous or turbulent action within the length of the pipe or by shock loss across a valve or other obstruction.

For flow through a water pump or turbine, we can apply the left and right sides of equation (18.52). However, the change in kinetic energy and the difference in elevation across the device are both negligible, giving

\[ W_{12} = (H_2 - H_1) - q_{12} \]

\[ = V(P_2 - P_1) + (U_2 - U_1) - q_{12} \]

\[ = \frac{(P_2 - P_1)}{\rho_w} + C_w (T_2 - T_1) - q_{12} \quad \text{J/kg} \quad (18.59) \]

If there is negligible heat exchange across the casing of the pump or turbine, then we may assume adiabatic flow \( (q_{12} = 0) \) giving the temperature rise as

\[ T_2 - T_1 = \frac{1}{C_w} \left( W_{12} - \frac{(P_2 - P_1)}{\rho_w} \right) \quad ^\circ \text{C} \quad (18.60) \]
While $W_{12}$ is the total work applied to a pump impeller, the term $(P_2 - P_1)/\rho_w$ represents the output of useful mechanical energy. The difference between the two indicates the degradation of some of the input power to thermal energy and produces a measurable temperature rise. In a turbine, it is the other way round with $(P_1 - P_2)/\rho_w$ representing the work input and $-W_{12}$ being the output work.

For a pump, we can also write that the actual work output is given by

$$W_{12} = \frac{(P_2 - P_1)}{\rho_w} \eta \quad \text{J/kg}$$  \hspace{1cm} (18.61)

and, for a turbine, the actual work output is

$$W_{12} = \frac{(P_1 - P_2)}{\rho_w} \eta \quad \text{J/kg}$$  \hspace{1cm} (18.62)

where $\eta = \text{efficiency of the device}$

Substituting for $W_{12}$ in equation (18.60) gives the temperature rise across a pump:

$$(T_2 - T_1) = \frac{(P_2 - P_1)}{C_w \rho_w} \left[ \frac{1}{\eta} - 1 \right] \quad \text{°C}$$  \hspace{1cm} (18.63)

and across a water turbine

$$(T_2 - T_1) = \frac{(P_2 - P_1)}{C_w \rho_w} (\eta - 1) = \frac{(P_1 - P_2)}{C_w \rho_w} (1 - \eta) \quad \text{°C}$$  \hspace{1cm} (18.64)

In these analyses, it has been assumed that water is incompressible. In fact, the slight compressibility of water does give it a small isentropic temperature change with respect to pressure (Whillier, 1967). This is given approximately as

$$\frac{At}{AP} = (0.759 t - 0.2) \times 10^{-6} \quad \text{°C per kPa}$$  \hspace{1cm} (18.65)

where $t = \text{initial temperature (°C)}$

This effect may be neglected for most practical purposes. However, if equations (18.63) or (18.64) are transposed in order to determine the efficiency of the device from temperature observations, then the measured temperature rise should be corrected according to equation (18.65) (negatively for a pump and positively for a turbine).

**Example 1**

The barometric pressures at the top and bottom of a 1500 m deep shaft are 90 and 107 kPa respectively. Chilled water at 3°C and ambient atmospheric pressure enters the top of a 150 mm diameter steel pipe that is suspended in the shaft. A pressure reducing valve at the bottom of the pipe controls the water flow rate to 35 litres/s and reduces the water pressure to that of the ambient surroundings. Determine the pressures and temperatures of the water at the inlet and outlet of the pressure reducing valve. Assume that no heat exchange occurs across the pipe wall.
Solution

The frictional pressure drop in the pipe is given by equation (18.51)

\[ p = \frac{4 f L \rho_w u^2}{2d} \text{ Pa} \]

The known values are

\[ L = 1500 \text{ m} \quad \rho_w = 1000 \text{ kg/m}^3 \quad d = 0.15 \text{ m} \]

where \( Q = \) flowrate = \( 35/1000 = 0.035 \text{ m}^3/\text{s} \) and \( A = \) c-s area = \( \pi \times (0.15)^2/4 = 0.01767 \text{ m}^2 \)

giving \( u = \frac{Q}{A} u = 1.981 \text{ m/s} \)

Allowing for aging effects, we shall assume the coefficient of friction for the steel pipe to be \( f = 0.005 \).
We shall also allow an additional pressure loss of 10 percent to allow for bends and fittings.
Then

\[
\frac{4 \times 0.005 \times 1500 \times 1000 \times (1.981)^2}{2 \times 0.15} \times (1 + 0.1) = 431.68 \times 10^3 \text{ Pa}
\]

Water pressure at shaft top = \( 90 \times 10^3 \text{ Pa} \)

Water pressure at shaft bottom (inlet to pressure reducing valve)

\[
= 90 + \rho_w g L - p
\]

\[
= 90 \times 10^3 + (1000 \times 9.81 \times 1500) - 431.68 \times 10^3
\]

\[
= 14,373 \times 10^3 \text{ Pa or } 14,373 \text{ kPa}
\]

At outlet from the valve, the water pressure is reduced to the shaft bottom barometric pressure of 107 kPa.

Hence,

pressure drop across the valve = gauge pressure at valve inlet

\[
= 14,373 - 107 = 14,266 \text{ kPa}
\]

The temperature rise in the shaft pipe is given by equation (18.58) as

\[
\Delta T_{sh} = \frac{p}{\rho_w C_w} = \frac{431.68 \times 10^3}{1000 \times 4187} = 0.103 \text{ °C}
\]

Temperature at valve inlet = water temperature at shaft top + 0.103 = 3.103 °C

As no energy is added or extracted by the pressure reducing valve, the flow is purely frictional.
Hence, equation (18.58) gives

\[
\Delta T_{valve} = \frac{\text{Pressure loss across valve}}{\rho_w C_w} = \frac{14,266 \times 10^3}{1000 \times 4187} = 3.407 \text{ °C}
\]

giving the water temperature at the valve outlet to be

\[
3.103 + 3.407 = 6.510 \text{ °C}.
\]
The same total increase in water temperature of 3.51°C from shaft top to outlet of the valve at the shaft bottom would occur whatever the cause of the restriction on flow - a valve, rougher lining of the pipe or a smaller diameter of pipe, provided that no energy of any kind is added or extracted from the water and that the water is emitted at the shaft bottom barometric pressure.

As a check, the overall temperature rise of the water between the shaft top and the outlet of the shaft bottom valve can be calculated directly. From the steady flow energy equation (18.52) with no work input \( (W_{12} = 0) \), for an incompressible fluid \( \dot{V} = 1/\rho = \text{constant} \), and ignoring changes in kinetic energy

\[
(Z_1 - Z_2)g = \dot{V}(P_2 - P_1) + F_{12}
\]

or

\[
\rho_{\text{tot}} = \rho_w F_{12} = (Z_1 - Z_2)g \rho_w - (P_2 - P_1)
\]

\[
= 1500 \times 9.81 \times 1000 - (107 - 90) \times 1000
\]

\[
= 14.715 \times 10^6 - 0.017 \times 10^6
\]

\[
= 14.698 \times 10^6 \text{ Pa}
\]

(The difference in barometric pressure is shown to have very little effect on \( \rho_{\text{tot}} \).)

Then

\[
(T_t)_{\text{tot}} = \frac{\rho_{\text{tot}}}{\rho_w C_w} \text{ (equation (18.58))}
\]

\[
= \frac{14.698 \times 10^6}{1000 \times 4187}
\]

\[
= 3.51 \degree \text{C}
\]

This check calculation illustrates that when the water is emitted from the pipe at the underground barometric pressure, the temperature rise is given approximately as

\[
\frac{(Z_1 - Z_2)g}{4187} = 3.51 \degree \text{C over the 1500m depth of the shaft, or 2.34 \degree \text{C per 1000 m depth.}}
\]

In the past, this has mistakenly been referred to as an "autocompression temperature rise." The misconception arose because in the special case of a pipe open at the bottom, the frictional pressure drop in the pipe approximates the static pressure caused by a column of fluid in the shaft pipe.

Example 2

If, in Example 1, the pressure reducing valve at the shaft bottom were replaced by a water turbine of efficiency 75 percent, what would be the temperature of the water leaving the turbine? Calculate, also, the power produced by the turbine. Assume that all other parameters, including the flowrate of 35 litres/s, remain unchanged.

Solution

As the condition and flow of water in the pipe have not been altered, the increase in water temperature down the pipe remains at \( \Delta T_{\text{tot}} = 0.103 \degree \text{C} \) and the water enters the turbine at a pressure of \( P_{\text{in}} = 14373 \text{ kPa} \) and a temperature of 3.103°C.
At outlet from the turbine, the water pressure has fallen to the shaft bottom barometric pressure of \( P_{\text{out}} = 107 \text{ kPa} \). The temperature rise across the turbine is given by equation (18.64).

\[
\Delta t_{\text{turb}} = \frac{(P_{\text{in}} - P_{\text{out}})}{C_w \rho_w} (1 - \eta) = \frac{(14373 - 107)10^3}{4187 \times 1000} (1 - 0.75) = 0.852 \text{ °C}
\]

Then the temperature of the water leaving the turbine is

\[
3.103 + 0.852 = 3.955 \text{ °C}.
\]

Comparing this with the temperature of water leaving the pressure reducing valve of Example 1, i.e. 6.510 °C, shows that replacing the valve by a water turbine not only produces useful power but reduces the temperature of the chilled water leaving the shaft station by 6.510 - 3.955 = 2.555°C.

This represents an equivalent refrigeration capacity of

\[
m_w C_w \Delta t = 35 \times 4187 \times 2.555 = 374.5 \times 10^3 \text{ W or 374.5 kW}
\]

The output work from the turbine is given by equation (18.62)

\[
W_{12} = \frac{(P_{\text{out}} - P_{\text{in}}) \eta}{\rho_w} = \frac{(107 - 14373)10^3}{1000} \times 0.75 = -10700 \frac{\text{J}}{\text{kg}}
\]

(negative, as work is leaving the water). To transform this to an output power, we simply multiply by the mass flowrate, i.e.

\[
10700 \times 35 = 374.5 \times 10^3 \text{ W or 374.5 kW}
\]

We have, therefore, demonstrated that the power output from the turbine is equal to the equivalent refrigeration capacity that has been produced by the system. This suggests a rapid means of approximating the overall temperature rise of the water from the shaft top to the turbine outlet at shaft bottom barometric pressure:

\[
\Delta t_{\text{tot}} = \frac{(Z_1 - Z_2) g}{C_w} - \frac{\text{turbine output power}}{m_w C_w} = \frac{1500 \times 9.81}{4187} - \frac{374500}{35 \times 4187} = 3.514 - 2.555 = 0.959 \text{ °C}
\]

This method remains acceptable provided that the frictional pressure losses are small compared with the static pressure caused by the column of water.

### 18.3.4. Energy recovery devices

The trend towards siting main refrigeration plant on the surface of mines accelerated during the 1980’s. This was caused in large part by the utilization of energy recovery devices, particularly water turbines. Prior to that time, the provision of chilled water at the surface had three major drawbacks. First, it caused high water pressures at shaft bottoms which necessitated a choice between high pressure heat exchangers and associated pipework or, alternatively, to pass the water through pressure reducing valves. The latter choice not only resulted in high operating costs of pumping to return the water to surface, but also converted the potential energy of the water to thermal energy, giving a rise in temperature and eroding the cooling capacity of the chilled water.
Energy recovery devices address each of these problems and divide into two major groups. First, water turbines or hydraulic motors employ the potential energy of shaft water lines to drive electrical generators or other devices. Secondly, hydrolift systems utilize the weight of descending chilled water to help raise heated return water back to surface. We shall discuss each of these systems in turn.

18.3.4.1. Water turbines

As illustrated by Example 2 in section 18.3.3.3, shaft bottom turbines provide:

- work output that may be used to drive pumps, generators or any other mechanical device; this work may provide part of the energy required to raise return water back to surface;
- reduced temperature of the subsurface supply of chilled water, and hence, savings in refrigeration costs;
- a low pressure water system underground.

Two types of turbines are in use. Figure 18.20 illustrates an impulse turbine, commonly known as a Pelton Wheel. In this device, the pressure of the supply water is used to produce kinetic energy by means of one or more nozzles. The high velocity jets are directed at cups attached to the periphery of a rotating wheel. Inside the casing of the device, the jets operate at the ambient atmospheric pressure. Hence, the water leaves at zero gauge pressure. Control of the mechanical output may be achieved by a needle valve in the nozzle which changes the water flowrate. Alternatively, if the flow is to be maintained, deflector plates may be used to direct part of the jets away from the cups.

The advantages of a Pelton Wheel are that it is simple and rugged in construction, reliable and may reach efficiencies of over 80 percent. The speed control is effective and no high pressure seals are required on the casing.

![Diagram of a Pelton Wheel](image-url)
The second type of water turbine is the reaction device and is often referred to as a Francis or Kaplan turbine. This is essentially a centrifugal runner (Figure 10.2) with the impeller designed to operate in reverse and incorporating guide vanes to minimize shock losses. Indeed, a reversed centrifugal pump may be used as a reaction turbine although at a considerably reduced efficiency. Control of the device is by valves located upstream and/or downstream. Unlike the Pelton Wheel, a reaction turbine operates in a "flooded" mode giving rise to its primary advantage, i.e. the water pressure at exit can be maintained above the ambient air pressure and, hence, may be utilized further downstream to overcome pipe losses, provide the required pressure at spray chambers or to activate further hydraulic devices. Pelton Wheels give better efficiencies at the higher duties, typically 2 MW. The Francis turbines are better suited for lower duties.

In addition to shaft bottom stations, water turbines may be used for more local applications. For example, if the water pressure available at a spray chamber exceeds that required at the spray nozzles, then a turbine can be sited in the water supply line and used to power local pumps or fans. This is particularly advantageous for multi-stage spray chambers (Ramsden, 1985).

18.3.4.2. Hydrolift systems

The principle of operation of a hydrolift system is illustrated on Figure 18.21. The device is sometimes also known as a hydrochamber or hydrotransformer. The chamber is excavated or constructed underground and is alternately filled from opposite ends with chilled water delivered from surface, then with hot water returned from the mine.

Let us assume that the chamber is initially filled with warm water. Both of the automated valves A are opened and valves B are closed. The pump at the surface cold water dam causes chilled water to flow down the shaft pipe and into the chamber. This, in turn, forces the hot water to exit the chamber and ascend the return shaft pipe. As the two columns of water are nearly balanced, the duty of the pump is simply to overcome pipe friction. When the chamber is filled with chilled water, valves A are closed and valves B are opened. The underground hot water dam pump then forces the hot return water into the chamber, expelling the chilled water to the underground cold water dam.

The cyclic nature of the system can be negated by having three hydrolift chambers with the sets of valves phased to give a continuous flow into and out of the water dams. The size of chambers and water dams should be determined on the basis of the cooling requirements of the mine. An inherent disadvantage of the hydrolift system is that the rate of chilled water flow must be equal to that of the heated return water. It is, therefore, advisable that the water dams should be sized to accommodate several hours supply.

A loss of efficiency occurs because of heat exchange with the walls of the chamber and due to mixing and thermal transfer at the hot and cold interface within the chamber. The latter may be minimized by inclining the chamber such that the hot water end is at a higher elevation.

---

2 As a guide, if the pressure differential available across the turbine (in kPa) is less than $70 m_w^{0.67}$ (where $m_w$ is the flowrate in litres/s), then a Francis turbine will give a better performance.
18.3.5. Design of mine cooling systems

The demand on a subsurface cooling system is dictated by the magnitude and distribution of the heat load and the surface climate. The availability of cold natural water at the surface can result in large savings in cooling costs. Similarly, passing the water supply through a surface cooling tower either prior to or instead of a refrigeration plant can produce low-cost cooling.

Section 18.3.1 gave an overview of mine cooling systems and introduced spot coolers for isolated heat loads (Figure 18.8), water chiller plant for sections of a mine (Figure 18.9) and a large centralized system (Figure 18.10). In this Section, we shall concentrate on the latter and discuss the alternative arrangements by which the hardware components may be integrated into a unified mine cooling system.

18.3.5.1. Location of main plant

Figures 18.22 to 18.24 give examples of system configurations using main refrigeration plant located on surface, underground and a combination of the two. In these figures, the ancillary components of pumps, valves and retention dams are not shown.
Surface plant
The ease of maintenance, construction and heat rejection, coupled with the utilization of energy recovery devices have led to a preference for the main refrigeration units to be located on surface. Figure 18.22(a) illustrates the simplest system that employs surface plant. Chilled water from the refrigeration plant passes down an insulated pipe in the shaft to a high pressure water to water heat exchanger at one or more shaft stations. This heat exchanger is a source of potential loss of system efficiency and care should be taken with the design and choice of tube materials to minimize problems of corrosion, scaling and erosion. Figure 18.22(b) shows the water to water heat exchanger replaced by a turbine. This configuration requires the water flow in the shafts to approximate that cycling through the mine, with capacitance provided by water dams.

In both cases, the heated water ascending to surface may be sprayed through a pre-cooling tower in order to lower its temperature before returning to the refrigeration plant. This can be advantageous in reducing the required capacity and operating costs of the refrigeration plant. However, the effectiveness of a pre-cooling tower is dependent upon the difference between the temperature of the return water and the wet bulb temperature of the surface atmosphere. An "approach" (see Figure 18.18) of 2°C is typical. For this reason, a pre-cooling tower may be utilized only during the colder seasons.

Figure 18.22(b) also shows part of the chilled water being used for bulk cooling of the air before it enters the downcast shaft. The vertical packed spray chambers used for this purpose are normally of the induced-draught type with the fan located between the spray chamber and the shaft. This helps to prevent fogged air from entering the shaft. If the experience of any given mine is that a heat problem exists during the summer months only, then all that may be necessary is to install a surface bulk air cooling system and utilize it during the warm season. However, if the workings are distant from the shaft bottoms and the heat production is concentrated heavily within those working zones, then bulk air cooling on surface will have greatly reduced effectiveness.

Underground plant
Figure 18.23 illustrates the principle of the system favoured by the South African gold mines prior to the 1980's. The refrigeration plant is located entirely underground. Hot water from the condensers is cycled through open spray cooling towers situated at or near the base of upcast shafts and supplied by air returning from the mine. The advantages of this system are that it eliminates the need for surface-connecting pipe ranges and the associated pumping costs. It also avoids any environmental problems that may arise from surface plant. The major disadvantage of the system is that its duty is limited by the capacity of the return air to accept heat rejected in the underground cooling towers. Furthermore, saturated air subjected to decompression in the upcast shaft can cause problems of heavy fogging, water blanketing in the shaft (Section 9.3.6) and reduced life of main exhaust fans.
Figure 18.22 Examples of system configurations using surface refrigeration plant.
Combinations of surface and underground plant

As the depth of mining increases, so also does the severity of geothermal and autocompressive heat problems. Unfortunately, increasing depth also exacerbates the costs of installing, maintaining and operating pipelines in shafts and, hence, eroding the advantages of a surface plant. Figures 18.24 (a, b and c) illustrate examples of alternative means of combining surface and underground refrigeration plant into an integrated system. In each of these configurations, the essential role of the underground plant is to concentrate the heat in the return mine water into a lower flow, higher temperature system for transmission to surface.

This enables smaller shaft pipes to be selected and also reduces the pumping costs. Furthermore, the increased temperature of the water returning to surface improves the effectiveness of the pre-cooling tower, allowing it to be utilized throughout the year. Another advantage of the combined system is that it adds considerably to the flexibility and future upgrading of the cooling duty. Surface plant alone is ultimately limited by the flow capacity of the shaft pipes. Similarly, underground plant alone is limited by the thermal capacity of the return air. However, with a combined system the cooling duty can be more easily upgraded by adding to the capacity of both surface and underground plant and, hence, increasing the temperature difference between the descending and ascending shaft pipes.
Figure 18.24 Examples of system configurations for combined surface and underground refrigeration plant.
The simplest combined system is shown on Figure 18.24(a) where the surface plant is used to cycle water through the shaft pipes and to remove heat from the subsurface condensers. Figure 18.24(b) shows a system of potentially improved efficiency in which the chilled water from surface first passes through a water-to-water heat exchanger before entering the condensers. The warm return water from the mine is pre-cooled in the water-to-water heat exchanger prior to passing through the evaporators.

Figure 18.24(c) shows the water-to-water heat exchanger replaced by a turbine. In this case, the chilled water emerging from the turbine passes on to the mine via a cold water dam. The return water in the hot water dam is split. Part of it removes heat from the condenser and returns to surface. The remainder is re-cooled through the subsurface evaporator and returned to the cold water dam.

In each of the three examples of combined plant shown on Figures 18.24 (a, b and c), some of the chilled water produced on surface may be used for bulk cooling the downcast air. Similarly, in all three cases, if mine return air is available for heat rejection, then some of the hot water from the underground condenser may be cycled through a subsurface cooling tower before returning to the hot water dam.

18.3.5.2. Service water cooling

Service water is supplied to mining equipment primarily for dust suppression purposes and perhaps, also, as a means of transmitting hydraulic power (Section 18.3.5.3). Heat and dust problems are generally greatest in the working areas and, particularly, where rock breaking equipment is in use. The heat emitted by the machines combines with that from the newly exposed strata to cause rapid increases in air temperature (Section 15.2.2). Uncooled water supplied at dry bulb temperature to dust suppression sprays exacerbates the problem by increasing the humidity of the air. However, if the service water is chilled, then it will both cool and dehumidify the air through which it is sprayed. This effect will continue until the effluent water running into drains or on rock surfaces attains the ambient wet bulb temperature.

The effect of chilled service water in a local workplace can be quite dramatic, depending upon the temperature and flowrate of both the water and the ventilating airstream. Indeed, it is recommended that the temperature of the service water should not be less than 12°C to prevent cold discomfort of personnel. A primary advantage of chilled service water is that it is applied where and when cooling is most needed, i.e. where and when rock breaking or rock transportation is taking place.

The cooling efficiency of chilled service water is greater than that normally obtained from heat exchangers because of the higher temperature of the return water. For example, water supplied to sprays at 12°C may eventually leave the district at the prevailing wet bulb temperature of, say, 27°C, giving a range of 15°C. For comparison, typical temperature ranges of water across local heat exchangers may be of the order of 6 to 10°C. It follows that for the same cooling duty, lower water flowrates can be used when the service water is chilled. This, in turn, reduces the required size of pipes and pumping costs. Furthermore, the increased temperatures of return water enhance the utilization of the low-cost pre-cooling towers on the mine surface.

The effectiveness of air cooling at the entrance to a stope or face is diminished by a large heat flux from newly exposed or fragmented strata. However, such heat can be partially counteracted by the use of chilled dust suppression water, enabling intake air cooling to be employed more effectively.

The benefits to be gained from chilled service water clearly depend upon how much water is required by the equipment, and any adverse effects on the strata or travelling conditions that may arise from too copious a supply of water. The practical approach is to determine the rate of water flow required for dust suppression or other machine purposes and the expected temperature range. The fraction of the total heat load removed by service water can then be estimated.
Example
Dust suppression water is supplied to a workplace at a rate of 100 litres per tonne of rock mined. If the water is supplied to the machine at 12°C and leaves the district at 26°C, determine the cooling provided by the service water at a mining rate of 5 tonnes per minute.

Solution
Water flow rate, \( m_w = \frac{100 \times 5}{60} = 8.333 \) litres/kg/s

Rate of heat removal = \( m_w \cdot C_w \cdot \Delta t \)

\[ = 8.333 \times 4187 \times (26 - 12) \]
\[ = 488.5 \times 10^3 \text{ W or } 488.5 \text{ kW} \]

The amount of service water used may vary from 20 to 3000 litres per tonne depending upon the mining method and the mineralogical constituents of the dust particles. There is a corresponding large range in the rate of possible heat removal by dust suppression water. However, due to the high efficiency of the technique and the utilization of existing water lines, it may be regarded as the first approach to the supply of “coolth” to a rock breaking area.

The fraction of total heat load that can be removed by chilled service water is eroded at greater depths due to the increased effects of autocompression and geothermal heat, while the supply of service water remains constant for any given mining method.

18.3.5.3. Hydropower
In section 18.3.4, we discussed the utilization of potential energy made available in shaft water pipes. In addition to driving shaft station turbines or hydrolift systems, the water pressure available at spray chambers can also be used to activate fans and pumps at those locations. The concept of hydropower takes this idea further and combines it with service water cooling (Section 18.3.5.2).

The use of electrical or diesel power involves the production of heat. Expenditure is then incurred to remove this heat from the mine by ventilation and/or cooling systems. The utilization of machine power by such means involves both an environmental and an associated cost penalty. However, if the machine is activated by chilled water under pressure provided by the shaft column, then the hydraulic machine power and climatic control act in concert rather than in opposition. In a hot and deep mine, chilled hydropower provides, simultaneously, a means of machine power, cooling and dust suppression, all from a single system.

For relatively low powered units such as drills, hydraulic motors may be employed directly while, for larger machines, turbine-electric combinations may be more practical. Furthermore, differing combinations of water flow, pressure and temperature may be required for the demands of power, cooling and dust suppression. A hydropower system should be designed and balanced to satisfy those varying demands.

18.3.5.4. Ice systems
The limitations of heat rejection capability underground, together with the use of energy recovery devices, have promoted the trend away from subsurface refrigeration plant towards surface units. However, despite water turbines and hydrolift devices, pumping costs remain the factor which limits the mass flow of water that can be circulated through the shaft pipelines of deep mines. Recalling the
relationship $q = m_w C_w \Delta t$, heat removal from a mine can be enhanced by increasing the temperature differential, $\Delta t$, between supply and return water. This can be accomplished by combinations of surface and underground refrigeration plant (Section 18.3.5.1). Despite such advances in mine cooling technology, heat remains the primary limitation on the depth at which mining can take place.

In order to proceed beyond the physical restraints imposed by the circulation of fluids, other means of transmitting "coolth" have to be considered. One of these is to employ the latent heat of melting ice. Although the use of ice in mines is not new (Section 18.1), the first pilot plant employing an ice pipeline was constructed in 1982 at the East Rand Proprietary Mines in South Africa (Sheer, 1984). Further full scale applications followed soon afterwards (Hemp, 1988).

Ice has a specific heat varying from 2040 J/(kg °C) at 0°C to 1950 J/(kg °C) at -20°C. It has a latent heat of melting of 333.5 kJ/kg. The mass flow of water circulated around a mine for cooling purposes can be reduced by a factor of over 5 if it is supplied in the form of ice.

Example
A mine requires 10 MW of cooling. Calculate the mass flow of water involved

(a) if the water is supplied at 3°C and returns at 20°C
(b) if the water is supplied as ice at -5°C and returns at 20°C.

Solution
(a) for the water system:

\[ q = m_w C_w \Delta t_w \]

or

\[ m_w = \frac{q}{C_w \Delta t_w} = \frac{10 \times 10^6}{4187 \times (20 - 3)} = 140.5 \text{ kg/s or litres/s} \]

(b) for the ice supply:

Heat absorbed as the temperature of $m_i$ kg of ice increases from -5 to O°C,

\[ q_{ice} = m_i C_i \Delta t_i \]

Choosing the specific heat of ice as 2030 J/(kg °C) gives

\[ q_{ice} = m_i \times 2030 \times 5 = 10 150 m_i \text{ W} \]

Latent heat absorbed as $m_i$ kg of ice melts at 0 °C

\[ q_L = L_{ice} m_i = 333 500 m_i \]

Heat absorbed as $m_i$ kg of liquid water increases in temperature from 0 to 20°C:

\[ q_w = m_i C_w \Delta t_w = m_i \times 4187 \times 20 = 83 740 m_i \text{ W} \]

Then, total heat absorbed

\[ q = q_{ice} + q_L + q_w = 10 \times 10^6 \text{ W} \]

\[ m_i (10 150 + 333 500 + 83 740) = 10 \times 10^6 \text{ W} \]

giving $m_i = 23.4 \text{ kg/s}$
Hence, in this example, the volume of cooling water is reduced by a factor of 6 when it is supplied as ice at -5°C. The calculation also illustrates the dominant amount of heat involved in the change of phase from ice to liquid, \( q_L \).

There are four primary items to consider when investigating an ice system for mine cooling:

- the large scale manufacture of ice,
- the means of transporting it underground,
- how it is best incorporated into a mine cooling system, and
- the economics of the system.

Each of these factors will be considered in turn.

**Manufacture of ice**

The ice that is utilized in a mine cooling system may be supplied either as particulate ice at sub-zero temperature or as a slurry of ice crystals within liquid water or brine.

**Particulate ice** can be manufactured as cubes, cylinders or tubes of clear ice, or as ice flakes that can be compressed into pellets for transportation. Water is sprayed or caused to flow over the surfaces of plates, tubes, concentric cylinders or drums that are cooled on their opposite sides by a refrigerant fluid. The formation of ice creates an additional thermal resistance to heat transfer. This necessitates an evaporator temperature of some -15 to -30°C giving rise to coefficients of performance in the range 2.3 to 3.5, lower than might be expected from a well designed water chiller.

The ice is usually removed from the cold surfaces by mechanical scrapers or by periodic cycling of hot refrigerant through the freezer unit. In the production of flake ice, a thin film of ice is removed continuously from the surface of a rotating drum or metal belt. The particles of ice fall into a hopper that feeds into a screw or belt conveyor for removal from the unit.

As an example of large-scale particulate ice manufacture, a plant at East Rand Proprietary Mines (South Africa) produced 6000 tonnes of ice per day from six 1000 tonne/day units. Each tubular unit consisted of eighty double walled tubes with ice formation on both the inner (99 mm dia) and outer (508 mm diameter) surfaces of the 4.5 m long tubes. The refrigerant passed through the annulus of each tube. The 15 minute cycle consisted of 13.5 minutes of freezing and 1.5 minutes harvesting (Hemp, 1988). The refrigerant was ammonia and ice harvesting was achieved by passing hot ammonia liquid through the annuli in the icemaker tubes. This arrangement also improved the efficiency of the unit by subcooling the liquid refrigerant before it returned to the evaporators.

**Slurry icemakers** promote the formation of microcrystals of ice within a stream of water or brine and are at an earlier stage of development for mine cooling systems. Slurry ice has one significant advantage over particulate ice. All industrial and domestic supplies of water contain many impurities including dissolved salts of calcium, magnesium and sodium. The freezing process favours pure water. Hence, if the freezing occurs sufficiently slowly or if the ice/water interface is continuously washed with water, then the ice crystals will have a greater purity than the water in which they grow. The dissolved salts remain in the liquid and, hence, become more concentrated. In contrast, the formation of plate, tube or flake (particulate) ice occurs quickly and with a more limited washing action by the water, giving rise to the entrapment of salts within the ice matrix. If the microcrystals, usually of less than 1 mm in size, are removed from the ice slurry and washed, then they will be relatively free from impurities. This process is known as freeze desalinization. The provision of purified water to a mine cooling system assists greatly in minimizing corrosion and scaling within pipes and heat exchangers.

There are at least three methods that have the potential for large-scale manufacture of slurry ice (Shone, 1988). In the **indirect process**, water or brine is passed through tubes that are surrounded by cold refrigerant. Microcrystals of ice are formed within the moving stream of water. Careful control of the salinity, flowrate and temperature is required to prevent the formation of solid ice on the sides of the tube. Scraper devices can also be used to prevent ice from accumulating on the walls.
The vacuum ice-making process involves the evacuation of water vapour from a vessel that contains brine. The triple point of water is reached at 0.6 kPa and 0 °C when boiling and freezing occur simultaneously (ref. Figure 18.1). Ice, liquid water and vapour coexist at this point. The ice slurry that is formed is kept in motion by an agitator until it is pumped from the vessel. About 1 kg of water vapour must be removed for each 7.5 kg of nucleated ice. The vapour is compressed and condensed for recycling. In the vacuum ice-making process, the water acts as its own refrigerant fluid (Section 18.2.2).

An even less developed method is the direct process in which a mixture of brine and immiscible liquid refrigerant are sprayed through a nozzle into a receiving chamber. The evaporation of the refrigerant cools the mixture and promotes the nucleation of ice crystals within the brine. The ice slurry collects at the bottom of the vessel and is pumped out. The refrigerant vapour is evacuated from the top of the vessel for compression, condensing and recycling.

In all three processes, the salinity of the brine has an important influence on the manufacture of ice slurry. In general, a higher concentration of dissolved salts will give a smaller production of ice crystals but at greater purity. The freezing temperature will also be lower, resulting in a reduced coefficient of performance for the unit. For mine cooling systems, a salt concentration between 5 and 15 percent would appear to be appropriate. However, the optimum salinity for any given installation should be determined from tests on the actual water to be used.

Future employment of ice slurries in mine cooling systems may be combined with energy recovery devices. The duties of direct-contact heat exchangers are considerably enhanced when the sprays are supplied with ice slurry of up to 20 percent ice (Gebler, 1988). Droplets in the spray chamber remain at 0°C until the ice crystals are melted. Tests have indicated that with an ice fraction of only 40 percent, pipeline pressure drops are about three times those for water (Shone, 1988).

Transportation of ice

The mass of ice required for a mine cooling system is such that it must be supplied continuously rather than in a batch transportation system. While slurry ice can be pumped through pipes, particulate ice must be transported from the ice-making plant to the shaft by conveying systems or by a hydraulic or pneumatic pipeline. In the latter case, the supply air temperature should be not more than 8°C in order to prevent undue agglomeration of the ice particles.

The particulate ice falls through the shaft pipeline either as dilute flow in which the particles are separated and retain their individual identity or as dense or slug flow. In the latter case, the ice particles agglomerate into discrete slugs separated by air spaces. If the ratio of ice to air is too great, then extrusion flow will develop accompanied by a greatly increased risk of pipe blockage. The factors involved in the type of flow through an ice pipeline are the ice/air ratio, the shape, temperature and size of the particles, and the type and size of pipe. Unplasticised (U) PVC has been found to be a satisfactory pipe material for ice lines. However, changes in cross section or lips at expansion joints should be avoided. At shaft stations, radii of curvature of 3 m or more allow the pipelines to be extended horizontally for several hundred metres. To cite the East Rand Mines example again, four uninsulated UPVC pipes of inside diameter 270 mm and wall thickness 22.5 mm were employed (Hemp, 1988). The longest pipe extended through a vertical distance of approximately 2900 m. Each pipe was capable of carrying 200 tonnes per hour (55.6 kg/s) allowing for considerable future expansion of the mine cooling duty.
Incorporation into the mine cooling system

Figure 18.25 gives a simplified representation of one layout that utilizes particulate ice.

Return water pumped from the mine passes through a surface pre-cooling tower and is further cooled in a water pre-chiller. This may be supplied by “coolth” from the ice-making refrigeration plant or, alternatively, be a separate water chiller package. The particulate ice falls through the shaft pipeline and is discharged into an ice/water mixing dam or a silo with a perforated base. Water from the hot water dam is also sprayed into the ice silo. The discharge from the silo enters the cold water dam at 0°C.

The bed of ice in the silo should be maintained at a thickness of 1 to 2 m. The supply of ice to each silo must be matched to the required cooling duty at that level. Position transducers can be used to monitor the ice level in each silo. The transmitted signals may be employed to control ice feed rates. It is advisable to provide sufficient ice bunkerage to handle short term fluctuations in demand. Longer
term variations can be handled by activating an appropriate number of individual sets in the ice-making plant or by compressor control on the refrigeration units (Section 18.2.3).

**Economics of ice systems**

The benefits of ice as a medium of heat transfer in mines may be listed as follows:

- greatly reduced water flow in shaft pipelines; therefore - smaller pipes and lower pumping costs
- water is available at 0°C underground rather than the 3 to 6°C common with conventional chilled water systems; therefore there is less flow and reduced pumping costs in the subsurface (secondary) circuits, and improved performance of heat exchangers.
- “coolth” is stored in ice bunkers and silos to satisfy short term variations in demand
- the system is more easily capable of being uprated for future increases in cooling load
- the subsurface system is simpler having reduced or eliminated the need for turbines
- the quality of the water is improved by ice-making.

The major disadvantage of the system is the considerable increase in both capital and operating costs of the ice-making plant. As the depth of a mine increases and the heat load and associated pumping costs both rise, then, at some point, it becomes economically advantageous to convert from water chilling to an ice system. A study in South Africa showed that the crossover point occurs at a depth of approximately 3000 m (Sheer, 1984). However, this will be influenced not only by the cost of hardware but also power charges, surface climate, geothermic gradient, rock properties and heat from equipment or stored materials (in the case of subsurface repositories). It is, therefore, prudent to conduct a site-specific economic study on any proposed ice system.

**18.3.6. Summary of design process**

Although this chapter has concentrated on the hardware associated with mine air conditioning systems, it is pertinent to remind ourselves that the design of such systems cannot be separated from ventilation planning (Sections 9.1 and 16.3.5). In particular, the initial question is how much heat can, or should, be removed by ventilation and how much by cooling systems.

This leads us into the first stages of the design procedure for a mine cooling system.

1. Establish the expected heat load for the entire mine and for each identifiable district, level or zone in the mine. The techniques for doing this are discussed in Chapters 15 and 16.

2. Again, for the whole mine and each individual area, determine the airflows required to deal with dust and polluting gases, taking into account air velocity limits (Section 9.3). Ventilation network analyses should be conducted in order to establish airflow distribution and leakage patterns (Chapter 7).

3. Determine the heat removal capacities of the airflows (Section 9.3.4). If this is greater than the corresponding heat loads for individual sections of the mine, then the heat can be removed entirely by ventilation and there is no need to consider a cooling system.

4. Conduct exercises to determine the feasibility of removing the excess heat by increased airflows without exceeding velocity limits set by physiological, economic or legislative considerations.
5. Determine the heat to be removed by cooling as the difference between the heat load and the heat removal capacity of the air (Section 9.3.4). Preferably, a climatic simulation program should be used in order to conduct a detailed analysis of the positions and duties of heat exchangers and application of chilled service water (Chapter 16).

Interactive studies should also be conducted to optimize between ventilation and air conditioning (Figures 9.1 and 16.4) (Anderson, 1986). These investigations will establish the required distribution of air cooling devices. If this is limited to a few localized areas, then the feasibility of employing spot or district coolers (Section 18.3.1) should be investigated. If, however, the heat problem is more widespread, then the study should be widened to examine centralized refrigeration plant.

6. Investigate the alternative locations of refrigerating plant (Section 18.3.5.1) and the feasibility of employing energy recovery devices (Section 18.3.4).

7. By summing heat exchanger capacities and allowing for line losses, establish the duty of the refrigeration plant.

8. Estimate evaporator and condenser temperatures on the basis of the desired temperature of the cooled medium and the heat rejection facilities respectively.

9. Determine the required flowrates of chilled water and, hence, the sizes of pipelines (Section 18.3.3.1) and pumping duties.

10. Invite tenders from manufacturers of refrigeration plant and ancillary equipment, including valves, pipelines, pumps, heat exchangers, instrumentation and controls.

11. Establish capital, installation and operating costs of the cooling system.

18.4. AIR HEATING

In the colder countries of the world, the temperature of intake air entering a mine may fall well below 0°C for a major part of the year. This can result in severe problems along intake routes, and particularly in surface connecting downcast shafts, slopes or adits. First, a build-up of ice on shaft walls and fittings can cause a significant increase in resistance to airflow. Furthermore, large pieces of ice dislodged by shaft operations or melting can present a hazard to personnel working in or near the shaft. Repeated cycles of freezing and thawing occurring on a daily or seasonal basis can result in severe damage to concrete linings in shafts or other intake airways, particularly when there is a seepage of ground water through the lining.

Secondly, sub-zero temperatures can have adverse effects on the operation and maintenance of equipment. This may result in poor productivity as well as a reduction in safety.

Thirdly, personnel who are inadequately protected against a cold environment will suffer the symptoms described in Section 17.6. Here again, this will lead to diminished levels of productivity and a deterioration in safety standards.

Winter temperatures can fall below -40°C at the surfaces of mines in cold climates (Hall, 1988; Moore, 1985). In these conditions, it becomes necessary to heat the intake air. This is the situation that obtains at many Canadian mines (Hall, 1989).
The methods of air heating suitable for mine applications may be listed as follows:

- utilization of waste heat produced from plant or processes on the mine surface
- heat recovery from warm return air
- direct heating
- indirect heating
- ice stopes
- geothermal or cycled storage heating.

Each of these techniques is examined in the following subsections. To choose the method most applicable for any given facility, it is necessary to consider technical feasibility, reliability, flexibility and economics. Because of the large airflows that are usually involved, the costs of heating may far exceed those of producing the ventilation (Hall, 1985). In very cold climates, it is often the aim simply to increase the intake air temperature to about 0°C and to allow autocompression, geothermal and other sources of heat to produce more comfortable temperatures in the main working areas.

In order to minimize operating costs, air heating systems might be employed only during main working shifts and when the surface temperature drops below a preset value. Furthermore, two or more of the systems listed above may be combined to minimize energy demand or to provide additional capacity for periods of abnormally low temperature.

There is, however, one situation in which air heating should be avoided or used with great caution. This occurs when the mine workings are in permafrost. Allowing air temperatures to exceed 0°C may cause partial melting of the permafrost and give rise to problems of ground control in the mine. Indeed, at the Polaris Mine, 1400 km from the North Pole, refrigeration techniques have been employed during the short summer in order to maintain the intake temperatures below the melting point of permafrost (Van der Walt, 1984).

### 18.4.1. Utilization of waste heat

The first matter to investigate for any proposed air heating project is whether heat is available from other plant or processes on the surface of the mine. Secondly, the costs and technical feasibility of using that thermal energy to raise the temperature of intake air should be studied. Common sources of waste heat are compressor stations, generators and mineral processing operations (de Ruiter, 1989). If the waste heat is in the form of hot water or steam, then it may be pumped directly through insulated pipes to heat exchanger coils located within all, or part, of the intake airstream. Ice accumulations on the heat exchanger can be avoided by using a single array of coils only.

In water systems, an automatic procedure should be incorporated in order to empty the coils and associated pipework during a shut-down period or if the water temperature approaches freezing point. A further safeguard against frost damage is to employ a glycol-water mixture. A 50 percent glycol mixture lowers the freezing point to -40°C.

If the available heat source provides relatively low temperature thermal energy, then it may be necessary to employ a heat pump (Section 18.2). Heat pumps may also allow the utilization of natural heat sources such as a large lake or the ocean, provided that these remain unfrozen through the winter months.

### 18.4.2. Heat recovery from exhaust air

This method may be considered if the downcast and upcast shafts are close to each other and there is a significant difference between the intake and return air temperatures. Figure 18.26 shows that during cold periods, the warm air returning from the mine is diverted through a direct contact spray chamber, transferring useful heat to the descending water droplets. Water from the sump is pumped
through a heat exchanger located within the intake airstream. Again, it is advisable to employ a single array of coils to minimize ice build-up, and to allow for automatic dumping of water from the heat exchanger and pipework when the flow ceases.

The spray chamber may be replaced by a tube coil heat exchanger. However, this reduces the overall heat transfer efficiency of the system and also introduces the dangers of ice accumulation and corrosion of the tubes, particularly if the return air is humid.

Heat exchange from upcast to downcast airflows is seldom practicable by itself. However, it may be employed in combination with an external source of heat as shown on Figure 18.26. If that source is waste heat from other plant, then it may be kept in continuous operation. However, if the external source of heat is gas, oil or electricity, then it may be used only during working shifts and when the surface air temperature falls below a set value.

A more direct means of recovering some of the heat from warm return air is by controlled partial recirculation (Hall, 1988, 1989). This technique gives one hundred percent efficiency of heat transfer from return to intake for that fraction of the air which is recirculated. This is limited by the rate at which gaseous and particulate pollutants are added to the mine air. The system should be capable of reverting to normal through-flow ventilation should monitors indicate a trend towards unacceptable concentrations of pollutants or following blasting operations. The volume of air that is recirculated may be controlled automatically by the pollutant concentrations in the return air or set at a fixed
fraction of the return airflow. The recirculating cross-cut should be located as close to the surface connection as practicable in order to take advantage of the pollutant dilution arising from air leakage.

If the level of contamination by airborne pollutants is sufficiently low to allow a reduction in the through-flow of fresh air, then the costs of heating the intake air will also be reduced. However, any proposed system of controlled partial recirculation must incorporate all of the safeguards and precautions discussed in Section 4.5.

18.4.3. Direct heating

The employment of electrically heated elements to raise the temperature of mine air is limited by high operating costs. It is practical only for very light duties or where electrical power is particularly cheap.

Natural gas, propane or other light hydrocarbon fuels may be injected through nozzles and burned in flame jets within the intake airstream. As no intermediate heat exchangers are involved, this is the most efficient way of using the fuel. The products of combustion, carbon dioxide and water vapour, are added to the airstream in relatively small amounts. However, the gas nozzles should be well maintained to prevent the formation of carbon monoxide through incomplete combustion. A prudent precaution is to locate a gas monitor downstream from the jets with alarms and automatic fuel cut-off at a predetermined concentration of carbon monoxide.

The rate of fuel consumption for any given heating duty may be calculated from the calorific value of the fuel. At the temperatures to which the products of combustion are cooled in mine air, the majority of water vapour will condense. Hence, it is the gross calorific value that should be used. Table 18.4 gives the gross calorific values of several gases. For natural gas and other mixtures, an approximate value may be obtained by summing the calorific values of the constituents weighted according to percentage composition.

<table>
<thead>
<tr>
<th>Gas</th>
<th>Gross calorific value MJ/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>carbon monoxide</td>
<td>11.8</td>
</tr>
<tr>
<td>hydrogen</td>
<td>11.9</td>
</tr>
<tr>
<td>methane</td>
<td>37.1</td>
</tr>
<tr>
<td>ethylene</td>
<td>58.1</td>
</tr>
<tr>
<td>ethane</td>
<td>64.5</td>
</tr>
<tr>
<td>propylene</td>
<td>85.7</td>
</tr>
<tr>
<td>propane</td>
<td>93.9</td>
</tr>
<tr>
<td>butane</td>
<td>121.8</td>
</tr>
<tr>
<td>natural gas</td>
<td>(depends upon composition)</td>
</tr>
<tr>
<td>(depends upon composition)</td>
<td>26 to 56</td>
</tr>
</tbody>
</table>

Table 18.4. Gross calorific values of gases
based on a temperature and pressure of 15°C and 101.3 kPa respectively.
For petroleum fuel oils, gross calorific values vary from 42 to 47 MJ/kg.

Example

An airflow of 100 m³/s and density of 1.4 kg/m³ is to be heated from -20°C to 2°C by propane jets.

(a) Determine the rate of fuel consumption in m³ per hour.

(b) If the air is supplied at 21 percent oxygen and 0.03 percent carbon dioxide, determine the corresponding concentrations downstream from the jets.
Solution

(a) Mass flow of air \( m_a = Q \rho \)
\[
= 100 \times 1.4 = 140 \text{ kg/s}
\]

Heat required, \( q_a = m_a \times C_p \times \Delta t \)
\[
= 140 \times 1005 \times (2 - (-20))
\]
\[
= 3.095 \times 10^6 \text{ W}
\]
\[
or \quad 3.095 \text{ MW}.
\]

[To be precise, we should take the presence of water vapour into account. However, the effect is very small at these low temperatures and may be neglected.]

The gross calorific value of propane is selected from Table 18.4 as 93.9 MJ/m\(^3\). Hence, heat produced at the jets

\[
q_p = 93.9 \times \text{propane flowrate} \quad \frac{\text{MJ}}{\text{m}^3} \text{ or } \text{MW}
\]

As \( q_p = q_a = 3.095 \text{ MW}, \) the required propane flowrate \( = \frac{3.095}{93.9} = 0.0330 \text{ m}^3/\text{s} \)

\[
or \quad 0.0330 \times 3600 = 118.7 \text{ m}^3 \text{ per hour at standard temperature and pressure.}
\]

(b) The chemical balance for burning propane is

\[
\text{C}_3\text{H}_8 + 5\text{O}_2 \rightarrow 3\text{CO}_2 + 4\text{H}_2\text{O}
\]

\[
1 \text{ m}^3 \text{ propane} + 5 \text{ m}^3 \text{ oxygen} \rightarrow 3 \text{ m}^3 \text{ carbon} + 4 \text{ m}^3 \text{ water}
\]

\text{dioxide vapour}

Therefore 0.0330 m\(^3\)/s of propane will consume

\[
5 \times 0.0330 = 0.165 \text{ m}^3/\text{s} \text{ oxygen to produce}
\]

\[
3 \times 0.0330 = 0.099 \text{ m}^3/\text{s} \text{ carbon dioxide}
\]

At entry, the airflow includes

\[
100 \times 0.21 = 21 \text{ m}^3/\text{s} \text{ oxygen and} \quad 100 \times 0.0003 = 0.03 \text{ m}^3/\text{s} \text{ carbon dioxide.}
\]

Downstream from the jets,

\[
\text{oxygen flowrate} = 21 - 0.165 = 20.835 \text{ m}^3/\text{s} \text{ and}
\]

\[
\text{carbon dioxide flowrate} = 0.03 + 0.099 = 0.129 \text{ m}^3/\text{s}
\]

As the airflow is 100 m\(^3\)/s these same values give close approximations to the concentrations of the gases downstream from the jets.

18.4.4. Indirect heating

In order to prevent products of combustion from entering the mine ventilation system, fuels may be used in a separate burner to heat a glycol-water mixture. This is then recirculated through an indirect heat exchanger located within the intake air. Again, a single array of coils will minimize the risk of ice accumulation on the tubes. This technique of indirect or offline heating inevitably gives a lower thermal efficiency than open flame jets within the intake air. However, the flowrate of the glycol mixture provides an added measure of control and flexibility.
18.4.5. Ice stopes

Figure 18.27 illustrates a metal mining technique that allows air to be heated from sub-zero temperatures to approaching 0°C at very low cost. Water is sprayed into the top of an abandoned open stope. Cold air coming directly from the downcast shaft or slope enters near the top and circulates through the stope, freezing the water droplets, before exiting at the other end.

The air is heated by three mechanisms:

(a) directly from the liquid water droplets,

(b) by latent heat of fusion (333.5 kJ/kg) as the water freezes, and

(c) by strata heat from the wall rock.

The ice particles fall and form an ice bed that gradually accumulates in the stope through the winter months.

In summer, warm intake air may be cooled by circulating it through the same stope and melting the ice. A drain level at the base of the stope carries water to a main sump. However, the air cooling may be enhanced by circulating the liquid melt back through the sprays.

The ice stope technique has been used very successfully at the Stobie Mine of Inco Ltd. in Canada (Stachulak, 1989). In this mine an airflow of 307 m³/s is heated in two ice stopes at a cost estimated at about one-ninth that for equivalent gas heating. In that installation, water is supplied to the nozzles
at a pressure of between 550 and 760 kPa. Compressed air is added to the line shortly before the nozzles. This provides a finely divided spray and prevents icing up of the jets. The compressed air is also used to clear water from the pipes when the system is to be shut down.

The disadvantages of the ice stope technique are that it requires the availability of large open excavations and also necessitates an increased pumping capacity for the mine drainage system.

Example
Water at 7°C is sprayed at a rate of 30 litres per second into an ice stope. Air enters the stope at -22 °C and is required to be heated to -2 °C. Calculate the maximum air flowrate through the stope that can attain this 20 °C increase in air temperature.

Solution
Mass flowrate of water, \( m_w = 30 \text{ kg/s} \)

Heat released by water in cooling from 7°C to 0°C
\[ q_w = m_w \ C_w \Delta t_w \]
\[ = \frac{30 \times 4187 \times 7}{1000} = 879 \text{ kW} \]

Heat of fusion released by ice formation (latent heat of fusion, \( L_i = 333.5 \text{ kJ/kg} \))
\[ q_i = L_i m_w = 333.5 \times 30 = 10 005 \text{ kW} \]

Total heat transferred to air
\[ q_a = q_w + q_i = 879 + 10 005 = 10 884 \text{ kW} \]
(This illustrates the dominant effect of the latent heat of ice formation.)

However, this is also equal to \( m_a \ C_p \Delta t_{air} \)
giving \( m_a = \frac{10 884 \times 1000}{1005 \times 20} = 541 \text{ kg/s} \)

If the air density were 1.3 kg/m³, this would give an air volume flowrate of 541/1.3 = 417 m³/s.

18.4.6. Geothermal and cycled storage heat

Another low cost method of controlling the temperature and humidity of intake air is to course it through one or more sets of old workings or fragmented strata before admitting it to current work areas. This takes advantage of a combination of natural geothermal energy and the "thermal flywheel" effect of heat storage within the envelope of rock surrounding a mine opening (Section 15.2.2).

The flow of geothermal heat into a mine airway is greatest at the moment of excavation and reduces with time until near equilibrium is attained when temperatures in the surrounding envelope of rock no longer change significantly with respect to time (Figure 15.3). If, however, the inlet air is cycled between hot and cold, then the surrounding rock will act as a storage heater, absorbing heat during the hot periods and emitting it during the cold periods. This cyclic behaviour is superimposed upon the longer term flux of geothermal heat.
Let us assume a situation in which the virgin rock temperature (VRT) at a given level is higher than the mean annual dry bulb temperature of air entering that level from the downcast shaft or slope. In order to moderate the extremes of winter temperature, the air is passed through a set of old workings that we may refer to as the control district. Figure 18.28 illustrates the seasonal variation of air temperature at inlet and exit of a control district, and ignoring the shorter-term diurnal changes. The moderating effect of the control district is shown clearly. The inlet peaks and troughs of summer and winter temperatures are damped very considerably by the time the air reaches the exit.

However, the figure also shows a longer term trend over which the average exit temperature falls until a repeated cyclic variation is attained. If the winter temperatures of the air leaving the control district are acceptable in this condition of dynamic equilibrium, then that district can be used indefinitely to moderate the extremes of seasonal variations. If, however, after a few years the winter exit temperatures fall below tolerable levels, then the intake air should be diverted through a younger control district.

Control districts that are no longer effective may be allowed to regenerate by sealing them off. The surrounding rock temperature will gradually tend towards the VRT. After a regeneration period which may be several years, the panel can again be opened up as a control district. If several old districts are available, then some of them may be used as control districts at any one time while others are regenerating (Moore, 1985).
The actual form of Figure 18.28 for any given mine depends upon the extremes of surface climate, the rate of airflow through the control district and the thermal properties of the surrounding strata. Simulation programs (Chapter 16) are most helpful in predicting the cyclic behaviour of the air temperatures (Moore, 1985; MVS, 1990). Manual methods of estimating the effects of varying inlet temperatures on strata heat flow have also been developed (Hemp, 1982).

Although the use of a control district can yield a useful degree of air conditioning, it does have some disadvantages and limitations. First, the passage of air through old workings requires additional fan energy. This can be minimized by arranging for the air to move fairly slowly along multiple flow paths, requiring only a small applied pressure differential. Simulation tests coupled with site-specific measurements allow the optimum layout to be determined.

Secondly, old workings are liable to changes in resistance due to strata closure and falls of roof. Hence, in order to ensure continuity of ventilation, control districts should be capable of being bypassed at short notice. Neither should they be relied upon as emergency escapeways.

Thirdly, the conditioning of intake airflow through old workings is untenable where there is a danger of spontaneous combustion or significant emissions of strata gases.

In addition to moderating temperatures, control districts can also serve a useful purpose where variations in intake humidity cause roof control problems. This can occur in deposits of salt or some shales. The hygroscopic nature of these rocks results in absorption of water vapour and a loss of mechanical strength (Robinson et al., 1981).

Where caving techniques in a metal mine have left a fragmented and permeable rock mass that extends to the surface, then winter air may be heated and summer air cooled by drawing intake ventilation through the broken rock. This technique can combine the phenomena of storage heating and the latent heat of ice formation. One of the best known examples of this essentially free form of air conditioning is at the Creighton Mine in the Sudbury Basin of Canada (Stachulak, 1989). Intake air is drawn through a bed of broken rock that is approximately 244 m by 183 m in plan and 137 m high. Precipitation and seepage results in ice formation during the winter and melting in the summer. The air progresses through old drawpoints, slusher drifts and ore passes where it is regulated according to its temperature. This technique results in intake air of near constant temperature reaching the deeper current workings throughout the year.

References


