USBM Contract No. G0122137

# A STUDY OF UNDERGROUND MINE HEAT SOURCES

Phase I, Survey of Underground Mine Heat Sources, Master's Thesis by James L. Fenton, March, 1972

Phase II, Evaluating Underground Heat Sources In Deep Mines, Master's Thesis by Walter I. Enderlin, April, 1973

Phase III, Underground Mine Air-Cooling Practices, Floyd C. Bossard and Koehler S. Stout, June 30, 1973

MONTANA COLLEGE OF MINERAL SCIENCE AND TECHNOLOGY FOUNDATION

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DEPARTMENT OF THE INTERIOR
BUREAU OF MINES
WASHINGTON, D.C.

# UNDERGROUND MINE AIR-COOLING PRACTICES

prepared by

Floyd C. Bossard

and

Koehler S. Stout

Montana College of Mineral Science and Technology

Butte, Montana

June 30, 1973

Special Report on USBM Sponsored Research Contract GO 122137

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# FOREWORD

This report was prepared by the Montana College of Mineral Science and Technology Foundation, Butte, Montana under USBM Contract No. G0122137. The contract was initiated under the Metal and Nonmetal Health and Safety Research Program. It was administered under the technical direction of the Pittsburgh Mining and Safety Research Center with Mr. Fred Kissell acting as the technical project officer. Mr. J. A. Herickes was the contract administrator for the Bureau of Mines.

This report is a summary of the work recently completed as part of this contract during the period July 1, 1971 to July 1, 1973. This report was submitted by the authors on June 30, 1973.

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## INTRODUCTION

As mines extend deeper, or encounter rock formations of high thermal gradient, the temperature of the mine environment increases to a point where ordinary ventilation methods are inadequate to keep the working place temperature within productive and safe working limits. Mr. Fenton (1972, p. 36-93) discusses sources of underground mine heat in his thesis. Briefly these sources of heat come from hot wall rock, hot water issuing from underground sources, adiabatic compression of the inlet air down deep shafts, operation of electro-mechanical equipment, heat exchange between hot-compressed air lines and hot mine-water drainage lines and the cool ventilation air in the entrance airway or shaft, oxidation of timber and sulfide minerals, heat liberated by blasting, friction and shock losses of moving air, body metabolism, and friction heat from rock movement.

Although preventative measures could be taken to minimize some of the heat sources enumerated above, the additive heat values of these sources are usually such that the ventilation air arrives at a working face at an uncomfortably elevated temperature.

Labor in the U.S. enjoys the highest standard of living in the world and a wide choice of employment. It is not reasonable to expect him to work under extreme temperature conditions. If a reasonably comfortable ambient environment is not provided in underground mines, we shall see a continued and possibly accelerated drift away from the industry. Additionally, under conditions of elevated temperatures, designated rest-periods will probably become mandatory under law, resulting in production losses and greatly increased production costs.

ABSTRACT

Major air-cooling facilities have been installed and are operating underground in U.S. copper mines (at Butte, Montana and Superior, Arizona), South African gold mines, European coal mines, Zambian copper mines, Kolar gold mines in India, Coeur d'Alene mines of Idaho, plus other locations. Mining districts that utilize spot cooling and local air-cooling facilities would increase the above listing many times.

The principal objectives of underground mine air cooling systems are to;

1. Extract heat and water vapor from the mine air and provide for their removal from the mine via (a) water pumped to surface or (b) discharged mine exhaust air.
2. Provide acceptable artificial ambient conditions for efficient working mine development and production areas.
3. Recondition used air, thus supplementing the limited mine ventilation balance.

In deep South African gold mines, when rock temperatures exceed about 105°F, cooling of underground air becomes necessary for the poorly ventilated sections of the mine. By U.S. standards, air conditioning in poorly ventilated mines usually commences when the virgin rock temperature exceeds 100°F. On the other hand, well ventilated deep vein-type mines in the U.S. have been able to forego general cooling of mine air until the mines have been extended to such depths that the virgin wallrock temperatures have exceeded 115° - 120°F.

It must be noted that spot-cooling of selected development headings may be required in many mines where rock temperatures exceed 100°F. Such local air-cooling practices permit the driving of long headings for exploration or extended development prior to holings which result in establishment of primary ventilation circuitry.

The following summary reports on current practices of cooling and dehumidifying mine air. General system descriptions with important details will hopefully permit the reader to gain an understanding of the subject. A list of references is provided for the serious investigator who desires a more thorough knowledge-base and who is responsible for the design of a mine air-cooling system.

## ENTHALPY AND PSYCHROMETRY OF MINE AIR

Texts on mine ventilation give detailed analysis of mine air and most of its properties which are of value to the mine ventilation engineer. Only the essential physical properties of mine air will be discussed in this section which relate to the problems of air-conditioning. The following properties of air will be defined as they will be used in this discussion.

| | |
|---|---|
| Density (w). | Weight of air per unit volume. Measured in lb/cu ft. |
| Enthalpy (h). | Total heat content of air - includes enthalpy of dry air and water vapor correlated to the unit weight of dry air. Measured in Btu/lb. |
| Entropy (s). | Ratio of heat transferred at a constant absolute temperature. Measured in Btu/lb. |
| Flow Rate. | Weight of air flowing per unit of time. Measured in lbs/min or lbs/hr. |
| Heat content change (q). | Usually rate of change of Enthalpy per unit of time. measured in Btu/minute or hour. |
| Horsepower (Hp). | Rate of performing work. A horsepower is equal to performing 33,000 ft. lbs of work per minute. Horsepower often rated as horsepower hour. ie. the average amount of horsepower required per hour. |
| Latent Heat (q) | Is considered the enthalpy of evaporation and is the amount of heat necessary to vaporize water at the wet bulb temperature. |
| Pressure (p). | Force exerted by the air on a unit area. Base reference may be absolute (0) pressure or atmospheric. Expressed in psi, psf, inches of mercury, or inches of water. |
| Pressure Head (H). | Height of a column of water equivalent to the pressure of air. Often used in ventilation work and checked with a manometer. Measured in./in. (water or mercury). |
| Quantity (Q). | Volume of air flowing per unit of time. Measured in cubic feet per Minute (CFM) |
| Relative Humidity(RH). | Ratio of vapor pressure of air at given conditions. Measured in percent. |
| Saturation, Degree of: | Ratio of weight of water vapor in air at given conditions compared to weight of water vapor at saturation at some given conditions. Measured in percent. |

Sensible Heat (qs) — The amount of heat necessary to raise one lb. air from 0°F to the dry bulb emperature and also to super-heat the vapor from the wet bulb temperature to the dry bulb temperature.

Specific Gravity(s) — Is the ratio of densities of a gas and air. Based on dry air as equal to 1.

Specific Heat (c) — Heat required to raise one lb. of a substance one degree F. The specific heat at constant pressure is usually used in air conditioning ($C_{pa}$). Measured in Btu/lb/°F.

Specific Humidity(W) — Another name for absolute humidity which is weight of water vapor contained per unit weight of dry air. Measured in grains or lb/lb.

Specific Volume(v) — Volume per unit weight of dry air. Measured in cu ft./lb.

Temperature Dew point $t_{dp}$ — Temperature at which water condences from air or the saturation temperature. Measured in °F.

Temperature Dry Bulb (td) — Temperature indicated by the regular dry bulb thermometer. Measures the sensible heat of the air and is expressed in °F.

Temperature Wet Bulb (tw) — Evaporation temperature of air. Indicated by a thermometer with a wetted wick & expressed in °F.

Total Heat ($Q_t$) — The sum of the latent and sensible heat measured above the 0°F datum. Expressed by carrier (1911,p 1005) as

$$Q_t = C_{pa}t + r'w + C_{ps}(t - t')w$$

$Q_t$ = total heat in Btu/.b.

$C_{pa}$ = Specific heat of dry air at constant pressure.

$C_{ps}$ = Specific heat of superheated steam at constant pressure.

r' = Latent heat of vaporization at wet-bulb temperature.

w = Weight of vapor in air in lb.

t = Dry-bulb temperature in °F.

t' = Wet-bulb temperature in °F.

From this equation the:

Sensible heat = $C_{pa}t + C_{ps}(t-t')w$. The small amount of heat required to heat the water from 0° to wet-bulb temperature is neglected. The first term represents the amount of heat necessary to raise dry air from 0°F to the dry-bulb temperature. The The last term is the heat required to heat the vapor from wet-to dry-bulb temperature.

Latent heat = r'w. This is the heat necessary to vaporize the water present at the wet-bulb temperature.

Vapor Pressure (pv) Partial pressure of water vapor and dry air equals the Barometric pressure. Measured in in. (mercury) or psi (absolute).

Velocity (v) Linear flow rate of air per unit of time. Measured in feet per minute (fpm)

Useful air relationships

| | |
|---|---|
| Atomic weight (assumed) | 29 |
| Specific gravity | 1 |
| R = Gas constant | 53.2 |
| Density (standard dry air at 60°F, Sea level, 14.7 psia) | 0.0761b/cu ft. |
| Barometric pressure (standard at sea level, 60°F) | 29.92"/Hg or 14.7psi |
| Specific heat at constant pressure | 0.24 Btu/ lb/°F. |
| n = ratio of specific heats at constant pressure & volume | 1.4 |

Pressure Conversion.

1 in. water = 5.2 psf = 0.036 psi

1 psi = 2.036 in mercury = 27.7 in. water

1 in. Mercury = 0.491 psi = 13.6 in. of water

$p = w. H. = w_2 H_2$

where p = pressure

w = density

H = head

## Heat Effects of Air-Vapor Mixtures in Underground Mining

Air has three temperatures, dry bulb, wet bulb and dew point. These temperatures are the same when the air is completely saturated with water vapor. If the air is not completely saturated, the dry bulb is the ordinary temperature. The wet-bulb temperature is found in the following manner; the bulb on the wet-bulb thermometer is covered with a wick made of silk or cotton. The wick is immersed in water so that it becomes completely saturated. The thermometer with the attached wick is then whirled or swung in the air until the temperature reaches a steady state. When swung in air that is not saturated, the wet-bulb temperature will be lower than the dry-bulb temperature. This can be explained as follows: The water vapor pressure in the wicks is greater than the partial pressure of the vapor in the air, hence water will pass by diffusion from the wick to the air. This causes a loss of heat from the thermometer liquid, so its temperature drops. But when the temperature of the thermometer liquid drops below air temperature, heat will flow from the air to the liquid. The greater the temperature difference, the greater the heat flow. Eventually a steady-state condition will exist where the heat leaving the wick is equal to the heat entering the wick. This temperature is known as the wet-bulb temperature and is the temperature water will reach as it is evaporated. In unsaturated air, the wet-bulb temperature lies between the dew-point and dry-bulb temperature.

If heat is added to the air with no increase of moisture, both dry-bulb and wet-bulb temperatures rise; if heat is taken away, both fall. The dew point will remain constant as long as the moisture content is constant. If moisture is added, the dew point and wet-bulb temperature will rise and the dry-bulb temperature will drop. If both heat and moisture are added, all three temperatures will rise. Because deep mines are both hot and moist, moisture and heat are common additives to mine ventilation air. Air conditioning in mines consists of the control of air velocity, and the dry-bulb and wet-bulb temperatures.

Fenton (1972) mentions that there is no standard base of measuring workers comfort and heat endurance. However, in most criteria the wet-bulb temperature is at least one of the limiting temperatures if not the major one. Warm air can carry much more moisture than cooler air. The following table shows this relationship for one pound of saturated air at 29.92 in. Hg.

| Temp., °F | Sensible Heat | Latent Heat | Enthalpy of Mixture Btu/lb | Grains of Moisture per lb. |
|---|---|---|---|---|
| 60 | 14.46 Btu/lb. | 11.97 Btu/lb. | 26.43 | 77.29 |
| 70 | 16.87 | 17.17 | 34.04 | 110.4 |
| 80 | 19.28 | 24.35 | 43.63 | 155.8 |
| 90 | 21.69 | 34.16 | 55.85 | 217.6 |
| 100 | 24.10 | 47.52 | 71.62 | 301.5 |
| 110 | 26.51 | 65.68 | 92.19 | 414.9 |

The thermal capacity of saturated air is not a constant figure, but varies considerably with temperature and slightly with pressure. At 30 in. Hg., the thermal capacity of saturated air (sigma heat increase per °F) is approximately;

| Btu/lb.air/°F | Deg.F. |
|---|---|
| 0.5 | 40 |
| 0.7 | 60 |
| 1.0 | 80 |
| 1.7 | 100 |
| 3.0 | 120 |

Figure 1 graphically demonstrates the psychometric and volumetric relationships of saturated air at 27.00 in.Hg.

At shallower depths, heat problems could be overcome by bringing in fresh ventilation air and exhausting the warm air. But at great depths and because of limited airway size, auto-compression of the ventilation air and heat build up from other fources (Fenton's Thesis), the fresh air brought in from the surface is at an uncomfortably high temperature. Any air that

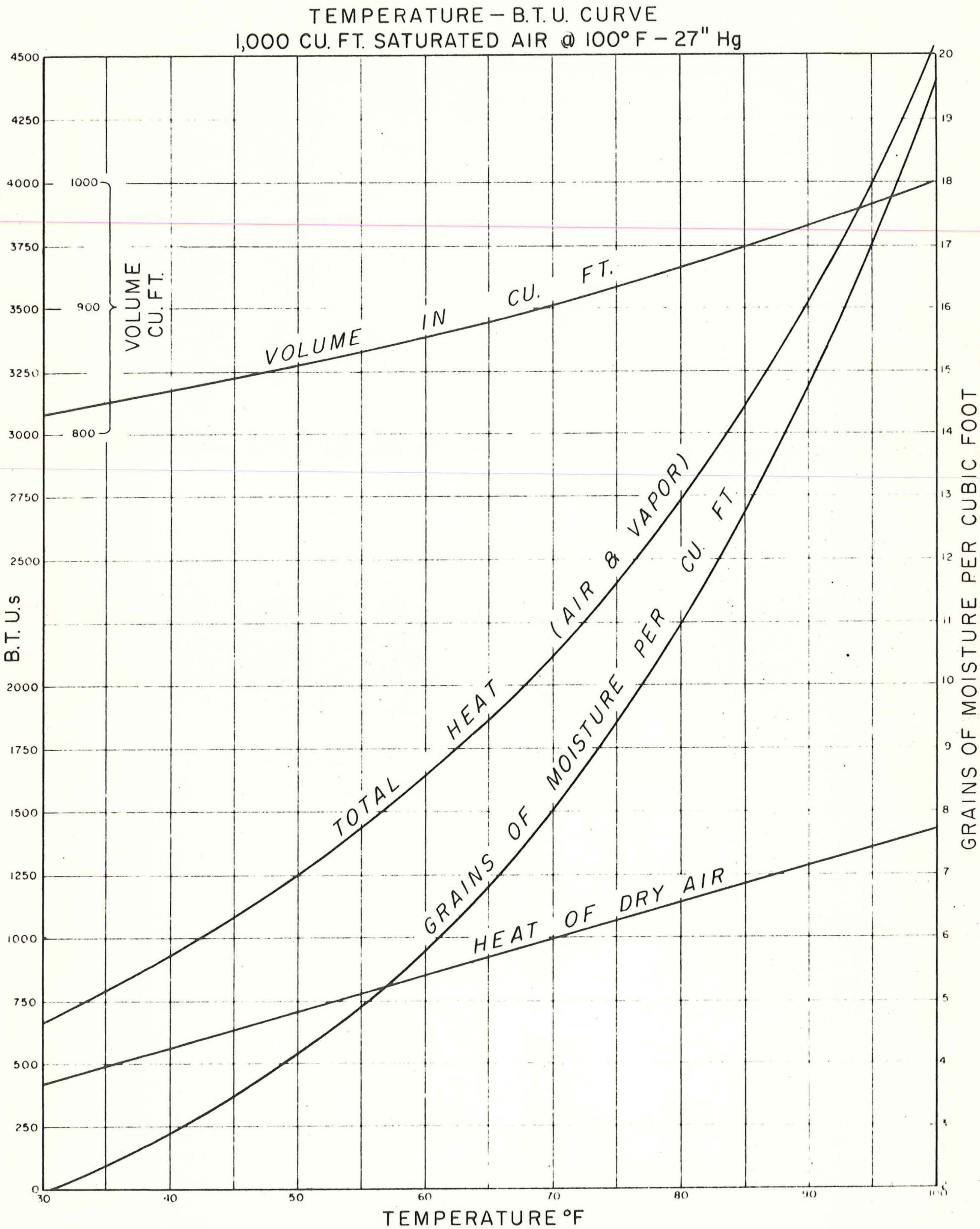


Figure 1 — Temperature — B.T.U. curve

moves along the levels or working places is subject to both moisture gain and heat sources, so heat build-up in the air is almost inevitable. The problem then consists of cooling the ventilation air so that when introduced into the working places, the air will be comfortable.

Cooling hot-saturated air will cause some of the moisture to condense out, which is one way of dehumidification. This action is beneficial because water droplets tend to form around fine-grained dust which then drop out of the air. This is advantageous in air conditioning because the dust count must be held below certain threshold limits. The underground procedure for air conditioning is to cool the ventilation air through a heat-exchange unit to about 70°F after washing it in a water spray and dust collector. The air as it comes out of the plant is saturated.

In spot cooling systems, every effort is made to keep additional moisture out of this air stream by confining it in a separate air duct. The temperature will rise in the duct, because of the hot mine air surrounding the duct. If no moisture is added and the air reaches the working place at a dry-bulb temperature of 90°F the wet-bulb temperature would be 76°F which is a tolerable working temperature. However if moisture were allowed to collect in the ventilation air, the wet-bulb temperature would approach 90°. This would be a very uncomfortable working temperature and the efficiency of the men would be greatly reduced.

Air velocity has a considerable effect on rate of cooling and comfort as mentioned in Fenton's (1972) thesis, but when the wet-bulb temperature approaches 90°F., velocity doesn't appreciably aid in the cooling power of air and hence the comfort of men.

In summary, an understanding of air enthalpy and psychrometry is essential for any engineer working in the area of underground mine cooling. Additionally, he has to appreciate the relationship of worker efficiency to ambient air temperature, humidity and velocity.

MINE AIR COOLING AND DEHUMIDIFICATION PRACTICES

When air is cooled underground, the heat taken from this air must be disposed of in some manner. In a water-air heat exchanger, the water is warmed while the air is cooled, and the warm water is either returned to the cooling plant (cooling tower, underground mechanical refrigeration unit, or heat exchangers) in a closed circuit or discharged to the mine drainage system, or sprayed into the mine exhaust system.

Water is an ideal heat transfer medium because one cubic foot of water with a 1°F. temperature rise will remove 62.4 Btu. One cubic foot of warm air may weigh about 0.075 pounds per cubic foot. The specific heat of air is 0.24. Hence one cubic foot of air would only remove 0.24 x 0.075 = 0.018 Btu per lb. per °F. Thus water is able to remove 62.4/0.018 = 3,467 more units of heat than a comparable volume of dry air. Water, if available, is undoubtedly the best heat transfer medium in a mine. A relatively small pipeline, servicing air-cooling plants, can remove as much heat from the mine as would be removed by a large volume of air which requires a large shaft and inlet airways.

Mine depth has a considerable influence on the need for cooling mine air. The energy content of the inlet air increases 1.285 Btu/lb. per 1000 ft. decrease in elevation. Simultaneously the air dry-bulb-temperature theoretically rises approximately 5.4°F/1000 ft. of depth. Geothermic gradients of the world's major mining districts vary from approximately 0.5°F to 4°F per 100 ft. of elevation decrease and even higher in local situations.

In summary, the major factors contributing to the necessity for general mine air cooling is a combination of increasing adiabatic heat, and increasing virgin rock temperature along with associated ground water. They make impressive contributions to the sensible as well as the latent mine air heat loads. The significance of these factors increase with depth.

To gain some appreciation of the installed mine air cooling capacities around the world,

1. In 1961 the South African gold mining operations had a total of sixty installations, with a capacity in excess of 30,000 tons of refrigeration per day.
2. The Anaconda Co's. Butte operations have an installed air cooling capacity of about 4,000 $T_r$/day.
3. The Magma Mine in Superior, Arizona has installed new facilities with a planned potential air cooling capacity of 5000 $T_r$/day.

A ton of refrigeration ($T_r$) is equivalent to the removal of 200 Btu/min. or 288,000 Btu/day.

## Surface Plants

With this type of installation, air is cooled on the surface and delivered to the intake shaft for delivery underground. The air is chilled by cold water or brine produced from mechanical refrigeration units. The refrigerant may be ammonia or freon and the condenser water is cooled in towers or spray ponds. Installations of this type have been used in Morro Velho, Brazil; Robinson Deep Mine, Rand, South Africa; Ooreyan Mine, Kolar, India; and Rieu du Coeur, Belgium. The major drawback to this type of installation is its poor positional efficiency. The air is heated up considerably by auto compression going down deep shafts and by increased heat flow from the wallrock of the inlet airways. By the time the air reaches the working headings, it has lost the bulk of its cooling capacity. Another disadvantage is that the average plant annual operating capacity is approximately 60 percent, thus requiring a large plant.

There have been no new systems such as these installed in the past 20 years, an indication of their deficiencies, plus the development of improved underground air-cooling equipment. The present trend is to install air-cooling plants underground with the cooling coils positioned as close to the working places as possible.

## Evaporative Cooling Of Mine Chilled Water

Geographic areas that enjoy a combination of low average winter temperatures and/or low relative humidity during the summer months are provided with unlimited natural cooling capacity at a temperature range adaptable to mine air-cooling methods.

A system in such an area cools water or brine at a surface cooling tower, from where it flows through a closed-circuit pipeline to an underground air-cooling plant close to the working zone, and returns to the cooling tower through a second pipe line. At the mine air-cooling plant, heat is absorbed by the water circulating through the system, and is dissipated to the surface atmosphere when the water is again cooled at the surface cooling tower. The pipe lines and underground plants form a closed-circuit piping system and the hydrostatic head is balanced. Power, furnished by a pump in the system, is required only to overcome frictional resistance to the flow of water.

A further advance in evaporative cooling of water in a tower was developed at the Anaconda Co. operations in Butte, Montana. This "dew point" cooling system reduces the temperature of the cooling medium to below the wet-bulb temperature of the surface atmosphere. See Fig. 2

Pre-cool coils are installed between the fan and cooling tower. Some cool water is pumped from the sump at the base of the cooling tower in counter-current flow through the tubes of the heat absorbers and on to the top of the cooling tower where this heated solution joins the warm water from the mine.

The cold solution within the heat absorbers is in indirect contact with the air flow. There is no addition or subtraction of moisture in the air stream passing over the heat absorbers. Therefore, the dew-point temperature remains constant. The heat content of the air is reduced and the equivalent heat is added to the solution circulating within the heat absorbers. The dry-bulb and wet-bulb temperature of the air entering the bottom of the cooling tower is less than the temperatures of the air entering the fan. Within limitations of economic design, the water leaving the tower will approach the wet-bulb temperature of the surface atmosphere.

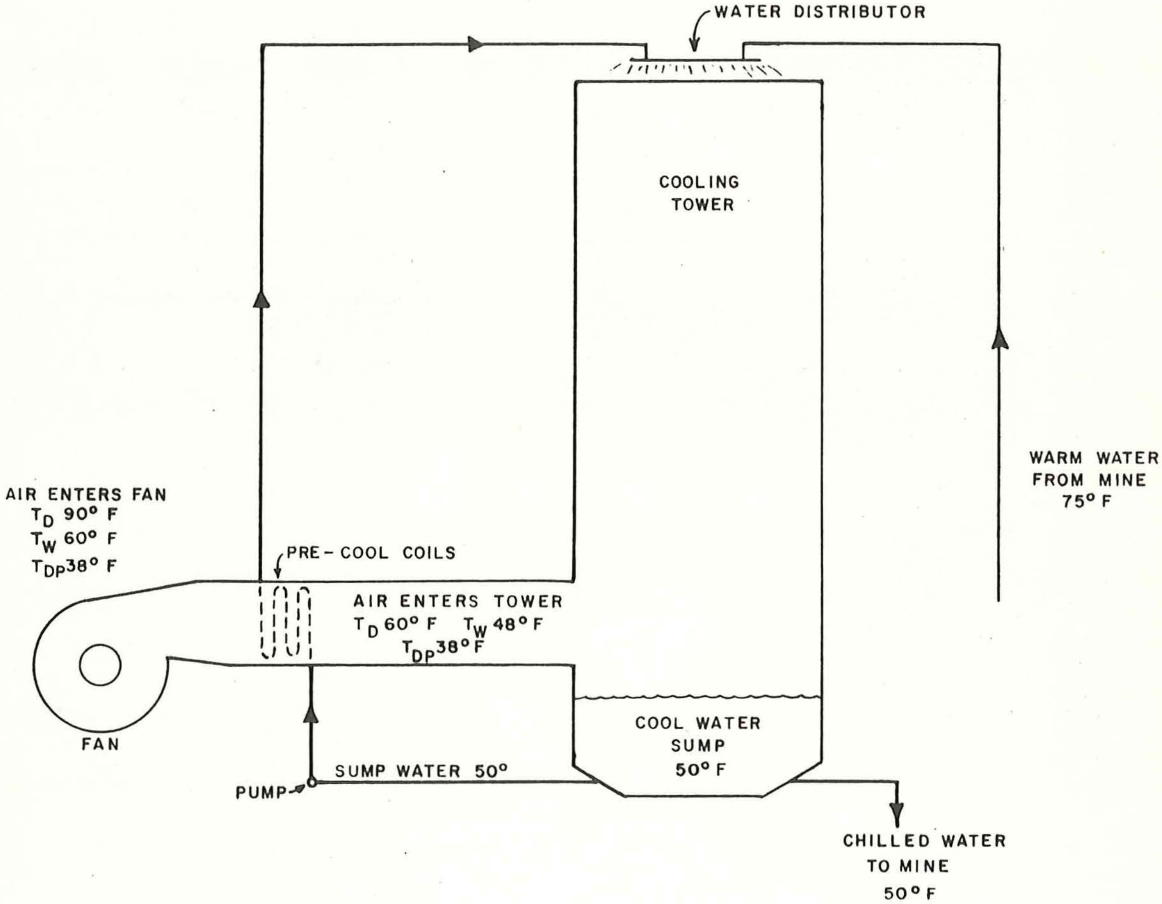


FIGURE 2- EVAPORATIVE COOLING TOWER SYSTEM IN BUTTE, MONTANA (RICHARDSON, 1950 p. 151)

Evaporative Cooling Plus Mechanical Refrigeration.

Some mines incorporate surface evaporative cooling towers and mechanical refrigeration units in series. The heat from the condenser is dissipated by water cooled in cooling towers or surface spray ponds.

In evaporative cooling towers, the entering wet-bulb air temperature is considered the evaporative temperature. However, on humid summer days, the wet-bulb temperature may increase over extended periods, severely hampering the effectiveness of the cooling tower. This factor plus the high-temperature air entering the mine may necessitate the series installation of a mechanical refrigeration unit cooling the chilled water delivered to underground. Facilities of this type are installed at operations of the Anaconda Company, Butte, Montana and the Magma Copper Corp., Superior, Arizona.

Underground Heat Exchangers

Chilled water in a closed circuit from surface cooling towers to underground is delivered to air-cooling plants at extremely high pressures, depending on the elevation difference from surface to underground plant locations. This high static head plus safety factor requirements necessitate that pipelines and heat absorbers be constructed of expensive, heavy duty materials. High construction costs place constraints upon frequent plant movement, made desirable by shifting mining operations.

In order to increase the flexibility of such a system, the Magma mine has installed heat exchangers underground at the mining horizon. A tube-and-shell heat exchanger converts the high-pressure chilled water system to a low-pressure system on the mine production levels. Air-cooling plants and chilled-water lines can be constructed of standard materials, permitting frequent relocation. The net result is to increase the plant positional efficiencies.

## Underground Mechanical Refrigeration Units

Mechanical refrigeration units are commonly installed underground to provide chilled water to low-pressure air-cooling plants. The source of condenser water is potable mine water or mine drainage water. Condenser heat is dissipated by pumping the condenser water to surface or to an exhaust airway where the heat is removed and the water recirculated to the mechanical refrigeration unit.

## Underground Heat Exchange

Three methods of transferring heat from the hot condenser water to the exhaust air or from hot ventilating air to chilled water include;

1. A vertical cooling tower.
2. A bank of sprays in a horizontal or inclined airway.
3. Pipe coils or tube-and-fin heat absorbers through which the water is circulated.

Pipe coils are more effective for cooling air than for cooling water unless they are kept wet. Tube-and-fin heat absorbers loose efficiency as scale builds up on inside and as dirt and scale accumulate on the outside. Coils suffer from similar problems, but on the whole are less susceptable as they are more easily cleaned. However when the two are compared efficiency-wise, a much larger coil installation is required to provide the same heat transfer.

The plants may also be constructed to perform as very effective dust collectors. The lowering of the mine air temperature causes condensation of water vapor. Dust particles assume weight by the condensation of the water vapor upon the dust particles acting as a nuclei and are removed from the air stream. This type of dust collecting requires the installation of sprays and impact plates or similar effective impingement system in series upstream from the heat absorbers. Aside from the capital cost, this feature increases operating power costs, as approximately one inch of water gage shock loss is experienced across the impact plates. This system has greatest advantage when the air conditioning plants are recirculating a sizeable air component.

The design of air-cooling plants should be directed toward installing an efficient heat transfer system so that the plants will cool 25,000 cfm of saturated air (density = 0.075 lbs/c.f.) from 85°F when provided with 125 gpm of water at 60°F. The system's tons of refrigeration capacity is not the most important factor. Of very considerable significance is the temperature range of the heat being removed. A $T_r$ when cooling air to 75°F is not as beneficial as a $T_r$ that is cooling air to 65°F.

## MECHANICAL REFRIGERATION

The mining industry uses mechanical refrigeration units in four general ways to provide underground air cooling;

1. Located on surface to provide a chilled fluid used for cooling entering mine air at the top of inlet shafts.
2. Located on surface separately, or in series with evaporative cooling towers, to provide a chilled-water source for delivery underground to air-cooling plants. This chilled-water refrigeration system transfers the heat of the ventilating air to the refrigerant via a chilled-water system.
3. Located underground to provide a chilled-water source for delivery to underground air-cooling plants. See Fig. 3.
4. Located underground in series with refrigerant-expansion coils. Air is passed over the coils, cooled and delivered to a working heading (s). This direct-expansion refrigeration system transfers the heat removed from the ventilating air directly to the refrigerant.

The leaving condenser water from surface-located units is pumped to associated evaporative cooling towers or ponds. The water is cooled and returned to the condenser for reuse in a closed system.

Underground mechanical refrigeration plants are located entirely underground and hence avoid any heat addition caused by auto compression of air coming down the shafts. The main disadvantage of these units is the disposal of heat from the condensers. In most underground plants, the hot condenser water is either run into the mine drainage system, returned to surface for cooling, or the hot water is sprayed into the mine discharge air system.

The hot water discharged from the condenser of underground mechanical refrigeration units may be rejected into the sumps of the mine-water pump-

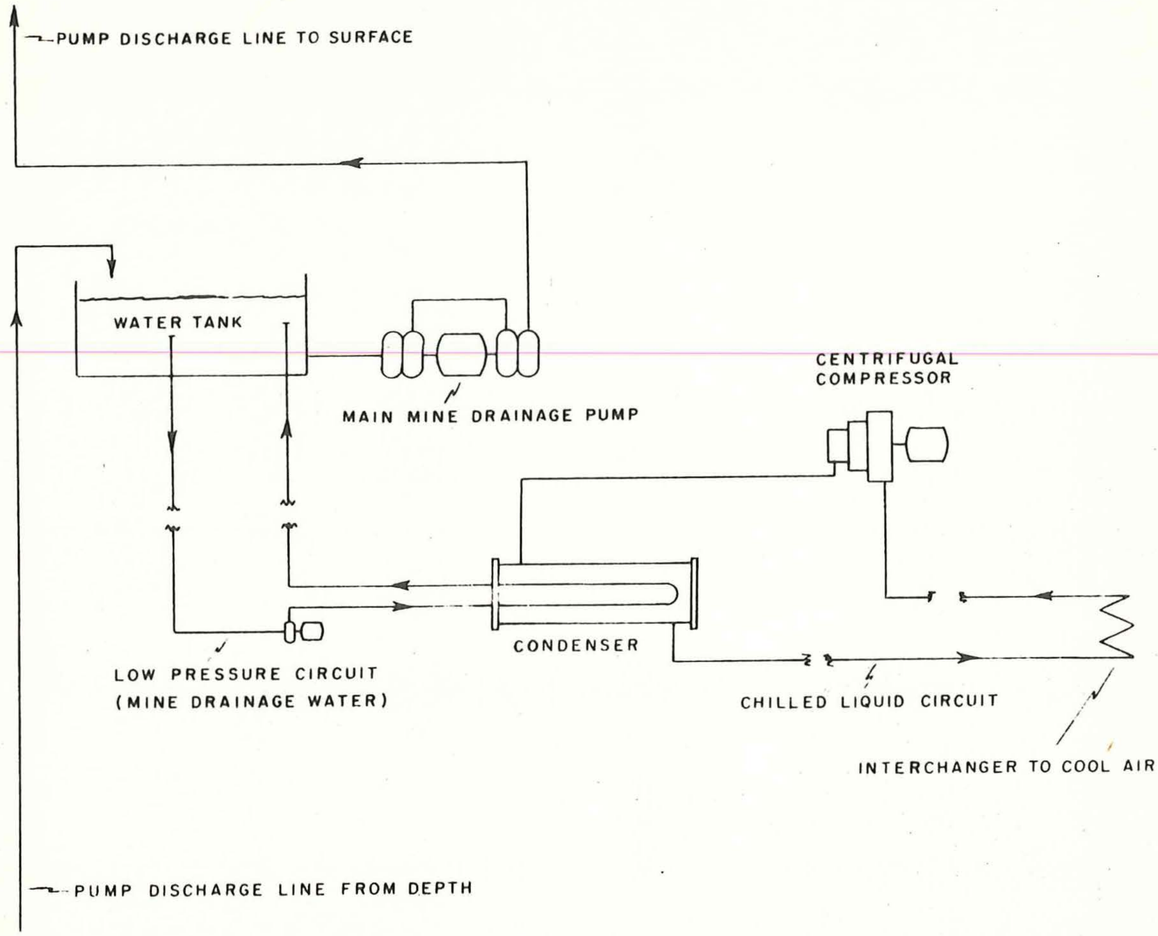


Figure 3 – Diagramatic sketch of mechanical refrigeration

ing system, necessitating a constant resupply of condenser water from fresh water lines or mine drainage sources. This former method requires the assumption of increased mine pumping costs.

Another technique is to recirculate condenser water from underground plants by rejecting the heat into the exhaust air by spraying the water into a return airway. This system requires close control of dissolved and suspended solids in order to limit condenser scale build-up to an acceptable rate. Intimate contact between the air and water requires a system of fine sprays to provide the greatest efficiency for heat exchange.

The smaller mechanical refrigeration units, up to about 100 tons of refrigeration capacity, may incorporate a reciprocating compressor or a centrifugal compressor. For larger air cooling systems, the industry currently favors the centrifugal compressor. Units are ordinarily semi-hermetic or hermetic type, where the electric motor and compressor are within the same housing and in the refrigerant atmosphere. This latter arrangement reduces the problem of maintaining shaft seals for refrigerant containment.

The refrigeration cycle works as follows. See Fig. 4 . Low pressure vapor at point (1) is compressed to a high pressure vapor at point (2) by the compressor. It enters the condenser where it is cooled by the cooling water circulating in a separate circuit in the condenser. The condenser cooling water is warmed while the hot-compressed refrigerant is cooled. The cooled refrigerant condenses to a liquid, giving off the heat of condensation which the condenser cooling water absorbs and carries away. The condensed or liquid refrigerant (3) now passes through an expansion valve to a low pressure side. (4) Because the liquid refrigerant will boil at a much lower temperature in a low pressure atmosphere, it then boils to a vapor in the evaporator taking heat from the water entering the evaporator. Because it requires heat to boil or vaporize the refrigerant, the evaporator cooling is cooled. The cooled vapor is now picked up at (1) by the compressor and the cycle is repeated.

The chilled water or brine from the evaporator is the medium pumped into the mine level for use in heat exchangers in the hot working parts of the mine. Another type of mechanical refrigerator, commonly called a spot cooler, is used where instead of water, air is used to supply heat to the evaporator. The cooled air from the evaporator is then carried into the working headings through ventilation lines. See Fig.5 . The principle of operation is the same as in Figure 4 except for the substitution of air as a heat source in the evaporator.

The refrigerant is any substance which acts as a cooling agent by absorbing heat from another body or substance. It is the working fluid of the cycle which alternately vaporizes and condenses as it absorbs and gives

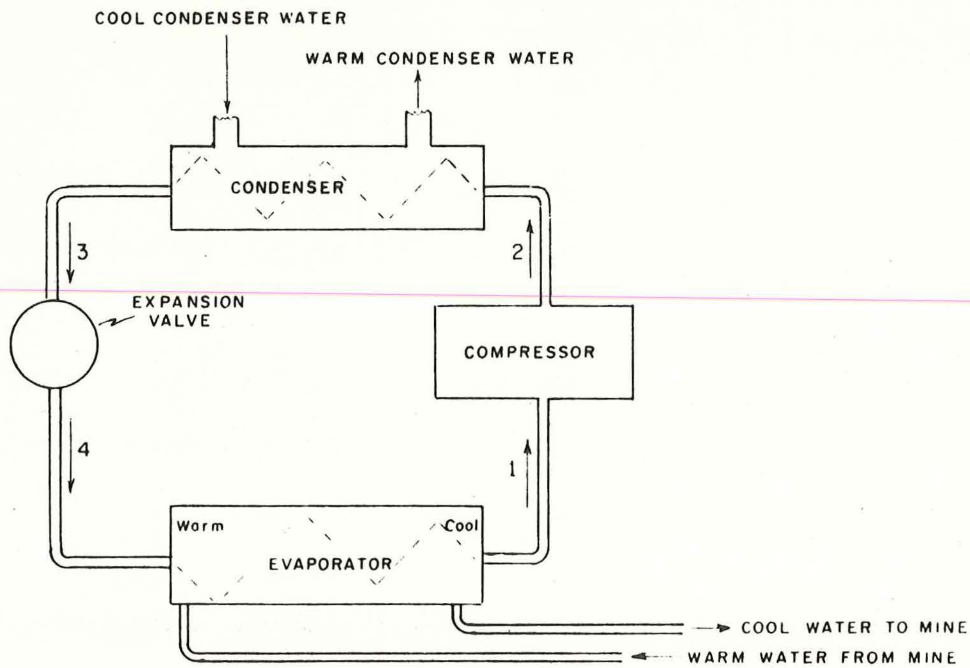


Figure 4 — Vapor-mechanical refrigerator with water chill evaporator for mine use

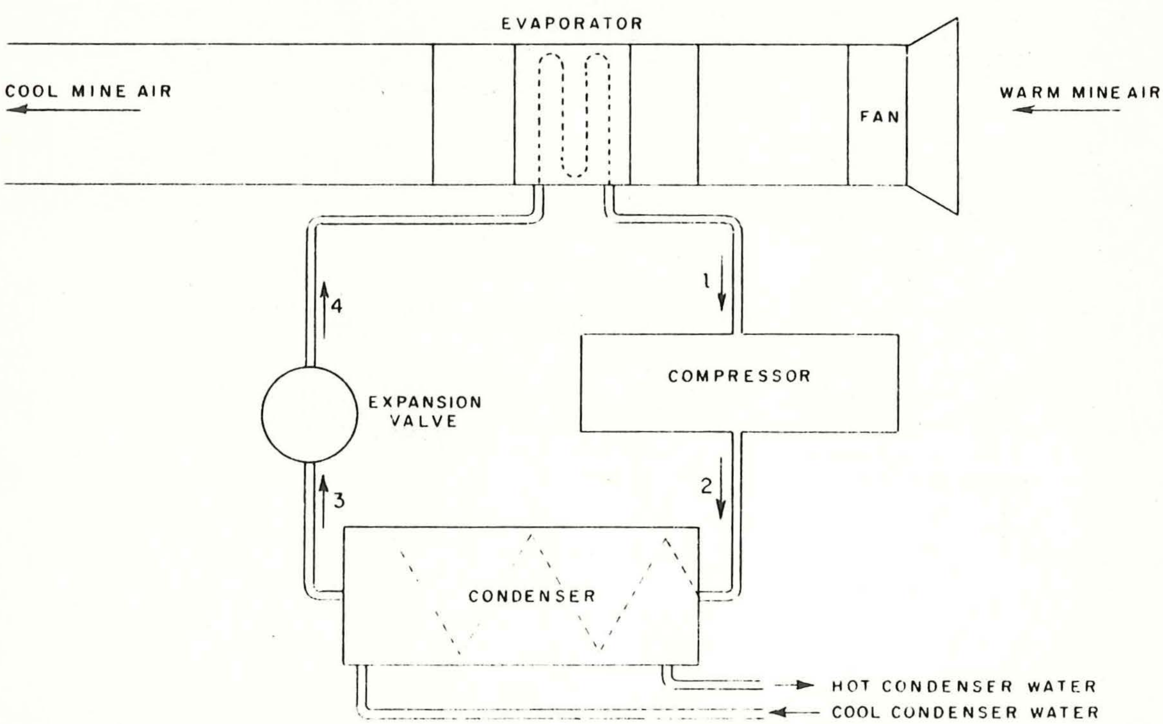


Figure 5 — Vapor-mechanical refrigerator with air chill evaporator for mine use

off heat respectively. A refrigerant must possess certain chemical, physical and thermodynamic properties that make it both safe and economical to use. There is no ideal refrigerant that is applicable to all situations. Ammonia probably comes closest to the ideal refrigerant because it has the highest refrigerating effect per pound of any refrigerant, but it is toxic and therefore cannot be used underground. The freon 11,12, and 22 type of refrigerants are popular for underground use because they are non-toxic.

A typical general specification sheet on a hermetically sealed, centrifugal refrigeration machine would include the following.

| | | |
|---|---|---|
| DIMENSIONS: | Approximate Over-all: | Length = 18'6" |
| | | Width = 10'6" |
| | | Height = 9'6" |
| COOLER: | Tons | 775 |
| | Entering Water Temp (°F) | 80 |
| | Leaving Water Temp (°F) | 55 |
| | Fouling Factor | .001 |
| | GPM | 744 |
| | Passes | 4 |
| | Pressure Drop (Ft.) | 26.5' |
| CONDENSER: | Entering Water Temp. (°F) | 95 |
| | Leaving Water Temp. (°F) | 110.4 |
| | Fouling Factor | .002 |
| | GPM | 1500 |
| | Passes | 4 |
| | Pressure Drop (Ft.) | 14' |
| COMPRESSOR MOTOR: | | |
| | Full Load Amps | 104 |
| | Current Characteristics | 4160/3/60 |
| PURGE UNIT MOTOR: | | |
| | Horsepower & Characteristics | 1/2 220/440 |
| OIL PUMP MOTOR: | | |
| | Horsepower & Characteristics | 1/2 220/440 |

## High-Pressure Heat Exchange System

This section of the report will discuss high-pressure heat exchange systems installed at the Butte, Montana and Coeur D'Alene, Idaho mining districts.

### Butte Mining District

The Steward Mine of the Anaconda Mining Co's Butte operations incorporates an air conditioning system where water is cooled by an evaporative cooling tower and an 800-ton mechanical refrigeration unit in series at surface. The water flows through a partially insulated pipeline in closed circuit to underground air cooling plants (up to 4500 ft. below surface), and returns to the surface cooling tower through another pipeline in a closed-loop system. The chilled-water passes through heat absorbers while warm mine air passes on the outside in a countercurrent manner. Heat is absorbed from the mine air into the water circulating through the pipe system and dissipated into the surface atmosphere by cooling the water in the surface cooling tower and refrigerating machine. The pipelines form a closed-circuit; hydrostatic head is balanced; and power, furnished by a pump on the return line, is required only to overcome frictional and shock resistance to the flow of water.

The main pipelines down the shaft and on the mining levels are weldless tubing, grade B low carbon steel with wall thickness sufficient to sustain the hydraulic pressure exerted by 5000 ft. water head, with a minimum safety factor of 4.

At the underground plant, air cooling occurs when the air is passed over "extended surface type heat absorbers." Cold water is circulated through copper coils countercurrent to the air flow. Copper sheets or "fins" are mounted on the copper coils, forming extended surfaces for transfer of heat and increased air contact. The heat absorbers are designed to withstand the 5000 ft. static head, with a safety factor of 4.

The following data table summarizes performance of the Steward Mine air-cooling system on a typical spring day.

Surface Water-Cooling System

| | |
|---|---|
| 1224 $T_r$/Day, | Evaporative Cooling Tower |
| 816 " ", | Mechanical Refrigeration Unit |
| 2040 " ", | Total |

1750 gpm = Water volume circulated

68°F = Water Temperature leaving mine

40°F = " " entering "

Air-Cooling Plants Underground

358,000 cfm, Total air volume cooled

80.3°F D.B., 74.3°F W.B., Ave. entering air temp.

56.4 " " , 56.4 " " , " disch. " "

42°F, Entering water temp.

67°F, Leaving " " .

364,000 BTu/min., Heat extracted from air

1820 $T_r$/Day, " " " "

## <u>Coeur D'Alene District</u>

The Crescent Mine of the Bunker Hill Company has been operating a 300-ton air-cooling system since August 2, 1967. Three hundred gallons per minute of water are pumped in an insulated line from a surface well a distance of 4400 feet horizontally (primarily through an adit) and dropped 3100 feet down a shaft, where it travels horizontally for another 2500 feet. At this point the water is circulated through cooling coils, and then returned to surface by a second return pipeline. The pipelines are unbroken from surface to underground and the system operates at a maximum head approaching 1400 psi. See Fig. 6.

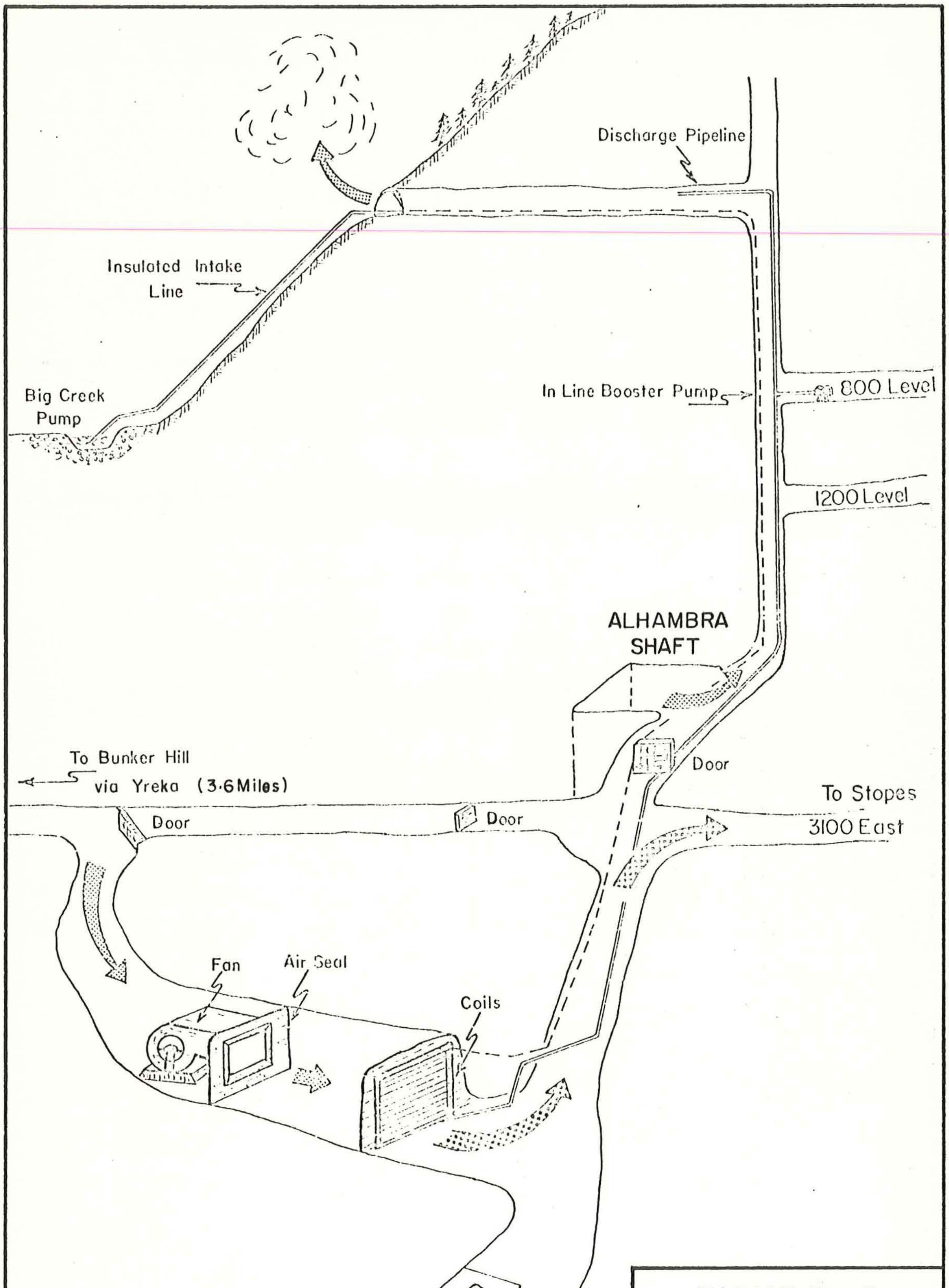


FIGURE 6
CRESCENT MINE
Pictorial Sketch
AIR COOLING PLANT



Reference 28

Heat transfer takes place in a bank of heat absorbers 10 feet wide, 12 feet high, and 1 foot deep. The heat absorbers consist of eight rows of copper tubes horizontally, with 14 copper fins per inch.

The following data summarizes performance of the air-cooling plant during October.

46,000 cfm of air was cooled

89°F saturated, entering air temperature

75°F " , discharge " "

54°F,entering water temperature

83°F, leaving " "

60,000 Btu/min., heat extracted from air

300 $T_r$/Day, " " " "

The temperature of the cooling water entering the mine at surface was 52°F.

Low-Pressure Heat Exchange System

Mechanical refrigeration plants may be located on a level where exhaust-air cooling of condenser water is practical or condenser water is available. Chilled water from the refrigeration unit is ordinarily delivered to heat exchangers situated near the production headings. The vertical distance over which the equipment is installed is normally limited to several levels so that standard mine piping is sufficient for making up the installation.

The Star Mine of the Hecla Mining Co., Burke, Idaho, has vapor-type mechanical refrigerators on several levels where most of the underground coolers are located so the chilled water is available at a low pressure circulating throughout the system. Hence their design, see Figure 5, can be light weight because the coils do not have to withstand high fluid pressures.

The latest installation, on the 7500 level, includes a rotary-screw mechanical refrigeration chiller package, 230 $T_r$/day capacity, with a

300 h.p., 2300 volt, 3600 rpm motor. The chilled water system uses a 40 h.p. pump to circulate 500 gpm of chilled water via 6-inch steel water lines to individual cooling coils. Operating data for the system include the following.

Mechanical Refrigeration Chiller Unit.

56°F, entering chilled-water temperature

45°F, leaving " " "

80°F, entering condenser water temperature

104-114°F, leaving condenser water temperature

68 psi, suction pressure

250 psi, discharge pressure

Cooling Coil Performance (typical)

1. Level development

   30" x 48" copper tube, aluminum fins, 6-row, 8 fins/inch.
   40 $T_r$/day, rating

   80°F saturated, entering air temperature
   70°F " , leaving " "
   12,000 cfm, air volume
   45 $T_r$/day, calculated air-cooling load.

2. Shaft ventilation

   Same unit as 1, above
   80°F saturated, entering air temperature
   64°F " , leaving " "
   12,000 cfm, air volume
   76 $T_r$/day, calculated air-cooling load

3. Production heading

   24" x 36" copper tube, aluminum fins, 6-row, 8 fins/inch
   20 $T_r$/day, rating
   80°F saturated, entering air temperature
   68°F " , leaving "
   6,000 cfm, air volume
   22.5 $T_r$/day, calculated air-cooling load

4. Production heading

Same unit as 3, above

80°F saturated, entering air temperature

68°F " , leaving " "

6,000 cfm, air volume

31.5 $T_r$/day, calculated air-cooling load

The mechanical refrigeration chiller unit was not operating at full rated capacity as additional cooling coils were scheduled for installation. Chilled water flow would be adjusted as more coils are added.

---

Latest E.R.P.M. underground cooling plant (Ref. 25)

Point of installation, 68 level 'H' Shaft

Make, Hitachi

No. of units, 2

Rated tonnage of each unit, 495 refrigeration tons

Refrigerant, F11 (C.C.L.3.F)

Freon charge per unit, 2000lb

Compressor data

| | hp | Speed |
|---|---|---|
| Electric motor | 750 | 1500 |
| Compressor gear | 13 | 1500-5600 |
| Compressor | 700 | 5600 |
| Capacity control | 30 to 100% | |

| Tubes | Evaporator | Condenser |
|---|---|---|
| External diameter | 5/8 in. | 5/8 in. |
| Internal diameter | 35/64 in. | 35/64 in. |
| No. of tubes | 564 | 660 |
| Tube material | 90/10 Cupro nickel | 90/10 Cupro nickel |
| Water velocity | 5.7 ft/sec | 7.1 ft/sec |
| Refrigerant temperature | 36°F | 116°F |
| Water temperature in | 68.9 | 105.1 |
| out | 43.5 | 113.7 |
| Gal/min | 330 | 1400 |

Recent heat balance: heat exchangers to Hitachi cooling plant evaporator

| Position of heat exchanger | Heat added, tons of refrigeration | | |
|---|---|---|---|
| | At heat exchanger | Insulated chilled-water columns to heat exchangers | Uninsulated return-water columns from heat exchangers |
| 66 Level No. 1 | 51.6 | 1.8 | 5.9 |
| 66 Level No. 2 | 37.5 | 9.4 | 2.8 |
| 68 level East | 56.5 | 23.6 | 30.0 |
| 71 level East | 94.5 | 8.2 | 6.9 |
| 71 level West | 97.3 | 8.5 | 6.7 |
| 73 level East | 45.8 | 2.3 | 2.3 |
| 73 level West | 77.8 | 2.3 | 2.3 |
| 74 level East | 57.0 | 2.2 | 1.8 |
| 74 level West | 52.5 | 1.5 | 1.0 |
| 75 level Station | 54.0 | 0.7 | 0.3 |
| 75 level Cross-cut | 31.5 | 0.5 | 0.5 |
| Total | 656.0 | 61.0 | 60.5 |

| | |
|---|---|
| Total heat pick-up accounted for | 777.0 tons of refrigeration |
| Evaporator tonnage at plant | 812.0 tons of refrigeration |
| Refrigeration tons unaccounted for | 35.0 (4.3%) tons of refrigeration |

Reduced-Pressure Heat Exchange System

To overcome the problem of high-pressure in the cooling system on the levels, a heat exchanger was installed on the 3200 level at the Magma copper mine in Superior, Arizona.

A diagram of this system is diagrammatically illustrated in Figs. 7 & 8.

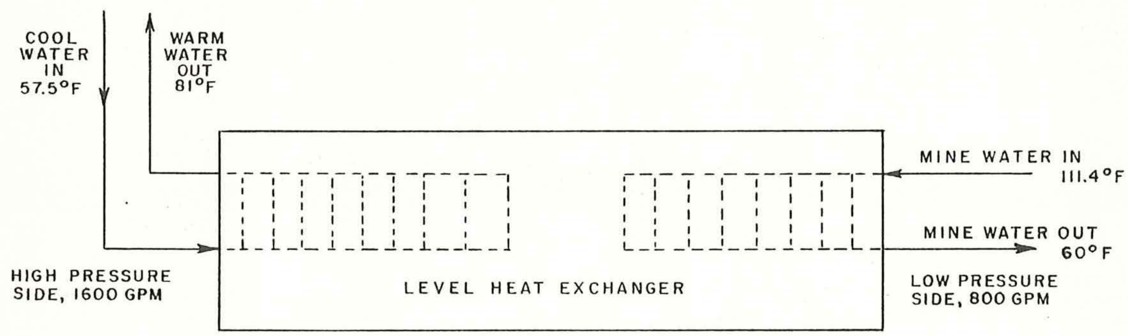


FIGURE 7—LEVEL HEAT EXCHANGER IN MAGMA MINE, SUPERIOR, ARIZONA

The installation included three sets of heat exchangers, and consisted of four shells per unit. Utilizing 1600 gpm of 57.5°F water from the high-pressure system, 800 gpm of water in the mine low-pressure system was to be reduced in temperature from 111.4°F.to 60°F. (design specifications). The units incorporated copper tubes and a steel shell.

The heat exchangers are shell and tube construction. High-pressure water from surface flows inside the tubes. Low-pressure water serving the mine air cooling plants is circulated in the volume between the shell and the tubes. The major reasons for using water-to-water heat exchangers in place of utilizing chilled water from surface directly into high pressure coils are (1) elimination of the expense of installing high-pressure lines throughout the mine, and (2) reducing the prohibitive cost of limited-life heat absorbers designed for a working pressure of 2500 psi.

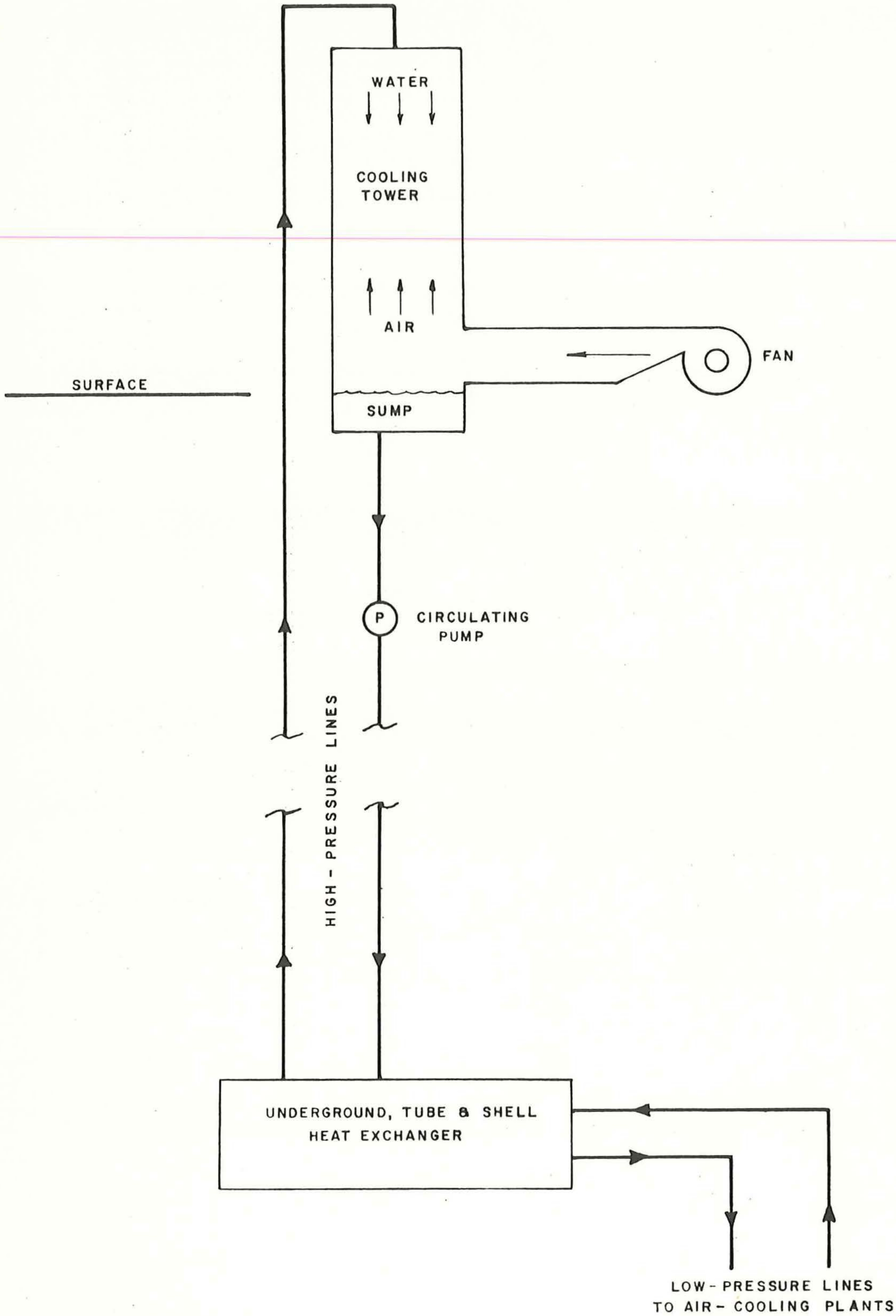


FIGURE 8.—UNDERGROUND HEAT EXCHANGER, PRESSURE-REDUCTION SYSTEM

The above described system was replaced by a 6300 $T_r$/day facility at a new section of the Magma Mine. This new system again incorporates surface cooling towers that cool the return water from the mine. Refrigeration units, interposed on surface in series with the towers, are utilized during the warm summer months to maintain a chilled-water product to underground. The chilled-water line and warm return line are installed in the No. 9 shaft. Chilled water in the high-pressure insulated shaft lines is passed through a tube-and-shell heat exchanger underground. A low-pressure, chilled-water system will (1) service the air-cooling plants and (2) provide the condenser water to six 140-ton carrier refrigeration units servicing additional air-cooling plants. The condenser water is the warm discharge from the original cooling plants. Condenser discharge water will then course to the heat exchanger, loosing accumulated mine heat to the return high-pressure insulated pipeline to surface.

## Direct-Contact Heat Exchange System

Chilled water may be used to cool mine air in two ways, by (1) direct contact, or (2) by indirect contact. The first method consists of spraying the water through multiple stages of banks of sprays countercurrent to the airstream. Ordinarily, if three stages of sprays are utilized, the water is successively sprayed into the airstream, picked up again in a collecting sump by a circulating pump and sprayed into the airstream (twice). Countercurrent exposure of air to the water means that the coldest stage of spray water is the last to come in contact with the air stream. Banks of sprays, in addition to cooling the airstream, perform the function of partially removing gases and dusts from the mine air.

Warm, humid mine air is dehumidified and cooled through transfer of of latent and sensible heat to the sprayed water. Efficient heat transfer by a direct-contact spray plant is more difficult to obtain than with banks of coils, which the mining industry relies upon to do the bulk of air cooling and dehumidification underground.

Spray cooling is ordinarily cheaper to install, but more expensive to operate than banks of coils, and offers less resistance to the air flow. Cooling plants with a potential long life should be installed with heat absorbers, all factors considered.

An example of an individual four-stage air conditioning plant, using acid mine drainage water in counter current flow to the air component to be conditioned has been described as follows by Mr. J.W. Warren (Ref. 34).

The acid mine drainage water is accumulated and impounded to permit removal of particulate material. The water is coursed through non-corrosive or plastic pipe to the location of the air conditioning plant.

The hydraulic head within the water transmission line furnishes the pressure for proper atomization through the correct number of sprays in the initial stage of the air conditioning plant. A pump, non corrosive construction, safeload type, draws the water from the initial sump and delivers water volume at least equal to and not less than the volume of water entering the initial stage to sprays in the subsequent stage. Three pumps perform the indentical operation in each of the three subsequent stages. The initial volume of water plus the water of dehumidification is discharged through a water seal to the mine drainage system.

A safeload fan induces airflow through the plant. The fan has a pressure characteristic sufficiently high enough in magnitude to overcome the resistance of the plant, about 1.5 inches water gauge, and the series portion of the ventilation circuit.

A representative performance tabulation follows:

The heat transfer balance on this plant shows that 41,000 cubic feet of air per minute, having a density of 0.0646 pounds per cubic foot is cooled from 85°F. dry bulb and 84°F. wet bulb, to 61°F. saturated. Two hundred and forty gallons of acid mine drainage water per minute, having an intake pH of 2.8 is raised in temperature through the four stages of the plant from 53°F. to 80°F. or 27° increase. The wet bulb temperature of the air is lowered 23°. The temperature difference of the air entering (wet bulb), and water leaving the plant is 4° and the temperature difference between the air leaving the plant and the water entering is 8°F. The mean temperature difference is 6°F. With a four stage plant this is considered to be good approach temperatures.

The plant performance indicates that approximately 55,000 B.T.U. per minute were interchanged from the air to water. This performance is approximately 275 tons of refrigeration per day.

The water vapor content of the air was lowered from 12.23 grains per cubic foot at the intake of the plant to 5.94 grains per cubic foot at the discharge of the plant. The extraction of 6.29 grains of moisture in each cubic foot of air through the plant accounts for 4.4 gallons of water per minute as dehumidification. The air entering the plant has a relative humidity of 96% and the discharge air was saturated at 100% relative humidity.

## Indirect Contact Heat Exchange System

The indirect contact method utilizes heat absorber units similar in construction to tube and fin radiators. The chilled water is circulated through the heat absorber tubes, again in a counter current manner relative to the airstream. Heat transfer is enhanced by thin-wall tubes and close-fin spacing and high thermal-conductivity metal construction. Copper is a very efficient conductor material. Because of the tendency of finned coils to become blinded with airborn dust and requiring scheduled cleaning maintenance, heat exchangers with bare coils, although both larger in size and less efficient, have found application in the industry.

### Butte, Montana

The underground air-cooling plant consists of a fan, water spray section, impact-scrubber plates, and a heat absorber section. A typical cooling plant used in Butte, Montana is shown schematically in Fig. 9.

Warm-humid mine air passes through the fan to an air-washing chamber, where the air is thoroughly scrubbed. The scrubbing section serves the two-fold purpose of increasing the removal of noxious gases and dusts, and maintaining the inlet sections of the heat absorbers in a wetted condition, resulting in a higher heat-transfer coefficient.

A series of impact plates (incorporating offset, slotted holes) removes the airborn water and a portion of the soluble noxious gases and dusts. The low-pressure scrubbing water returns to the sump, from where it will be pumped back to the atomizing sprays.

The scrubbed air then passes over fin-coil type heat absorbers where it is cooled down to the 60-65°F temperature range before it is discharged into the mine ventilation circuit again. Air velocity is in the range of 1000 fpm. The copper heat absorbers, with a heat transfer surface of approximately 9000 sq. ft. per 25,000 cfm of air, compose two parallel banks with two banks in series.

Because the air entering has a high humidity, cooling of the air causes moisture to condense and the plant will make water. Therefore, the low-pressure scrubbing water is continually being diluted by the water of dehumidification. Excess water continually overflows to mine drainage.

The following data illustrates the high performance characteristics possible in an air-cooling plant of The Anaconda, Co.;

1. Fan Inlet → Fan Disch. → Plant Disch.

| Fan Inlet | | Fan Disch. | | Plant Disch. | |
|---|---|---|---|---|---|
| 79.0 / 76.5 | → | 82.0 / 77.5 | → | 58°F D.B. / 58°F W.B. | Air |
| | | 75°F | ← | 55°F | Water |

2. 94,400 Btu/min., calculated heat removed from air

   472 $T_r$/day., " " " " "

3. 566 gpm of chilled water to plant.

4. 76,000 cfm of air through plant

## South African Operations

The air-cooling practices of the South African gold mining industry are reported on in Reference 25, and others.

Chilled water temperatures range between 45 and 70°F. Cooling coil units are usually designed for air capacities from 7500 to 15,000 cfm while maintaining air velocities of 400-500 fpm. Typical duty will cool air from 88-90°F W.B. to 75-80°F W.B. Cooling plants are relocated as mining requirements demand (from a few months to a few years).

Both large unfinned coils and smaller finned coils are utilized. Finned coil corrosion and fouling are matters not yet fully resolved. The choice of unfinned coils was considered of great importance because experience with closely-packed finned tubes in spot coolers proved that the ease of cleaning was essential. Despite size and weight disadvantages these coils have proven very satisfactory in past years. However, since the smaller and lighter finned coils (same performance efficiency) are so much easier to handle and install, newer units purchased are generally of this type, with tube and fins lead-coated.

## Cooling Coils & Fan Position

In visiting underground cooling plants, the position of the fan in relation to the cooling coils was noted. In Figure 9, the fan is upstream from the wet scrubber and cooling coils while in Figure 10. the fan is downstream from the cooling coil.

By theoretical adiabatic compression, compressing air 1 in. water gauge will raise the air temperature about 0.375°F. The calculations are as follows:

$$T_1 = T\left[\frac{P_1}{P_o}\right]^{\frac{n-1}{n}}$$

where $T_1$ = Final Temp Abs.

$T$ = Initial Temp. Abs.

$P_1$ = Final Pressure Abs.

$P_o$ = Original Pressure Abs.

$n$ = 1.406

Initial Conditions

$T$ = 535.000°F

$P$ = 14.700 psia

1" water = 0.036 psi

$P_1$ = 14.736 psia

$$T_1 = 535.00 \left(\frac{14.736}{14.700}\right)^{\frac{0.406}{1.406}} = 535 (1.0025)^{0.289} = 535.375°F$$

Fenton (1972, p. 80) states that one researcher found that the temperature rise across a fan is about 0.8°F per inch water gauge which includes both heat from the fan motor and adiabatic compression.

For steady state unidirectional heat flow, the Fourier's equation is in general use, which is:

$$Q = -KA\left(\frac{dt}{dL}\right) \text{ Btu/hr.}$$

where
Q = heat transfer Btu/hr.
K = thermal conductivity
A = Area
dL = Wall thickness
dt = Tem. difference

The efficiency of a cooling coil depends, among other things, on the temperature difference between the air flowing over the coils and the water circulating in the coils. Therefore the greater the difference in temperature, other things being equal, the greater the heat flow. Because a fan warms the air, and it is placed ahead of the coil, there would be a greater temperature difference between the air and water in the coils so a greater heat flow would result. Assume that the water in the coil was at 55°F and the air was at 80°F. If the fan is working against 6" of pressure, the air entering the coil would be about 80 + 6 x .8 = approp. 85°F. No effect of scrubber sprays is taken into account for this example. The heat transfer, being more efficient because of the greater temperature difference may cool the air to 65°. When the fan is downstream, the air enters the cooler at 80°. The resulting air temperature may only be 62°F because there isn't as great a temperature spread as in the first case. However the fan is still working against the 6" head so it adds approx. 5° temp. to the air or 62 + 5 = 67°F. Hence the position of the fan makes a few degrees difference to the output temperature of the cooling system. From this calculation it would appear that the fan should always be upstream to the cooling coils.

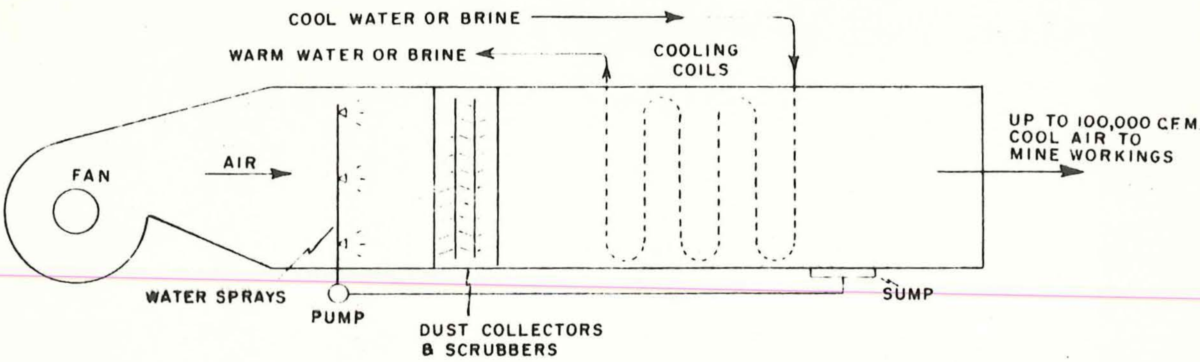


Figure 9 — Underground cooling plant

## Spot Cooling

Extension of long primary development headings will be best accomplished by the utilization of spot coolers. These spot coolers are just what the name implies. Utilizing mechanical refrigeration systems in conjunction with indirect air-freon heat absorbers, the function of a spot cooler is to cool a small ventilation component that services a single heading or localized mining area. They furnish air at a reduced and more desirable temperature, thus allowing the mine to advance development headings more rapidly and under more desirable conditions even though the spot coolers may replace more heat into the air stream than extracted in the heat balance of the cooled air. For example the heat extracted from the air stream by the refrigerant is dissipated at the condenser into the cooling water provided. In addition to this heat balance is the mechanical heat that is transferred into the condenser water and surrounding air by the inefficiency of the compressor and driving motor. See Fig. 5.

Because of their convenience, these units will undoubtedly find wider application in underground use. The units of large capacity are relatively small in size and can be conveniently installed in underground locations with only a minimum of extra excavation. Two problems present themselves in the installation of these units underground: one is getting rid of the heat from the condenser and the other is the noise generated by the compressor and motor.

Any condenser water discharged from these units in the vicinity of the spot cooler may dissipate its heat into the airstream on the level where the unit is in operation. In any type of underground air cooling it is necessary to dissipate the extracted heat into the mine exhaust or to surface. In a large underground cooling installation of this type it can be readily seen that it would be necessary to transmit the condenser water upward to an elevation above the mining zone where the heat contained in the water can be dissipated. Experience indicates that some mine drainage waters utilized as condenser water medium may result in difficulties with scale build-up.

Spot coolers purchased should be constructed with the circulating fan ahead of the expansion-coil heat absorbers. The fan heat of compression should be added to the air prior to cooling. This is the most efficient system from the standpoint of heat extraction.

<u>Operating Performance Of A Typical Spot Cooler</u>

Operating Suction Pressure = 66 p.s.i.g.

Operating Discharge Pressure = 180 p.s.i.g.

Operating Oil Pressure = 125 p.s.i.g.

Fan Inlet Temperature = 91° - 83°F

Fan Discharge Temperature = 95° - 84 1/2°F

Plant Discharge Temperature = 75° - 73°F

Air volume at standard temp. and press. = 5040 c.f.m.

Fan Static Pressure = 11 IN.W.G.

Condenser Water, Inlet Temperature = 63°F

Condenser Water, Discharge Temperature = 81 1/2°F

Condenser Water Volume = 43 gallons per minute

Cooling Capacity = 4012 B.T.U. per minute

Cooling Capacity = 20 Tons Refrigeration per day.

## Area-Cooling Vs. Working-Place Cooling

Hot mines utilize underground plants for "bulk cooling" of air by chilled water plants. However when working-headings are distant, this arrangement results in low-positional efficiency. This means that the air is heated up in transit and its effective cooling power in the production area is severely limited. Under such physical conditions, it is preferable to pump the chilled water in closed circuit to cooling plants close to or serving individual productive headings. Such a low-pressure system is very flexible and is not made obsolete by low-positional efficiency as the operating working places become more distant.

There are several designs of such underground cooling plants. The one in use at the Star mine of the Hecla Mining Co. in Wallace, Idaho use the arrangement shown in Figure 10. These units are light, portable and are attached to the normal auxiliary ventilation system, requiring no extra excavation.

In this system, the condensed water is allowed to drip out of the duct. The fan gives good service downstream from the cooler, and no separate spray and dust collection system is used in this installation. The source of chilled-water is a mechanical refrigeration unit, utilizing potable water in the condenser, and discharging the same to the mine water pumping system.

Air is not cooled and exposed to wallrock as it is delivered many hundreds or several thousand feet down an airway for pick-up by auxiliary ventilation equipment. The cooling coils are normally located within 200-300 ft. of the working place. Dramatic evidence of the results of the high-positional efficiencies of the Hecla system of air-cooling at the working place are manifest in two ways; low effective temperatures in the working place, and reduced overall air-cooling plant requirements.

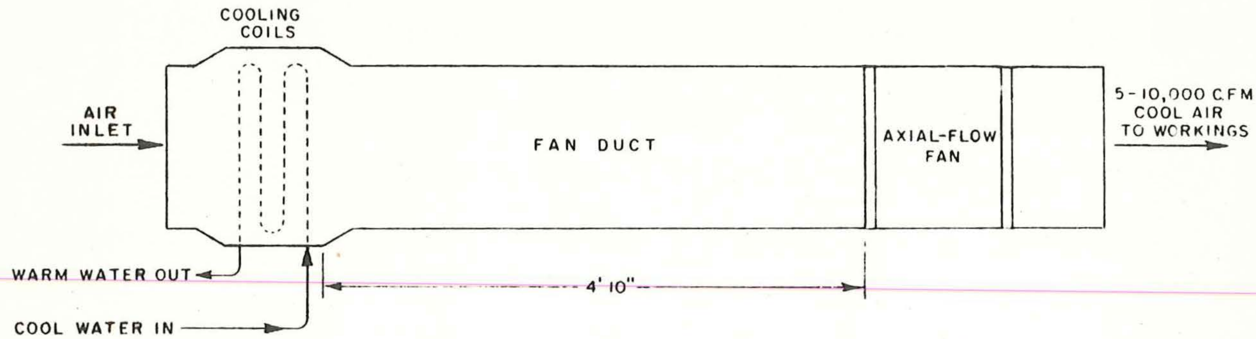


Figure 10. Underground cooling plant, Star Mine

## OTHER AIR-COOLING AND HEAT INSULATION TECHNIQUES PRACTICED

### Vortex Tube

A recent development in spot cooling is the Vortex tube, a form of pneumatic refrigeration. One of the manufactures of this tube is the Vortec Corporation, 1067 North Bend Road, Cincinnati, Ohio 45224. Literature on the operation and performance of this tube is readily available.

The main advantage of this system is that it is small, compact and can be used on individuals wearing a heat protective helmet or suit. The cold air portion from the tube can be directed into the suit, while the hot fraction of the air from the tube is discharged to the mine atmosphere.

In most working places underground, compressed air is available, so the source for driving the vortex tube is readily available.

However, there are some disadvantages of the system. An air hose must be attached to the tube which could be both a nuisance and safety hazard (to miners working in a confined space around machinery) because of catching and snagging on projections. The protective suit or helmet could be a restrictive inconvenience for a full eight-hour shift. Compressed air often contains some oil mist which may be a health hazard to some individuals.

The cooling capacity of the 208-PAC-25 tube is about 1600 Btu per hour, which should be ample cooling for one man because the average metabolism heat rate is about 1200 Btu per hour. The best application would be where the miner could operate equipment with a minimum of movement. Such equipment would be jumbo drills controlled from one position, mucking machines controlled from one position and tunnel borers or other types of continuous miners. However a quick disconnect coupling should be used so that the miner could easily break the air hose connection in case of an emergency.

High-Horsepower Rated Underground Stationary Motors

As the depths of mines increase, the numbers of large-horsepower-rated motors installed underground generally increase. Underground hoists, compressor plants, refrigeration units, diesel equipment, and fans are common examples of large electric-motor installation. Essentially all of the inefficiency of these motors add heat to the mine. Most of these electric motors are air cooled and the cooling air from these motors is discharged into the intake mine ventilation air system because they are frequently installed near the intake air shaft. Although these motors are efficient, ie 80 to 95%, if large horsepower motors are installed underground, they can contribute substantially to the heat source of the mine.

For example at one mine inspected, the shaft station at the level contained a 225 horsepower refrigeration motor and a pump station with two 450 hp. rated pumps. Generally only one pump motor ran at a time and then, only intermittently, but when one pump motor and the refrigeration motor was running, a total of 675 hp. of energy was being consumed. Estimating 85% efficiency, about 101 horsepower was added as heat to the mine intake system because both machines discharged their warmed air exhaust into the intake ventilation system of the mine level. One horsepower is equivalent to 42.4 Btu so the heat added by these motors is equal to $\frac{42.4 \times 101}{200}$ = 21.4 tons of refrigeration which is approximately 10% of the output of the refrigeration plant on this level. In 1967, Fenton (1972 p. 79) reports that the Anaconda Co. had 14,000 hp operating underground. Taking the same inefficiency figure of 15%, this would contribute $\frac{15\% \times 14{,}000 \times 42.4}{200}$ = 445 tons of refrigeration.

These two examples cited show that stationary electric motors contribute substantially to the heat source in a mine. The next question that arises, what can be done about them?

Although electric motors are not often water cooled, this possibility should be investigated. The heat would be carried away by water rather than air, and the water could be discharged into the mine drainage system or piped into a closed circuit cooling system. Most of the large motors are located near the shaft station so this is feasible. Of course, it would be necessary to approach motor manufactures for water cooled designs, but this is one way to carry away heat from these motors.

One other possibility is to install an underground cooling plant at the motor location to cool the motor discharge air before it enters the level intake system. The heat could be transferred to the water of the cooling system rather than to the ventilation intake air. Perhaps other schemes could be devised to keep the motor heat from entering the working level.

## Insulating Hot Water Pipes

Mine drainage lines often go up the main downcast air shaft of a mine. If the mine water is warm, heat is transferred from the warm water to the cooler intake air. At one mine visited, the water temperature at the bottom of the shaft entered the pump column at about 98°F while at the surface it came out at 84°F. A total of 380 gallons/minute was being pumped. The amount of heat exchanged per minute is as follows: 380 gal/Min x 8.33 lb/gal x 14°F x 1 Btu/°F lb = 44,316 Btu/Min or 44,316 ÷ 200 Btu/Min/ton refrigeration = 220 tons. Although there may have been some water mixing on upper levels, with larger flows of water not at 98°F, this is still a substantial quality of heat lost to the shaft air stream.

Fenton (1972, p. 88) reports that one mine pumping 5,000 gpm lost 83,000 Btu/min to the intake air stream or 415 tons of refrigeration.

Insulating hot water lines or hot air lines should be investigated so that this heat loss could be substantially reduced. Although insulation is expensive, the cost of refrigeration is likewise expensive. In recent years, there has been great strides made in insulation of very warm and very cold lines, so adequate and effective insulation should be available which would withstand rough mine usage.

## Compressed Air Service Lines

When air is compressed it becomes very hot. For efficiency, air is usually compressed in two stages with the air being cooled to near atmospheric temperature by an aftercooler in some compressor installations because this does slightly increase the efficiency of the plant. Fenton (1972 p. 87 & 88) gives an approximate formula for estimating the heat loss in a compressed air line going down a vertical airway; on p. 88, he states that it has been estimated that up to 16% of the total heat entering an intake shaft can come from compressed air lines. With this much heat being added to the ventilation circuit, two methods of reducing this heat flow could be investigated. The first one is to insulate all compressed air lines, but this is impractical because compressed air lines run over the levels and they are subject to abuse and abrasion from mine equipment.

The second approach is to aftercool the compressed air on the surface. Aftercoolers are now available from compressor manufacturers, but they are infrequently installed. If a mine is using 10,000 cfm of compressed air (corrected to free air equivalent) and is entering the shaft, it would lose approximately 10,000 cfm x .065 lbs/ft$^3$(au) x 0.24 Btu/lb (specific heat of air) x 1°F=158 Btu/°F, or 158/200 = 0.79 ton of refrig./1°F. It is not uncommon for compressed air to be delivered to the shaft at 200°F if aftercoolers are not used and delivered underground at 100°F. Hence this would amount to 79 tons of refrigeration. The installation of an aftercooler could become very economical when comparing it to the cost of underground mine refrigeration. If the compressed air could be cooled to below mine air temperature, it would act as a refrigerant.

## Covering Ditches Running With Hot Mine Water

Considerable study should be made on either covering ditches running with hot mine water or picking up the hot mine water and pumping or carrying it through insulated lines to the mine water disposal area.

If the hot water is issuing from rocks or stopes above the main pumping level of the mine, methods should be studied on ways to collect this water and transfer it through insulated pipe lines, covered ditches, or flumes so that the hot water is not in intimate contact with the ventilation air. Air that is not saturated will readily pick up water from a hot water source thereby increasing both the dew-point and wet-bulb temperature of the air even if the dry-bulb temperature and the ditch water temperature are similar. Because the wet-bulb temperature is one of the comfort criteria, all means should be taken to keep from exposing the ventilation air to hot water.

This is much easier said than done because water in a wet mine is everywhere and control is most difficult. However, with study, undoubtedly hot water courses could be channeled so that it would not be in intimate contact with ventilation air except on the bottom levels. On bottom levels, and especially where development headings are being driven, water presents a real problem as levels are usually not dry and water is often standing in low places in the bottom of headings. Most mine operators use ventilation bags or pipes to carry ventilation heading air past these areas of exposed water. However the air flowing out picks up water as vapor and is usually both heated and saturated by the time it comes out of the heading.

Some mines make a special effort to dig small sumps and collect the hot water by means of sand pumps discharging into a low-heat conductivity pipe which carries the water to the mine sump. Although this appears to be a sensible solution, it would undoubtedly require considerable maintenance to keep in operation. Therefore the economic gain of this system versus the added cost of refrigeration required to recool the hot return mine air would have to be compared for adoption in a mine heading.

## Sump Location

Often sumps are located close to the shaft level station and in or near the main ventilation intake on the level. Thought should be given to sump location in regard to the heat contribution to the mine ventilation system by the hot sump water. Air near the hot water is both moisturized and heated which mixes with the intake air ventilation system. The hot mine water and the heat from the pump motors can contribute considerable heat to the mine level unless they are disposed of in some other manner.

The sump should be located away from both the shaft and the main intake air source so that heat transfer from the sump into the air stream will be minimized.

## Reducing Heat Effects Of Oxidation

When timber or sulfide minerals are present either in mine workings or worked out areas, oxidation may take place if air is allowed to filter through these areas. Oxidation produces heat, and in some cases has produced enough heat to start mine fires with disasterous consequences.

Mined out stopes filled with mine waste offer small resistance to mine air flow. Needed ventilation air is often lost by leakage through these areas. Although these areas may be sealed off, the seals once made are often left unattended whence they deteriorate and in time become ineffective.

Tailings fill in stopes has reduced this fugitive air loss to practically nothing, although some authorities still feel that a sand fill stope will pass some air especially if the pressure differential across the stope is large enough.

The only effective way to stop this source of heat is to effectively seal off all mined out areas. Some mines do use mined out areas as a source of fresh ventilation air but the heat build up and fire potential is an extreme hazard by this practice.

Miscellaneous

A unique mine air-cooling system is practiced at the Creighton Mine, INCO, Sudbury, Ontario, Canada. During the summer, as much as 1500 to 2000 tons of refrigeration per day of air-cooling results when a portion of the mine inlet air is passed over an ice field. The ice field forms during the winter time when cold surface air is drawn into mine inlet airways after having passed through a mined-out sub level caving area. Surface ground water percolating through the cracked and disturbed ground comes in contact with the cold air and ice is formed. Heat is given off by the heat of fusion of ice forming and the cold winter air is warmed. The reverse happens during the summer time. Warm surface air is pulled through the cave area, coming in contact with ice and cold wallrock and is cooled before entering the mine. About 400,000 cfm of air passes through the icefield, leaving at an average temperature of 32-35°F throughout the entire year.

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Investigations of the Anaconda, Co. at Butte, Montana were directed to developing an insulated air-transmission duct for use in underground auxiliary ventilation systems. The general results are as follows.

| Description of Ventilation Duct | Diameter | K. Coeff. of Heat Transf., Btu/Hr./$ft^2$/°F MTD |
|---|---|---|
| 1. Naylor lockseam spiral welded pipe | 21 1/4" I.D. | 1.94 |
| 2. Spunstrand Fibreglass air transmission duct | 24 1/4" I.D. | 0.92 |
| 3. One-half inch of polyurethane sandwiched between two-ply spunstrand outer surface and one-ply inner surface | 21 1/4" I.D. | 0.41 |
| 4. Dupont 5740 ventube, exterior insulated with one-inch of polyurethane | 21" | 0.30 |
| 5. DuPont 5740 ventube, exterior insulated with 1 1/2" fibreglass building insulation. | 28" I.D. | 0.30 |
| 6. Two-inches of polyurethane sandwiched between two spunstrand fibreglass layers | 30" I.D. | 0.20 |

## RECOMMENDED AREAS OF STUDY

### Synthetic Liquid Air

Liquid oxygen and nitrogen are in widespread use and are commercially available in most areas of the United States. The trucking industry uses liquid nitrogen as both a cooling agent and a means to provide an inert atmosphere so perishable products such as fruits and vegetables will stay fresh longer. Apparently from discussions with people associated with the trucking industry, the cost of this system, with its inert atmosphere advantages is comparable to the truck mounted mechanical refrigerator system.

The following properties of these gases are as follows at the boiling temperature under 1 atmosphere of pressure:

| | Occupies | Weight/gallon | Heat of vaporization | Cp | Boiling Temp. |
|---|---|---|---|---|---|
| 1 lb $O_2$ | 0.105 Gal. | 9.51 lbs. | 91.7 Btu | 0.218 | -297°F |
| 1 lb $N_2$ | 0.148 Gal. | 6.73 lbs. | 85.2 Btu | 0.248 | -320°F |

With properly insulated tanks, liquid oxygen and nitrogen can be transported considerable distance at a low pressure of about 20 psi. Hence high pressure tanks are not necessary for the underground environment. Insulated tanks are available with capacities up to 1900 liters, or approximately 500 gallons capacity. If this system is considered for spot cooling underground, it would be necessary to make synthetic air from combining oxygen and nitrogen. The two liquids have different boiling points with $N_2$ boiling first. These tanks are constructed so that they maintain the liquid at the freezing point. But the gases could be readily combined in the proper proportions in a simple mixing chamber. The amount of cooling available is as follows:

| | | | |
|---|---|---|---|
| Oxygen | heat of Vap. = | | 91.7 Btu lb. |
| | heat to raise from -297 to | | |
| | 75°F = 372 x 0.218 (Cp) | = | 81.0 Btu lb. |
| | Total | | 172.7 Btu lb. |
| Nitrogen | heat of Vap. | | 85.2 Btu/lb. |
| | heat to raise from -320° to | | |
| | 75° = 395 x 0.248 | | 98.0 Btu/lb. |
| | Total | | 183.2 Btu/lb. |

Synthetic air at the proportion 23% $O_2$ and 77% $N_2$ by weight would have the following properties:

$$\left.\begin{array}{l} 172.7 \times .23 = 39.72 \text{ Btu from } O_2 \\ 183.2 \times .77 = 141.06 \text{ Btu from } N_2 \end{array}\right\} = 180.78 \text{ Btu/lb.}$$

To provide one ton of refrigeration it would require 200 ÷ 180.78 = 1.11 lbs. of synthetic air per minute or 66.6 lbs/hr. In a heading, it may be possible to isolate the immediate working area by curtains, brattice cloths or some other system so that men could move around unrestricted, yet be cooled by artificial air. Of course the warmed $N_2$ & $O_2$ would be breathable so a minimum of outside air would be needed for this particular heading. In a working heading, assume that it would take 5 tons of refrigeration which would require 333 lbs. of combined $N_2$ & $O_2$ per hour. One ton of gases would run the heading for about 6 hours.

The cost of these gases, if a plant is available near the mine would be about \$20.00/ton delivered to the mine surface. Estimating handling cost, it would probably be about \$24.00 a ton delivered to the working heading. This would cost about $\frac{24.00}{5 \times 6}$ = .80¢/ton hour of refrigeration.

At first glance, this cost may appear excessive when compared to other refrigeration systems. But if this system would make a suitable atmosphere so that men could work at full rather than partial efficiency, which is not unlikely, the cost of refrigeration by this method can be justified. Labor is one of the most expensive cost items in a mine. This system is by no means considered to be a substitute for one of the systems discussed in the previous section but it is a method of introducing cooling and an air supply at the same time for those difficult-to-service zones by ordinary refrigeration systems.

Of course it would be necessary to meter the proper amount of $O_2$ & $N_2$ into the working area, but the trucking industry has this sort of equipment available and it is a low cost simple unit.

Thermoelectric Cooling

A new space age development in thermoelectric cooling may eventually find application in mine cooling, although at this time, it does not appear practical. Actually the discovery of this principle was made in 1822 when Seeburg noted that if a closed circuit was made of two dissimilar metals, an electric current flowed in the circuit when the two junctions were maintained at different temperatures. This is the principle used today on thermocouples which measure temperature. In 1834, Peltier observed that if an electric current flowed across the junction between two dissimilar materials, heat was either absorbed or evolved. Except for the development of thermocouples, little was done with this phenomena until the discovery of the transistor and other semi-conductor materials which exhibit thermoelectric properties of sufficient magnitude so that fabrication of useful devices has become a reality. The basic ida of the thermoelectric cooling couple is shown in Figure 11.

The thermoelectric materials are represented by n & p where n-type of material has an excess of electrons and p type has a deficiency of electrons. The current is shown flowing in the conventional direction but actually the electrons are flowing in the opposite direction with the

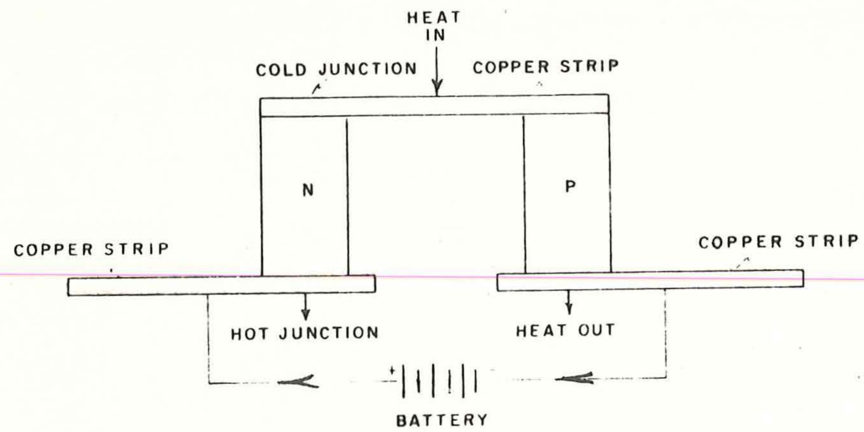


Figure II.— Peltier Thermoelectric Couple

current flowing as shown, the upper junction is the cold one while the lower junction is the hot one.

This system can be compared to a vapor-mechanical type of refrigerator with the battery comparable to the compressor, the cold junction comparable to the evaporator and the hot junction comparable to the condenser. In a thermoelectric cooler, the capacity is adjusted by varying the electric current.

The cooling capacity of one thermoelectric couple is small so large capacity units are made by using many couples. Presently the capacities are small, but one such system of 9 ton capacity is used as an air conditioning unit for a small submarine. Maximum size for other systems are about one ton in capacity.

If this system could be developed, it may have potential use underground because it is compact, noiseless, and requires only electric power. However like a mechanical refrigerator unit, the heat from the hot junction must be disposed of. This could be a troublesome task.

### Dry Ice And Ice

Dry ice (frozen $CO_2$) is used as a commercial refrigerant where it is necessary to keep products frozen. For example, frozen food lockers can be kept frozen while being transported with no source of electricity available; by packing with dry ice. This has had limited use underground as a refrigerant because when it melts or sublimes, $CO_2$ passes into the mine atmosphere and in an enclosed space, an oxygen depleted atmosphere could result. Except for very special cases, it is doubtful if dry ice would be used much for underground cooling and in that case, auxiliary ventilation would be required to carry away the $CO_2$ gas.

Ordinary ice may be a suitable refrigerant in some cases. The latent heat of vaporization is 144 Btu/lb and raising the water from 32°F to 75°F would require 43 Btu. Melting 1 lb. of ice per minute would supply 144 + 43 = $\frac{187}{200}$ = 0.935 tons of refrigeration, or it would require $\frac{200}{187}$ = 1.07 lbs of ice/minute to provide 1 ton of refrigeration. This figure compares to liquid synthetic air. The water vapor is not injurious to health, and except for the logistics involved in getting the ice into the working heading and placing it so it would not be in the way of traffic, it may be a temporary way of providing refrigeration to a heading. This system has had limited use underground. Ice delivered to the mine site would be costly but in bulk quantities may be cheaper than liquid air or other forms of spot cooling.

## Reducing High Pressure Water or Brine

Water or brine that is standing in a vertical column will develop a static pressure head equal to $\gamma h$ where $\gamma$ is the density of the water and h is the head. For example in a pipe 4,000 ft. high filled with water, the pressure in psi would be 62.4 x 4,000 ÷ 144 = 1732 psi. With water flowing up the pipe the head would actually be higher at the bottom of the column because of pipe and flow resistance. If the water is flowing down the pipe, however the head is less at the bottom of the column because resistance dissipates some of this head. Pipe and coils which contain this water in the mine must be selected with thick walls to resist this pressure. This not only costs more per foot of length, but the thermal conductivity characteristics of the coil pipe are reduced. Fittings and other pipe connections are more costly and considerable safety precautions and care must be taken with high pressure equipment.

Closed circuit systems are used in Butte and the Crescent mine in Idaho to overcome the cost of pumping brine or cooling water out of the mine. If the water is reduced to low pressure for level use and then discharged to the mine drainage system, the costs for pumping are high. For example, with a power cost of 5¢/horsepower hour, it would cost to pump 100 gpm against a head of 4,000 ft; $\frac{100 \text{ gpm} \times 8.33 \text{ lbs gallon} \times 4{,}000 \text{ ft} \times \$0.005}{33{,}000 \text{ ft lbs/min/hp.}}$ = \$0.505/hr. With vapor-mechanical types of refrigerators underground, the rule of thumb is that it takes 1 gallon of water per ton of refrigeration per minute. Thus for a mechanical refrigerator dumping condensor waste water into the mine drainage system, the power cost for a 200 ton unit would be about \$1.00 an hour for pumping against a 4,000 ft. head. The convenience of a low pressure mine cooling system is offset by a high pumping cost. At Superior, Arizona, the Magma Mine uses a heat exchanger at the shaft to provide cool low-pressure water to the level coolers. published reports indicate that this system has some operational problems and it has not been popular with other companies.

If a method could be devised so that the cooling water could be used at low pressure on the level and some means found to help with the work of pumping the low pressure water to the surface, a considerable power savings would result. One way of doing this is to introduce a fluid motor attached to the pump so that the energy of the high pressure water flowing to a lower pressure could be used to drive the pump which is necessary to increase the pressure of the low pressure water to the high static head in the pump column.

The following suggestion is made as illustrated in Figure 12 for a pressure reducing and pressure gain system. A piston type pump and motor is illustrated however turbine or centrifugal types could also be used.

These ideas have been forwarded to pump and turbine manufactures and they feel that the idea is wholly practical. The Gardner-Denver Co. of Peoria, Ill. makes high-pressure plunger pumps for the oil industry and they believe they can convert one of their triplex pumps into a fluid plunger-type motor with no great problem and attach it to a regular triplex pump. The engineering department of this company said that the efficiency of the pump is estimated to be about 90% and the fluid motor about 88%.

The water horsepower required to pump 100 gpm from the 4,000 ft. level, and assuming a total head of 4200 ft. to account for pipe friction, etc. calculates as follows:

$$\frac{4200 \text{ ft.} \times 100 \text{ gpm} \times 8.33 \text{ lbs/gal}}{33{,}000} = \frac{4200 \times 833}{33{,}000} = \frac{3498600}{33{,}000}$$

= 106 hp. If 90% pump efficiency is used, the hp. requirement would be 106 ÷ .90 = 118 hp. to pump this water to the surface. Assuming 85% efficiency of the hydraulic motor, to account for head losses in the pipe leading to it, it would develop 106 x .85 = 91 hp.

If the hydraulic motor is used, it would save 91 hp per 100 gallons over the cost of pumping the water from the level by ordinary pump means. At .5¢/hp hour it would amount to \$.005 x 80 hp x 24 hrs = \$9.60/day or \$3,504.00/year. Actually in an existing closed system such as in Butte or the Crescent mine in Idaho, the pump horsepower requirements at the level would not be as large as calculated because there is a circulating pump installed to overcome pipe friction of the present system. Hence the effective head on the pump would not be as high as chosen for the cal-

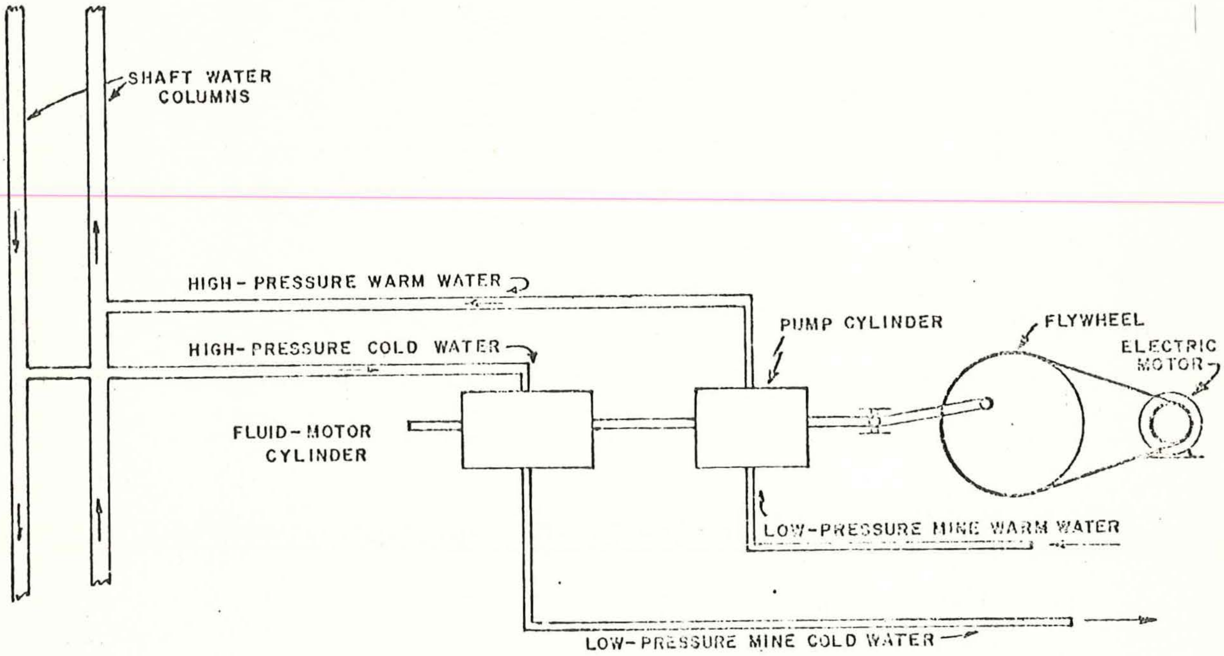


FIGURE 12—SCHEMATIC DIAGRAM OF REDUCING & REGAINING PRESSURE ON REFRIGERATION-WATER SYSTEMS (PISTON TYPE)

culation. If there is no circulating pump in the system, the calculations shown would be valid for the conditions chosen.

Plunger type motors and pumps were chosen for this example because the Gardner-Denver Company feels this conversion can be made on their present high pressure pumps. For smaller capacities, this system is feasible. The pumps that they show in their sales literature do not exceed 200 gpm for continuous service.

Actually for size requirements and convenience, a rotary type fluid motor, centrifugal pump and electric motor could be readily made into the combination as shown in Figure 13.

Rotary-type water pumps are now developed to pump against a 5,000 ft. head, and water turbines are also available to operate under this head.

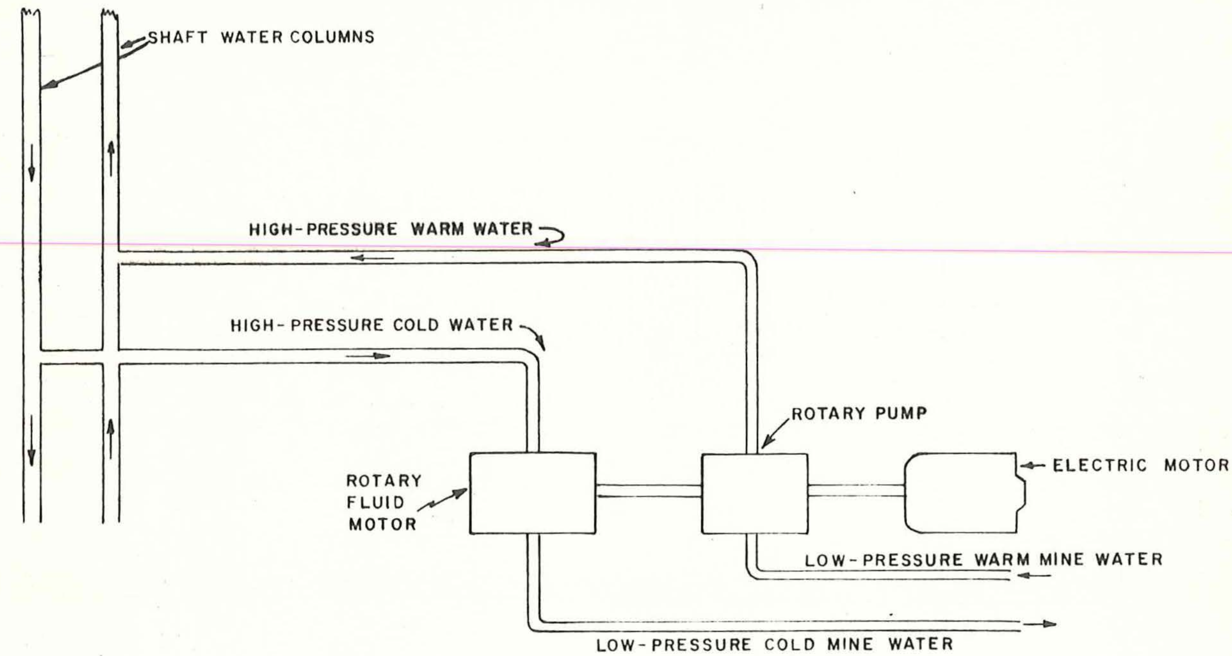


FIGURE 13-SCHEMATIC DIAGRAM OF REDUCING & REGAINING PRESSURE ON REFRIGERATION-WATER SYSTEMS (ROTARY TYPE)

The James F. Leffel & Co. of Springfield, Ohio will make an impulse type turbine which will operate under 5,000 ft. of head and as low as 300 gpm. The Bingham-Willamette Pump Co. of Portland Oregon have successfully used centrifugal pumps as high head turbines but the efficiency is very low, below about 800 gpm. Operating under idal conditions the efficiencies are not as good as the plunger type of pumps.

This system would have to be balanced, that is the same amount of water must flow into the pump as flows out of the hydraulic motor. Therefore pressure regulating systems must be designed and installed for electric motor overload protection. Possible problems that could occur are as follows: If a leak occurred in the low pressure side, water would be lost so a sensing switch would be necessary to shut off the unit in this event because the pump would not receive water. If the fluid motor failed, the electric motor would rapidly be overloaded so electric motor over-load protection would be required.

This system has the advantage that the only high strength pipe needed is in the shaft and in the pipe and fittings to the unit on the level. It should be applicable to a refrigeration unit placed underground where cool condenser water is taken into the mine and the warm condenser water is discharged into the mine drainage system. At the cost saving of this system over pumping condenser discharge water from the bottom levels, it would appear feasible.

## Air Compression - Air Motor System

Air compression and Air expansion systems for cooling are used in airplanes for air conditioning of the cabins. (See ASHRAE, Systems & Equipment, 1967 p. 147-158). The advantage of this system is that it is light in weight and the only refrigeration medium used is air. A system such as this may have application in a mine in certain cases. Compressing air to 100 psi requires considerable power, but of course compressed air is in widespread use in most metal mines. Air when it expands, if it does work, will cool to a very low temperature. In fact,in compressed air drills and motors, it is common to see frost on the air discharge ports and often time they freeze completely over.

A schematic diagram for a proposed air refrigeration system is shown in Figure 14.

In a deep mine the air would be both hot, humid and near sea level pressure. From table 8.6 in the Compressed Air handbook (1966, p. 8-51) the specific gravity of moist air at 90°F and 100 saturated is 0.9821. This would correspond to standard air at about 500 ft. in elevation. Table 8.31 in the Compressed Air handbook (1966 p. 8-90) gives the approximate horsepower required to compress 100 cu. ft. of free air per minute to 100 psig as 21.9 which include motor and compressor inefficiencies. At sea level and 60°F, one cubic foot of air weighs

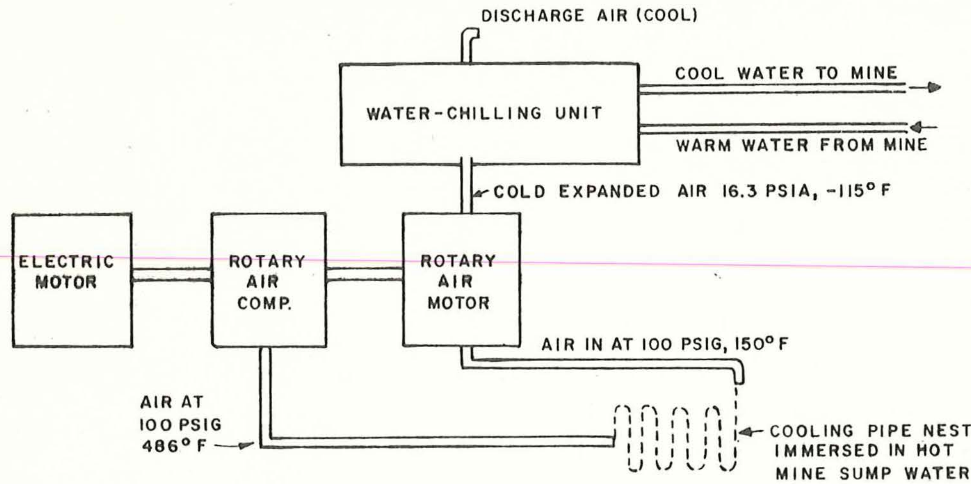


FIGURE 14-SCHEMATIC DIAGRAM OF COMPRESSOR EXPANDER AIR-REFRIGERATION SYSTEM

0.07658 lbs. (Compressed Air handbook, table 8.39, 1966, p. 8-97). The air chosen in this example weighs 0.07658 x 0.9821 = 0.075 lbs. Hence 1 lb. of air would occupy 0.075 = 13.3 cubic feet. The horsepower requirements necessary to compress 1 lb. of air would be (21.9 ÷ 100) x 13.3 = 2.91. The temperature of the compressed air is given by Simons (1922, p. 164) as + 485°F.

Equations for air motors are difficult to find in the literature. However Simons gives one (1922, p111) of a partial adiabatic expansion engine as

$$Hp = \frac{144}{33{,}000}\left[P_2 V_2 + \frac{P_2 V_2 - P_1 V_1}{n - 1} - P_a V_1\right] \quad \text{where}$$

$P_2$ = Abs. pressure at air intake

$V_2$ = Volume of compressed air in $ft^3$/min. taken into the cylinder/min.

$P_1$ = Abs. pressure at exhaust port

$V_1$ = Volume of air exhausted at a pressure $P_1$ in $ft^3$/min.

$P_a$ = Atm. pressure in psia = 14.7 psia

$n$ = 1.406

After several trial calculations, it appears that an air motor with 1/4 cut-off gives the best horsepower output. Thus if $V_1$ = 1 Cubic foot, $V_2$ = 4 cu.ft. $P_1 = P_2(\frac{V_2}{V_1})^n$ so $P_1 = (100 + 14.7)(\frac{1}{4})^{1.406} = 16.3$ psia

$$Hp = \frac{144}{33,000}\left[114.7 + \frac{114.7 - 16.3 \times 4}{0.406} - 14.7 \times 4\right] = \frac{144 \times 178}{33,000} = 0.777$$

The general gas law for dry air states that

PV = RT for 1 lb. of air where

P = pressure in $lbs/ft^2$

V = volume of 1 lb. of air in $ft^3$

T = abs. temperature

R = 53.2 lb ft/dry °F.

$$V = \frac{53.2 \times (460 + 150)}{144 \times 114.7} = \frac{53.2 \times 610}{16516.8} = \frac{32452}{16517} = 1.96 \text{ cu. ft.}$$

The expansion of 1 lb. of air would develop 1.96 x 0. 777 = 1.52 horsepower under the above listed conditions. Assuming that the air engine is 85% efficient, the expected horsepower output would be 1.52 x .85 = 1.3.

The net horsepower imput by the electric motor would be 2.91-1.30 = 1.6 horsepower per lb. of air compressed and expanded. The temperature of the air from the compressor is 485°F so the hot mine water, even if it is 140°F could cool the compressed air to 150°F. This would correspond to the condenser of a vapor-mechanical refrigerator circuit and because the air is so hot, the hot mine water would still act as a cooling agent.

The equation for adiabatic expansion of air is $T_1 = To\left[\frac{P_1}{Po}\right]^{\frac{n-1}{n}}$

where $T_1$ = Final temperature absolute

To = Original temperature absolute

$P_1$ = Final pressure absolute

$P_o$ = Original pressure absolute

n = 1.406

$$T_1 = (460 + 150)\left(\frac{16.3}{114.7}\right)^{.29} = 610(.14)^{.29} = 610 \times .565$$

$T_1 = 345°R = -460 + 345 = -115°F$

Assume that 50° water is wanted in the chiller, the temperature rise of the air would be +115 + 50 = 165°F. The specific heat of air is 0.24 so 1 lb. of air heated to 50°F would give 165 x .24 = 40 Btu/lb. For one ton of refrigeration it would require 200 ÷ 40 = 5 lbs of air. This would require 5 x 1.6 = 8.0 horsepower per ton of refrigeration.

Of course this is a very large power requirement, when compared to the vapor-mechanical refrigeration unit where about 1 hp per ton of refrigeration is required. However if 1 gallon of cooling water is needed per ton of refrigeration and it is pumped from a 4,000 ft. deep shaft, the horsepower requirements for pumping the water comes out to be equal to

$$\frac{1 \times 8.33 \times 4,000}{33,000} = 1 \text{ hp/ton}$$

In a vapor mechanical refrigerator, auxiliary equipment is needed for the condenser cooling either in the way of a pipe line or added pump capacity to handle the condenser cooling water, hence in certain cases, air plants of this type may have some application.

A plant of this type would probably be impractical over 25 tons of refrigeration but it would be independent of any service in the mine except for the hot discharge water and power lines for the motor. However the motor should be a water cooled or a low heat motor because a 25 ton plant would require a 200 hp motor and motor inefficiency could seriously impair the efficiency of the plant. The cooled compressed air from the plant could also be a ventilation aid.

In summary, even though the horsepower requirements are extremely high by this system, some hot isolated areas may use this plant as a small zone or spot cooler where only a hot water source for cooling and electric power is available.

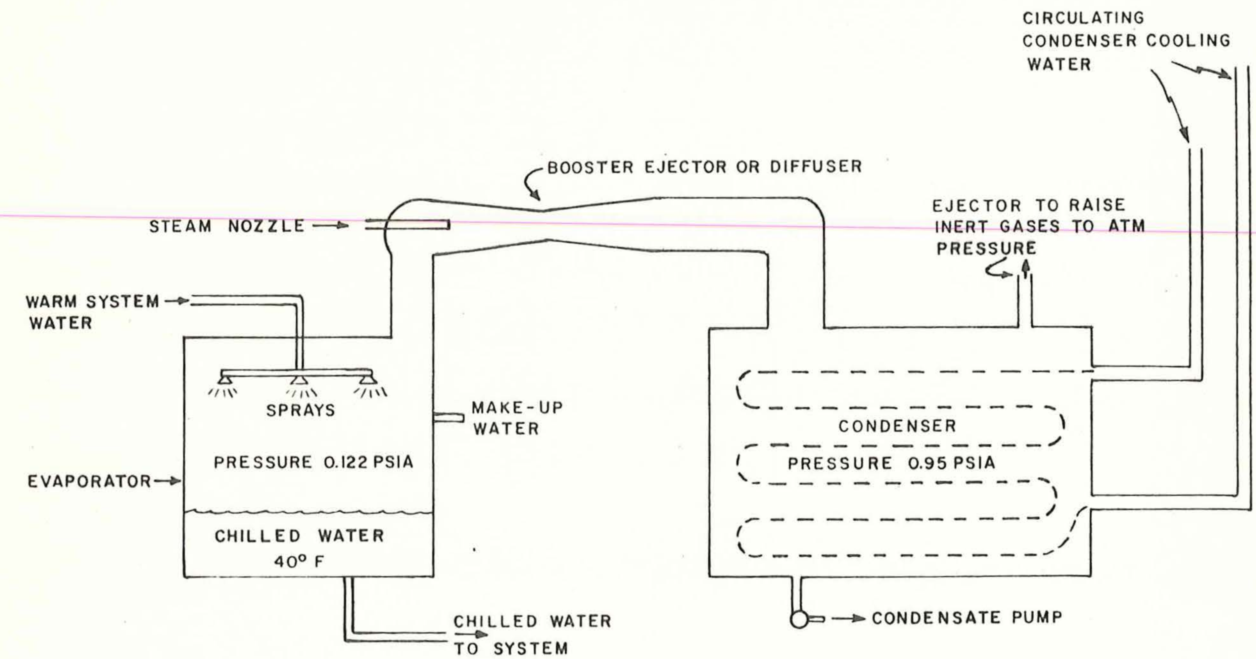


FIGURE 15-STEAM-EJECTOR VACUUM REFRIGERATION SYSTEM

Vacuum Cooler

Steam jet refrigeration is successfully used where large supplies of steam are available. The steam-jet principle is shown in Figure 15.

Steam enters the steam nozzle, and expands in the booster ejector to the low pressure of 0.148 psia. This expansion action entrains the vapor formed from the evaporation of the water in the evaporator and carries it with the expanded steam to the condenser where the combined steam and entrained water vapor is condensed at a temperature of about 95°F at an absolute pressure of 0.95 psia. This low pressure in the condenser is maintained by other ejectors. With the steam ejector discharging into a low pressure condenser, the work required to maintain the vacuum in the evaporator is much less than if the discharge were to the atmosphere. The evaporative temperature of water at 0.248 psia is 40°F and the condensensation temperature at .95 psia is 100°F, so the water can boil and condense in this temperature range. One of the problems encountered in this system is the large volume of water vapor that must be transferred to the condenser.

If a suitable evacuator could be found, perhaps a vacuum cooler could be installed underground. A schematic diagram is shown in Figure 16 where the hot mine water provides heat for the condenser.

Instead of a steam ejector, perhaps some type of rotary vacuum pump would be utilized to maintain the vacuum in the evaporator. Power requirement calculations show that this system is feasible.

The big draw back to this system is the large compression ratio required and the large capacity of low pressure steam. The compression ratio would be about 117 to 1, the volume of steam per/ton of refrigeration would be 456 cu. ft/minute which for a 200 ton plant, would require approximately 100,000 cfm. This would require a very high capacity machine even though the power requirements are low. A centrifugal machine would be necessary and undoubtedly it would be necessary to stage compress in at least 3 stages.

With the development of high capacity centrifugal compressors and evacuators, this system may be possible. It would be necessary to contact compressor and evacuator manufacturers for help in the design of this equipment. Perhaps efficiency could be improved if stage cooling between the compressors could be achieved or if a condenser was used with hot mine water as the coolant, and a small pump to pump the condensate water to atmospheric pressure and a small evacuator pump installed to handle any entrained inert gases.

Some of the advantages of this system would be as follows:

1. Only water is used as a refrigerant and in a 200 ton-a day plant only 200 x 0.224 = 45 lbs of water per minute is needed. This is 45 ÷ 8.33

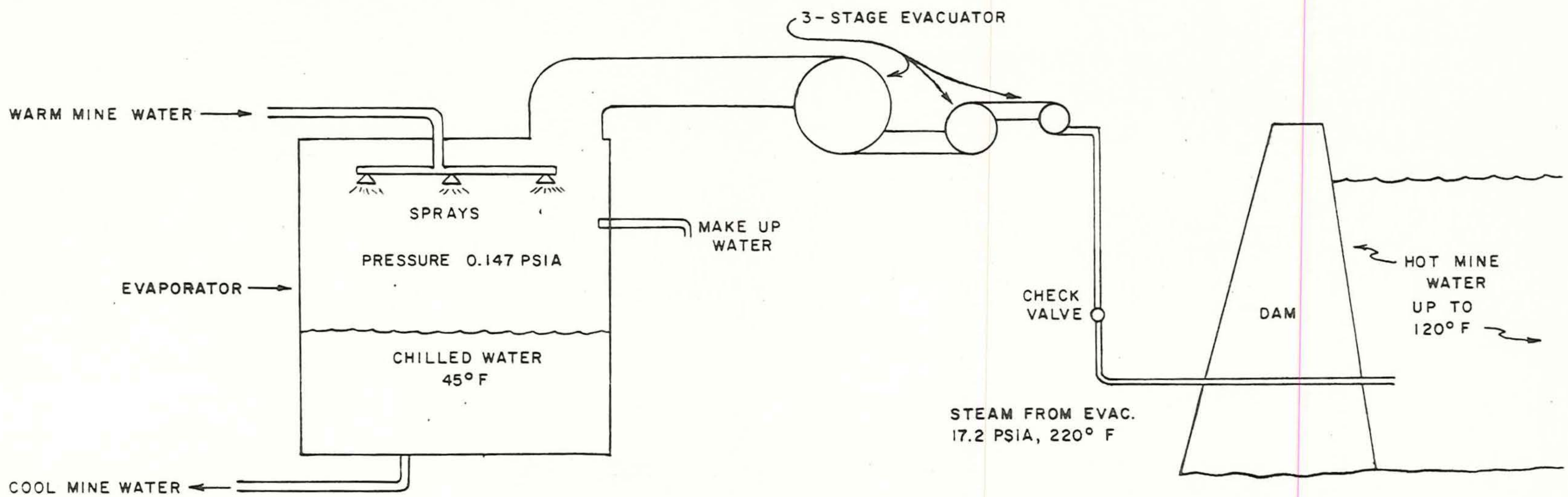


FIGURE 16– PROPOSED VACUUM COOLER FOR A MINE

= 5.4 gpm which could easily be disposed of in the mine drainage system. There is no fire or toxant possibilities by this system.

2. There is no high-pressure equipment involved and the only units required are the evaporator and evacuator. After initial development, the cost per unit should be less than the more complicated vapor-mechanical type of refrigerator. It could be installed anywhere there is a heat sink available such as a return airway or hot mine water sump.

3. Horsepower requirements compare favorably with vapor-mechanical refrigerators because the calculated horsepower is less per ton of refrigeration and only a very small horsepower requirement is needed to pump the condensate from this system to the surface of the mine.

4. Although the capacities of the evacuator are extremely high, the pressure differences are small and it is possible that the machine could be made light weight and keep its size to a minimum. Compressors for turbine jet engines are light weight and have high capacity so the know-how for this machine is available.

5. As long as compression is carried out in the superheat region, no water should condense out in the evacuators so corrosion should not be a problem.

6. Because the exhaust steam is at such a high temperature, heat sinks at high temperature such as hot mine sump water or hot discharge airways could be used to cool the hot steam so that it would either condense or be carried out the exhaust air way system as a vapor.

Some experimental work has been done at this college on vacuum coolers using water jets instead of steam jets. Water jets have given vacuum good enough to reach 65 to 70° water in the evaporator but much more experimental work is required. Some ideas have been advanced which will improve the efficiency of this experimental system.

References


1. American society of heating, refrigerating, and air-conditioning engineers, Inc. ASHRAE, Handbook of Fundamentals 345 E. 47th Street, New York, N.Y. 1967.

2. American society of heating, refrigerating and air-conditioning engineers, Inc. ASHRAE, Systems and Equipment, 345 E. 47th. Street, New York, N.Y. 1967.

3. Bell, A.R.; Ventilation and Refrigeration As Practiced At Rhokana Corporation Ltd., Zambia; J. Mine Ventil. Soc. of So. Africa, Vol. 23 No. 3, pp. 29-35, March, 1970.

4. Beskine, J. M.; Priorities In Deep Mine Cooling; Mine & Quarry Engineering, pp. 379-384, Dec., 1949.

5. Bromilow, J. G.; Ventilation Of Deep Coal Mines; Iron And Coal Trades Review; Part I, pp. 303-308, February 11, 1955
Part II, pp. 367-376, February 18, 1955
Part III, pp. 427-434, February 25, 1955.

6. Brown, U. E.; Spot Coolers Increase Comfort of Mine Workers; Eng. And Min. J.; Vol. 146 No 1., pp. 58-59, January, 1945.

7. Carrier, W.H. "Rational Psychrometric Formula", Trans. Am. Soc. Mech. Eng. vol. 33 (1911) p. 1005.

8. Carrier, W.H. Air Cooling in the Gold Mines on the Rand: Trans. Amer. Inst. Mng. Met Engrs. 141 pp 276-287.

9. Caw, J.M.; Some Problems Raised By Underground Air Cooling On The Kolar Gold Field; J. Mine Ventil. Soc. of So. Africa, Vol. 2 No 2, pp. 83-137, Feb., 1953.

10. Caw, J.M.; Air Refrigeration; Mine & Quarry Engineering, pp. 111-117, March, 1957 pp. 148-156, April, 1957.

11. Caw, J.M.; Current Ventilation Practice In Hot Deep Mines In India;

References continued,

J. Mine Ventil. Soc. of So. Africa, Vol. 11 No. 8, pp. 145-161, August, 1958.

12. Caw, J.M.; Observations At An Underground Air Conditioning Plant; J. Mine Ventil. Soc. of So. Africa, Vol. 12 No. 11, pp. 270-274, November 1959.

13. Compressed Air & Gas Inst. Compressed Air & Gas Handbook, 122 E. 42nd. Street, New York, N.Y. 10017.

14. Dossat, Roy J. Principles of Refrigeration John Wiley & Sons, Inc. New York, N.Y. 1961.

15. Faires, Virgil M. Thermodynamics, The Macmillan Co. New York, 4th. Ed. 1962.

16. Fenton, James L. Survey of Underground Mine Heat Sources, Masters' Thesis, Montana College of Mineral Science & Technology, March 1972.

17. Field, W.E.; Combatting Excessive Heat Underground At Bralorne; Mining Engineering, pp. 76-77, December, 1963.

18. General Motors, A to Zero of Refrigeration, 5th. printing, Revised 1956, Detroit 2, Mich.

19. Hartman, Howard L. Mine Ventilation and Air Conditioning, the Ronald Press Co. New York, N.Y. 1961.

20. Hill, M.; Refrigeration Applied to Longwall Stopes And Longwall Stope Ventilation; J. Mine Ventil. Soc. of So. Africa, Vol. 14 No. 5, pp. 65-73, May, 1961.

21. Kock, H.; Refrigeration In Industry; The So. African Mech. Engineer, pp. 188-196, November,1967.

22. LeRoux, W.L.; Heat Exchange Between Water And Air At Underground Cooling Plants; J. Mine Ventil. Soc. of So. Africa, Vol. 12, No. 5 pp. 106-119, May, 1959.

References continued,

23. Marvin, C.F. Psychrometric Tables, U.S. Dept. of Commerce, Weather Bureau, U.S. Govt. Printing Office Washington 1941.

24. Minich, G.S.; The Pressure Recuperator And Its Application To Mine Cooling; The So. African Mech. Engineer, pp. 57-78, Oct., 1962.

25. Muller, F.T. & Hill, M.; Ventilation And Cooling As Practised On E.R.P.M. Ltd., So. Africa; J. Inst. of Min. & Met. of So. Africa, October, 1966.

26. Richardson, A.S. A Review of Progress in the Ventilation of the Mines of the Butte, Montana District, Quarterly of the Colorado School of Mines, Golden, Colorado April, 1950.

27. Sandys, M.P.J.; The Use Of Underground Refrigeration In Stope Ventilation; J. Mine Ventil. Soc. of So. Africa, Vol. 14 No. 6, pp. 93-95. June, 1961.

28. Schlosser, R.B.; The Crescent Mine Cooling System; Northwest Mining Assoc. Convention, December 2, 1967.

29. Short, B.; Ventilation And Air Conditioning At The Magma Mine; Mining Engineering, pp. 344-348, March 1957.

30. Simons, Theodore, Compressed Air, McGraw-Hill Book Company, Inc. New York, N.Y. 1921.

31. Thompson, J.J.; Recent Developments At The Bralorne Mine; Canadian Min. And Met. Bul., pp. 1301-1305, November, 1967.

32. Torrance, B. & Minich, G.S.; Heat Exchanger Data; J. Mine Ventil. Soc. of So. Africa, Vol. 15 No. 7, pp. 129-138, July, 1962.

33. The Trane Co., Trane Air Conditioning Manual, LaCrosse, Wisconsin, 1965.

34. Warren, J.W.; The Science of Mine Ventilation; Presented at the American Mining Congress, San Francisco, CA., Sept. 23, 1958.

References continued,


35. Warren, J.W.; Supplemental Cooling For Deep-Level Ventilation; Mining Congress Journal, pp. 34-37, April, 1965.

36. Weeks, Walter S., Ventilation of Mines, McGraw-Hill Book Co. Inc. New York & London, 1926.

37. Whillier, A.; Heat - A Challenge In Deep Level Mining; J. Mine Ventil. Soc. of So. Africa; Vol. 25 No. 11, pp. 205-213, November, 1972.