MINE AIR CONDITIONING AND VENTILATION

IN underground mines, excess humidity, high temperatures, inadequate oxygen, and excessive concentrations of dangerous gases can lower worker productivity and can cause illness and death. Air cooling and ventilation are needed in deep underground mines to minimize heat stress and remove contaminants. As mines become deeper, heat removal and ventilation problems become more difficult and costly to solve.

Caution: This chapter presents only a very brief overview of the principles of mine ventilation planning. The person responsible for such planning should either be an experienced engineer, or work under the direct supervision of such an engineer. Seven English-language texts have been written on mine ventilation since 1980 (Bossard 1982; Hall 1981; Hartman et al. 1997; Hemp 1982; Kennedy 1996; McPherson 1993; Tien 1999). The ventilation engineer is strongly encouraged to study these references.

DEFINITIONS

Definitions specific to mine air conditioning and ventilation are as follows.

Heat stress is a qualitative assessment of the work environment based on temperature, humidity, air velocity, and radiant energy. Many heat stress indices have been proposed (see Chapter 8 of the 2001 *ASHRAE Handbook—Fundamentals* for a thorough discussion); the most common in the mining industry are effective temperature (Hartman et al. 1997), air cooling power (Howes and Nixon 1997), and wet-bulb temperature. The following wet-bulb temperature ranges were derived from experience at several deep western U.S. metal mines:

$$t_{wb} \leq 80°F \quad \text{Worker efficiency 100\%}$$
$$80 < t_{wb} \leq 85°F \quad \text{Economic range for acclimatized workers}$$
$$85 < t_{wb} \leq 91°F \quad \text{Safety factor range; corrective action required}$$
$$91°F < t_{wb} \quad \text{Only short-duration work with adequate breaks}$$

Heat strain is the physiological response to heat stress. Effects include sweating, increased heart rate, fatigue, cramps, and progressively worsening illness up to heat stroke. Individuals have different tolerance levels for heat.

Reject temperature, based on the heat stress/strain relationship is the wet-bulb temperature at which air should be rejected to exhaust or recooled. Reject temperature ranges between 78 and 85°F wet bulb, depending on governmental regulation, air velocity, and expected metabolic heat generation rate of workers. Specifying the reject temperature is one of the first steps in planning air-conditioning systems. The ventilation engineer must be able to justify the reject temperature to management because of the economics involved. If too high, work productivity, health, safety, and morale suffer; if too low, capital and operating costs become excessive.

Critical ventilation depth is the depth at which the air temperature in the intake shaft rises to the reject temperature through autocompression and shaft heat loads. Work areas below the critical ventilation depth rely totally on air conditioning to remove heat. The critical ventilation depth is reached at about 8000 to 10,000 ft, depending on surface climate in the summer, geothermal gradient, and shaft heat loads such as pump systems.

Base heat load is calculated at an infinite airflow at the reject temperature passing through the work area. The temperature of an infinite airflow will not increase as air picks up heat. **Actual heat load** is measured or calculated at the average stope temperature. It is always greater than the base heat load because the average stope temperature is lower than the reject temperature. More heat is drawn from the wall rock. **Marginal heat load** is the difference between base and actual heat loads. It is the penalty paid for using less than an infinite airflow (i.e., the lower the airflow, the lower the inlet temperature required to maintain the reject and the higher the heat load).

Temperature-dependent heat sources (TDHs) depend on the temperature difference between the source and air. Examples include wall rock, broken rock, and fissure water (in a ditch or pipe). **Temperature-independent heat sources (TIHs)** depend only on the energy input to a machine or device after the energy required to raise the potential energy of a substance, if any, is deducted. Examples include electric motors, lights, substation losses, and the calorific value of diesel fuel.

Passive thermal environmental control separates heat sources from ventilating airflows. Examples include insulating pipes and wall rock, and blocking off inactive areas. **Active thermal environmental control** removes heat via airflow and air conditioning quickly enough so that air temperature does not rise above the reject.

Positional efficiency, an important design parameter for mine cooling systems, is the cooling effect reaching the work area divided by the machine evaporator duty. The greater the distance between the machine and work area, the more heat that the cooling medium (air or water) picks up en route.

Percent utilization is the ratio of the evaporator duty of the refrigeration plant over a year in energy units to the duty if the plant had worked the entire year at 100% load. This consideration becomes important when evaluating surface versus underground plants.

Coefficient of performance (COP) usually is defined as the evaporator duty divided by the work of compression in similar units. In mines, the overall COP is used: the evaporator duty divided by all power-consuming devices needed to deliver cooling to the work sites. This includes pumps and fans as well as refrigeration machine compressors.

A **shaft** is a vertical opening or steep incline equipped with skips to hoist the ore, and cages (elevators) to move personnel and supplies. Electric cables and pipes for fresh water, compressed air, cooling water, and pump water are installed in shafts. **Drifts** and **tunnels** are both horizontal openings; a tunnel opens to daylight on both ends, whereas a drift, deep underground, does not. In metal mining, a **stope** is a production site where ore is actually mined. In coal mining, coal is usually produced by either longwall (one continuous production face hundreds of feet long) or room-and-pillar (multiple

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production faces in a grid of rooms with supporting pillars in between) methods.

SOURCES OF HEAT ENTERING MINE AIR

Adiabatic Compression

Air descending a shaft increases in pressure (because of the mass of air above it) and temperature. As air flows down a shaft, it increases in temperature as if compressed in a compressor because of conversion of potential energy to internal energy, even if there is no heat interchange with the shaft and no evaporation of moisture.

For dry air at standard conditions (59°F at 14.696 psia), the specific heat at constant pressure c_p is 0.24 Btu/lb·°F. For most work, c_p can be assumed constant, but extreme conditions might warrant a more precise calculation: 1 Btu is added (for descending airflow) or subtracted (for exhaust) to each pound for every 778 ft. The dry-bulb temperature change is $1/(0.24 \times 778 \times 1) = 0.00535$°F per foot, or 1°F per 187 ft of elevation. The specific heat for water vapor is 0.45 Btu/lb·°F. So, for constant air-vapor mixtures, the change in dry-bulb temperature is $(1 + W)/(0.24 + 0.45W)$ per 778 ft of elevation, where W is the humidity ratio in pounds of water per pound of dry air.

The theoretical heat load imposed on intake air by adiabatic compression is given in Equation (1), which is a simplified form of the general energy equation:

$$q = 60Q\rho E \,\Delta d \qquad (1)$$

where

    q = theoretical heat of autocompression, Btu/h
   60 = 60 min/h
    Q = airflow in shaft, cfm
    ρ = air density, lb/ft^3
    E = energy added per unit distance of elevation change, 1 Btu/778 ft·lb
  Δd = elevation change, ft

Example 1. What is the equivalent heat load from adiabatic compression of 300,000 cfm at 0.070 lb/ft^3 density flowing down a 5000 ft shaft?

Solution:

$$q = (60)(300,000)(0.070)(1/778)(5000) = 8,097,686 \text{ Btu/h}$$

The adiabatic compression process is seldom truly adiabatic: *autocompression* is a more appropriate term. Other heating or cooling sources, such as shaft wall rock, introduction of groundwater or water sprayed in the shaft to wet the guides, compressed-air and water pipes, or electrical facilities, often mask the effects of adiabatic compression. The actual temperature increase for air descending a shaft usually does not match the theoretical adiabatic temperature increase, for the following reasons:

- The effect of seasonal and daily surface temperature fluctuations, such as cool night air on the rock or shaft lining (rock exhibits thermal inertia, which absorbs and releases heat at different times of the day)
- The temperature gradient of ground rock related to depth
- Evaporation of moisture within the shaft, which suppresses the dry-bulb temperature rise while increasing the moisture content of the air

The wet-bulb temperature lapse rate varies, depending on the entering temperature and humidity ratio, and the pressure drop in the shaft. It averages about 2.5°F wet bulb per 1000 ft, and is much less sensitive to evaporation or condensation than the dry bulb.

Electromechanical Equipment

Electric motors and diesel engines transfer heat to the air. Loss components of substations, electric input to devices such as lights, and all energy used on a horizontal plane appear as heat added to the mine air. Energy expended in pumps, conveyors, and hoists to increase the potential energy of a material does not appear as heat, after losses are deducted.

Vehicles with electric drives, such as scoop-trams, trucks, and electric-hydraulic drill jumbos, release heat into the mine at a rate equivalent to the nameplate and a utilization factor. For example, a 150 hp electric loader operated at 80% of nameplate for 12 h a day liberates (150 hp)(42.4 Btu/min·hp)(0.80) (12 h/day)(60 min/h) = 3,663,360 Btu/day. Dividing by 24 h/day gives an average heat load over the day of 152,640 Btu/h. During the 12 h the loader is operating, the heat load is doubled to 305,280 Btu/h. The dilemma for the ventilation engineer is that, if heat loads are projected at the 152,640 rate, the stope temperature will exceed the reject temperature for half the day, and the stope will be overventilated for the other half; if projected at 305,380 Btu/h, the stope will be greatly overventilated when the loader is not present. Current practice is to accept the additional heat load while the loader is present. Operators get some relief when they leave the heading to dump rock, at which time the ventilation system can partially purge the heading.

Diesel equipment dissipates about 90% of the heat value of the fuel consumed, or 125,000 Btu/gal, to the air as heat (Bossard 1982). The heat flow rate is about three times higher for a diesel engine than for an equivalent electric motor. If the same 150 hp loader discussed previously were diesel-powered, the heat would average about 458,000 Btu/h over the day, and 916,000 Btu/h during actual loader operation. Both sensible and latent heat components of the air are increased because combustion produces water vapor. If a wet scrubber is used, exhaust gases are cooled by adiabatic saturation and the latent heat component increases even further.

Fans raise the air temperature about 0.45°F per in. of water static pressure. Pressures up to 10 in. of water are common in mine ventilation. This is detrimental only when fans are located on the intake side of work areas or circuits.

Groundwater

Transport of heat by groundwater has the largest variance in mine heat loads, ranging from essentially zero to overwhelming values. Groundwater usually has the same temperature as the virgin rock. Ventilating airflows can pick up more heat from hot drain water in an uncovered ditch than from wall rock. Thus, hot drain water should be stopped at its source or contained in pipelines or in covered ditches. Pipelines can be insulated, but the main goal is isolating the hot water so that evaporation cannot occur.

Heat release from open ditches increases in significance as airways age and heat flow from surrounding rock decreases. In one Montana mine, water in an open ditch was 40°F cooler than when it flowed out of the wall rock; the heat was transferred to the air. Evaporation of water from wall rock surfaces lowers the surface temperature of the rock, which increases the temperature gradient of the rock, depresses the dry-bulb temperature of the air, and allows more heat to flow from the rock. Most of this extra heat is expended in evaporation.

Example 2. Water leaks from a rock fissure at 20 gpm and 125°F. If the water enters the shaft sump at 85°F, what is the rate of heat transfer to the air?

Solution:

$$\text{Heat rate} = (20 \text{ gpm})(60 \text{ min/h})(8.33 \text{ lb/gal})(1 \text{ Btu/lb·°F})$$
$$\times (125 - 85\text{°F}) = 399,840 \text{ Btu/h}$$

Wall Rock Heat Flow

Wall rock is the main heat source in most deep mines. Temperature at the earth's core has been estimated to be about 10,300°F. Heat flows from the core to the surface at an average of 0.022 Btu/h·ft^2. The implication for mine engineers is that a geothermal gradient exists: rock gets warmer as the mine deepens. The actual gradient

Table 1 Maximum Virgin Rock Temperatures

Mining District	Depth, ft	Temperature, °F
Kolar Gold Field, India	11,000	152
South Africa	12,000	135
Morro Velho, Brazil	8,000	130
North Broken Hill, Australia	3,530	112
Great Britain	4,000	114
Braloroe, BC, Canada	4,100	112.5
Kirkland Lake, Ontario	6,000	81
Falconbridge Mine, Ontario	6,000	84
Lockerby Mine, Ontario	4,000	96
Levac Borehold (Inco), Ontario	10,000	128
Garson Mine, Ontario	5,000	78
Lake Shore Mine, Ontario	6,000	73
Hollinger Mine, Ontario	4,000	58
Creighton Mine, Ontario	10,000	138
Superior, AZ	4,000	140
San Manuel, AZ	4,500	118
Butte, MT	5,200	150
Homestake Mine, SD	8,000	134
Ambrosia Lake, NM	4,000	140
Brunswick #12, New Brunswick, Canada	3,700	73
Belle Island Salt Mine, LA	1,400	88

Source: Fenton (1972).

Table 2 Thermal Properties of Rock Types

Rock Type	Thermal Conductivity, Btu/h·ft·°F	Diffusivity, ft²/h
Coal	1.27	0.050
Gabro	1.37	0.092
Granite	1.11	0.129
Pyritic shale	2.11	0.078
Quartzite	3.18	0.090
Sandstone	1.14	0.065
Shale	1.38	0.035
Rhyolite	2.00	0.043
Sudbury ore	1.50	0.049
North Idaho metamorphic	2.95	0.109

Source: Mine Ventilation Services Inc., Fresno, CA. Reprinted with permission.

varies from approximately 0.5 to over 4°F per 100 ft of depth, depending on the thermal conductivity of local rock. Table 1 gives depths and maximum **virgin rock temperatures (VRTs)** for various mining districts. Table 2 gives thermal conductivities and diffusivities for rock types commonly found in mining. These two variables are required for wall rock heat flow analysis.

Wall rock heat flow is unsteady-state: it decays with time because of the insulating effect of cooled rock near the rock/air boundary. Equations exist for both cylindrical and planar openings, but this section discusses cylindrical equations (Goch and Patterson 1940). The method can solve for either instantaneous or average heat flux rate. The instantaneous rate is recommended because it is better used for older tunnels or drifts. For newer drifts, a series of instantaneous rates over short time periods is equivalent to the average rate. The Goch and Patterson calculations are easily performed on a computer using the following variables and equations:

$$\text{Fo} = \frac{\alpha\theta}{r^2} \qquad (2)$$

$$\varepsilon = \{1.017 + 0.7288\log_{10}(\text{Fo}) + 0.1459\left[\log_{10}(\text{Fo})\right]^2$$

$$- 0.01572\left[\log_{10}(\text{Fo})\right]^3 - 0.004525\left[\log_{10}(\text{Fo})\right]^4 \qquad (3)$$

$$+ 0.001073\left[\log_{10}(\text{Fo})\right]^5 \}^{-1}$$

$$\text{Heat Flux, Btu/h·ft}^2 = \frac{k(t_{vr} - t_a)(\varepsilon)}{r} \qquad (4)$$

$$\text{Total Heat Flow, Btu/h} = (\text{Heat Flux})(L)(P) \qquad (5)$$

where

Fo = Fourier number, dimensionless
k = thermal conductivity of rock, Btu/h·ft·°F
L = length of section, ft
P = perimeter of section, ft
r = radius of circular section, ft, or equivalent radius of rectangular section; $r = (A/\pi)^{1/2}$, where
  A = cross-sectional area of section, ft²
t_a = air dry-bulb temperature, °F
t_{vr} = virgin rock temperature, °F
α = thermal diffusivity of rock (equals $k/\rho c$), ft²/h, where
  ρ = rock density, lb/ft³
  c = heat capacity, Btu/lb·°F
ε = function of Fourier number for instantaneous rate, dimensionless (Whillier and Thorpe 1982)
θ = average age of section, h

Example 3. A 500 ft long section of drift, 12 ft high by 15 ft wide, was driven in quartzite with a VRT of 110°F. The drift was started 20 days before the face was reached, and the face is 1 day old. One design criterion is keeping the average dry-bulb temperature of the air in the drift at 80°F. How much heat will flow into the section?

Solution: From Table 2, the thermal conductivity of quartzite is 3.18 Btu/h·ft·°F and the diffusivity is 0.090 ft²/h. The average age of the section is $(20 + 1 \text{ days})/2 = 10.5$ days, or 252 h. The cross-sectional area of the drift is $12 \times 15 = 180$ ft² and the perimeter is $(12 + 15) \times 2 = 54$ ft. The equivalent radius of the drift is $(180/\pi)^{1/2} = 7.57$ ft. The following equations are then applied:

Using Equation (2), $\text{Fo} = \dfrac{\alpha\theta}{r^2} = \dfrac{(0.090)(252)}{7.57^2} = 0.396$

Using Equation (3), $\varepsilon = 1.336$

Using Equation (4),

$$\text{Heat Flux} = \frac{k(t_{vr} - t_a)(\varepsilon)}{r} = \frac{(3.18)(110 - 80)(1.336)}{7.57}$$

$$= 16.84 \text{ Btu/h·ft}^2$$

Using Equation (5),

$$\text{Total Heat Flow} = (\text{Heat Flux})(L)(P)$$

$$= (16.84)(500)(54) = 454{,}700 \text{ Btu/h}$$

Thus, to keep the average temperature of the drift section at 80°F db, 37.9 tons of refrigeration are needed.

The Goch and Patterson method lacks a convective heat transfer coefficient at the rock/air boundary, and overestimates heat transfer in a dry drift by 8 to 15%. It also does not have a wetness factor. Because a drift with water on the perimeter draws more heat from wall rock, the method underestimates heat flow. Almost all drifts have some wetness on the floor, back, and side walls, though it may not be visible. Comparisons of the Goch and Patterson method with field measurements and results from commercial software under typical conditions (a drift with 20 to 60% of the perimeter wetted) indicate that the overestimate is nearly equal to the underestimate. When using Goch and Patterson for drift heat loads, keep drift section lengths under 200 ft and do not apply any contingency factor to the calculated heat load.

Heat load calculations for stoping require a large number of variables. Irregular shapes, sporadic advance rates, intermittent TIH sources, fissure water, and nonhomogeneous or anisotropic (with directionally differing heat conducting properties) rock are difficult to model (Duckworth and Mousset-Jones 1993; Marks and Shaffner 1993). For cut-and-fill stoping with a sand floor, measured heat

loads are about 70% of the heat loads predicted by Goch and Patterson. Other stoping methods such as room-and-pillar or tabular reef mining are more amenable to planar heat load equations. Patterson (1992) gives empirical graphs relating heat load to productivity and depth.

Ventilation engineers needing to project heat loads for new mines or extensive tunnel projects can write their own computer program using the Goch and Patterson equations, or use a commercial software package. These programs account for convective heat transfer, wetness, elevation changes, and TIH sources that can make hand calculations tedious. However, program input must be carefully derived or the output will be misleading.

Heat from Broken Rock

Freshly blasted broken rock can liberate significant amounts of heat in a confined area. The broken rock's initial and final temperatures, and where the rock is cooled en route from the face to the hoisting facility, must be estimated.

$$\text{Heat, Btu} = (\text{mass})(\text{specific heat})(\text{VRT} - \text{final temperature}) \quad (6)$$

$$\text{Heat load, Btu/h} = \frac{\text{Heat}}{(\text{time, h})} \quad (7)$$

Example 4. A 12 ft high by 15 ft wide by 10 ft long drift round is blasted in quartzite where the VRT is 120°F. Quartzite has a 168 lb/ft³ density and a 0.2 Btu/lb·°F specific heat. By the time the rock is hoisted to the surface 4 h later, it has cooled to 90°F. What is the heat load imposed on the drift and shaft?

Solution:

$$\text{Heat} = (12 \text{ ft} \times 15 \text{ ft} \times 10 \text{ ft} \times 168 \text{ lb/ft}^3)(0.2 \text{ Btu/lb·°F})$$
$$\times (120 - 90°F) = 1.8144 \times 10^6 \text{ Btu}$$

$$\text{Heat load} = \frac{1.8144 \times 10^6 \text{ Btu}}{4 \text{ h}} = 453,600 \text{ Btu/h}$$

Heat from Other Sources

Heat produced by oxidation of timber and sulfide minerals can be locally significant and even cause mine fires. Fortunately, timber is seldom used for ground support in modern mines. Heat from blasting can also be appreciable. The typical heat potential in various explosives is similar to that of 60% dynamite, about 1800 Btu/lb. This heat is usually swept out of the mine between shifts and thus is not tallied in heat load projections. Body metabolism is only a concern in refuge chambers and is rarely if ever included in heat load projections. Although these heat sources are usually neglected, the ventilation engineer must remain vigilant for cases where local effects might be significant.

Summation of Mine Heat Loads

Mine cooling requirements should be estimated after mining methods, work sites, production rates, and equipment are specified, and heat sources identified. The time frame, during which the ventilation and cooling systems must provide an acceptable work environment, is normally 10 years, but can vary.

Total heat for a mine or mine section is the summation of all TIH and TDH sources. It helps to plot heat sources on a schematic. The heat load from the surface to the entrance of the stope is assessed first, starting with TIH sources because they influence TDH sources. Shaft heat loads, autocompression, and drift heat loads are added to the air en route to stopes. The process should take only one iteration to find a stope entering temperature. Stope heat load is calculated by assuming that the wet bulb leaving the stope equals the design reject temperature. The air temperature entering the stope is estimated, and heat load equations are used to

calculate the exit temperature. If this exit temperature exceeds the reject temperature, a lower stope entering temperature is assumed and a new exit temperature is calculated. The process is repeated with new stope entering temperatures until the calculated stope exit temperature equals the design reject temperature.

If the entering stope temperature calculated from the surface is greater than the entering stope temperature calculated from the reject temperature, higher airflow or air conditioning will be needed. Psychrometrics can determine the size of the airflow increase or cooling required.

HEAT EXCHANGERS

Underground heat exchangers can be water-to-refrigerant, air-to-refrigerant, water-to-water, air-to-water, or air-to-air. Brine can be used instead of water where freezing might occur. Heat exchangers can be direct (e.g., spray chambers) or indirect (e.g., conductive heat transfer through tubes or plates).

See Chapters 16 to 25 and 34 to 44 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* for general guidelines when designing large cooling plants for mine duty.

Shell-and-Tube and Plate Heat Exchangers

Shell-and-tube heat exchangers are the mainstay of refrigeration machines used in mines. Machines in the 200 to 400 ton range may use either direct expansion (DX) or flooded evaporators. In both cases, the working fluid (refrigerant or water) is circulated through the tubes.

South African mines often use plate-and-frame evaporators in large surface chilled-water plants (van der Walt and van Rensburg 1988). These machines can cool water to within a degree of freezing without danger of rupture. In contrast, shell-and-tube evaporators should not be expected to chill water below 38°F. Manufacturers must be consulted.

Shell-and-tube water-to-water heat exchangers have been used in mine cooling systems to avoid pumping return water against high heads (the U-tube effect). Chilled water from the surface is sent down to the high-pressure (tube) side of the exchanger. Water on the low-pressure side operates district chiller systems or spot coolers. The shell-and-tube water-to-water heat method is not very popular, perhaps because it requires high-pressure supply and return piping. The Second Law of Thermodynamics limits the approach temperature of the outlet high-pressure water to the inlet low-pressure water temperature. This tends to limit heat removal in deep mines that would require at least three heat exchanger stations, in series, in the shaft.

Cooling Coils

Cooling coils can be DX or chilled-water coils. DX coils are used with spot coolers and typically range from 15 to 60 tons. Some modern spot coolers use dual coils in parallel for compactness. Chilled-water coils, used in district chiller systems, are also used in a wide range of sizes.

Air-side fouling is the main operational problem with cooling coils in mines. Coils with fin spacing tighter than 6 fins per inch are not recommended. Water-side fouling is minimal if the water is of fair quality, the circuit is closed, and a corrosion inhibitor is added to the circuit.

Small Spray Chambers

Small spray chambers can be used as an alternative to cooling coils. Heat transfer is direct, air-to-water. Spray chamber maintenance is minimal, and the amount of water sprayed is typically one-half to one-third that required for a cooling coil at the same duty. Some washing effect also occurs in the chamber.

Spray chambers are open systems that dump water into a ditch or collection pond after spraying. This water drains to the dewatering

system or is pumped back to the chiller plant. Small spray chambers are still popular for small duties in mines that chill service water, but the pumping system must be able to handle the increased service water requirement.

Cooling Towers

When heat loads are large, the full capacity of a mine's heat removal system will probably be needed. A key component of the heat removal system is exhaust air. The ventilation engineer for a deep, hot mine must be proficient at designing underground cooling towers for condenser heat rejection, and spray chambers for cooling airflows.

Rather than using the standard HVAC&R method of assessing cooling tower performance, as described in Chapter 36 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*, mining engineers use the South African factor-of-merit method, developed in the 1970s for designing direct-contact heat exchangers (Bluhm 1981; Burrows 1982; Whillier 1977). This method requires a psychrometric program and the following equations:

$$\Sigma = h - (c_{pw})(W)(t) \tag{8}$$

Note that t is either the wet-bulb temperature for air, or the water temperature, depending on whether Σ_{ai} or Σ_{wi} is calculated.

$$q = (M_w)(c_{pw})(t_{wi} - t_{wo}) \tag{9}$$

$$q = (M_a)(\Sigma_{ao} - \Sigma_{ai}) \tag{10}$$

$$n_w = \frac{t_{wi} - t_{wo}}{t_{wi} - t_{ai}} \tag{11}$$

$$R = \frac{(M_w)(c_{pw})(t_{wi} - t_{ai})}{(M_a)(\Sigma_{wi} - \Sigma_{ai})} \tag{12}$$

$$N = \frac{F}{(1 - F)(R^{0.4})} \tag{13}$$

$$n_w = \frac{(1 - e^{-N(1 - R)})}{(1 - Re^{-N(1 - R)})} \text{ (for counterflow towers)} \tag{14}$$

where

a = air
c_{pw} = specific heat of water at constant pressure, 1 Btu/lb·°F
F = factor of merit, roughly equivalent to UA factor in conductive heat transfer, dimensionless; ranges from 0 (no heat transfer) to 1 (as much as heat transfer as allowed by Second Law takes place)
h = enthalpy of moist air, Btu/lb
i = inlet
M = mass flow rate, water or air, lb/min
N = number of transfer units, an intermediate factor for calculating water efficiency
n_w = water efficiency, dimensionless
o = outlet
q = heat rate to be transferred in chamber, Btu/lb
R = tower capacity factor, dimensionless; ratio of heat capacity of water to heat capacity of air under limits of Second Law
t = temperature, °F
W = humidity ratio of moist air, lb water per lb air
w = water
Σ = energy of air, Btu/lb; total enthalpy minus enthalpy of liquid water evaporated into air [approximated by $(c_{pw})(W)(t)$, where t is wet bulb]; dependent only on wet bulb and barometric pressure

Designing an underground cooling tower requires the exhaust air mass flowrate M_a, the wet- and dry-bulb temperatures available at

the tower, and the ambient barometric pressure. A psychrometric program is needed to calculate enthalpy, density, humidity ratio, and specific volume from the wet bulb, dry bulb, and barometric pressure. Tower airflow is taken from measurements or from the mine plan. The temperature, if not measurable (e.g., in a new mine), can be assumed to approach the reject temperature. Experience shows that air usually enters exhaust at about 82 to 83°F wb. Then,

1. Calculate heat rejection rate in the tower (evaporator duty × condenser heat rejection factor, typically between 1.2 and 1.4). Discuss with the manufacturer.
2. Select condenser water flow and use Equation (9) to calculate Δt_w in the tower and machine condensers.
3. Specify cooling tower diameter and tower air velocity.
4. Calculate M_w/M_a.
5. Calculate heat rejection rate per cfm in tower.
6. Calculate air enthalpy h and use Equation (8) to calculate Σ_{ai}.
7. Select a factor of merit for tower using Table 3.
8. Estimate tower capacity factor R (e.g., $R = 0.5$). Using $R = 1$ in Equation (14) will result in division by zero. Skip to Step 10 if $R = 1$, and use the value of F for n_w.
9. Use Equation (13) to calculate N.
10. Use Equation (14) to calculate water efficiency n_w.
11. Use Equation (11) to calculate inlet water temperature t_{wi}.
12. Use Equation (8) to calculate Σ_{wi}, the energy of air at inlet water temperature t_{wi}.
13. Use Equation (12) to calculate a new tower capacity factor R.
14. Compare the R calculated in Step 13 with the R estimated in Step 8. If different by more than 1%, return to Step 8 and reestimate R. Repeat Steps 9 to 14 until the calculated R is within 1% of the estimated R.
15. Calculate air and water temperatures leaving the tower and the evaporation rate.

Keep the following empirical design criteria in mind during Steps 1 to 5:

- Realistic Δt_w in tower is 12 to 16°F
- Realistic water loading in tower is 6 to 8 gpm per square foot
- Optimum water velocity in machine condenser tubes (3 to 13 fpm, per manufacturer's recommendations) is based on tubing material and water quality
- Realistic maximum air velocity in tower is 1600 fpm
- Realistic ratios of the mass flows of water to air M_w/M_a range from 0.5 to 2.5
- A realistic heat rejection rate in tower is 32 to 65 Btu/h per cfm

Values outside these design parameters are sometimes used (especially when plant duty is increased at a future date), but the penalty paid is a higher condensing temperature and lower COP. Once q, F, M_a, t_{ai}, and M_w are specified, only one t_{wi}, t_{wo}, and t_{ao} will balance all equations.

Example 5. Design a cooling tower for a 1000 ton refrigeration plant planned for a deep, hot mine. Exhaust airflow for heat rejection is

Table 3 Factors of Merit

	Factor of Merit Range	Source
Vertical counterflow, open, unpacked	0.50 to 0.70	Hemp 1982
Horizontal cross-flow		
Single-stage	0.40 to 0.55	Hemp 1982
Two-stage	0.57 to 0.72	Marks 1988
Three-stage	0.69 to 0.81	Marks 1988
Four-stage	0.76 to 0.87	Marks 1988
Commercial packed cooling		
Counterflow tower	0.68 to 0.78	Patterson 1992
Cross-flow tower	0.55 to 0.65	Patterson 1992

250,000 cfm at 83°F saturated. Barometric pressure is 15.226 psia (31 in. of mercury, or 1000 ft below sea level). What size cooling tower is needed, how much condenser cooling water is required, what are the inlet and outlet air and water temperatures, and how much makeup water is needed?

Solution:

Step 1. For a refrigeration plant to produce 1000 tons of cooling, it must reject about 1000 tons × 12,000 Btu/h·ton × 1.25 condenser heat rejection factor = 15,000,000 Btu/h.

Step 2. Select a condenser water flow. For this example, start with 1 gpm per 12,000 Btu/h rejected. The condenser flow is thus 1250 gpm, or 10,413 lb/min at 8.33 lb/gal. The change in water temperature is calculated from Equation (9):

$$\Delta t_w = \frac{15,000,000 \text{ Btu/h}}{(10,413 \text{ lb/min})(60 \text{ min/h})(1 \text{ Btu/lb·°F})} = 24°F$$

That exceeds the realistic 12 to 16°F Δt_w, so arbitrarily increase the water flow to 2000 gpm and recalculate Δt_w. In practice, selecting the condenser water flow is anything but arbitrary. Generally, the higher the flow, the better, but higher flows require larger condensers to keep tube velocity within design limits, larger cooling towers with more nozzles, and significantly larger pumps. Actual condenser water flow is a compromise between machine and tower performance, capital cost, and overall plant COP (operating cost). At 2000 gpm for this example, $\Delta t_w = 15°F$, which is acceptable.

Step 3. Specify cooling tower diameter by using a midrange value to 12 gpm/ft^2.

$$(\pi/4)d^2 = \frac{2000 \text{ gpm}}{12 \text{ gpm/ft}^2} \qquad \therefore \quad d = 14.57 \text{ ft} \approx 15 \text{ ft}$$

$$\text{Air velocity} = \frac{250,000 \text{ cfm}}{(\pi/4)(15^2)}$$

$$= 1415 \text{ fpm} (< 1600 \text{ fpm; acceptable})$$

Step 4. Calculate M_w/M_a.

$$M_w = (2000 \text{ gpm})(8.33 \text{ lb/gal}) = 16,660 \text{ lb}_w/\text{min}$$

The specific volume for 83°F saturated inlet air at 15.226 psia is 13.71 ft^3/lb$_a$. M_a is therefore

$$M_a = \frac{250,000 \text{ cfm}}{13.71 \text{ ft}^3/\text{lb}_a} = 18,235 \text{ lb}_a/\text{min}$$

$$M_w/M_a = \frac{16,660}{18,235} = 0.914 \quad (0.5 < M_w/M_a < 2.5; \text{ acceptable})$$

Step 5. The heat rejection rate in the tower is

$$\frac{15,000,000 \text{ Btu/h}}{250,000 \text{ cfm}} = 60 \text{ Btu/h per cfm}$$

This rate is approaching the upper acceptable limit. Consideration should be given to routing more air through the tower if possible. All design criteria have now been met.

Step 6. Σ_{ai} at 83°F wb and 15.226 psia = 45.95 – (1)(0.0237)(83) = 43.98 Btu/lb.

Step 7. Select a factor of merit for the tower. From Table 3, an open, unpacked, vertical counterflow cooling tower can conservatively be expected to have a 0.55 factor of merit. If the tower is well designed and actually has a higher factor, the tower will return cooler water to the plant and COP will increase.

Step 8. Estimate $R = 0.5$ (first pass).

Step 9. Calculate N from Equation (13):

$$N = \frac{0.55}{(1 - 0.55)(0.5^{0.4})} = 1.613$$

Step 10. Calculate n_w from Equation (14):

$$n_w = \frac{1 - e^{-1.613(1 - 0.5)}}{1 - 0.5e^{-1.613(1 - 0.5)}} = 0.713$$

Step 11. Calculate t_{wi} from Equation (11) (after manipulation, and assuming that $t_{wi} - t_{wo} = \Delta t_w$):

$$t_{wi} = \frac{\Delta t_w}{n_w} + t_{ai} = \frac{15}{0.713} + 83 = 104.04°F$$

Step 12. Σ_{wi} at 104.04°F and 15.226 psia = 77.11 – (1)(0.047) (104.04) = 72.22 Btu/lb.

Step 13. Calculate the new R using Equation (12):

$$R = \frac{(16,660)(1)(104.04 - 83)}{(18,235)(72.22 - 43.98)} = 0.681$$

Step 14. The new R is higher than the 0.5 R estimated in Step 8. Return to Step 8 and iterate until the R calculated in Step 13 equals the R projected in Step 8. This occurs at $R = 0.662$.

Step 15. All other values can now be calculated.

Per Step 11, $t_{wi} = 106.09°F$

$$t_{wo} = 106.09 - 15 = 91.09°F$$

$$\Sigma_{ao} = \Sigma_{ai} + \frac{15,000,000 \text{ Btu/h}}{(18,235 \text{ lb/min})(60 \text{ min/h})}$$

$$= 43.98 + 13.71 = 57.69 \text{ Btu/lb}$$

$$t_{ao} = 94.5°F \text{ (via psychrometric iteration)}$$

The water evaporated in the tower is the difference in humidity ratios ΔW × the mass flow of dry air. From psychrometric equations, $W_{83°F} = 0.0237 \text{ lb}_w/\text{lb}_a$ and $W_{94.5°F} = 0.0347 \text{ lb}_w/\text{lb}_a$.

$$\text{Evaporation rate} = (18,235 \text{ lb/min})\frac{0.0347 - 0.0237 \text{ lb}_w/\text{lb}_a}{8.33 \text{ lb}_w/\text{gal}}$$

$$= 24.1 \text{ gpm}$$

Total makeup water depends on evaporation rate, water carryover (if any), and blowdown used to control dissolved solids in the condenser circuit. Leakages and carryover can be deducted from the blowdown. Makeup water is usually planned at 1 to 3% of the condenser water flow, depending on the quality of the makeup water, allowable cycles of concentration of dissolved solids, and water treatment plan.

Vertical unpacked cooling towers in mines often use clog-resistant full-cone nozzles circling the top of the tower, at least 40 ft above the pond. South African mines tend to use ham-type sprayers. Nozzle pressure of 30 psig is typically specified: lower water pressures do not generate the fine water droplets preferred for heat transfer, and higher pressures increase pumping costs. Higher pressures can also impinge water drops into side walls, where the water runs in sheets down the sides. This drastically reduces the surface area of the water flow, which reduces heat transfer. Rings circling the tower are recommended to kick water running down the sides back into the airstream. Unpacked towers do not have as high a factor of merit as towers with film packing or splash bars, but they are virtually maintenance-free and have low resistance to airflow. Figure 1 shows a typical underground vertical counterflow cooling tower.

After a cooling tower begins operation, the actual factor of merit should be determined. This is accomplished by measuring air and water flow rates and temperatures at the tower inlet and outlet, and then working the cooling tower equations in reverse. The actual factor of merit can be used to determine performance at other inlet conditions. This applies to mine, industrial, and commercial cooling towers.

Large Spray Chambers (Bulk Air Coolers)

The procedure for designing spray coolers is the same as for cooling towers, with the following minor changes:

$$q = M_w c_{pw}(t_{wo} - t_{wi}) \qquad (15)$$

$$q = M_a(\Sigma_{ai} - \Sigma_{ao}) \qquad (16)$$

$$n_w = \frac{1 - e^{-R(1 - X)}}{R} \qquad (17)$$

where $X = e^N$ for horizontal crossflow chambers.

A perfect counterflow tower has a factor of merit of 1.0, but the factor of merit for a single cross-flow chamber cannot exceed 0.63 (Bluhm 1981). Two-stage cross-flow chambers are most often specified. Counterflow performance is approximated, and the counterflow equation for water efficiency can be used. Three-stage chambers can be designed when water flow must be limited to control pumping costs. Four-stage chambers are rarely cost-effective in mining applications.

Spray chambers often use vee-jet nozzles at 30 psig, placed uniformly along the chamber length and designed to cover the cross section evenly. Sprays should just reach the back of the chamber. Mist eliminators are usually installed at the chamber exit. Whereas cooling towers need makeup water to replace evaporated water, bulk air coolers gain water through condensation. This water can be sent to the condenser side as makeup. Figure 2 shows a typical two-stage horizontal cross-flow spray chamber.

MINE COOLING TECHNIQUES

A mine cooling system typically sends air, water, or ice into the mine at a low enthalpy state and removes it at a higher one. In hot mines, heat is typically rejected to water being pumped to the surface, and to exhaust air being drawn from the mine. There are many combinations and variations on how this is accomplished. Economics and site-specific conditions determine the optimum methods.
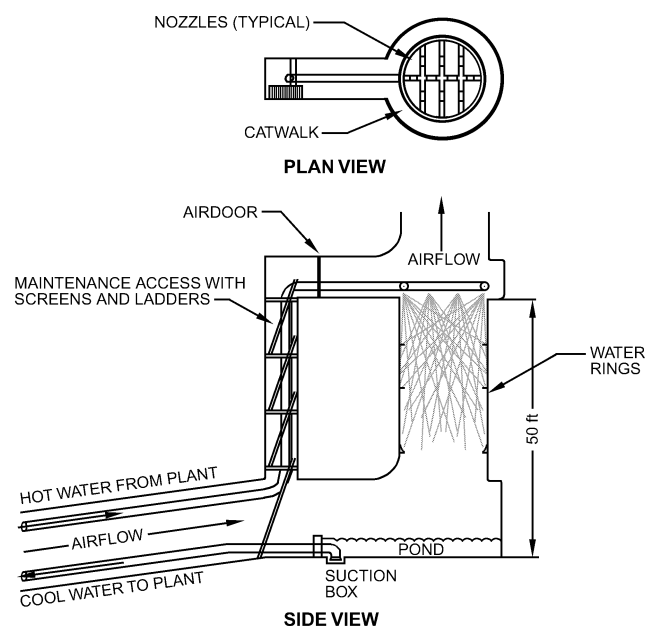
Increasing Airflows

This alternative should be considered first: it is usually less expensive to moderately increase airflows than to install refrigeration if the mine is above the critical ventilation depth. Increasing airflows also helps remove diesel fumes, which is increasingly important for modern mining. However, in deep, usually older mines with small cross-sectional airways, airflow increases may not be practical because of the cube relationship between fan power and airflow increase through a given resistance. Circuit resistance reduction via new airways or stripping existing airways is very expensive.

Chilling Service Water

When a mine requires additional heat removal, and airflow increases are not practical, chilling service water should be considered. Most mining methods require that water be sprayed on rock immediately after blasting to control dust; chilled water can intercept rock heat before the heat escapes into the air. This is a very flexible method of heat removal because it is applied when and where it is needed the most. After blasted rock is removed, the water is turned off and routed elsewhere. Main water lines should be insulated when service water is chilled.

Water is usually chilled in surface plants. A single-pass system is used if regional water supplies are plentiful. If scarce, water is pumped to the surface, recooled, and returned underground. Recycling can ease discharge permit requirements and thus save on treatment costs. Some mines have zero discharge permits. Regions with low winter temperatures and low relative humidities during summer have a natural cooling capacity that is adaptable to water chilling on the surface. Warm mine water is precooled in an evaporative cooling tower before being sent to the refrigeration plant.

Reducing Water Pressure and Energy Recovery Systems

All mines send water underground for drilling, cleaning, suppressing dust, and wetting broken rock. Hot mines using extensive cooling systems often send large volumes of water underground solely for air conditioning. The pressure of descending water must be broken periodically. The most common methods are the open cascade system and pressure-reducing valves. Turbines can also break the pressure and recover a significant portion of the potential energy that would otherwise be lost. Two types of turbines are suitable for mine use: the Pelton wheel, which is most often used, and a centrifugal pump specially designed to run in reverse. The Pelton
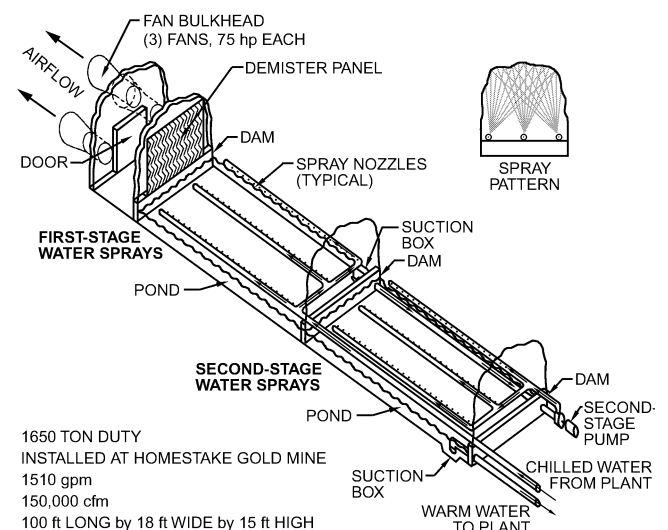

Fig. 1   Underground Open Counterflow Cooling Tower

Fig. 2   Two-Stage Horizontal Spray Chamber

wheel rotor is shaped like the spokes of a wheel, with cups attached to the ends of the spokes. One or two nozzles shoot high-pressure water onto the cups, spinning the wheel. It is at least 80% efficient over a range of flows, simply constructed, and readily controlled. A wide operating range is important because water demand fluctuates. A turbine can turn either a generator or pump. Turning a generator is preferred because it separates service and cooling water from the mine dewatering system so that downtime in one system is less likely to disrupt the other.

Besides providing power to help return service water to the surface, turbines have another advantage: unrecovered potential energy is converted to heat at a rate of 1 Btu/lb per 778 ft of depth. If, for example, a 6000 ft deep mine uses 1000 gpm for air conditioning without energy recovery, the water will heat by 7.71°F. If 80% efficient turbines are used, the water temperature rise is about $0.2 \times 7.71 = 1.54$°F. The refrigerating effect lost is only 64 instead of 321 tons.

Other energy-recovery devices include hydrotransformers (large pistons transfer force from the high-pressure side to the lower-pressure side), and three-pipe feeder systems that deliver chilled water on one side while pumping out crushed ore on the other. These concepts have been tested in Europe and South Africa.

Bulk Cooling Versus Spot Cooling

Engineers must balance bulk cooling and spot cooling. **Bulk cooling** using a centrally located plant cools the entire mine, or a large section of it. Benefits are lower cost per ton installed, generally better maintenance, and lower temperatures in non-stoping areas such as haul drifts. Bulk cooling intake air is often done at warm-climate mines to provide winter-like or better conditions year round. Air is cooled in large direct-contact spray chambers adjacent to the shaft and then injected into the shaft below the main landing.

Cooling the entire mine draws more heat from surrounding wall rock, so a larger system must be designed to ensure proper stope cooling (i.e., positional efficiency suffers). When a multilevel mine is bulk cooled, cooling may be wasted on upper levels where heat load is low.

Spot cooling provides adequate temperature control in exploration and development headings, and in stopes on the fringes of mining activity. Total heat load is lower, but cost per ton is higher and temperatures in some areas might exceed design limits.

Combination (Integrated) Surface Systems

Combination (or integrated) systems can cool both air and water. Surface plants devote a higher fraction of cooling capacity to bulk cool intake air in the summer. In winter, a higher fraction is used to chill service or air-conditioning water. Water is delivered underground via open or closed systems, with or without energy recovery. Figure 3 shows components of an integrated mine cooling system.

Underground Refrigeration

Larger refrigeration machines also can be located underground. They usually produce chilled water for cooling air in spray chambers, and heat is rejected to exhaust air via cooling towers. Another method is to operate district cooling systems, using a chiller to produce water for a closed network of cooling coils installed in parallel. These coils can be used in auxiliary systems at individual work areas, or installed in a bank. As with spot coolers, coils should be installed upwind of blasting to limit air-side fouling. Condenser heat from district chiller systems is rejected either to service water or to the mine dewatering system.

Ice Plants

For ultradeep mines (>12,000 ft), or those at the performance limits of existing water and airflow heat rejection systems, ice cooling should be considered. In going from 32 to 90°F before being pumped out of the mine, cooling water starting as ice can remove about 4.5 times the heat as the same mass flow of chilled water in going from 45 to 90°F:

Heat removal (Btu/lb) = Sensible + Latent

Heat removal of chilled water = (1 Btu/lb·°F)(90 − 45) = 45 Btu/lb

Heat removal of ice = (1 Btu/lb·°F)(90 − 32) + 144 Btu/lb = 202 Btu/lb

Heat removal factor increase of ice over water = 202/45 = 4.5

South African mines have been at the forefront in this application (Sheer et al. 2001). Both chunk and slurry delivery methods send ice to underground chambers, where it mixes with warm water returning from the mining area. The cold mixed water is then sent back to the mining area.

Several successful systems have been installed. Cost has dropped as technology improves; the overall COP of ice systems for ultradeep mines is now competitive with traditional cooling methods.

Thermal Storage

This Canadian innovation uses near-surface ice stopes or rock rubble to effectively and inexpensively heat intake air in the winter and cool it in the summer (Stachulak 1989).

Controlled Recirculation

This technique, used in conjunction with bulk air cooling, can reduce ventilation and air-conditioning requirements in older, deep mines, especially heavily mechanized ones (Tien 1999). Besides increasing air velocities in work areas without drawing more surface air through a high-resistance circuit, controlled recirculation reduces the heat load caused by autocompression. Using Equation (1), for every 100,000 cfm at standard density brought from the surface, the lost cooling capacity per 1000 ft of descent is

$$
\begin{aligned}
\text{Heat Load} &= (100{,}000 \text{ ft}^3/\text{min})(60 \text{ min/h})(0.075 \text{ lb/ft}^3) \\
&\quad \times \left(\frac{1 \text{ Btu}}{778 \text{ ft/lb}}\right)(1000 \text{ ft}) \\
&= 578{,}406 \text{ Btu/h, or 48.2 tons of cooling lost} \\
&\quad \text{per 100,000 cfm, per 1000 ft}
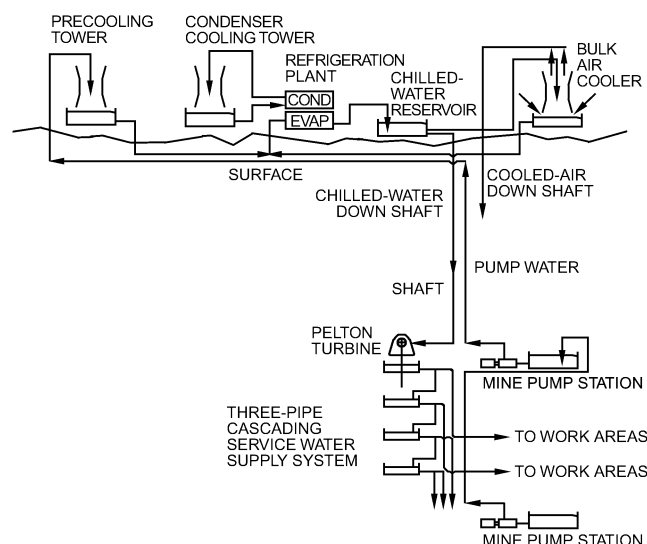\end{aligned}
$$

Fig. 3   Integrated Cooling System

Controlled recirculation systems must be designed very carefully, with stringent monitoring and control safeguards.

Cabs and Vests

Mechanization can add significant heat, especially in confined auxiliary-ventilated spaces. Most noncoal mines have converted to diesel equipment in the last 20 years, although some use electric loaders and trucks. Coal mines are also mechanizing, but more slowly. The biggest problem with diesel engine heat is related to the greatest advantage of these vehicles: mobility. Heat and emissions from a diesel vehicle in a confined area can tax almost any ventilation system. Increasingly, cabs are being specified for large diesel vehicles. Cabs come equipped with window-type air conditioners and HEPA filters to capture diesel particulate and dust. After the diesel vehicle has left the heading, the mine ventilation and cooling system can provide an acceptable environment for other personnel. Cabs are expensive; however, because a loader often visits multiple headings in a week, the cost of a cab is much less than maintaining the design reject wet-bulb temperature in all headings at all times.

Cooling vests are not popular in mining. They are bulky, reduce mobility, and are time-consuming to prepare and use. Vests have limited application for mechanics, electricians, pipe-fitters, and others who must enter hot areas to set up ventilation and cooling. Vests using blue-ice packs or dry-ice last two to three hours; those using compressed-air venturi-type coolers require an umbilical cord.

Other Methods

Other methods being developed include the air cycle (air, compressed on the surface and sent underground to a turbine, turns a generator and exits at –40°F), and the ammonia cycle (sending down liquid ammonia, evaporating it, and sending the vapor back to surface condensers). These methods may be best suited to ultradeep mines where other cooling methods are already fully developed.

Transferring heat from current stopes to wall rock or rock rubble in previously worked-out stopes (the only method that does not remove heat from the mine) has also been considered. Refrigeration equipment would have to operate at a high condensing temperature to produce water hot enough to transfer heat to worked-out stopes.

SELECTING A MINE COOLING METHOD

After mine cooling and ventilation requirements have been projected, the designer must analyze and select the best method(s) for meeting those requirements. Cost-benefit analysis is the most widely used, but hardware reliability, dependency on outside factors, flexibility, safety, and technological level are just as important. Some factors to consider include the following:

- **Seasonal ambient conditions.** Warm-climate mines tend to bulk-cool air on the surface in industrial direct-contact spray chambers located close to the intake shaft.
- **Orebody and mining methods.** The more massive the orebody, the more ideal bulk cooling becomes. When stopes are scattered and continuously advanced into new areas, district or spot cooling might be better.
- **Mining rate.** This is critical: heat removal is energy related, not necessarily power related. A fast mining rate prompts a high instantaneous heat load (Btu/h), but less heat energy (Btu) per ton of production. This is because wall rock is covered by fill or isolated before its total heat energy has escaped into the airstream. The Btu per ton of production incurs the air-conditioning costs. Leave as much heat in the wall rock as possible.
- **Size and condition of major airways.** In older mines, small airways often limit airflow increases. This may prompt the need for air conditioning sooner than it normally would have been necessary.

- **Heat sources.** The contribution of TIH and TDH sources to the total can help determine the balance of passive to active thermal environmental controls, the ratio of airflow to air conditioning, and whether cabs should be specified.
- **Cost of power, water, labor, and supplies.** Knowing these costs is critical for assessing optimum capital expenditure to control operating costs. For example, if power cost is high, spending extra for a higher-COP system may be warranted.
- **Governmental regulations.** Heat stress standards can influence the size of the system. Other safety issues may constrain design, such as not using combustible pipe insulation or ammonia machines underground.

Basic cooling alternatives for specific cases are summarized in Table 4. Airflows are described as limited, medium, or large. One way to express airflow for a given mine is the ratio of tons of airflow per ton of ore. Limited airflow is defined as less than 8 tons of air per ton of ore, medium airflow is 8 to 16 tons per ton, and large airflow is over 16 tons per ton.

These ranges are based on an unpublished study of approximately 100 mines of all types, worldwide. The ranges discussed are for heat removal only. Additional airflow for methane or radon removal must be addressed separately.

MECHANICAL REFRIGERATION PLANTS

Surface Plants

Centrifugal or helical rotary screw machines are typically used in surface plants to chill water or bulk-cool air. Banks of machines are usually installed in parallel: plant design must accommodate one machine being down at any given time for maintenance while others operate. Shell-and-tube heat exchangers are standard, although plate-and-frame are used if water close to the freezing point is specified. The most common refrigerant for positive-displacement compression is HCFC-22. Ammonia is also commonly used in surface plants. Absorption machines can be considered if external waste heat is available.

Table 4 Basic Cooling Alternatives

	Warm Climate	Cool/Cold Climate
Massive ore-body (deep)	Large or medium airflow Chill service water Bulk-cool air on surface Bulk-cool air underground	Medium airflow Chill service water Bulk-cool air underground Thermal storage
Massive ore-body (shallow)	Large airflow Bulk-cool air on surface Chill service water Shell and tube	Large or medium airflow Chill service water on surface Shell and tube
Scattered orebody (multilevel)	Large airflow if not too deep Chill service water Bulk-cool air on surface District chiller systems Spot coolers/spray chambers	Large or medium airflow Chill service water District chiller systems Thermal storage
Ultradeep ore-body (massive and/or multilevel)	Limited airflow Chill service water District chiller systems Ice cooling Controlled recirculation	Limited airflow Chill service water District chiller systems Ice cooling Controlled recirculation
Small orebody	Bulk-cool air on surface Chill service water District chiller systems Spot coolers	District chiller systems Chill service water Spot coolers
Porous rock	District chiller systems Spot coolers	District chiller systems Spot coolers

Underground Plants

Large underground plants do the same work as surface plants, but are closer to work areas. Better positional efficiency and percent utilization are the advantages. Whereas surface plants use atmospheric air for heat rejection, underground plants use mine exhaust air, which raises natural ventilation pressure and aids circuit fans. Components for underground machines must be disassembled for transport down the shaft.

The main disadvantage of underground refrigeration is that heat rejection is limited by the amount of available exhaust air. Excavating underground refrigeration rooms and spray chambers is more costly than erecting prefabricated surface buildings. Maintenance is also more difficult because of shaft logistics. Power is more difficult to supply to an underground facility, and subject to more disruptions.

Spot Coolers

Spot coolers with 15 to 100 ton capacity allow driving long development headings, or cooling exploration sites before installing primary ventilation and cooling equipment. Development headings can be advanced more rapidly and under more comfortable conditions. Condenser heat is most often removed by service water, although some air-to-air condensers are used.

Spot coolers use reciprocating, scroll, or small screw compressors. Hermetic scroll compressors are becoming more popular because they handle liquid slugging better than reciprocating compressors and are less expensive. Spot coolers use direct exchange (DX) air cooling coils. The packaged unit includes a fan, which draws air through the coil (or coils, in a dual-coil unit) and then blows it through duct to the heading. Spot coolers must be compact and portable because they are moved often. The service water required is typically 1.1 gpm per ton, but can be less if the water temperature is under 55°F, or more if it is over 70°F. A return drain pipe is recommended to prevent contact between hot discharge water (often over 100°F) and ambient air. Coils sometimes receive dusty air immediately after blasting; if so, coils must be washed at least every other day.

Spot coolers are expensive, but often are the only choice for cooling exploration, development, and small-scale stoping on the fringes of mining activity.

Maintenance

Mines with extensive systems (e.g., a large chiller plant, or over 10 spot coolers) should employ a mechanic specializing in refrigeration. Mines with over 2000 tons probably need a second mechanic. These persons should be factory-trained and must be certified to handle refrigerants. Refrigeration specialists can be assisted periodically or full-time by apprentice mechanics. Another viable approach is a maintenance contract with the equipment manufacturer or supplier, or an independent HVAC&R shop. Some mines have a full-time person cleaning coils.

A fouling factor ($ft^2 \cdot h \cdot °F/Btu$) should be calculated from lab analysis of the condenser water, especially for district chillers using sump water. Planning a tube-cleaning regimen, either manual, acid circulation, or automatic with brushes and a flow reversal valve, is critical. Underground condensers can become plugged within a couple of weeks without cleaning, depending on the fouling factor. Water treatment is needed to control scale, corrosion, and organisms in surface or underground plants with cooling towers.

MINE AIR HEATING

Cold-climate mines typically heat intake air in the winter. In Canada, heating intake air can cost more than all other ventilation costs combined (Hall 1989). However, without heat, water in the shaft will freeze, disrupting hoisting operations and damaging shaft

support members, cables, and pipes. Very cold air and icy floors are safety and health hazards; heavy gloves and other protective clothing required can make routine tasks difficult. Intake air is typically heated to just above the freezing point. Autocompression and shaft heat loads further temper the air as it downcasts into the mine.

Steam coils operated by boilers burning wood, coal, fuel oil, or natural gas often served as shaft heaters in the past. Electric resistance heaters have also been used, but they are expensive to operate. Waste heat from compressor stations has also been used.

When exhaust and intake shafts are located close together, a circulating glycol or heat pump system can be used to transfer heat from exhaust air to intake. For every degree of total heat (sensible plus latent) given up by warm saturated exhaust air, the same mass flow of cold intake air can be heated sensibly by 4 degrees. Either coils or a cooling tower extracts heat from exhaust air, then coils transfer this heat to the intake air.

Controlled recirculation (up to 25% of total airflow) can also be applied to heat intake air (Hall 1989). The system is temporarily shut down during blasting.

Some cold-climate mines isolate the primary production shaft from the ventilation circuit. A slight upcast flow of uncontaminated air maintains good conditions in the shaft for hoisting ore and moving personnel and supplies. The disadvantage of this method is that ventilation duties of the production shaft must be transferred to one or more expensive stand-alone intake airways.

Natural gas and/or propane heaters are typically used at modern mines. Natural gas is preferred because it is less expensive and it burns more cleanly. Where natural gas is not available, propane must be trucked to the mine site. The same heater can burn either natural gas or propane; thus, propane can be used for back-up in case natural gas is cut off. Direct-fired heaters are usually preferred because the entire calorific value of the fuel enters the intake airstream. If indirect heaters are used, roughly 15 to 25% of the heat is lost up the flue pipe.

Two types of natural gas or propane heaters have been used to heat intake air: (1) a grid of burner bars installed in a housing at the intake shaft, sometimes with louvers to adjust the flow of intake air and to mix air from the heaters with outside air; and (2) a crop dryer type of burner. Temperature sensors installed downstream can modulate both heater types to ensure that no more heat is applied than necessary to bring the temperature of the mixed intake air to 34°F.

Carbon monoxide sensors should also be installed downstream of the heaters. Experience at two mines in the western United States shows that the CO content of intake air heated by direct-fired burners can reach 10 to 20 parts per million.

Equation (18) is used to calculate the total heat required, assuming that the air has a low humidity ratio (which is the case for very cold air), and that no water is evaporated in the heater. Heating values for different fuels are given in Table 5.

$$\text{Heat, Btu/h} = (\text{Airflow, cfm})(60 \text{ min/h})(\text{density, lb/ft}^3)$$
$$\times (0.24 \text{ Btu/lb} \cdot °F)(\Delta t, °F) \tag{18}$$

where Δt = 34°F minus the intake air temperature.

Example 6. A mine is located where the atmospheric air temperature can drop to –20°F for two or more weeks per year. Occasionally the tem-

Table 5   Heating Values for Fuels

Fuel	Value	Source
Natural gas	1000 Btu/ft^3	Kennedy 1996
Propane	90,000 Btu/gal	Kennedy 1996
Bituminous coal	12,300 to 14,400 Btu/lb	Abbeon Cal 2001
Fuel oil	143,000 Btu/gal	Abbeon Cal 2001
Wood	15,000,000 to 31,000,000 Btu/cord	Abbeon Cal 2001

perature drops to –30°F. An intake shaft handles 400,000 cfm, and the density of air entering the shaft in winter is 0.070 lb/ft³. What heating should be installed at the shaft intake to keep the shaft free of ice?

Solution: Sizing heaters is usually based on average cold periods, not extreme cold snaps. Here, a direct-fired heater is sized to raise –20°F air to 34°F. When the temperature drops to –30°F for short periods, intake airflow should be temporarily reduced. Using Equation (18),

Heat = (400,000)(60)(0.070)(0.24)[34 – (–20)] = 21,800,000 Btu/h

If natural gas is used, the volume required is

$$\frac{21{,}800{,}000 \text{ Btu/h}}{1000 \text{ Btu/ft}^3} = 21{,}800 \text{ ft}^3/\text{h}$$

If propane is used, the gallons required are

$$\frac{21{,}800{,}000 \text{ Btu/h}}{90{,}000 \text{ Btu/gal}} = 242 \text{ gal per hour}$$

MINE VENTILATION

Mine ventilation supplies oxygen to underground facilities, and removes dangerous or harmful contaminants such as methane, radon, strata gases, dust, blasting fumes, and diesel emissions. Ventilation also removes heat and helps control humidity in hot mines. Planning a ventilation system consists of five basic steps: (1) determining airflows, (2) planning the primary circuit, (3) specifying circuit fans and their installation, (4) determining auxiliary system requirements, and (5) assessing health and safety aspects.

Determining Airflows

Mining operations generate differing types and amounts of contaminants, and airflows dilute and remove these contaminants. The ventilation engineer must work closely with mine planning staff to understand where and how much production will take place, and what contaminants will be generated. The federal Mine Safety and Health Administration (MSHA) regulates contaminant concentrations to limits specified in the Federal Register, CFR 30. Controlling the most problematic contaminant normally keeps all others within their legal limits. For coal mines, contaminants of concern are typically methane and coal dust; for uranium mines, radon gas; for non-dieselized hard rock mines, usually silica dust and blasting fumes; for dieselized mines, typically diesel emissions. Design airflows for dieselized nonuranium metal mines range from 75 to 150 cfm per diesel horsepower, depending on the reference cited. With the current emphasis on controlling diesel emissions, start planning at 100 cfm per horsepower.

Total airflow is a summation of airflows for individual work areas, plus a leakage factor. Leakage is defined as airflow that does not ventilate any active work area or permanent site such as a pump room. A "tight" system minimizes leakage through well-constructed doors and seals, by minimizing the number of possible leakage paths, and by careful fan placement. Leakage can range from 10% of total airflow at a tight metal mine to 80% at some coal mines.

The ratio of tons of air per ton of ore production is about 2 to 4 for block cave mines, 6 to 8 for nondieselized cut-and-fill metal mines, and 9 to 16 for dieselized metal mines. Gassy coal and uranium mines can have significantly higher ratios, depending on the methane or radon generation rate.

Example 7. A new mechanized cut-and-fill gold mine is planned. Ore production is expected to be 1,200,000 tons per year. Intake air density is 0.070 lb/ft³. What is the rough airflow required for ventilation?

Solution: The airflow range is 9 to 16 tons of air per ton of ore for dieselized metal mines. For a first-pass guess, assume an average 12.5 tons per ton. The total weight of the air through the mine in a year is

(1,200,000 tons ore per year) × (12.5 tons air per ton ore)

= 15,000,000 tons air per year

$$\text{Airflow, cfm} = \frac{(15{,}000{,}000 \text{ tons/yr})(2000 \text{ lb/ton})}{(0.070 \text{ lb/ft}^3)(525{,}600 \text{ min/yr})} = 815{,}400 \text{ cfm}$$

Ratios provide a good first guess. However, the ventilation engineer should derive the total airflow by listing all operations, estimating leakages, and adding the specific airflows required to ventilate each operation (zero-based planning). As with reject temperature, the total airflow selected should be economically justifiable to management.

Airflow specification may change with time because of production, equipment, or mining method changes.

Planning the Circuit

With airflow specified and work sites plotted, the ventilation engineer must lay out the primary circuit. The three basic types of airways are intake, work area, and exhaust. Sizing airways is normally based on keeping velocity within acceptable limits: If velocity is too low, the airway is oversized and thus costs more than necessary; if it is too high, pressure drop is too large and raises operating costs. Air velocity should not exceed 1200 fpm in production shafts and haul drifts. Higher velocities can create dust problems and lead to employee discomfort. However, velocities in bare circular concrete exhaust shafts can approach 5000 fpm if necessary. Air velocity in vertical upcast exhaust shafts should avoid the 1400 to 2300 fpm range because water sheets can form, causing surging at the main fan.

Resistance to airflow is calculated using Atkinson's (1854) and McPherson's (1993) equations:

$$\Delta H = RQ^2 \frac{d}{0.075} \tag{19}$$

where

ΔH = pressure drop, in. of water
R_t = resistance, in. of water·min²/ft⁶⁴
Q = airflow, cfm
d = actual air density, lb/ft³
0.075 = standard air density, lb/ft³

$$R = \frac{kLP}{5.2A^3} \quad R_t = \frac{fLP}{2A^3} \tag{20}$$

where

k = friction factor, lb·min²/ft⁴ (includes effects of pipes, ground support, and rock surface roughness)
L = length, ft
P = perimeter of opening, ft
5.2 = conversion factor, lb/ft²·in. of water
A = area of opening, ft²

For rectangular drifts, the k factor can range from 22×10^{-10} for a smooth, concrete-lined straight drift to 90×10^{-10} for an unlined, irregular curved drift. For shafts, the k factor can range from 12×10^{-10} for a smooth-sided borehole to over 500×10^{-10} for a heavily timbered rectangular shaft. See Hartman (1997), McPherson (1993), and Tien (1999) for more precise airway resistance specification.

Example 8. Mine plans call for 65,000 cfm to be sent through 2000 ft of 10 ft wide by 10 ft high drift. The k factor from measurements of similar drifts is 50×10^{-10} lb·min²/ft⁴. The average temperature is 75°F wb and 80°F db. The barometric pressure is 13.8 psia. What is the resistance of this drift, and what is the air pressure drop?

Solution: Using psychrometric equations in Chapter 6 of the 2001 *ASHRAE Handbook—Fundamentals*, the density is 0.0683 lb/ft³.

$$R = \frac{kLP}{5.2A^3} = \frac{(50 \times 10^{-10})(2000)(40)}{(5.2)(100)^3}$$

$$= 7.69 \times 10^{-11} \text{ in. of water} \cdot \text{min}^2/\text{ft}^6$$

$$\Delta H = RQ^2 \frac{d}{0.075} = (7.69 \times 10^{-11})(65,000)^2 \frac{0.0683}{0.075}$$

$$= 0.30 \text{ in. of water}$$

A mine ventilation circuit contains airways in series and in parallel. The overall resistance (Hartman 1997) is

For series:
$$R_T = R_1 + R_2 + R_3 + \cdots + R_n \qquad (21)$$

For parallel:
$$\frac{1}{\sqrt{R_T}} = \frac{1}{\sqrt{R_1}} + \frac{1}{\sqrt{R_2}} + \frac{1}{\sqrt{R_3}} + \cdots + \frac{1}{\sqrt{R_n}} \qquad (22)$$

Example 9. If Airway #1 has a resistance of 1×10^{-10}, Airway #2 has a resistance of 2×10^{-10}, and Airway #3 has a resistance of 3×10^{-10}, what is the resistance of these three branches in series and in parallel?

Solution:

Series:

$$R_T = 1 \times 10^{-10} + 2 \times 10^{-10} + 3 \times 10^{-10}$$

$$= 6.0 \times 10^{-11} \text{ in. of water} \cdot \text{min}^2/\text{ft}^6$$

Parallel:

$$\frac{1}{\sqrt{R_T}} = \frac{1}{\sqrt{1 \times 10^{-10}}} + \frac{1}{\sqrt{2 \times 10^{-10}}} + \frac{1}{\sqrt{3 \times 10^{-10}}}$$

$$R_T = 1.92 \times 10^{-10} \text{ in. of water} \cdot \text{min}^2/\text{ft}^6$$

Modern ventilation network computer analysis uses Kirchhoff's laws to balance airflows: (1) the summation of airflows into a junction equals the summation out, and (2) the summation of pressure drops around any enclosed mesh equals zero.

Computer simulation allows quick analysis of a wide range of scenarios. Most programs use a balancing algorithm based on the work of Hardy Cross in the 1960s and 1970s. The program iterates as it converges on final balanced airflows. Fan curves or regulators can be inserted in almost any branch.

Regulators or section booster fans control airflow in branches. Without regulation, too little or too much airflow may occur; nevertheless, circuits should be designed with a many free-split branches (branches without a fan or regulator) as possible to minimize overall resistance. Free-split branches are often located in circuit extremities.

A mine should have more intakes than exhausts. This enhances safety, because miners have more escape paths, and because more paths bring in fresh air, in case a fire occurs in one of the intakes. Also, exhaust shafts can generally handle greater air velocities and hence larger quantities, so fewer exhaust shafts are needed.

Metal mines often contain circuit booster fans. Underground boosters can create neutral points in the system where air short-circuits from intake to exhaust above the point, and recirculates from exhaust to intake below the point. Uncontrolled recirculation should be minimized.

Exhausting primary circuits are commonly used in both metal and coal mines (intakes do not have airlocks). Under normal operation, this produces a negative mine pressure gradient. If fans fail or are deactivated, barometric pressure in the mine rises, which temporarily helps keeps methane in coal mines from flowing away from gob (mining waste) areas (Kennedy 1996).

Specifying Circuit Fans

Primary fans are either centrifugal or axial. South African mines typically use large centrifugals, whereas most U.S. and Canadian mines use axials. Both types have advantages. Efficiency (up to 90%) is about the same with either type. Centrifugals are heavier duty, quieter, do not have a pronounced stall region, and can generate higher static pressures (over 30 in. of water). Axials are more compact, and airflows can be easily adjusted by blade angle changes. Primary fans range from 100 to over 3500 hp each. Surface installations with multiple fans are common for large airflows, and for back-up operation when one fan is turned off for maintenance. Circuit fans can also be installed underground, especially in metal mines.

Primary fans are specified while the circuit is designed. Engineers must often balance airway considerations (sizes and numbers) and fan specifications. A fan should be selected that will operate on an efficient part of its curve. Fan speed, quantity, pressure, and power are related in the fan laws equations, described in Chapter 18 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*. Care must also be taken to anticipate future circuit changes as mining operations advance.

The fan installation must be designed after primary fans are selected. Consequences of fan downtime must be carefully considered, especially for coal mines, because methane concentrations can increase when circuit airflows decrease. Fans exhausting a mine are typically mounted horizontally near a vertical shaft or borehole. A 90° transition turns the air into the fan inlet. An isolation door is installed between the transition and fan. Coal mines require a blast door to dampen a shock wave caused by a possible methane or coal dust explosion. An evasé, or diffuser, is attached to the fan outlet to recover part of the velocity pressure exiting the mine. A silencer can be added if surface noise reduction is desired. Increasingly, variable-speed drives are used with electric motors to turn the fans. These drives provide soft start, and speeds from 50 to 100% of synchronous speed, and even to 110% for temporary emergency duty. The installation must be designed for accessibility and ease of maintenance.

Determining Auxiliary System Requirements

Auxiliary fan and duct systems deliver air into dead-end headings. These systems are generally not permitted in coal mines but are common in metal mines. A blowing system is most often used. A fan is set in a fresh air base at the start of a drift, and duct is installed as the drift advances. For drifts under 1000 ft, flexible brattice-cloth duct can be used. Longer drifts that require booster fans need rigid duct because duct gage pressure can drop below atmospheric. Rigid duct also offers less resistance than brattice-cloth duct, but it is about eight times as expensive.

The air quantity needed at the face is determined by the equipment used and the rate at which blasting and diesel fumes must be removed. Ducts are sized for the air quantity needed and for space limitations in the drift. Fans are selected to provide the specified airflow. In general, a single-stage axial fan can generate up to 10 in. of water static pressure, which should deliver required airflow up to 2500 ft through properly sized duct. A larger duct or a two-stage axial fan is needed if distance is much longer. For very long drifts, booster fans are needed about every 2500 ft.

An exhausting system is often used for drifts requiring quick ingress after blasting. Air flows to the face through the drift, captures fumes, and is blown back to the circuit through duct. This keeps the drift clear of fumes. Disadvantages include: (1) the air picks up heat and humidity en route to the face, (2) rigid duct is required, and (3) the face is not swept by air as with a blowing system. A face overlap fan and duct can be installed.

Chapter 39 of the 2001 *ASHRAE Handbook—Fundamentals* provides a friction chart for round duct. However, it is better to acquire the friction chart of the specific duct being considered from the supplier. Shock losses through couplings and bends must be tallied. One important consideration is leakage through couplings. This can be minimized by careful installation, keeping duct pressure

under 10 in. of water static pressure, and installing longer pieces of duct (up to 100 ft for brattice cloth, or 20 ft for rigid).

Cassettes loaded with brattice-cloth duct are now used for drifts driven by rapidly advancing tunnel boring machines.

Duct damage is common in mines. Mobile equipment and fly rock from blasting can punch holes in the duct. These factors can drastically reduce airflow. Care must be taken to minimize damage, and to quickly repair or replace damaged pieces.

Assessing Health and Safety

Few aspects of underground mining have as direct an impact on health, safety, and morale as ventilation. No component of ventilation design should be undertaken without a rigorous review of health and safety aspects, including the risk of

- Fire and explosion
- Dangerous and toxic substances
- Heat
- Ventilation equipment usage

For metal mines, fire is the most significant potential ventilation hazard. Fuel, heat, and oxygen are required for combustion; removing any of these components will prevent combustion. Fuel sources such as oil, diesel fuel, and blasting agents are kept in special areas designed to keep out ignition sources. Sprinkler or chemical suppression systems can be installed in these areas as well as in repair shops. Mobile equipment fires are a special concern for modern mining: vehicles should be fitted with a dry chemical fire suppression system, triggered either automatically or by the operator. Electric substation and conveyor fires can be very dangerous.

Engineers should anticipate various scenarios in ventilation circuits. What are the fire risks in any given area? If a fire broke out in any location, how would circuits respond? Would fire-induced natural drafts change airflow quantities and directions? How would fire be detected, how would miners be notified, how would they escape, and how would the fire be fought? MSHA requires that refuge chambers be constructed if miners cannot be hoisted to the surface within 1 h of notification. The ventilation staff must work closely with the safety department and mine management in preplanning how to respond to different emergencies.

Spontaneous combustion is a problem for both metal and coal mines. Fortunately, timber is now seldom used for ground support, although many older mines have worked-out areas that contain timber. Coal, being combustible, can be particularly troublesome. Circuits must be designed so that spontaneous combustion fires will not contaminate active workings.

For coal mines, methane and coal dust explosions pose the greatest risk. Equipment must be rated "permissible," or non-sparking. Airways should be coated with rock dust to prevent a methane ignition from propagating. Methane is explosive in air from 5 to 15% concentration. Whenever methane reaches 0.25%, MSHA requires that changes be made to improve ventilation. At 0.5%, further steps must be taken, and no other work is permitted until the concentration drops. At 1%, all personnel except those working on ventilation must be evacuated from the affected area.

Ventilation is the first line of defense against toxic or asphyxiating gases. These can be generated by blasting (CO, CO_2, NH_3, NO_x) and by diesel engines (CO, NO_x, SO_2, various hydrocarbon compounds, and soot). The rock itself can release CO_2 and H_2S.

The relationship between heat stress and accident frequency has been clearly established in South African mines (Stewart 1982). Work area temperatures should be kept under 85°F wb, especially where heavy physical work is performed.

Ventilation and air-conditioning equipment may also pose health and safety risks. All fan inlets require screens. Fans should be equipped with vibration sensors that can deactivate the fan if neces-

sary. Silencers may be needed if personnel work nearby. Refrigeration rooms must be well ventilated in case of a sudden refrigerant release. Duct, pipe insulation, and other substances such as foam for seals should be approved by MSHA in accordance with 30 CFR Part 7. Electrical systems must meet rigorous MSHA codes.

REFERENCES

Abbeon Cal, Inc. 2001. *Pocket ref*, 2nd ed. T.J. Glover, ed. Sequoia Publishing, Littleton, CO.

Atkinson, J.J. 1854. *On the theory of the ventilation of mines*. North of England Institute of Mining Engineering.

Bluhm, S.J. 1981. Heat transfer characteristics of direct-contact crossflow heat exchangers. Master's thesis presented to the University of the Witwatersrand. Republic of South Africa.

Bossard, F. 1982. Sources of heat entering mine air. *Mine Ventilation Design Practices*, 1st ed. Butte, MT.

Burrows, J. 1982. Refrigeration—Theory and operation. *Environmental engineering in South African mines*, pp. 631-637. The Mine Ventilation Society of South Africa.

Duckworth, I. and P. Mousset-Jones. 1993. A detailed heat balance study of a deep level mechanized cut-and-fill stope. *Proceedings of the 6th U.S. Mine Ventilation Symposium*, pp. 441-447. SME, Littleton, CO.

Fenton, J.L. 1972. Survey of underground mine heat sources. Master's thesis, Montana College of Mineral Science and Technology, Butte.

Goch, D.C. and H.S. Patterson. 1940. The heat flow into tunnels. *Journal of the Chemical, Metallurgical and Mining Society of South Africa* 41.

Hall, A.E., D.M. Mchaina, and S.G. Hardcastle. 1989. The use of controlled recirculation to reduce heating costs in Canada. *Proceedings of the 4th International Mine Ventilation Congress*. Australasia Institute of Mining and Metallurgy, Melbourne, Australia.

Hall, C.J. 1981. *Mine ventilation engineering*. Society of Mining Engineers and Lucas-Guinn Co., Hoboken, NY.

Hartman, H.L., J.M. Mutmansky, R.V. Ramami, and Y.J. Wang. 1997. *Mine ventilation and air conditioning*, 3rd ed. John Wiley & Sons, New York.

Howes, M.J., and C.A. Nixon. 1997. Development of procedure for safe working in hot conditions. *Proceedings of the 6th International Mine Ventilation Congress*, pp. 191-197. SME, Littleton, CO.

Kennedy, W.R. 1996. *Practical mine ventilation*. Intertec Publishing, Chicago.

Marks, J.R. 1988. Computer-aided design of large underground direct-contact heat exchangers. Master's thesis, University of Idaho.

Marks, J.R. and L.M. Shaffner. 1993. An empirical analysis of ventilation requirements for deep mechanized stoping at the Homestake Gold Mine. *Proceedings of the 6th U.S. Mine Ventilation Symposium*, Salt Lake City, UT, pp. 381-385. SME, Littleton, CO.

McPherson, M.J. 1993. *Subsurface ventilation and environmental engineering*. Chapman & Hall, NY.

Patterson, A.M. ed. 1992. *The mine ventilation practitioner's data book*. The Mine Ventilation Society of South Africa.

Sheer, TJ., M.D. Butterworth, and R. Ramsden. 2001. Ice as a coolant for deep mines. *Proceedings of the 7th International Mine Ventilation Congress*. Research and Development Center for Electrical Engineering and Automation in Mining, Kracow.

Stachulak, J. 1989. Ventilation strategy and unique air conditioning at INCO Limited. *Proceedings of the 4th U.S. Mine Ventilation Symposium*, Berkeley, CA. Society for Mining Metallurgy and Exploration, Littleton, CO.

Stewart, J.M. 1982. Practical aspects of human heat stress. In *Environmental engineering in South African mines*, p. 574. Burrows, Hemp, Holding and Stroh, eds. The Mine Ventilation Society of South Africa.

Tien, J.C. 1999. Controlled recirculation. In *Practical mine ventilation engineering*, pp. 370-392. Intertec Publishing, Chicago.

van der Walt, J. and C.S.J. van Rensburg. 1988. The options for cooling deep mines. *Proceedings of the 4th International Mine Ventilation Congress*, p. 401. Australasian Institute of Mining and Metallurgy, Melbourne, Australia.

Whillier, A. 1977. Predicting the performance of forced-draught cooling towers. *Journal of the Mine Ventilation Society of South Africa* 30 (1):2-25.

Whillier and Thorpe. 1982. Sources of heat in mines. In *Environmental engineering in South African mines*, Burrows, Hemp, Holding and Stroh, eds. Mine Ventilation Society of South Africa.