Key engineering considerations in the specification and selection of mine refrigeration plants

By D J Brake[1], Fellow

INTRODUCTION

As mines become deeper, the problems of heat in the workplace become more acute. An important response from the industry has been the widespread adoption of microclimate cooling such as air-conditioned cabins. However, not all work in mines can be completed from inside an air-conditioned cabin, and in any event, it is still important to keep the ambient air in the mine outside of any microclimate to a safe temperature. An important method for bringing workplace temperatures under control is the appropriate use of refrigeration. The application of refrigeration in mechanised mines has been recently described (Brake, 2001; Brake and Fulker, 2000). This paper provides an overview of some of the important engineering considerations in the specification and selection of mine refrigeration plants. The purpose is to provide a basic understanding of refrigeration principles, practices, terminology, hazards and opportunities for mining engineers or other personnel who may need to be involved in the introduction of refrigeration onto a mine site.

HISTORY OF REFRIGERATION

Mechanical (non-electrical) refrigeration systems were originally introduced in the mid 1880s and within a few years were being used to preserve foods and documents (Howakowski and Busby, 2001). Willis Carrier patented his 'apparatus for treating air' in 1906. In the same year, the term 'air conditioning' was coined by engineer Stuart Cramer. The early applications included printing factories and textile mills, both of which required humidity control. The first 'human comfort' cooling systems were introduced by Alfred Wolff in the early 1900s in the New York area, with the New York Stock Exchange building being one of the first clients. This was soon followed by movie theatres. Carrier patented the centrifugal compressor in 1921, with theatres and factories continuing to be the largest market during the remainder of the 1920s.

Perhaps the earliest recorded use of cooling in mines was in Nevada in the 1860s, when blocks of naturally-produced ice were transported underground in ore cars to cool the miners. Vapour compression refrigeration was first introduced into mining at the Morro Velho mine in Brazil in 1921 (about 2400 m depth) and as an experimental system in a British coal mine in 1923 (Hancock, 1926). The first European plant was at the Radbod mine in 1924. It was first introduced into the South African and Kolar (India) goldfields in the 1930s. Refrigeration application in US mines includes the gold mine at Homestake, copper mines at Butte and Superior, uranium mines in New Mexico and silver-lead-zinc mines in the Coeur d'Alene district (Hartman *et al*, 1997). In total, over 15 countries have introduced refrigeration into one or more of their mines. However, it was only in the 1960s that refrigeration in mines started its exponential growth, led largely by the needs of the ever deeper and hotter metal mines in South Africa. Installed capacity in South Africa is currently several hundred MW(R), and mine refrigeration had reached 256 MW(R) in Germany alone by 1994 (almost all in coal mines). There are perhaps a few hundred MW(R) installed in the rest of the world.

In Australia, the earliest application of mine refrigeration was at Mount Isa, which installed a 3 MW(R) system in the early 1960s and sent cold water underground via a 'U-tube' piping system and through high-pressure (11 MPa) heat exchanger coils to provide cooling for the deep crushing facilities. Since then, surface chilled water plants have been installed at Mount Isa (feeding underground air coolers via surface and underground hot and cold water dams and a Pelton wheel), surface bulk air cooling at Mount Isa, Broken Hill and Telfer, and underground spot coolers at Olympic Dam and Telfer. Total installed capacity in Australia at present is about 50 MW(R), with most of this serving the deep Enterprise mine at Mount Isa.

One of the reasons that installed capacity in Australia is relatively low despite the size of Australia's mining industry and its hot climate, is that allowable workplace temperature limits in Australia are much higher than in the UK or Europe. Frequently, there are no upper temperature limits in Australian metal mines, whereas UK and Europe are generally limited to 28°C DB or 28° ET (whichever is the higher). A corollary to this is that great care must be taken in transposing heat stress standards and practices from the Northern Hemisphere, with its temperate climate and unacclimatised workforce, into Australian conditions.

Whilst the major reason to install mine refrigeration is to provide acceptable environmental conditions for workers and equipment, refrigeration (humidity control) has also been introduced into potash, salt and coal mines purely to help protect the 'product' (potash, salt) from deteriorating, or to avoid slaking of the roof strata in coal mines (Hartman *et al*, 1997).

A variety of exotic cooling systems have been used in underground mines; these have usually been possible where mines have been located in very cold climates, so that access to cold surface water (eg lakes) to feed underground cooling towers directly without refrigeration, or to cold surface air temperatures to create underground 'ice' stopes during winter or to obtain 'free' cooling in surface pre-cooling towers, has been practical. However, such solutions are rare and highly unlikely to be encountered in the much-hotter Australian climate.

REFRIGERATION PLANT OVERALL DESIGN

The operation of a standard vapour-compression[i] refrigeration machine is described in many text books (Stoeker and Jones, 1982). The physical layout of the main components is shown in Figure 1. The basic process is best seen on a pressure-enthalpy diagram (Figures 2). Here, a fluid[ii] (the 'refrigerant') is taken as a cool, low-pressure gas (point A on Figure 2) and compressed (Figure 3) to a high pressure (point B on Figure 2). This results in it becoming very hot. This very hot gas is then condensed to its liquid form (point C on Figure 2) in a type of heat exchanger called a *condenser* (Figure 4), with the heat of condensation being rejected (usually via a *cooling tower*, Figure 5) to the atmosphere. The high-pressure liquid drains into a *liquid receiver*

---

1. Principal, Mine Ventilation Australia. 12 Flinders Parade, Sandgate QLD 4017. E-mail: rick.brake@mvaust.com.au

i Virtually all large industrial refrigeration systems currently use the vapour-compression cycle. However, with continuing advances in refrigeration technology, it is possible that where a mine has access to large amounts of high-grade waste heat, an absorption refrigeration plant could be technically and economically viable. Such an system has recently been installed at a Polish underground coal mine. This mine uses methane drained from the coal seams to power a gas engine which generates electric power, which is then sold to a power utility. The waste heat from the gas motor is then used to 'power' a two-stage absorption refrigeration system generating cold water which is then sent underground and distributed to air coolers prior to being returned to the surface via a 3-chamber pipe pump.

ii 'Fluid' includes both the liquid and vapour (gas) phases of a substance.

(Figure 4) then allowed to pass through an *expansion valve* (Figure 6). The resulting drop in pressure (point D on Figure 2) causes some of the liquid to 'flash off' (evaporate) with the latent heat being drawn from the remaining liquid, causing a significant drop in temperature in the liquid. The now cold, low-pressure liquid is stored in a large pressure vessel called the *surge drum*[iii] (Figure 7). It then passes through another heat exchanger called the *evaporator* (Figure 7), in which the refrigerant boils (evaporates), with the latent heat required to achieve this coming from the air or water that is chilled in the process (the 'load'). The evaporated, cold, low-pressure refrigerant gas then re-enters the *compressor* (point A on Figure 2) to restart the cycle.

In a mine plant, the ultimate heat 'load' is usually hot fresh air. The evaporator can cool the air directly (but this is not very efficient due to the poor heat transfer coefficient between the evaporator and the air, and also significantly increases the hazard from leakage of refrigerant directly into the intake air) or can be used to chill water which is then used to chill the air in a more efficient air-water, direct-contact[iv] air cooler.[v] In this case, the water is called a 'secondary' refrigerant, as it is the intermediary mechanism to transfer the 'cooling' from the liquid refrigerant (the 'primary' refrigerant) to the air.[vi]

Likewise, the condenser can reject heat directly to the ambient air in a dry process, but this is usually an inefficient process. A variation on this is where air is passed through the condenser and water is sprayed on it at the same time. This hybrid process is called an evaporative condenser and can achieve high efficiencies. A further alternative is to use a secondary refrigerant (water) to remove the heat quickly from the condenser and then reject this into the atmosphere using a cooling tower.

Thus the vapour-compression refrigeration process is fundamentally one in which a pump (the compressor) is used to 'pump' heat from water in the evaporator (producing cold water) and reject this heat via the condenser (and usually a cooling tower) to the atmosphere. The condensers must therefore absorb not only the heat picked up by the refrigerant in the evaporator, but also the work of compression.

iii There are several alternate names used for the 'surge drum'. These include the *accumulator*, so-called because it allows the refrigerant vapour produced in the evaporator to 'accumulate' before being re-compressed by the compressor. Without such a device, the compressor could either be flooded or starved of vapour refrigerant as the load on the plant changes. The name '*surge drum*' follows a similar logic. Another name in less common usage is the '*separator*'. This term arises from the fact that the 'surge drum' also has an important role in allowing the liquid refrigerant to separate from the gaseous refrigerant, which must occur in all flooded evaporators, to ensure only vapour is sent to the compressor. However, this presents problems in distinguishing the vessel from the *oil separator*. Finally, the device is sometimes called a '*low pressure receiver*', as it receives the low-pressure liquid refrigerant after passing from the expansion valve, and before it enters the evaporator. However, this presents problems in distinguishing it from another important vessel called the *liquid receiver*.

iv Direct contact (sometimes called a *wet* air-cooler) means the air and water come into direct contact with one another; this can only happen in some form of cooling tower. In indirect contact (sometimes called a *dry* air-cooler), the two fluids are separated, eg by the steel plate or pipe in a heat exchanger and heat transfer must occur through the interspersed plate or surface.

v To confuse matters, bulk air coolers are merely a type of cooling tower, so that the nature of both the tower and the cooling duty in it (whether it the cooling of *air* that is the prime purpose, or the cooling of *water*) can only be obtained from the context.

vi A secondary refrigerant is a liquid used for transmission of heat without any change in its liquid state. The primary refrigerant in a vapour compression system always has two changes of state, liquid to vapour, and vapour to liquid.

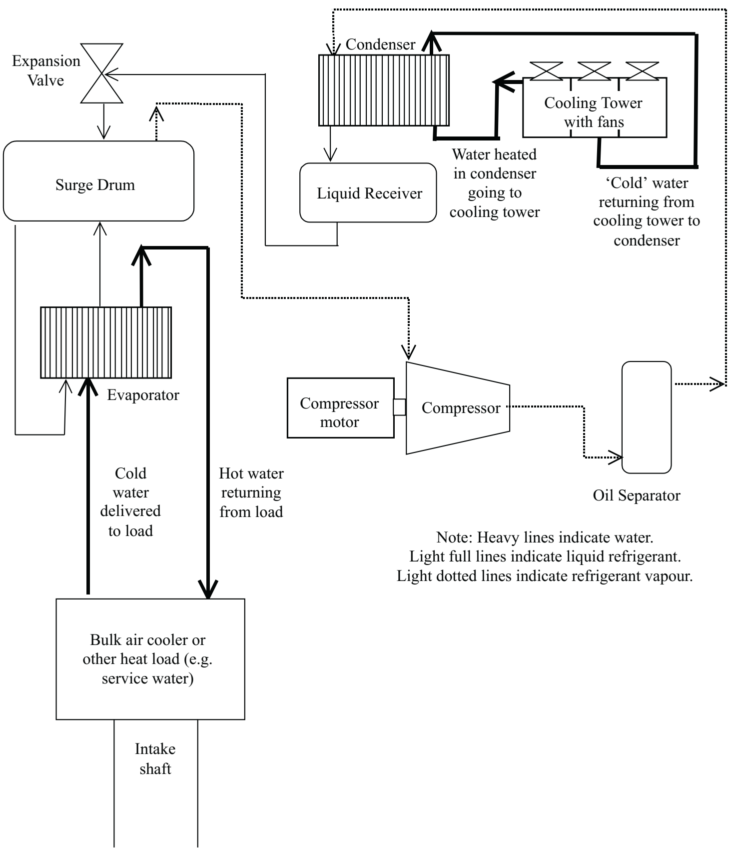

FIG 1 - Schematic of basic vapour-compression refrigeration machine.

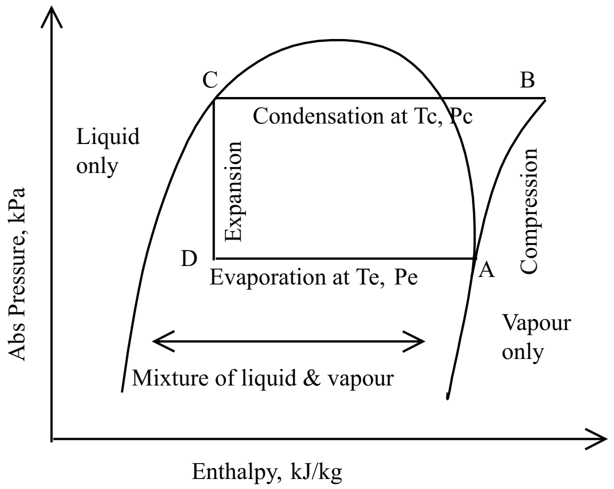

FIG 2 - Pressure-enthalpy diagram for typical single stage, vapour-compression, refrigeration cycle.

FIG 3 - Compressor (left), HV compressor motor (centre) and oil separator (right) (photo courtesy of MIM).

It can be shown that each kW of compressor power usually produces several kW of cooling (McPherson, 1993). The ratio of cooling effect (refrigeration, in kW(R)) to total electrical power input (in kW(E)) is called the coefficient of performance (COP) of the plant, and typically varies between about two and about six. The Ideal (or *Carnot*) coefficient of performance of the system (the theoretical maximum) is given by:

$$\text{Carnot COP} = T_e/(T_c - T_e) \qquad (1)$$

where T_e and T_c are the evaporating and condensing temperatures in Kelvin.

Carnot COPs cannot be achieved in a 'real' plant; however, they give a useful yardstick against which actual plant performances can be measured. In addition, a change in plant operating conditions that affects the Carnot COP will have a similar magnitude effect on actual plant COP.[vii]

Consider a plant in which the refrigerant is condensing at 40°C (313 K) and evaporating at 0°C (273 K):

- The maximum (Carnot) COP is 273/(313-273) = 6.825. This means that in a perfect, reversible and frictionless process, 6.825 kW of cooling is produced for each kW of compressor shaft power.
- If the condensing temperature increases by five degrees, without affecting the evaporating temperature, the COP falls to 273/(318-273) = 6.07.
- If the evaporating temperature decreases by five degrees, without affecting the original condensing temperature, the COP is 268/(313-268) = 5.96.

FIG 4 - Condenser PHE above (obscured) with liquid receiver below (photo courtesy of MIM).

A number of observations should now be made:

- As the condensing temperature increases, the COP decreases. Hence low condensing temperatures are desirable for high thermodynamic efficiency and low power costs. The lowest achievable condensing temperatures are as follows. If the heat is being rejected directly to the air in an air-cooled condenser (a dry heat transfer), it is the *dry bulb* temperature of the air. If a secondary coolant (water) is used to pick up the heat in the condenser and then reject it to atmosphere by evaporation, then it is the *wet bulb* temperature of the air. In an air-cooled heat exchanger, the only heat transfer is sensible heat, whereas when an evaporative process is used, the heat transfer is latent heat.[viii]

---

vii Note that the COP of a plant does not remain constant during the year. This is because any plant, while operating, has a certain electrical load that remains relatively constant. This includes water pumps and CCT fans. Therefore, when the refrigeration output is low but the plant is still 'on', the actual COP may also be low, depending on how large a proportion of the total electrical power requirement of the plant in incurred in the base load.

viii This is a function of the definition of the wet bulb temperature. Even with a perfect evaporation process in some sort of water-air heat exchange, the water temperature can never drop *below* the air WB temperature (assuming the water temperature was originally *above* the WB temperature, as in some sort of cooling tower), or can never get *above* the WB temperature (assuming the water temperature was originally *below* the WB temperature, as in some sort of bulk air cooler). The actual driving force for the heat transfer in an evaporation process is the difference between the vapour pressure of saturated air at the inlet water temperature and the vapour pressure of saturated air at the inlet wet bulb temperature.

FIG 5 - Condenser cooling tower (photo courtesy of MIM).

FIG 6 - Expansion valves (photo courtesy of MIM).

- As the evaporating temperature decreases, the COP decreases. Conversely high evaporating temperatures result in high COPs. In the limiting case, where $T_e=T_c$, the COP tends to infinity.

It is clear that where there is a large difference between evaporating and condensing temperatures, the maximum COP that is achievable will be low. Hence in refrigeration plants that evaporate at very low temperatures (eg most frozen food manufacturing plants, or in liquefying propane gas, both of which operate at –40 degrees or lower), multiple-stage refrigerant compression is frequently employed. The COP of each stage can then be higher, which usually results in overall plant COPs that are also higher. However, in mine refrigeration plants, the coldest evaporating temperatures are usually at most a few degrees below zero (eg to produce chilled water at +1°C); therefore single-stage refrigeration plants have acceptable thermodynamic efficiency.

In practice, even single stage mine refrigeration plants[ix] typically use at least two compressors to produce the required load. This is to provide more flexibility in operation and to provide at least partial capacity in the event of breakdown of one of the compressors. If each compressor has its own evaporator and these are run in series, not only can the above objectives be achieved, but also higher COPs can be attained. This means one compressor, sometimes designated the 'lead[x] compressor' chills the load (typically water) partway to its final required temperature, with the 'lag' compressor then taking the partially chilled water and chilling it to its final temperature.[xi] However, series operation is not always possible and sometimes there are greater benefits in operating the compressors in parallel.

'Fouling' is a process (discussed later) in which a heat exchanger becomes dirty and the heat transfer mechanism less efficient. Fouling, and particularly the necessary prudential measures to avoid it, have important impacts on plant design.

Further guidance in the engineering criteria and safety requirements relevant to refrigeration plants is provided in AS1677 'Refrigeration systems'.

REFRIGERANT SELECTION

The ideal refrigerant has the following properties (Stoeker and Jones, 1982).

- operates at relatively low pressures. This allows use of relatively lightweight vessels and pipes.
- suitable boiling and condensing temperatures.
- operates with both T_e and T_c above atmospheric pressures.[xii] Hence any leakage in the plant is *into* the atmosphere. This is usually less of a problem than leakage *from* the atmosphere into the circuit, which allows ingress of moisture and air and requires an air purger to be installed.
- low pressure ratio (ratio of discharge to suction pressures on the compressor). This allows for less expensive and less elaborate compressors.
- high COP. The COP of the refrigerant is determined by the ratio of the refrigerating effect (the difference in the enthalpy of the discharge and suction gases) and the work of compression (the compressor shaft input power).
- low flow rate of suction vapour per kW of refrigeration. This allows for smaller compressors or lower speed operation.

---

ix A refrigeration *plant* consists of the entire plant, including refrigeration machine, cooling tower and bulk air cooler (if required). A refrigeration *machine* consists of one evaporator and the other mechanical items required to sustain its refrigeration process (at least one compressor, condenser, liquid receiver, etc), but excludes the cooling tower and bulk air cooler. Whilst a *compressor* is the 'heart' of the refrigeration machine, it is no more the entire machine than the engine of a car is the entire car.

x There is no consistent use of the term 'lead' and 'lag' in refrigeration. Sometimes 'lead' can refer to the machine that first sees the 'hot water' returning from the load. At other times, 'lead' can refer to the machine that actually modulates to meet the water-off temperature.

xi Note that in multi-stage compression, the refrigerant *gas* goes through two or more stages of compression between T_e and T_c. With series chilling, the *water* goes through two or more stages of chilling, but the refrigerant gas only goes through one stage of compression between T_e and T_c.

xii Note that in certain applications where it is critical that the refrigerant does not leak into the atmosphere, a negative pressure system is required.

FIG 7 - Evaporator PHE with surge drum on top (photo courtesy of MIM).

- high latent heat of vaporisation, high thermal conductivity and low viscosity.
- low flammability.
- low toxicity.
- low miscibility and no chemical reaction with the compressor oil.
- compatibility with the remainder of the plant components, especially gaskets.
- availability/ozone depletion. Some refrigerants are scheduled for phase-out. Under the Montreal protocol, production of CFC refrigerants such as R11 and R12 ceased in 1996 and usage of HCFC refrigerants such as R22 is frozen at 1996 levels (ASHRAE, 1997) and is scheduled for complete phase out over the next 20 years. Production and usage of HFC refrigerants such as R32 and R134A are not regulated by the Montreal protocol, although there is now pressure in many European countries to phase-out HFC refrigerants also.

In practice, the choice of refrigerants for mine applications often reduces to two: ammonia (R717) and the HCFC refrigerant R22. Ammonia, which was used from the inception of mechanical refrigeration machines until fluorinated hydrocarbons were developed in 1928, has a slightly higher COP than R22 and is much cheaper, and is therefore often the refrigerant of choice. It is also the refrigerant of choice for most other industrial processes such as breweries, milk factories or abattoirs.

In addition, its high thermodynamic efficiency (high COP), small approach temperatures, low refrigerant circulation rate, high heat transfer coefficients, ease of leak detection, low refrigerant charge, low cost and 'ozone-friendly' nature all contribute to its choice. In particular, due to its high COP and heat transfer, a plant running on ammonia needs a much smaller evaporator to achieve water temperatures of 0.5°C (a common mine refrigeration requirement in South Africa), compared to plants running on other refrigerants. However, ammonia is toxic and a loss of containment must be carefully considered at the design stage (see below). Ammonia is not a sensible choice (and is frequently prohibited as a refrigerant) in residential or commercial buildings. Obviously, ammonia should *never* be selected for underground refrigeration plants.

Ammonia is not suited to high condensing temperatures. For any given temperature, the saturated vapour pressure of ammonia is substantially higher than for (say) R-134A. Therefore, for an efficient plant, ammonia needs to be coupled with an effective and low-temperature condensing system.

For safety reasons, it is desirable to operate a plant with the minimum amount ('charge') of refrigerant necessary to meet full compressor capacity.

COMPRESSOR DESIGN

Refrigeration plants vary greatly in size and application. The type of compressor best suited to each plant can therefore vary, with each type having its applicable niche.

In mines, *reciprocating* compressors are usually not considered because they are only available to a maximum load of about 800 kW(R). *Vane* compressors are designed for even smaller plants. The two remaining choices are *centrifugal* compressors and *screw* compressors and these two types constitute virtually the entire market for surface mine refrigeration plants around the world.

Centrifugal compressors

A centrifugal compressor is a dynamic turbo-machine. It operates on the same principles as a centrifugal pump or fan. This means it pressurises the fluid (in this case a gas) by inducing high velocities in the gas as the impellor rotates, and then converts a portion of this velocity to static pressure in the volute. It is *not* a positive displacement machine.

Centrifugal compressors have very high efficiencies at their design point. However, because they are not positive displacement devices, their efficiency drops off substantially as they move off design, and especially as they drop below about 40 per cent of the design volume flow rate (Figure 8). Centrifugal compressors are therefore ideally suited to applications where the load is relatively constant (eg underground refrigeration plants in very deep South African mines where the 'load' does not vary substantially between summer and winter). However, in most Australian applications, where the load would vary from 100 per cent in peak summer to nil in mid winter, centrifugal compressors would operate over a substantial portion of the year at relatively low efficiencies and hence result in high operating (power) costs. Moreover, varying the output of a centrifugal compressor must be achieved either by using variable inlet vanes or variable speed drives. Either option is expensive compared to the slide valve arrangement that is used to achieve capacity control in screw compressors. Another disadvantage of the centrifugal compressor is that, like pumps and fans, it has a pronounced surge (or stall) point (Figure 8). If the condensing temperature goes up (for example due to fouling in the condensers), the condensing pressures will also go up. This could send the compressor into stall, which would damage the impeller. To avoid this, a hot gas bypass is often installed. This operates at low refrigeration load (typically at less than 40 per cent load) to throttle part of the discharge gas back to the suction line in order to provide an artificial load to the compressor. This and other safety circuits must be incorporated in centrifugal compressors to protect them from stall or from high condensing temperatures unloading or tripping the compressor. This protects the compressor but results in a device that is less efficient than a

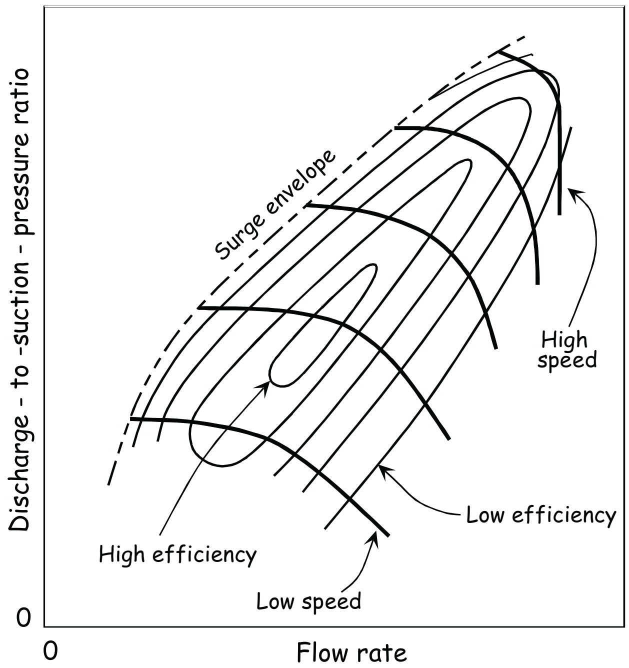

FIG 8 - Performance curve for centrifugal compressor (redrawn from Stoecker and Jones, 1982)

screw compressor and less adaptable to the rigour of the mining application.

Screw compressors

Screw compressors were originally developed in the size and configuration required for mining applications back in the 1960s. The compressor consists of a cylinder with twin, matched ‘screws’ (male and female rotors) mounted internally. The gas is taken in at one end (the suction end) and is compressed as the screws rotate, discharging the compressed gas at the other end. Because the rotors must not touch, a means was required to ensure the refrigerant gas did not flow back through the rotors during compression. It took a number of years to find oils that were compatible with the refrigerant, yet prevented the refrigerant from flowing backwards, and which could then be recovered from the hot, high pressure refrigerant gas at the discharge of the compressor. Oil lubrication is usually via high-pressure oil injection into the rotors. This allows exceptionally high pressures to be developed in one stage, as the oil seals the clearances between the rotors and also absorbs much of the heat of compression. It also allows the bearings to be located closer to the rotors, which keeps rotor deflections low, which in turn also helps to achieve high pressure ratios. Screw compressors have been steadily gaining popularity with designers and operators over the past 25 years, in both mining and other industrial applications.

Screw compressors have an advantage over centrifugal compressors by virtue of their greater efficiency at part load and more straightforward capacity control over their output range (even down to as low as ten per cent of design output) (Figure 9). Screw compressors, being positive displacement devices, are less susceptible to reduced capacity when fouling occurs than are centrifugal compressors, as fouling, which reduces the heat exchange in the plant, lowers the evaporating (or raises condensing) temperatures and this increased pressure requirement can be more readily handled by a screw compressor. Furthermore most screw compressors can take a ‘slug’ of liquid refrigerant that would destroy a centrifugal compressor. The nature of a screw compressor with its heavy rotors results in a much more rugged and robust design than the impellor in a centrifugal compressor.

Compared to centrifugal compressors, screw compressors are also ideally suited to direct drive without gearboxes and have lower starting torque due to the low inertia of moving parts.

Size of compressors

The size of the compressor depends on the required volumetric flow rate of refrigerant (and hence the choice of refrigerant), and the suction and discharge pressures. Pressure losses around the circuit also add to the size of the compressor (for example, high pressure drops across the evaporator or the condensers) as this increases the pressure differential that must be achieved by the compressor. Furthermore, compressors are only manufactured in certain sizes, so this also limits the selection.

With both screw and centrifugal compressors, the designer has the option of either selecting a small machine that operates at high speed, or a larger machine that operates at lower speed. Small, high-speed machines are cheaper but usually less efficient and noisier than larger, slower rotating machines. For reliability and robustness, industrial refrigeration machines are often slower speed than ‘packaged’ commercial units.

COMPRESSOR MOTORS

Hermetic (where the motor is enclosed with the compressor, to provide cooling of the motor windings) and semi-hermetic motors are common in refrigeration applications that do not use ammonia. This allows a much smaller motor size but at the expense of some loss of refrigeration capacity and slightly higher overall power costs. However, because of ammonia’s corrosive nature on copper, motors used to drive ammonia compressors are inevitably ‘open’ types. In this case, motor cooling is either via induced airflow or, on larger motors, by using other sources of water.

HEAT EXCHANGERS

A *closed* plate heat exchanger (PHE) consists of pairs of rectangular pressed metal plates that are gasketed or welded together at their periphery, with each pair sometimes called a ‘cassette’. Plates are separated by small distances, typically 3 to 6 mm and are very thin, typically 0.6 to 0.8 mm, which results in very effective heat exchange in a very small volume. The plates have ribs (also called corrugations or grooves) pressed into them. The ribs help to reduce short-circuiting within the cassette, promote turbulent flow (which improves the heat transfer) and increase the surface area (also improving the heat transfer). Individual cassettes are then clamped together in a rack (or ‘frame’) to form a ‘plate pack’. By employing suitable gasketing arrangements,[xiii] the refrigerant can be introduced into every *second* passage and the water to be chilled can enter the *alternate* passages.

An *open* PHE is an exposed pair of plates with horizontal ribs. The fluid to be chilled trickles down over the *outer* surface of the plate whilst the refrigerant passes through the inner passages. Open PHEs were historically used in breweries, dairies and other food-industry applications to directly chill a product such as beer or milk; however, mining (and most other modern) applications use closed PHEs.

A shell and tube heat exchanger consists of a large, heavy-plate, fully-enclosed ‘shell’ (shaped much like a boiler)

xiii Semi-welded construction is usually preferred, particularly in condensers as these operate at much higher pressures than evaporators, as welding prevents problems with leakage on the refrigerant side (also the side with the greatest hazard). Semi-welded construction means that each ‘cassette’ is fully welded, and a variable number of cassettes are bolted together using rubber gaskets. The ammonia side of each cassette is the welded part with the water flowing through the gasketed region between each pair of cassettes.

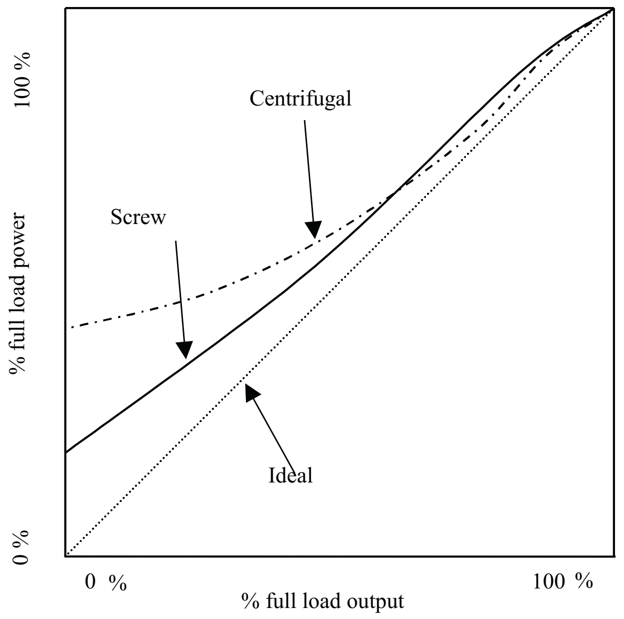

FIG 9 - Efficiency of centrifugal versus screw compressor.

with dozens or hundreds of heavy pipes running the length of the inside of the shell. Typically the water to be chilled flows through the pipes with the refrigerant flowing inside the shell over the pipes.

Closed plate heat exchangers are almost universally used in mine refrigeration plants for a variety of reasons.

In addition to better heat transfer characteristics and better resistance to fouling, PHEs are also much smaller and lighter (typically 1/3rd the size and 1/6th the weight) – which was the original reason they were adopted in mines (easier to get the heat exchangers down a shaft). They usually have smaller system refrigerant charges (due to their high efficiency), which is a significant safety benefit. They also allow boiling to take place at temperatures below the freezing point of water, because accidental freezing of the water is less likely to damage the PHE. This robustness to near-zero temperature conditions allows water to be produced down to 0.5 degrees C. Evaporation at below freezing temperatures cannot be tolerated in shell and tube heat exchangers due to the potential damage that would occur if the large-diameter tubes in the evaporator were to freeze up. Low water temperatures are a major advantage of PHEs where water is being sent underground, as it can save as much as 30 per cent on return water pumping costs from the mine, compared to shell and tube exchangers. PHEs can also achieve a closer 'approach' (difference between evaporating/condensing temperature and the water temperature) and thus achieve a higher COP than a plant using shell and tube exchangers (refer to Equation 1). Maintenance is much less frequent on plate heat exchangers. One limitation on PHEs is that they are usually restricted to maximum pressures under 25 bar (2500 kPa), which is not a limitation on a properly-designed shell and tube exchanger. However, pressures exceeding 2500 kPa would be unlikely in mine applications.

One advantage that shell-and-tube heat exchangers do have over PHEs, is that shell and tube exchangers can be mechanically cleaned on their water sides without opening the refrigerant (eg ammonia) circuit, because the end of the shell can be removed exposing the tubes only. Fully welded PHEs cannot be mechanically cleaned at all, and semi-welded PHEs can only be mechanically cleaned at major overhauls when the PHE is disassembled. As the gaskets can usually only be used once, major overhauls occur typically only every five years or even less frequently.

Two key design criteria for PHEs are the pressure drop between the inlets and outlets of the plate pack, and the thermal 'length' of the plate. When the ribs or grooves in each plate are parallel to the fluid flow, the plate has a low pressure drop, but also a low temperature 'pick-up' as water or refrigerant passes along the plate. This is termed a thermally short or 'soft' plate. Conversely, a plate that has corrugations at right angles to the flow has a high pressure drop but also has a high temperature 'pick-up'. This is termed a thermally long or 'hard' plate. In practice, corrugations are usually at varying combinations of oblique angles to achieve the correct combination of pressure drop and thermal length for the application.

The two critical selection parameters for PHEs are

- water pressure drop across the plates. This must be high enough to prevent *water-side* fouling but not so high as to impose a serious capital and operating cost penalty on pumps, and
- 'oversurfacing' or the provision of additional plate surface area in the plate pack to allow for (primarily) *refrigerant-side* fouling and also for 'off-design' evaporating or condensing conditions.

Choice of materials for heat exchangers

The choice of materials in a heat exchanger (galvanised mild steel, stainless steel, titanium, etc) depends largely on the type of heat exchanger, the choice of refrigerant and the quality of the water to be chilled.

COOLING TOWERS

The main choices in direct-contact cooling towers are between counterflow (or counter-current) and crossflow towers. Counterflow means the air and water flow in opposite directions; clearly as water droplets can only flow vertically down (due to gravity acting only downwards), the air in a counterflow tower must flow vertically up.

In a crossflow tower, the water falls vertically, but the air moves horizontally, entering the tower at one side and leaving at the opposite side. In this case, only the air entering the side of the tower at the top comes into contact with the cold water entering the top of the tower. The air entering (and leaving) the bottom of the crossflow tower has only been in contact with water that has already given up some of its cooling effect to the air above it, ie is 'hot' water.

Counterflow towers are therefore inherently more efficient than crossflow towers. This means the cooling duty can usually be achieved in a single-stage whereas a crossflow design may require two or more stages to achieve the same duty with the same efficiency. However, counterflow towers tend to be physically taller than crossflow towers to achieve the necessary contact time between air and water in a single stage, and also because the air must ultimately be introduced into a downcast shaft (the intake to the mine), which requires a 180° turn for the airflow. This increases the height (and cost) of the tower and also increases the overall mine resistance.

Conversely, the crossflow tower, being multi-stage, is typically lower in height (its overall height being the height of only one stage, with the several stages side-by-side), which usually lowers the capital cost, and also the tower only needs a 90-degree bend to introduce the air into the mine, which can lower frictional losses.

The actual choice of tower depends on local geometry of the shaft collar and the design conditions.

Cooling towers have several zones within them. Using a counterflow tower as an example, the zones are as follows (from bottom to top).

- the basin or sump where the 'hot' water is collected to be pumped away.
- the rain zone above the basin but below the packing. This is the area in which the air is generally introduced to the tower.

- the zone of packing or fill (if any) where the water droplets are broken up and the airflow is disrupted to ensure good contact between air and water.
- the spray zone between the packing and the water spray nozzles above the packing. This allows the water spray to be disseminated before it makes contact with the fill, to help reduce channeling of water through the fill.
- the water distribution level with piping and sprays.
- the drift eliminators above the sprays, which greatly reduce the actual liquid water losses from the tower.
- the zone above the drift eliminators (sometimes called the plenum) where the air is directed into a shaft or other place.

Cooling towers are common in mine refrigeration plants. Typical uses are

- for bulk air cooling (BAC) of air before it enters the downcast shaft,
- for cooling of underground air, and
- to cool the condenser cooling water (CCT).

Factors affecting cooling tower performance include

- the relative motion between the air and the water,
- the contact time between air and water,
- the contact area between air and water,
- the air flow rate,
- the temperature difference between air and water,
- the relative humidity (or wet bulb temperature) of the air, and
- the water loadings within the tower (L/sec of water per m^2 of contact surface area).

Air is moved through a cooling tower using one of three methods: natural draft (no fans), forced draft (fans at air inlet) and induced draft (fans at air outlet). Induced draft towers are typically used for condenser cooling towers as mounting the fans on the top of the tower makes for easy maintenance. Forced draft towers are typically used for bulk air coolers (if fans are required at all), as it is often less practical and more corrosive to position the fan in the narrow zone of saturated air between the cooling cells and the shaft intake. Induced draft towers are typically used in power station cooling towers where the massive heat rejection quantities require very tall towers which can then take advantage of their height and the air temperature and air density difference between the top and bottom of the tower to effectively induce a draft.

Cooling towers can be ‘packed’ (ie include highly permeable ‘fill’ or ‘packing’ to improve air-water contact area and time) or ‘open’ (with no fill). The airflow in a counterflow packed cooling tower is typically 1.5 to 3.5 m/s, which increases to 4 to 6 m/s for an open cooling tower. The uninterrupted transit of air and water in an open tower results in short contact times. Therefore open towers are very inefficient for a given size. However, they avoid any maintenance on the packing.

High-density fill[xiv] increases the air-water contact time *and* area and thus allows smaller, and lower cost, towers. However, higher fan power and operating costs may result. Splash-type fill reduces the potential for fouling as do ‘non-clogging’ (large orifice) nozzles to distribute the water on the top of the fill. Trickle and film types of fill packing tend to work well in Europe where ambient dust loadings are low (due to high rainfall and high vegetation) but can be prone to clogging in hot, dry climates and terrains such as in Australia.

Typically, packed cooling towers achieve 80 per cent to 85 per cent of the tower’s total heat exchange in the fill zone, ten per cent to 15 per cent in the spray zone between the fill and the drift eliminators above the fill, and about five per cent in the ‘rain zone’ between the fill and the water in the basin below.

Note that to achieve a close ‘approach’ temperature in a condenser cooling tower, a low water to air loading is required, ie more air contacting less water. Most commercial towers are designed to minimise the fan power required by having much higher water to air loadings. The resulting higher approach temperature results in higher condensing temperatures and higher compressor power costs.

Low maintenance construction materials, both for the structure and the internal components are also essential for a cooling tower. Non-clogging nozzles and non-corroding fill material is required. To improve the ‘turndown’ capacity of the plant in off-season, and to allow for maintenance downtime, it is desirable to have at least two cells in the tower. Each basin must be able to be isolated separately so as to remove mud in the basin to conform to protocols designed to maintain safe *legionella*, other bacteria and algal levels.

As condenser cooling towers evaporate water, salt and dissolved solids are gradually concentrated. To maintain satisfactory water quality, a fraction of the circulating water must be ‘blowndown’ or bled-off. The blowdown rate is typically between 50 per cent and 100 per cent of the evaporation rate. Water losses or ‘carryover’ due to liquid water droplets leaving the tower should be less than 0.05 per cent of the cooling tower circulating water flow and with well-designed drift eliminator plates can be as low as 0.00001 per cent. Water losses due to evaporation can be calculated by a suitable thermodynamic analysis but are typically one per cent of the circulating water flow per seven degrees of water temperature change in the tower.

Care must be taken in the design of cooling towers to ensure ‘recirculation’ of air from the discharge to the intake is not significant. Otherwise a substantial loss of thermal performance will occur as the air entering the tower will be hotter than it should be: at a temperature somewhere between the ambient air and the discharge air. A guideline is to ensure the discharge velocity in the plume is more than 8 m/s to avoid recirculation. The cooling tower inlet air WB temperature should be no more than 0.2° WB above the ambient WB in a well-designed tower.

Pre-cooling towers

When a bulk air cooler is used to chill intake air, the temperature of the water returning from the cooler to the plant can never exceed the surface ambient wet bulb temperature. However, if the chilled water is not being used for surface cooling but is being sent to underground air coolers, then the return water temperature will be somewhere between the wet and dry bulb temperature of the underground air, which is usually well *above* the surface wet bulb temperature. In this case, a simple direct-contact, water-air ‘pre-cooling’ tower can be used to drop the water temperature to within about two degrees of the surface wet bulb temperature. As this pre-cooling can be achieved without refrigeration, it has a very high COP (typically over ten, a function of only the pumping power to circulate water through the tower) and therefore improves the overall COP of the surface refrigeration plant.

EVAPORATORS

The evaporator is the ‘business end’ of the refrigeration machine itself. It is here that the cooling effect from the cold liquid refrigerant is transferred to the hot water. Any problems in the evaporator therefore have an immediate impact on plant output.

PHE evaporators are usually plumbed with the cold liquid refrigerant entering the bottom of the PHE, and the cold liquid/gas refrigerant mix leaving from the top. The hot water (the thermal load) also enters the bottom of the PHE and leaves from the top, ie in co-current flow with the refrigerant. Bringing the hot water together with the cold liquid-phase refrigerant

xiv In this context, ‘high density’ is not referring to the density of the actual substance used to manufacture the fill, but rather to the proportion of the overall packing volume that is taken up by the fill material itself, compared to the amount that is void and therefore available for passage of the air and water.

ensures that violent boiling of the refrigerant occurs, which is important in thermosiphon evaporators (described later).

CONDENSERS

Condenser performance is an often-neglected aspect of plant design. An undersized condenser results in high condensing temperatures, with commensurate high condensing pressures and compressor power costs, and often a reduction in plant capacity as well.

Condensers must remove not only the heat being picked up in the evaporator but also from the work done by the compressors.

Condensers for mine refrigeration plants usually rely on rejecting heat to the ambient air, although rejection of heat into water (eg a lake or the ocean) is practiced in some other countries and industries. Heat rejection into the air can be achieved directly from the refrigerant to the air in an *air-cooled condenser*, or by a combined heat and mass transfer in a *cooling tower* or *evaporative condenser*.[xv] As discussed earlier, the condensing temperature in an air-cooled condenser can never be lower than the *dry bulb* temperature of the air, whereas the condensing temperature in a cooling tower or evaporative condenser can approach the *wet bulb* temperature of the ambient air. As lower condensing temperatures result in a better plant COP, and the wet bulb temperature is always lower than the dry bulb, cooling towers or evaporative condensers have the better thermodynamics (and hence lower power costs), but sometimes a higher capital cost. Air-cooled condensers do have the advantage that *legionella*, which inhabits the cooling water systems of water-cooled condensers, is much less of a risk with air-cooled condensers. Typical water-cooled condenser configurations are as follows.

Plate heat exchangers and cooling towers

In this option, a plate heat exchanger is used to reject the heat from the refrigerant into cooling water.[xvi] The cooling water is then circulated though a cooling tower, where heat from the water is transferred to the ambient air by evaporation of a portion of the water. Thus there are two physically separate heat exchange processes in this option, firstly refrigerant-to-water in the PHE and secondly, water-to-air in the cooling tower. Each of these heat transfers has its own thermal and other inefficiencies.

One advantage in keeping the two heat transfer processes separate is that it allows the heat rejection facility to be located away from the refrigeration plant. This is an advantage where the air available for heat rejection is at a different place to the desired location for the refrigeration plant. In this case, it would be hazardous and expensive to run the necessary distance of piping containing refrigerant from the compressor to a heat exchanger at the reject air.

Because the condensed refrigerant must drain under gravity from the condenser, the refrigerant gas enters at the top of the PHE and the condensed liquid drains out the bottom. The cooling water enters the PHE at the bottom and leaves at the top, in counterflow fashion. This ensures the best approach and lowest possible condensing temperatures.

Evaporative condensers

In this option, the hot refrigerant gas from the compressor is discharged through inexpensive galvanised[xvii] steel coils over which air is drawn and water is simultaneously sprayed.[xviii] An evaporative condenser therefore involves a single-stage, simultaneous heat and mass transfer process (evaporation of the water). In principal, lower condensing temperatures should be achievable in an evaporative condenser, compared to a PHE with a separate cooling tower. Also, water flow rates are lower than in the evaporative condenser. Therefore it often has the potential for a lower capital cost and better efficiency than the PHE and cooling tower option.

However, evaporative condensers also have disadvantages including the following.

- Because of the relatively large diameter refrigerant tubes in an evaporative condenser (to handle the pressure) compared to a plate heat exchanger, the necessary charge of refrigerant is higher. If the refrigerant is toxic and the plant located near a mine air intake, then evaporative condensers have a higher risk potential due to leakage than plate condensers.
- As the chilled water plant and the evaporative condenser are separate units, high-pressure refrigerant lines must be used to join the two. This means that ‘factory assembled’ units need skilled labour in the field that can weld refrigerant lines and charge refrigerant systems. In addition, underground spot coolers should avoid evaporative condensers as these require the refrigerant system to be evacuated and then broken apart prior to transport. This is not the case with a PHE condenser and cooling tower as only water is pumped between the chilled water plant and the cooling tower.
- Because the evaporative condensers combine two heat transfer processes in one plant, they are more directly and immediately affected by changes in ambient wet bulb temperatures.
- They scale more readily than plate condensers, because the high shear stresses of water passing through the PHE tends to produce less scale than in evaporative condensers, where the water pressure of the sprays on the outside of the coils is low. Also the evaporation process (which tends to deposit salts from the water) occurs in the cooling tower in the PHE/cooling tower option, whereas the evaporation in the evaporation condenser occurs from the walls of the condenser tubes themselves.
- They can be more difficult to clean than plate condensers, partly because they are often made of galvanised steel, which can be attacked by some of the more commonly used cleaners.
- Galvanized steel is subject to corrosion from any ammonia leaks anywhere near the plant (which are inevitable, eg on maintenance); the alternatives – stainless steel or titanium evaporative condensers – are very expensive.
- Steel has much poorer thermal conductivity than does copper, and this tends to negate some of the excellent thermodynamic properties of ammonia, unless the steel is very thin (which it is in plate-type heat exchangers).

---

xv Do not confuse *evaporative condenser* with *evaporator*. They are totally different devices.

xvi Rather confusingly, the water in the condenser circuit is called *cooling water*, with the water in the evaporator circuit called *chilled water*. Hence condenser water and cooling water are interchangeable terms; likewise evaporator water and chilled water are interchangeable terms.

xvii Steel is essential if the refrigerant is ammonia as copper cannot be used with ammonia. Galvanising on the outer surface of the coils helps to reduce corrosion from the condenser water. Galvanised steel has higher conductivity than stainless steel, but sufficient strength that tube thickness can be kept to a minimum. If a major ammonia leak occurs, the galvanising will be damaged; however, the piping can be replaced relatively easily and inexpensively and will continue to maintain its integrity (ie not allow leakage of ammonia) whilst replacement piping is being purchased.

xviii In some evaporative condensers, only a portion (typically about 50 per cent) of the air goes through the evaporative condenser, with the remaining air going through a conventional cooling tower through which the cooling water is also circulated. This improves the cooling process, therefore allowing less surface area for the coils than in standard evaporative condensers.

- Steel piping is much tougher than copper or thin-plate steel, which makes it difficult to manufacture high-efficiency heat transfer surfaces (eg fins, etc).
- The 'plumbing' of evaporative condensers is critical. Small errors in the discharge plumbing or in the vertical drop from the condenser to the receiver can have major detrimental effects on the operation of the condenser, often due to 'lock-up' of condensate in the coils.

Because of these problems, the potential for lower capital and operating costs from evaporative condensers may not be realised due to poor design, poor installation or poor operating practices.

Shell and tube and air-cooled condensers

The heat transfer coefficients in an air-cooled condenser are usually poor compared to the alternatives, because the driving force for the heat exchange process is the difference between the ambient DB temperature and the condensing temperature, T_c (which is always smaller than (T_c-WB)), and because of the very large differences in the thermal conductivity of water and air. The need to have 'fins' on the coils to improve the heat transfer coefficient also makes these more susceptible to fouling. Shell and tube condensers are difficult to clean (some having several thousand tubes) and often have relatively poor in-service heat transfer characteristics in a mine environment. For these reasons, these types of condensers are rarely selected for new installations in mines.

Non-condensables

'Non-condensables' (or 'inerts') are gases (typically air) that will not liquefy in the condenser. Non-condensables are caused by

- insufficient evacuation prior to refrigerant charging,
- leaks in the low side of the system, if it operates below atmospheric pressure,
- addition of poor quality refrigerant containing non-condensables, or
- chemical breakdown of oil or refrigerant.

These non-condensable gases reduce the capacity and surface area of the condenser available for heat exchange and may promote internal corrosion. Reduced heat transfer area can lead to higher condensing temperatures and reduced plant performance. For example, in a typical ammonia system, two per cent non-condensables can increase the condensing pressure by 20 kPa, which results in a power penalty of about 2.5 per cent on the compressor. Non-condensables must be purged from the system from time to time and a non-condensables test and purge valves are provided for this purpose.

Non-condensables are more detrimental to plate heat exchangers than shell and tube heat exchangers, as the former have a high surface area but low internal volume. The same volume of trapped non-condensables therefore has a disproportionate effect on loss of effective surface area in PHEs.

In ammonia refrigeration systems, non-condensables tend to collect in the coldest part of the condenser and/or at the highest elevation in the condenser, so purge valves are usually located at these points. Manual purging is usually satisfactory for plants that always operate under positive pressure with respect to atmosphere, with automatic purgers used otherwise.

A 'non-condensables' test typically involves running the plant in a stable operation, then turning it off and monitoring the condensing temperature and pressure. If this deviates significantly from the saturated vapour pressure of the refrigerant gas at that temperature (by more than 2° or 3°C), then non-condensables are probably present and must be purged.

Liquid receiver

The refrigerant gas that liquefies in the condenser drains into a high-pressure, liquid receiver. This device should have sufficient capacity to accommodate the full refrigerant charge, along with isolation and pressure relief valves. During maintenance on the ammonia side of the system, the refrigerant is usually pumped and drained into this receiver and isolated from the rest of the circuit.

DISCHARGE SUPERHEAT, DE-SUPERHEATING AND SUCTION SUPERHEAT

Superheating occurs when the temperature of a gas is above its saturated vapour temperature (the temperature at which liquid and vapour co-exist in equilibrium) at that particular pressure. For example, superheated steam means steam at more than 100°C (at sea level) – the steam has been heated to a temperature above that necessary for water to evaporate at that pressure. Superheating of refrigerant always occurs in the compressor discharge and to a lesser extent in the evaporator. Superheating does *not* necessarily mean the temperature of the gas has been increased, it may mean the pressure has been lowered, which also results in a 'superheated' gas.

At the *compressor discharge*, the refrigerant gas is often at temperatures of 65 to 75 degrees or more. The typical summer condensing temperature is about 30 to 35 degrees. As the gas will only start to condense when it reaches the condensing temperature (saturated vapour pressure), the de-superheating of the discharge gas must either occur after it reaches the condenser or can sometimes occur in a device called a de-superheater. Discharge temperatures are particularly relevant to a PHE condenser, as very hot refrigerant gas can damage the hundreds of gaskets used in the plate packs. In practice, formal de-superheaters are usually only used to recover useful, high-grade 'waste heat' in non-mining applications where discharge temperatures are often designed to be deliberately high for this particular purpose. However, where concerns exist about superheated gas damaging the gaskets in a PHE condenser, a de-superheater may be installed between the compressor discharge and the condenser.

In the low-pressure side of the system (after the expansion valve), the refrigerant gas that has boiled off in the evaporator will often increase in temperature or experience a pressure loss due to piping or valves as it is drawn to the compressor suction. This is called *compressor suction superheat* and is the difference between the *actual* compressor suction temperature and the *saturated* refrigerant temperature at the actual evaporating pressure.

SUPER-HEATING AND SUB-COOLING

Liquid refrigerant boils in the evaporator at the evaporating temperature. As the water being chilled is at a higher temperature than the liquid refrigerant, the temperature of the resulting refrigerant gas may increase beyond its boiling point into its *superheated* region. This incidental superheating is usually called non-useful superheating, as it has no thermodynamic benefit to the process. A small amount of non-useful superheating is prudent as it helps to prevent liquid refrigerant accidentally being drawn into the compressor. Useful superheating occurs where a heat exchanger is interspersed between the evaporator and the condenser. The low temperature gaseous refrigerant from the evaporator is then used to remove heat from the high temperature liquid refrigerant leaving the condenser. This superheats the vapour entering the compressor and also provides some 'useful' sub-cooling of liquid refrigerant. However, large amounts of suction superheat results in a reduced density of vapour entering the compressor, which merely acts to push the compressor into a less efficient region.

In flooded evaporators (ie those in which only a portion of the liquid refrigerant entering the evaporator boils, the rest returning to the surge drum), it is impossible to have superheated vapour leave the surge drum, and any superheat is only created by the pressure loss in the suction line to the compressor or any ambient heat gains in this line.

Sub-cooling is basically the difference between the saturated condensing temperature and the temperature of the liquid refrigerant just upstream of the expansion valve. When the gaseous refrigerant liquefies in the condenser, the resulting liquid will be at the condensing temperature. However, if the water temperature in the condenser is much lower than the condensing temperature, then sub-cooling of the liquid refrigerant may occur. A small amount of sub-cooling is helpful as it may improve the thermodynamics of the refrigeration process. However, PHE condensers, unlike shell and tube, do not provide for refrigerant storage within the condenser (this occurs in the high-pressure liquid receiver), therefore there is usually no sub-cooling of the condensed refrigerant as the refrigerant drains from the condenser (and hence from further contact with circulating water) as soon as it condenses. In practice, for a PHE to produce a sub-cooled liquid, it would need to trap liquid condensate within the PHE. This would cause control problems and would seriously reduce condenser capacity.

FOULING

Types of fouling in heat exchangers

Fouling can occur on either the *refrigerant-side* or the *water-side* (or air-side) of a heat exchanger.

The refrigerant side of the plant is totally sealed from the environment, so that the only material that can usually foul this side is the oil used to lubricate and cool the compressor, some of which is inevitably carried over from the compressor oil separator into the rest of the system. This is because oils used in ammonia refrigeration systems (traditionally inexpensive mineral oils) are not miscible with ammonia, and are heavier than liquid ammonia. Oil rarely fouls on the condenser as the high temperatures result in low oil viscosity and both gravity and the shear forces (the action of the flow of refrigerant) tend to push the oil out. Oil and oil-decomposition products can, however, collect in the surge drum and evaporators. For example, in a system using ammonia (which typically uses insoluble oils for lubrication), any oil which is carried over in the liquid refrigerant remains behind when the refrigerant evaporates in the evaporator. Here, the oil is cold and viscous, and other means are required to drain it from the evaporator back into the oil system.

The water-side of a heat exchanger is not sealed from the environment. It is exposed to dust in the cooling tower or bulk air coolers.

Fouling on the water side can be due to sticky products (such as oil and grease), scale (especially inorganic salts such as $CaSO_4$ or $CaCO_3$, which have inverted solubility curves[xix]), algae, bacteria, fungi and deposits of sand, etc.

Refrigerant side fouling can be reduced by properly designed oil separation, oil filtration and oil return systems and proper provision for chemical cleaning.

Water side fouling can usually be effectively controlled by strainers or filtration, water treatment, suitable geometry and keeping the pressure drop across the PHE high enough to ensure turbulence and high shear stresses to keep the surfaces clean. Provision for backwashing is also required.

Backwashing

Backwashing is the process of reversing the water flow, at full volume, through the water side of a heat exchanger, to dislodge and then remove any fouling on the water side. Typical backwashing time is three to five minutes. The water is then dumped to waste. The backwashing arrangements can be automatic or manual.

The plant running time between ‘backwashing’ (the process of cleaning heat exchangers) must also be realistically estimated. If cleaning is only required annually, and the plant can be shut down during winter (ie is not required in winter), then the expected fouling factor over the summer months can be used for plant design. However, this is not always the case, and in some instances the *minimum* (worst) fouling factor immediately prior to cleaning should be used for design purposes.

Chemical cleaning

Where chemical cleaning of heat exchangers is required, specialist advice must be sought. Some detergents contain ammonia (NH_3), which attacks copper, zinc, tin and any of their alloys; ammonia also forms explosive mixtures with silver, mercury and halide compounds. Other cleaning agents contain $NaOH^-$, which dissolves protein (including hands and eyes) and aluminium. Acids such as HCl, HNO_3 or H_2SO_4 should never be used.

Evaporator Fouling

Because the water in the evaporator is at low temperature, biological fouling such as by algae and bacteria is uncommon. Fouling on the water-side of evaporators is usually due to deposition of sludge and sediment picked up as the water mixes with dusty ambient air in the bulk air cooler. This can easily be removed by backwashing.

Condenser Fouling

The water circulating through the condenser is at moderately high temperature, which can promote biological fouling. In addition, as the water solubility of many naturally occurring salts decreases as the water temperature increases, chemical ‘scaling’ is a common problem in condensers. If scaling is not prevented, then once established, it can be difficult to remove. Backwashing is unlikely to remove scale from condensers. Chemical cleaning with a buffered acid solution is usually required (typically eight per cent nitric acid for ammonia/stainless steel combinations). If the condensers are of the evaporative type, then inhibitors must be added to the chemical treatment to protect the galvanised coating. Chemical treatment is both costly and results in down time on the plant. Therefore water conditioning in the condenser water circuit is critical. Filtering or straining of the condenser water is essential to reduce contaminants being lodged in the condensers, as is provision for a ‘bleed-off’ and ‘make-up’ with fresh water to keep dissolved solids to acceptable levels.

Significance of fouling

It is important to note that allowing the manufacturer to ‘select’ a fouling factor will possibly lead to an unrealistically small heat exchanger. This lowers the capital cost and improves the competitiveness of the bid. However, the plant will then only perform to specification when new. It is essential therefore that suitable fouling factors be specified for the bidder to use. To reject the same amount of heat from a fouled (compared to a clean) condenser, a higher condensing temperature (more ΔT) and therefore higher condensing pressure is required. This increase in $T_c – T_e$ reduces the plant COP and increases the motor power required to operate the compressor; however, a high condensing temperature usually has little direct effect on the evaporator (*providing* the compressor and motor have sufficient capacity to achieve the higher condensing pressures at full refrigerant flow) and hence on the overall refrigerating capacity of the plant.

Contrast this to fouling in the evaporator which, because it reduces the heat exchange between the refrigerant and the ‘load’ (eg the water to be chilled) requires lower evaporating temperatures to overcome the poor heat transfer because of the fouling. This low evaporating temperature results in similarly low COPs, high power consumption and reduced plant capacity.

xix A substance with an inverted solubility curve tends to precipitate out as the water temperature increases, explaining why scale forms when cold cooling water comes into contact with warm condenser surfaces.

It is clear that fouling in the evaporators will have a serious immediate detrimental impact on plant capacity. However, fouling in the condensers will result in additional power costs (and a minor reduction in COP) but only result in reduced refrigeration output if the compressor has limited capacity.

Fouling factors

Fouling factors are quoted in various units by different manufacturers and design engineers. One such unit is kW/(m^2.K); a common alternate is m^2.K/W. Typical water-side fouling factors in shell-and-tube heat exchangers in a mine environment are as follows.

- New surfaces 15 kW/(m^2.K)
- Clean surfaces 10 kW/(m^2.K)
- Typical operating conditions 5 kW/(m^2.K)
- Very dirty surfaces 2 kW/(m^2.K)

There is usually no allowance made for water-side fouling in PHEs as these are designed to remain relatively clean as discussed above, and some provision exists by virtue of the over-surfacing allowance.

For PHEs using ammonia, typical ammonia-side fouling factors (mainly due to oil carry-over) are

- Condenser 0.000035 m^2.K/W [28.6 kW/(m^2.K)]
- Evaporator 0.000020 m^2.K/W [50 kW/(m^2.K)]

The fouling factor is higher in the evaporator as this is where the oil tends to collect.

Because the nature of water-side fouling in condensers and evaporators is usually different, the prevention and treatment options are also different. Note that in both evaporators and condensers, some fouling is a 'normal' part of plant operation; it is not an 'abnormality'. Provision for the mitigation and treatment of fouling is therefore essential at both the design and operational stages. During design, it is essential to provide sufficient heat transfer surface to handle typical fouling. The more heat transfer surface area 'purchased' for the plant, the more fouling the plant can handle. However, too much surface area results in low refrigerant and/or water flows in the heat exchanger, low pressure drops and hence low shear stresses in the fluids. This tends to have the perverse effect of promoting fouling. Therefore specialist advice must be sought to obtain the correct compromise between sufficient surface area to accommodate heat transfer and operational fouling, without so 'over-surfacing' the heat exchanger as to promote excessive fouling.

Water quality and treatment

As mentioned earlier, water treatment and filtration are important in both the evaporator and condenser water circuits. This is to control the growth of algae and bacteria, to inhibit scaling, and to filter out suspended solids that otherwise can result in an increasing pH and also can foul any packed fill material. During the refrigerant plant design process, plant water supply should be checked for at least the following: pH, Cl^-, S^{2-}, NH_3/NH_4^+, SO_3^{2-}/SO_4^{2-}, $Fe^{3+}/Fe^{2+}/Fe$, O_2, CO_3^{2-}.

Suitable water 'bleed off' and 'make up' is also essential to control fouling and scaling. This can be a particularly serious problem in underground air coolers. Dust and diesel particulates in the air can form a sludge that can reduce the effectiveness of the air cooler, especially those that are 'packed' (ie those designed with internal fill material to improve the air-water heat transfer) compared to those that are 'open' (no fill material).

ECONOMISER

The hot, high-pressure refrigeration liquid in the receiver must pass through an expansion valve before entering the surge drum. The resulting drop in pressure causes a portion of the liquid to 'flash off' as a gas. It provides some benefit in this process, as the remaining liquid is further cooled by an amount equivalent to the latent heat of evaporation of the flash gas. This flash gas is at a temperature and pressure somewhere between the condensing and evaporating temperatures and usually passes into the surge drum along with the cold liquid refrigerant, increasing the mass flow in the evaporator. Another option, called an *economiser*, is to take a side stream of liquid refrigerant and pass it through a heat exchange device inserted in the liquid line between the condenser/receiver and the evaporator/surge drum. On evaporation, this feed subcools the main liquid feed. This then reduces the volume of flash gas at the evaporator, which decreases the required evaporator mass flow. The flash gas produced in the economiser (or *superfeed*) is taken directly into a port part-way along the length of the screw compressor, or injected between the stages of a multi-stage centrifugal compressor. In effect, this means the compressor needs to do less mechanical work to pass the same flowrate of liquid refrigerant through the evaporator. This has the effect of extending the capacity of the compressor, perhaps by as much as ten per cent to 12 per cent.

The main disadvantage of the economiser is that it only provides this benefit when the compressor is operating at more than about 80 per cent of its rated capacity. Since mine refrigeration plants are often required (for most of the year) to operate below their rated capacity, an economiser usually just adds to the capital cost and complexity of the plant, without contributing much to operational performance. Furthermore, because of the relatively high evaporating temperatures of mine plants, which already result in relatively high COPs, it is often cheaper to buy any additional capacity using a larger compressor than to buy the same capacity using a smaller compressor extended by an economiser.

INTERCOOLING (OR INTERSTAGE COOLING)

In multi-stage compression, the opportunity exists to reduce the work of compression per kilogram of vapour, and reduce the final condensing pressures, by *intercooling*. This involves positioning an intercooler (using liquid refrigerant) to cool the discharge gas from the first stage of compression before it enters the second stage. However, as multi-stage compression is rarely used in mine systems, and the benefit of intercooling is modest, intercoolers are not normally used in mine plants.

OIL SEPARATOR

As discussed earlier, screw compressors must inject oil into the suction end to lubricate and cool the rotors and to seal against refrigerant escape back along the rotor. This oil must then be removed from the refrigerant gas at the discharge end, otherwise it will foul the condenser or evaporator.

As the mass flow of oil is often similar to the mass flow of refrigerant, effective separation of oil and refrigerant is one of the most critical factors in effective plant design.

The oil separator is a large pressure vessel and usually relies on a combination of centrifugal action, impingement, straining and coalescing to remove the oil. This is effective in removing oil particles down to about one micron in size. Minor quantities of oil 'carryover' (which should be less than 5 ppm of the refrigerant flow) as a mist pass through the separator and eventually accumulate in the condenser, surge drum and even the evaporator, and suitable oil drainage or return arrangements need to be provided, based on the fact that the oil is usually heavier than the liquid refrigerant and can be caught in a trap, 'pot' or 'udder'. The oil separation efficiency falls off as the oil discharge temperature increases, as a higher proportion of the oil leaves the compressor as a finer mist fraction.

OIL COOLING AND LIQUID REFRIGERANT INJECTION

There are several mechanisms for oil cooling. Water heat exchangers (usually shell and tube) are used in some applications, or refrigerant thermosiphon oil cooling (see below) may be used. In some screw compressor systems, liquid injection oil cooling (or Liquid Refrigerant Injection, LRI) is used. This involves a variable amount of condensed liquid refrigerant from the receiver being injected through a separate port into the compressor to cool the refrigerant and oil leaving the compressor. Liquid injection oil cooling is of particular advantage to plate condensers. These often have elastomer gaskets that are prone to perishing if the compressor discharge temperatures are too high. It also assists with oil separation, as this is more effective at lower oil temperatures. However, LRI does result in a loss of capacity and efficiency in the compressor and an increase in plant power consumption.

It is always wise to carefully scrutinise the gasket-ageing characteristic prior to making a final decision on either the gasket material or the oil cooling arrangements. Gasket warranties should be sought.

THERMOSIPHONS

There are two main applications for thermosiphon devices in mine refrigeration plants.

In PHE evaporators, the refrigerant is usually not mechanically pumped through the evaporator, but a flow is induced by virtue of the evaporation (boiling) of a portion of the refrigerant. The surge drum containing the liquid refrigerant is mounted above the evaporator, creating a *flooded evaporator*. Liquid refrigerant is delivered from the surge drum to the bottom of the PHE, with a mixture of liquid and gaseous refrigerant leaving at the top of the PHE to return to the surge drum. The buoyancy of the gas within the plates of the PHE induces a natural, convective upflow. This upflow in the return leg, combined with the action of gravity in the supply leg, causes liquid refrigerant to circulate through the PHE. The process is self-starting and self-sustaining: as the thermal 'load' increases on the evaporator, more boiling occurs, which promotes more buoyancy and thus more refrigerant flow.

When the plant is not operating there is no water flowing through the evaporator. Hence, there is no temperature difference across the evaporator, no boiling occurs and therefore the flow of refrigerant stops. When the plant starts up, the water flow induces boiling that in turn induces the convective refrigerant flow.

Thermosiphon PHEs need careful attention to the following matters.

- 'Slug flow', which is caused when the thermal load on a thermosiphon PHE is so low that the boiling rate of refrigerant is very low, resulting in the accumulation of vapour in the plate pack. When the load on the plate pack reduces five-fold, the flow rate of refrigerant also reduces five-fold and the outlet leg refrigerant velocity reduces five-fold. The vapour velocity in the return leg can fall so low (typically below 12 m/s for ammonia) that it can no longer 'lift' the liquid fraction back to the surge drum. It falls back down the outlet leg and accumulates in the PHE. When a critical vapour fraction is finally reached, the liquid-vapour mixture becomes buoyant and a 'slug' rises rapidly through the plates, resulting in an unstable, unsteady flow regime.
- 'Burn out', which is caused when the thermal load on the PHE is so high that the refrigerant is boiling at an excessive rate. As the heat transfer coefficient for vapour is much lower than for liquid, the overall heat transfer in the PHE falls resulting in a rapid loss of heat exchange, an increase in the thermal load, and an escalation of the problem.

Also operating on the thermosiphon principle, an oil cooler can be placed at the discharge of the compressor to operate in tandem with the liquid refrigerant in the liquid receiver, forming a thermosiphon oil cooling system. The hot oil cools by boiling off the refrigerant in a self-sustaining process. Thermosiphon oil coolers are, however, vulnerable to low refrigerant charge and in any event, are not required where liquid injection oil cooling is employed.

As thermosiphon systems do not rely on mechanical pumps, any constriction in the system will introduce a pressure drop that could interfere with the thermosiphon effect. Care must be taken with the selection of any isolating valves or other installed devices to ensure this does not occur. Pumps may be required where a suitable geometry is not possible, eg where the surge drum and evaporator are too far apart or sufficient vertical height difference is not available.

LEGIONELLA, BACTERIAL AND ALGAL CONTROL

In all water systems, especially where chilled water is pumped underground, control of bacteria and algae is essential. *legionnaires* disease is a potentially fatal form of pneumonia caused by various species of *legionella* bacteria. It thrives in the hot water often associated with cooling towers. It can be a problem both for maintenance workers and also any persons who are downwind of the tower, as *legionella* has been frequently reported in the drift from cooling towers, particularly where these are near the intake to a building air-conditioning system. Good drift eliminators, proper water treatment, regular testing and cleaning of the tower to remove algae and sediment (which otherwise reduce the effectiveness of any biocide) and siting the tower so that the discharge is away from human habitation are all good-practice measures to avoid *legionella*.

Frequently, cold and hot water dams are also installed underground to provide some buffer in the event of pump or pipeline failures and short-term (<24 hour) variations in underground loads. Thermal insulation and bacterial and algal control are also required here. Chlorine or bromine is commonly used (bromine being more normal due to its lower toxicity).

CONTROL STRATEGY ON REFRIGERATION PLANTS

Control of a refrigerant plant is not a trivial matter. As the flow of refrigerant (with its various phase and specific volume changes) forms a completely closed loop, and must respond to wide fluctuations in the load, the process has the inherent risk of oscillations and 'hunting'.

In addressing this, many control systems are designed to minimise the capital cost of the plant and are relatively wasteful of power costs. This is particularly true of 'packaged' plants designed primarily for commercial applications.

Close control of plant operation is particularly important where the water is being chilled to a temperature close to freezing, else the heat exchanger can freeze, resulting in loss of capacity and potential damage to the plant. In general, where the evaporator water-off temperature is 3°C or less, the plant must be controlled on both the PHE leaving-water temperature and the compressor suction pressure to avoid freezing of the plates. Where the leaving-water temperature exceeds 3°C, control can be from the water-off temperature only.

Due to the elaborate nature of a refrigeration plant, careful selection of the various modulating valves, especially the expansion valve but also the liquid refrigerant injection (LRI) valve and others, is critical. These need to be sized so as to provide a controlled and linear response over the full operating range (flows and pressures) of the system requirements. In some cases, this requires more than one valve to be installed in parallel as no single valve can satisfactorily perform the duty in the required fashion (Figure 6).

Strategically, a mine refrigeration plant can potentially be operated under one of two broad control strategies.

Firstly, it can run with the compressor 'flat out' and fill up a cold-water dam, which in turn chills the heat load. If the dam becomes full, the plant turns off. This is called *batch* operation.

Alternately, it can run in *continuous* mode, where the evaporator partially unloads to produce the required chilled water outlet temperature, which will vary according to the evaporator load. Water flow rates are usually fixed. No cold water dam is required for continuous operation. A variation on the continuous mode operation is to leave the water temperature fixed and vary the water flow rate; however, this presents problems with the heat exchangers, which are not generally very tolerant, from a process and fouling point of view, to wide fluctuations in water flow rates.

Batch mode results in higher average condensing temperatures than continuous mode, because the compressors, while on, are always operating at full load. Despite the higher compressor efficiency at full load, the plant COP is usually lower than in continuous mode. However, the ability to position a cold-water dam between the plant and the load effectively ‘extends’ the plant capacity by uncoupling the mine load from the plant supply. Therefore during periods when the plant is operating near maximum capacity (ie summer), batch mode allows the plant to better cope with temperature excursions above its design capacity. Batch operation also better provides for maintenance by providing regular non-operational time. However, capital costs are higher due to the need to construct cold- and hot-water storage dams that are typically quite large if they are to be useful (ie must contain several hours of chilled and return (hot) water supply). The cold-water dam needs to be well insulated, including its roof.

By contrast, in continuous mode, the plant unloads the compressors to just meet the mine load. This results in average condensing temperatures below those of batch operation. In turn, this results in improved COPs and lower power costs. However, when ambient conditions exceed plant capacity, the plant can no longer simultaneously meet both the water flow demand and the water-off temperature. The outlet temperature must be allowed to rise (or water flow reduced) which can result in a serious reduction in heat exchange at the load. However, capital cost of continuous operation is lower as water storage dams are not required.

Typically, ‘batch’ operation is used when water is being sent underground. This is because

- high pumping costs back to surface mean that the water should be sent underground as cold as possible, and therefore the variation in load demand is met by reducing the volume but leaving the water temperature constant, and
- the system needs water dams typically on both surface and underground (eg to allow time for routine breakdown maintenance on supply and return water pipes or pumps) and this allows, in practice, a continuous underground load to be met by a discontinuous surface supply.

By contrast, ‘continuous’ operation is used for surface bulk air coolers. This is because

- the air load (the air going underground through the bulk air cooler) is continuous, and
- the bulk air cooler cannot tolerate large swings in water flow without creating problems with the spray patterns in the nozzles ‘collapsing’ and the water ‘channelling’ through the fill, with substantial loss of efficiency.

However, even continuous-operation plants need some water storage capacity for several reasons.

- The need to provide a supply of at least five to ten minutes of ‘backwash’ water. Backwashing may also require a dam for the dirty water discharge, ie some sort of anti-pollution pond.
- The need to provide for an initial water charge for the system. Without such storage, filling up the refrigeration system with water from the mains water line may take several days!
- Sometimes water storage is required to ensure water is available for fire-fighting, especially where the plant is remotely located and the towers are constructed from timber.
- In some installations, water storage may be required for ammonia containment systems.

It is often possible to provide the necessary water storage by designing deeper sumps in the tower.

THERMAL INSULATION

Cold dams need to be insulated depending on the expected cooling losses to the ambient air, which can otherwise take up a substantial portion of plant capacity. Likewise piping and some vessels within the plant (particularly the surge drum) that pass or store cold refrigerant or cold water need to be insulated. Most insulation must be covered with a water vapour barrier to prevent it becoming saturated with condensation and losing its insulating value. Phenolic foam, foamed glass or pre-cast polystyrene foam is a typical insulation, protected by a spiral-wound galvanised sleeve. Foamed glass, being a closed cell structure, is relatively unaffected by moisture. Water dripping from piping or from outside dam walls is a sure indication of significant cooling losses in the system, as the latent heat lost from the system when water condenses from the ambient air onto these surfaces is high. Insulation is especially critical where the surfaces are sub-zero due to the formation of frost.

Depending on condensing temperatures and other factors, some piping or vessels may also need to be insulated to protect maintenance or operating workers from receiving burns. High temperature insulation does not require a vapour barrier, as water will not condense on surfaces which are hotter than the ambient air.

It is sometimes said that insulation of chilled water lines in intake airways is not required, as any cooling loss into the intake air is a ‘useful’ effect and not a true waste. However, much of the cooling loss from uninsulated pipes in the hot, humid environments of an underground mine is via condensation of moisture from the air onto the pipe. The water subsequently drips away with only minor chilling of the intake air.

EXTREMITIES OF OPERATION

Many refrigeration plants are designed around a single, notional ‘duty’ point. This focus on a single operating duty frequently leads to a poor plant design, and in some cases, major modifications are required after installation, either to make the plant work, or to allow it to work in a stable condition. It is particularly important to consider the ‘extreme’ conditions under which the plant will be required to operate. These extremities include the following.

- *’Pull-down’ conditions*. For example, if the plant has been off-line for some time and the water to be chilled is warm, suction pressures and temperatures can be extreme, resulting in excessive thrust bearing pressures on the compressor.
- *’Above-design’* ambient conditions. For example, if the ambient wet bulb temperature exceeds the design figure, the plant will need to ‘unload’ using some method to ensure the entire plant output is not lost. This could be by bypassing cold water around the bulk air cooler or short-circuiting ambient air directly into the intake airway.
- *Upset conditions*. If part of the plant capacity is lost or taken off-line during summer conditions, and if there is no way to reduce the refrigeration demand on the remaining plant, the remaining capacity will be exceeded resulting in overload and a domino effect leading to total plant shutdown. A ‘loss of plant capacity’ in this case could be a single compressor (for essential maintenance), or part of a cooling tower or bulk air cooler (eg for de-mudding of basins as part of a *legionella* control program).
- *Extremities of air and water flow rates*. Large swings in water flow rates through plate heat exchangers must be avoided due to the potential for fouling and, if the water is used in a BAC, due to ‘channelling’ of water through the fill in the BAC. Large swings in airflow through the BAC may also result in problems, although these are usually less serious than off-design water flows.
- *Backwash operation*. The entire water circuit through the plant needs consideration not only for normal operation, but also for backwash operation. Water circuits must also have

provision for make-up water (including disposal of natural make-up in the BAC due to condensation) and 'blowdown' or 'bleed-off' water in the condenser cooling tower. On critical plants, the backwash operation must be designed to avoid lengthy plant downtime. Backwash is typically triggered when either the water flow falls below an acceptable level, or the pressure drop across the heat exchanger increases to a pre-determined limit.

- *Winter shutdown operation*. The refrigeration plant may need to be shutdown in winter. The compressor sets, the BAC (if installed) and the CCT must be able to be 'winterised' so as to avoid damage to the equipment and to minimise the risk of ammonia leaks and fire in the dry and flammable fill materials during the extended shutdown.
- *Water hammer and vacuum*. Where plants have large water flow rates, water circuit start-up, shut-down and upset need very careful consideration in terms of hydraulic (water) hammer, air locks, and vacuum. PHEs are comprised of very thin plates and can easily be damaged by hydraulic shock or vacuum. Generally water flows should be designed so that, under shock conditions, there is no separation of the water (as flow separation results in partial or full vacuum[xx]), and suitable air reliefs and vacuum breakers should be installed. Specialist studies may be required.

REFRIGERANT CONTAINMENT

Most refrigerants are harmful and some are highly toxic (eg ammonia). Persons at risk from a refrigerant leak are not only those in near proximity at the time, but particularly for plants that chill intake air, the entire underground workforce. Well-designed and fail-safe containment systems, commensurate with the level of hazard, are required. These must handle the loss of either the liquid or gaseous refrigerant in the system, plus the component of the liquid refrigerant that 'flashes off' when liquid escapes. Suitable containment dams and water scrubbing arrangements are usually included for ammonia systems, along with ammonia sensors. Some of the factors that need to be considered include

- prevailing wind direction – siting the plant downwind from the intake to the mine,
- surface topography and infrastructure – air drainage and inversions, water drainage, access roadways and nearby offices need to be considered,
- elevation – where possible, the plant should be elevated above the intake airway collar,
- distance from a downcast shaft – the 'safe' distance depends on the mechanism of release and other factors, but as a general rule, the plant should not be located closer than 200 m to the mine air intake, or in close proximity to the intake of any mine compressed air surface compressors,
- surface bulk air cooler – in particular, whether all the intake air passes through the cooler or not,
- access – access must be restricted by suitable fencing and gates to authorised personnel only,
- water treatment equipment – water treatment chemicals can be toxic or corrosive and must be positioned away from instrumentation and control equipment, and properly bunded, and
- electrical equipment – ammonia rapidly corrodes copper and its alloys and sub-stations and the like should be situated outside any zone of influence.

Where the refrigerant being used is ammonia (which is typically the case in industrial plants, including mines), additional precautions are required as ammonia is not only toxic, but also flammable and explosive (within certain ranges).

It is critical that the containment systems are *never* taken off-line or electrically or mechanically isolated for maintenance whilst maintenance is also being carried out on the *refrigerant side* of the plant.

Containment systems work best when they are an integral part of the production of cold air. In this case, they *must* be operational whenever the plant itself is operating. Where containment systems are *unrelated* to plant operation ('add-ons'), maintenance may not be satisfactory which could result in them being ineffective if a major leak was to occur.

As a refrigerant leak is possible even when the plant is non-operational (eg during winter), the containment systems must remain operational while the plant is off-line for winter overhauls, unless the refrigerant is removed from the system to a safe storage location.

Pressure relief valves on the plant must also have their own containment systems to avoid refrigerant being vented to atmosphere.

INSTRUMENTATION AND PERFORMANCE TESTING

The normal range of electrical and mechanical instrumentation is required for a refrigeration plant. In addition, instrumentation regarding the thermodynamic performance of plant components (evaporator, condenser, mine load, compressor) and refrigerant containment/leakage is essential.

It is also critical to consider plant performance and acceptance testing criteria at design stage. These criteria must be written and included in the tender specification, both to ensure the suppliers agree with the basis for performance testing, and also to ensure that the necessary measurement arrangements are included in the plant design. Manual measurements of critical pressures and temperatures must be possible by the inclusion of suitable tapping points and thermometer wells, etc. Unlike other items of fixed plant, such as a mine winder, there is little control over some of the most important 'drivers' of plant performance when a refrigeration plant is tested. These are the ambient wet bulb temperature, and often the mine heat load. In practice, in average summer conditions, the plant may only be required to operate at 50 per cent to 70 per cent of its full load, (full load being only required for short periods of time each summer). Therefore an under-performing plant can easily meet this average summer condition. An agreed process must therefore exist to establish whether the plant meets performance criteria using the measurements obtained at these 'off-duty' points. Accurate instruments must also be available for the testing.

It is also useful to ensure that the plant control system can and is set up to monitor the necessary data for plant performance, prior to attempting any performance test.

Relevant Australian or other (eg British, USA, South Africa) standards can be consulted for guidelines on performance testing (ISO, 1968; ASHRAE, 1978; ARI, 1992). The enhanced Thorp method as developed by Bailey-McEwan (Bailey-McEwan, 1995a and 1995b) is also helpful in understanding the issues in measuring true plant performance.

SUMMARY

xx Water hammer is a phenomenon in which water flow in a pipe separates, typically due to a valve closing too quickly. This results in a decreasing pressure in the separated portion. This decreasing pressure allows water to 'boil off' at decreasing temperatures, and at full vacuum, water will rapidly boil off. At some point, the water column stops moving and, under the influence of the partial or complete vacuum, starts to move back rapidly in the reverse direction. The pressure in the separated section increases. The accelerating water column then 'hits' the closed valve resulting in the characteristic 'hammer'. The collapse of the water vapour 'bubbles' in the separated portion (which reverts from vapour to liquid as the pressure increases) can also result in a hammer effect. Full vacuum will occur when the columns of water separate by more than 10 vertical meters (corresponding to a vacuum of 100 kPa, ie completely negating normal atmospheric pressure).

Refrigeration plants incorporating screw compressors, ammonia refrigerant, plate heat exchangers for the evaporator and condenser with cooling towers for heat rejection provide an effective design for surface refrigeration plants in mines. However, there are numerous variations depending on the application.

Safety considerations in the choice of refrigerants, and the subsequent design of the safety and containment systems, are essential.

Fouling on both the water and refrigerant sides of heat exchangers, and in the packing material and design of cooling towers, must be taken into account in component selection.

Allowance must be made in the overall specification of the plant both for estimation errors in the mine heat loads and de-rating of individual components.

The extremities of plant operation and the particular needs of remote mining sites must be very carefully considered during the specification and design phases of the project.

There are two major control strategies for mine refrigeration plants, each with major advantages and disadvantages. The correct selection depends on the application.

Performance and acceptance testing must be specified and agreed with the tenderer prior to placing an order, due to the difficult nature of testing of refrigeration plants. This includes instrumentation and process monitoring requirements.

REFERENCES

ARI, 1992. *Centrifugal and Rotary Screw Water Chilling Packages,* Standard 550-92 (Air-conditioning and Refrigeration Institute: Arlington).

ASHRAE, 1978. *Methods of Testing Liquid Chilling Packages,* Standard 30-78 (American Society of Heating, Refrigeration and Air-Conditioning Engineers: New York).

ASHRAE, 1997. *ASHRAE Fundamentals (SI edition)* (American Society of Heating, Refrigeration and Air-conditioning Engineers: Atlanta).

Bailey-McEwan, M, 1995a. An enhanced version of the Thorp Method of verifying surveys, in *J. Mine Vent Soc South Africa,* Vol 1, January: 1-24.

Bailey-McEwan, M, 1995b. Extension of the Enhanced Thorp Method beyond conventional water chilling machines operating normally, in *J. Mine Vent Soc South Africa*, February: 2-12.

Brake, D J, 2001. The application of refrigeration in mechanised mines, *Proc Australas Inst Min Metall,* 306(1):1-9.

Brake, D J and Fulker, R, 2000. The ventilation and refrigeration design for Australia's deepest and hottest underground operation: the Enterprise mine, in *Proceedings MassMin 2000*, pp 611-621 (The Australasian Institute of Mining and Metallurgy: Melbourne).

ISO, 1968. Recommendation 916. Testing of Refrigeration Systems (International Organisation for Standardisation: Geneva).

Hancock, W, 1926. Local air conditioning underground by means of refrigeration, *Trans IME*, 72:342-366.

Hartman, H L, Mutmansky, J M, Ramani, R V and Wang, Y J, 1997. *Mine Ventilation and Air Conditioning, 3rd Edition,* pp 585-586 (John Wiley and Sons: New York).

Howakowski, G A and Busby, R L, 2001. Advances in Natural Gas Cooling, *ASHRAE Journal*, April: 47-52.

McPherson, M J, 1993. *Subsurface Ventilation and Environmental Engineering*, p 378 ff (Chapman and Hall: New York).

Standards Association of Australia, 1998. AS1677, *Refrigeration Systems* (Standards Association of Australia: Sydney).

Stoeker, W F and Jones, J W, 1982. *Refrigeration and Air Conditioning*, pp 298-299 (McGraw-Hill).