OPTIMIZATION AND TEST OF WATER SPRAY COOLERS

**BUREAU OF MINES
UNITED STATES DEPARTMENT OF THE INTERIOR**

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abstract16. Abstract As mining develops into deeper and hotter regions, the safety and health of miners require that the workspace be cooled. Toward this end, this program has successfully developed a high performance, low maintenance spray cooler of 5000 cfm nominal capacity. Further, a series of four 25,000 cfm, nominal capacity, spray coolers has been built and mine tested with satisfactory results. The smaller coolers have been patented and become commercialized with 50 units currently in service. The concept has been developed independently overseas with an unknown number of units in operation.

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FOREWORD

This report was prepared by Foster-Miller, Inc.,
(FMI), Waltham, MA, under Bureau of Mines (Bureau)
Contract No. H0395058. The effort reported herein was
performed between July 1979 and June 1984. It was
administered with the technical direction of the
Pittsburgh Research Center. The original Contract Officer
was Alan G. Bolton, Jr. During the program, this
responsibility was assumed by Joseph A. Gilchrist.

The Technical Project Officer was Edward D. Thimons.
Mr. Thimons was an active technical contributor to the
program in addition to his role as technical monitor. His
contributions for this program, both at the conceptual
stage and during development and evaluation, were
significant in the overall success of this effort.

Dr. Austin Whillier from the Chamber of Mines of
South Africa acted as special consultant to FMI on this
program. Dr. Whillier brought to this program extensive
experience with knowledge of and unique insight into both
direct water to air heat exchanges and mine cooling. His
contributions were in the areas of thermodynamics,
physiology and economics. He was one of those rare
individuals who could not only see further and deeper into
a problem, but could patiently and kindly teach those who
were not as able as he. Dr. Whillier participated in this
program with enthusiasm until his untimely death.

The technical effort was performed by the Mining
Division, Terry L. Muldoon, manager, part of the
Engineering Systems Group, David A. Monaghan, assistant
director. The program manager was John F. McCoy. For
much of the effort, Kenneth S. Heller functioned as
principal investigator. Major FMI contributors to the
effort were: Joseph W. Valentine, Alve J. Erickson,
S. Gary Anderson, Muthiya Thangaraj, Thomas E. Gore and
Michael E. Demaree.

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ABBREVIATIONS AND SYMBOLS USED: DEFINED WITH UNITS OF MEASURE AND CONVERSION BETWEEN U.S. CUSTOMARY AND SYSTEMS INTERNATIONAL d'UNITES (SI UNITS)

Symbol	Definition	Customary unit(s)	SI unit(s)	Relationship between units
C^o	specific heat of water (a constant)	1 Btu/lb-F^o	4.18 kJ/kg-C^o	
C'	average equivalent specific heat of moist air with state change	Btu/lb-F^o	kJ/kg-C^o	1 Btu/lb-F^o = 4.18 kJ/kg-C^o, 1 kJ/kg-C^o = 0.2392 Btu/lb-C^o
E_w	water efficiency $(t_{ai} - t_{ao})/(t_{ai} - t_{wo})$ (unitless)	–	–	
F	factor of merit or F-factor (unitless)	–	–	
G	mass flow of air	lb/hr	kg/s	1 lb/hr = 0.0001264 kg/s, 1 kg/s = 7920 lb/hr
L	mass flow of water	lb/hr	kg/s	1 lb/hr = 0.0001264 kg/s, 1 kg/s = 7920 lb/hr
WAR	water air ratio (L/G) (unitless)	–	–	
N	coefficient, $(F/1 - F)/R^{0.4}$ (unitless)	–	–	
P	cooling power, $E_w LC(t_{wi} - t_{wo})$	Btu/hr or R-ton (12,000 Btu/hr)	kW	1 Btu/hr = 0.0002931 kW, 1 kW = 3412 Btu/hr
R	capacity ratio, LC/GC' (unitless)	–	–	
S	specific sigma heat	Btu/lb	kJ/kg	1 Btu/lb = 2.327 kJ/kg, 1 kJ/kg = 0.4298 Btu/lb
h	specific enthalpy	Btu/lb	kJ/kg	1 Btu/lb = 2.327 kJ/kg, 1 kJ/kg = 0.4298 Btu/lb
t $\begin{cases} t_{ai} \\ t_{ao} \\ t_{wi} \\ t_{wo} \end{cases}$	temperature $\begin{cases} \text{wet bulb in} \\ \text{wet bulb out} \\ \text{water in} \\ \text{water out} \end{cases}$	$\{ \, ^oF$	$^oC \}$	( )^oF = ( )^oC x 9/5 + 32, ( )^oC = 5/9[( )^oF - 32]
wb/db	wet bulb/dry bulb (temperature)	–	–	
gpm	U.S. gallons per minute	–	m^3, ℓ/s	1 gpm = 6.31 x 10^{-5} m^3/s = 0.0631 ℓ/s = 0.0631 kg(water)/s
hp	horsepower	–	kW	1 hp = 0.746 kW, 1 kW = 1.34 hp
psi	pounds per square inch	–	Pa	1 psi = 6891 Pa, 1 Pa = 0.0001451 psi
Pa	pascal	psi	–	1 Pa = 0.0001451 psi, 1 psi = 6891 Pa
lb	pound (mass)	–	kg	1 lb = 0.454 kg, 1 kg = 2.2 lb
kg	kilogram	pound (mass)	–	1 kg = 2.2 lb, 1 lb = 0.454 kg
Btu	British thermal unit	–	kJ	1 Btu = 1.055 kJ, 1 kJ = 0.9479 Btu
kJ	kilojoule	Btu	–	
R-ton	refrigeration ton (12,000 Btu/hr)	–	kW	1 R-ton = 3.517 kW, 1 kW = 0.2843 R-ton
ft	foot	–	m	1 ft = 0.3048 m, 1 m = 3.281 ft
in	inch	–	m	1 in = 0.0254 m, 1 m = 39.4 in
in water gage	inch (manometric) of water gage	–	Pa	1 in water gage = 249.1 Pa, 1 Pa = 0.004014 in water gage
s	second	s	s	
min	minute	–	s	1 min = 60 s
hr	hour	–	s	1 hr = 3600 s
C^o	Celsius degrees (differential temperature)	F^o	–	1 C^o = 9/5 F^o
F^o	Fahrenheit degrees (differential temperature)	C^o	–	1 F^o = 5/9 C^o
m	meter	ft	–	1 m = 3.281 ft, 1 ft = 0.3048 m
mm	millimeter (10^{-3} m)	in	–	1 mm = 0.0394 in, 1 in = 25.4 mm
μm	micrometer (10^{-6} m)	μin	–	1 μm = 39.4 μin, 1 μin = 0.0254 μm

ACKNOWLEDGEMENTS

EXECUTIVE SUMMARY

E.S.1   INTRODUCTION TO SUMMARY

As mines develop into deeper and hotter regions, new tools are increasingly in demand for cooling the underground workplace. Cooling is necessary to maintain productivity and, more importantly, to protect miners from the danger of the possibly life-threatening effects of heat stress. This program has successfully developed and patented a new spray cooler for mine cooling.

Prior to this program, Thimons, et al. (1) of the Bureau performed a series of laboratory and in-mine tests of a duct type spray cooler (see figure 1). The promising results of these preliminary evaluations prompted the work reported herein. Broad objectives for this program were:

a.   Optimization of a 5,000 cfm water spray cooler

b.   Extention to larger size cooler designs.

10 ft LONG, 16 in. DIAM FIBERGLASS TUBE
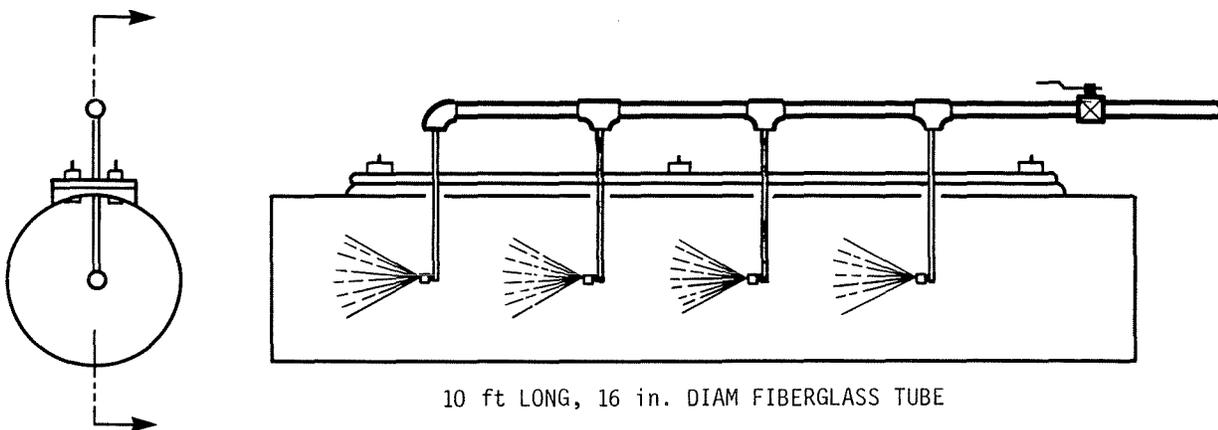
FIGURE 1. - Duct type spray cooler.

Optimization of underground equipment occurs in two phases. The first is performance optimization. In this case, the goal was optimum use of costly chilled mine service water to cool the workspace. The second goal is a "mineworthy" component or system - one that will perform reliably without malfunction and without frequent service. High reliability is important for all mining equipment, since maintenance is difficult underground. High reliability is particularly critical for environmental control equipment because the limited resources are applied first to keep direct production equipment operating, often ignoring inoperative or poorly performing environmental control equipment.

Early success in this program led to interest in increasing the capacity of coolers. This defined the second general objective - to scale up to larger coolers.

This program was among the first, if not the first, to use the factor of merit or F-factor method for analyzing spray cooler performance. First considered by Whillier (2) and developed by Bluhm (3,4) at the Chamber of Mines of South Africa, this technique uses data collected at a single operating condition to predict cooler performance over the full range of conditions within the allowable range of airflows and waterflows for the cooler.

The F-factor is a number between 0 and 1 inclusive. Higher numbers indicate higher performance. For operating conditions where airflows and waterflows have equal total heat capacity, the F-factor represents the cooling water efficiency (the fraction of "coolth" which may be theoretically extracted).

Use of the F-factor technique permitted considerable program efficiencies, and significantly aided development work. The amount of required laboratory testing was reduced, since it was unnecessary to duplicate all possible mine conditions to correlate laboratory test results with those obtained during the in-mine tests. Now, 5 years after the start of this program, many United States mine ventilation engineers at hot mines are conversant in the Factor of Merit method of rating spray coolers. For this reason, F-factors are used extensively in this report.

E.S.2 SUMMARY OF SMALL COOLER DEVELOPMENT

During this program, a new concept was found for mine spray coolers, developed for 5,000 cfm nominal airflow. The prime advantages of this spray cooler over available cooling methods are the very low maintenance requirements and high water efficiencies that are obtained. Hot, humid, dusty air is brought into the bottom of the spray

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cooler and directly contacted with cold water both in droplets and on mesh pads. Heat and moisture are then transferred from the air to the water, while dust and other particulates are removed by the waste water. The warmer water is drained out of the cooler. A mist eliminator pad removes the remaining liquid from the air so that cooler, drier, cleaner air leaves through the top of the unit. Figure 2 shows the Spray Cooler. This cooler was issued U.S. Patent No. 4,394,142 on July 19, 1983. In addition to the 12 units built under this program, 38 other units have been built and sold commercially by FMI. The 50 extant units are at the following locations:

a. 46 at the Homestake Gold Mine, Lead, SD
b. 2 at the Sunshine Mine, Kellogg, ID
c. 1 at the Mt. Taylor Mine, NM (now closed)
d. 1 at the University School of Nevada, Reno. NV.

Figure 3 shows the original prototype undergoing developmental testing at the test facility built under the program at FMI's Saxonville Laboratory. Figure 4 shows the same unit under initial tests at the Homestake Mine. Both in the laboratory and in mine tests these coolers achieved an F-factor of 0.626, which is very high for a single-stage cooler. The coolers, in fact, exceeded the desired operational specifications. Although performance degrades slightly with extended use and virtually no maintenance, these coolers continue to operate at satisfactory performance levels under these conditions. Informal communication from John Marks, Senior Ventilation Engineer at the Homestake Gold Mine, after a performance inventory in March of 1984, indicates the small spray coolers "...perform remarkably well at low levels of maintenance." This evaluation includes coolers that have operated almost continuously for about 4 years.

E.S.3   SUMMARY OF LARGE COOLER DEVELOPMENT

Success with the small cooler prompted development of a larger cooler with 25,000 cfm nominal airflow. The Mt. Taylor Mine expressed considerable interest in a cooler in this range.

The 25,000 cfm airflow requirements greatly exceeded the capacity of the laboratory test facility precluding a laboratory development program like that undertaken for the small coolers. For this reason, four alternate cooler configurations were designed and built for full scale testing. All designs were to fit within a roughly 11-1/4- long by 6- wide by 4-ft high envelope. This mine-specified size permitted only horizontal airflow which, in turn, precluded the mesh approach. To minimize air pressure loss, a wave plate mist eliminator was used in all designs. The designs are briefly described as:

WATER IN

SPRAY
MANIFOLD

AIR IN

EGG CRATE

AIR FLOW
STRAIGHTENER

PLASTIC MESH

WASH MANIFOLD

MIST ELIMINATOR

DRIP FINS

AIR OUT

WATER OUT

FIGURE 2. - New spray cooler.

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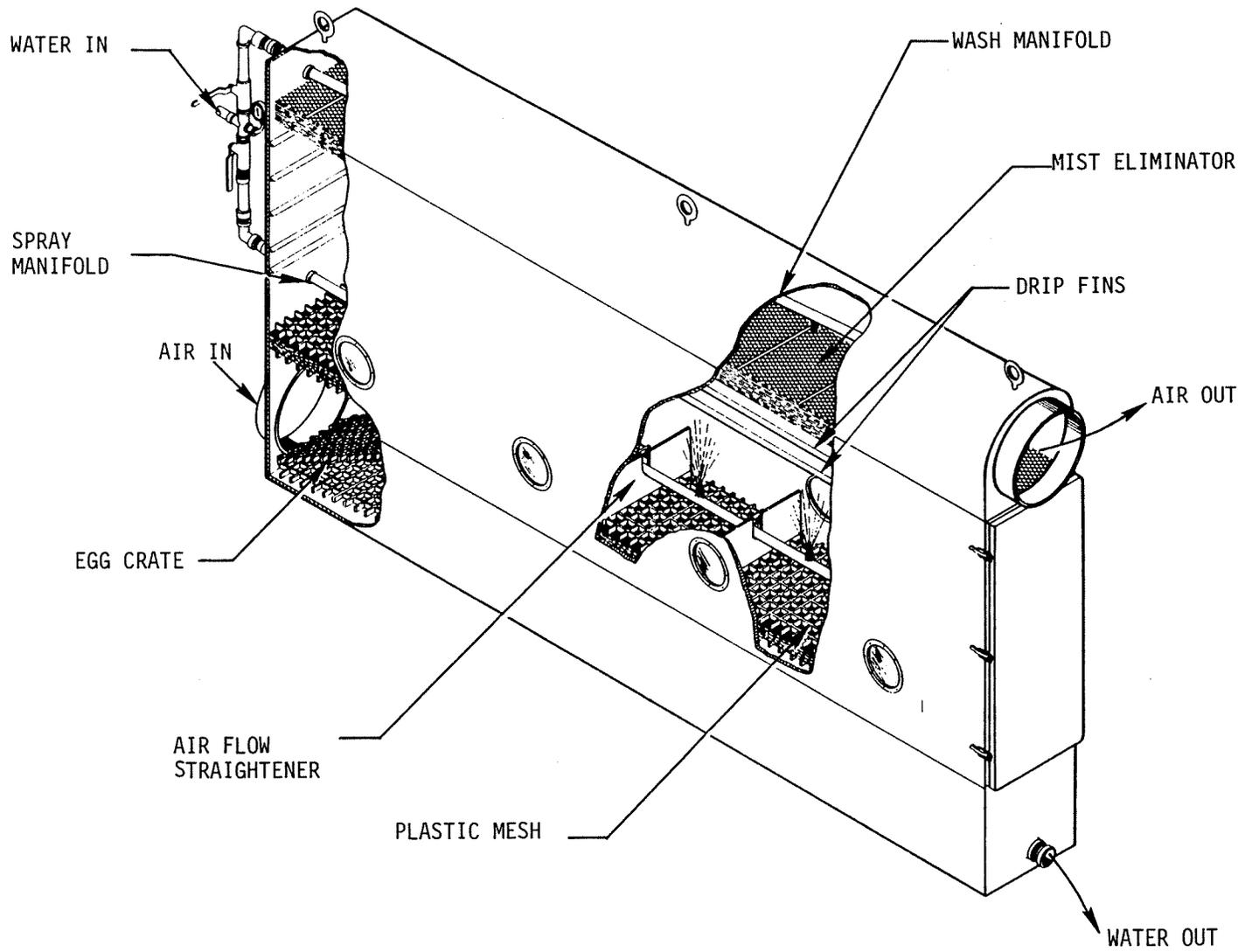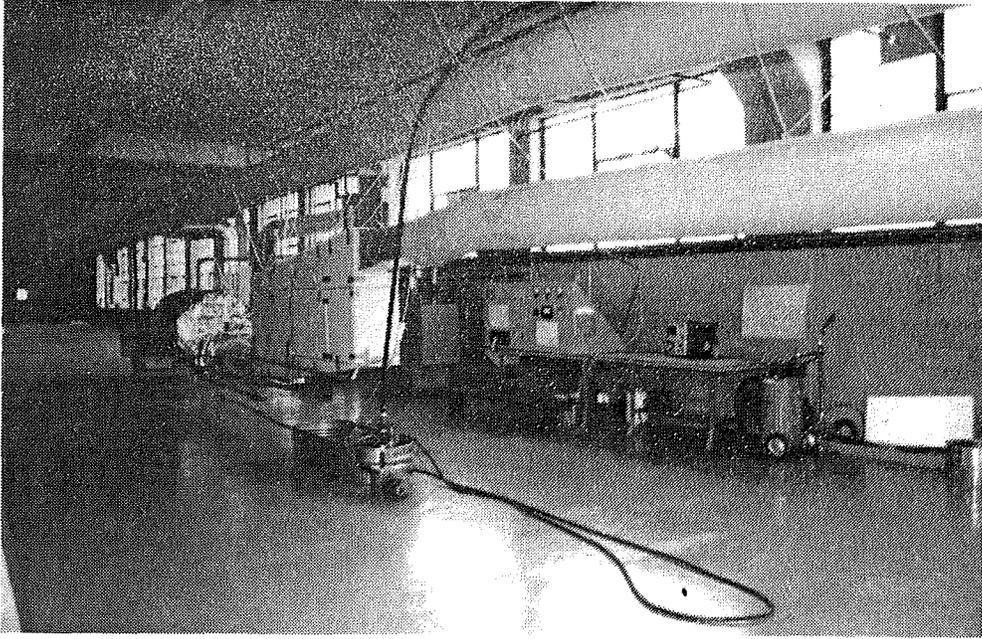
FIGURE 3. – Laboratory testing of prototype cooler.
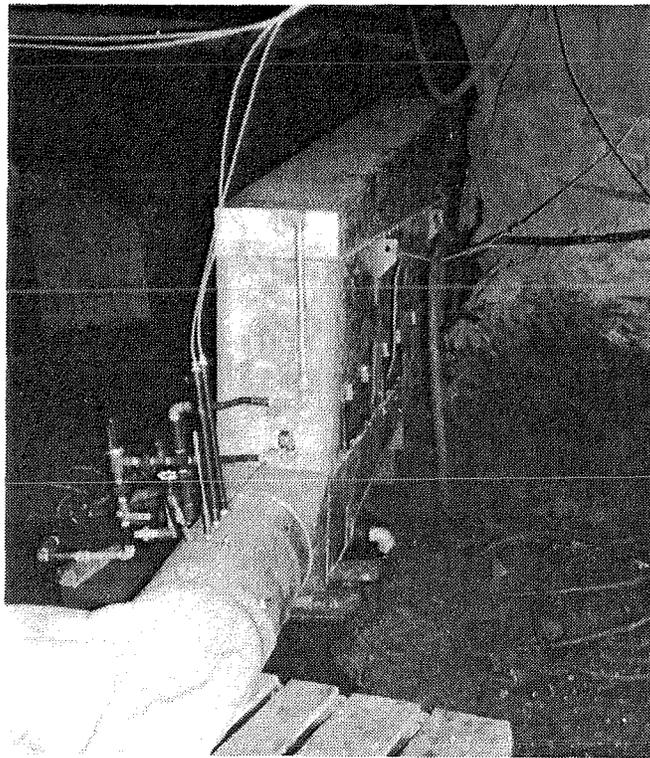
FIGURE 4. – Prototype cooler during in-mine tests.

a.   Single-stage unit with 40 spray nozzles spraying against the airflow. A stack of corrugated plates channel the spent spray water counter to the airflow.

b.   Two-stage[1] spray chamber with 72 spray nozzles per stage. Each stage consists of two manifolds of 36 nozzles each. The first (cold water stage) sprayed counter to the airflow. The second water stage was supplied with water pumped from the pump of the first stage. In this stage, water was sprayed both with and counter to the airflow.

c.   Three-stage spray chamber with each stage consisting of a single 40-nozzle manifold. Water from the second and third stages was pumped from the pumps of the preceding stages. Between the first and second stages, an extra mist eliminator was installed to aid heat transfer. Of course, a final mist eliminator was installed downwind of the first water stage.

d.   Three-stage unit similar to unit c, above, without the interstage mist eliminator.

The single-stage unit was tested in the Mt. Taylor Mine with an F-factor of 0.525 which is quite high for a single-stage, non-mesh type cooler. This unit performed well without maintenance for over 6 months before the mine closed for economic reasons.

The Homestake Gold Mine offered to cooperate with testing of the remaining coolers. However, only two units were installed and tested before the end of this program. During 1985, the Homestake Mine plans to install and test unit d. Data on this cooler will be provided to the Bureau by Homestake.

Both the two-stage cooler and the three-stage cooler with two mist eliminators have been tested. The two-stage unit tested out with F=0.5. The three-stage unit achieved an F=0.6, which is considered excellent. Both units are presently operating satisfactorily at the Homestake Mine.

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[1] In multistage coolers described in this report, the first water stage is where the coldest water is sprayed. This is the last spray the air passes through before leaving the chamber. This approach produces a counterflow heat exchanger effect.

E.S.4 SUMMARY OF CONCLUSIONS

Program goals were met. High performance, low mainte-
nance spray coolers were developed. At the beginning of
this program, the duct cooler conceived and evaluated
underground by the Bureau personnel had achieved an
F-factor approaching 0.5. This program has advanced the
state-of-the-art by producing mineworthy coolers with
cooling performances of F=0.626. As a result of the
Bureau in-house and contract research efforts, spray
coolers have been incorporated into United States mine
cooling system designs.

With additional development, the large, single-stage
cooler with corrugated plates offers considerable
promise. The F=0.525 measured for this cooler was
achieved with large temperature stratification inside the
cooler. Developmental effort to produce a nearly uniform
air velocity profile inside this type of cooler should
improve this F-factor considerably. The resulting cooler
should exhibit high cooling performance, and low air
pressure loss, be of compact design, and require little
routine maintenance. This is a promising direction for
future research.

1. INTRODUCTION

1.1 BACKGROUND

1.1.1 General

Improved methods for workspace cooling become necessary as mining progresses deeper into regions with higher rock temperatures. Cooling of workers is desirable to maintain productivity and essential to ensure against heat stroke and possibly death. This program has developed some new tools for mine ventilation engineers to use for the thermal protection of miners.

While heat problems frequently do not develop until mining operations descend below sea level, problems will arise any time mining penetrates high temperature rock. Hot fissure water will increase the severity of the situation, causing increases in wet bulb temperature - the temperature at which perspiring miners reject metabolic body heat. All energy expended in mining adds to the heat load which must be absorbed by the ventilation air. Just when the cooling capacity of ventilation air is needed most, adiabatic compression (autocompression) of air descending the shaft raises dry bulb air temperatures 2.5° F to 5.3° F/1,000 ft, depending on humidity. Increased temperatures decrease the ability of the air to absorb heat from the mine, causing work area temperatures to rise.

Once forced ventilation air can no longer maintain safe work area temperatures, refrigeration methods must be employed. Many mines have used cooling coils for this purpose. A cooling coil uses refrigerated or naturally cold mine water to cool the air by passing the air through banks of copper tubing passing this cold water. Heat from the air is transferred to the water through the pipe with the aid of fins. Conventional water-to-air heat exchangers prove troublesome for many mines because of the high levels of dust that are encountered, causing these exchangers to operate at relatively low efficiencies or with very high maintenance. A more satisfactory approach to workplace cooling is desired, one that offers reduced maintenance and increased cooling efficiency.

1.1.2 Historical

In the late 1970s, Thimons et al. ([1]) at the Bureau began experiments with small, moveable, direct water-to-air heat exchangers - spray coolers.

Development of these spray coolers began with testing
by the Bureau of a spray system that mounted in line in
the ventilation tubing. This system consisted of a series
of axially mounted spray nozzles in a 16-in diam duct as
shown in figure 5. Preliminary testing at the Homestake
Mine proved promising.

Based on the promising results of this spray cooler
testing, the Bureau contracted with FMI to optimize mine
spray coolers. Besides improving the thermal performance,
greatly improved mechanical design was also considered
necessary to make the coolers mineworthy.

1.2 PROGRAM OBJECTIVES

Initially the objective of this program was to
optimize and test a 5,000 cfm cooler. Success in meeting
this objective prompted an expansion of the objective to
include the development of larger 25,000 cfm coolers.

1.2.1 Optimization and Testing of 5,000 cfm Coolers

The first objective of this program was to optimize
and evaluate, through both laboratory and in-mine tests, a
spray cooler of 5,000 cfm and 12 gpm air and water rates,
respectively. Optimization of a mine cooler does not end
with maximum thermal performance. High thermal perform-
ance in extracting most of the "coolth" from the water is
necessary, but equally necessary is trouble-free operation
within the mine environment. The space limitations of the
underground environment make maneuvering of bulky objects
and servicing of any equipment difficult. Deficiencies in
reliability are intolerable.

1.2.2 Extended Objective

Successful completion of the objective for the
5,000 cfm spray cooler developed into an additional
objective for the development of a mineworthy 25,000 cfm,
100 gpm spray cooler. Novel concepts were considered as
the cooperating mine requirements precluded the applica-
tion of the concept used for the successful 5,000 cfm
cooler design.

10 ft LONG, 16 in. DIAM FIBERGLASS TUBE
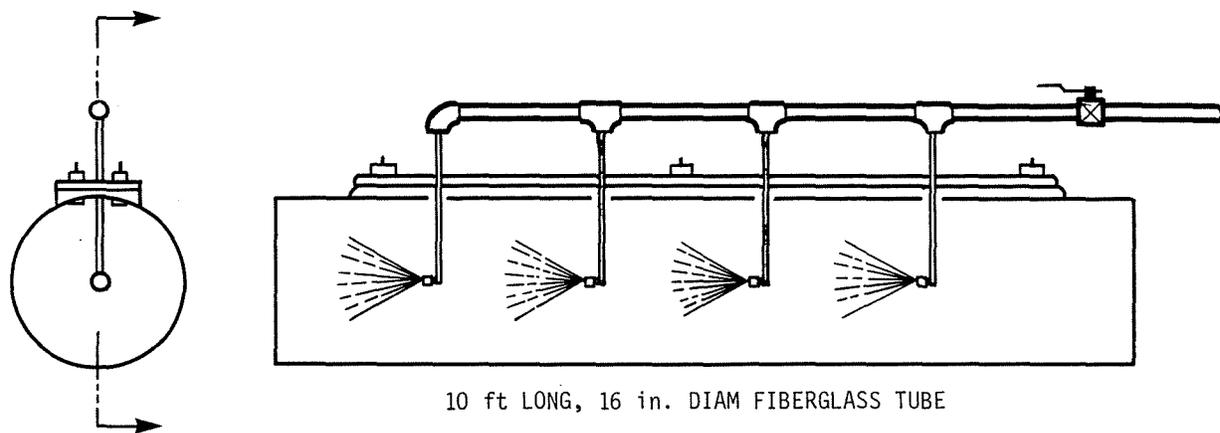
FIGURE 5 – Duct spray cooler as tested at the Homestake Mine.

2. ANALYSIS FOR THERMAL PERFORMANCE

2.1 FACTOR OF MERIT

Spray type cooling units have been most frequently characterized as having a cooling capacity or power of so many refrigeration tons (R-tons) or kilowatts. This is misleading. The cooling capacity of a unit depends on the inlet fluid temperatures and flow rates. This may be illustrated by considering the extreme case of a cooler supplied with water at the same temperature as the input airstream. Obviously, the cooling capacity under these conditions is nil.

In this development program, the performance of the cooling unit has been characterized by the factor-of-merit method, or F-factor, originally developed by Whillier ($\underline{2}$) and improved by Bluhm ($\underline{3}$-$\underline{4}$). The F-factor for a cooler is independent of operating temperature and barometric pressure. It is equivalent to knowing UA[1] for a conventional heat exchanger. The F-factor may also be thought of as the cooler efficiency when the capacity factor, R, is one.

Measuring the water efficiency at known arbitrary capacity factors is sufficient, as a semi-empirical relationship exists that relates to the F-factor. This is based on the hypothesis that for a given cooler (and by extention a given cooler design), a unique relationship exists between cooling water effectiveness or efficiency, E_W, and capacity ratio.

For a counterflow (airflow and waterflow in opposite directions) spray cooler, Bluhm ($\underline{3}$-$\underline{4}$) gives the relationships:[2]

$$E_w = \frac{1 - e^{-N\,(1-R)}}{1 - Re^{-N\,(1-R)}}, \quad R \neq 1$$

$$E_w = F, \quad R = 1 \text{ only}$$

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[1]The product of the overall heat transfer coefficient and the effective surface area of the heat exchanger.
[2]Terms are defined at the beginning of this report.

where

$$N = \left(\frac{F}{1 - F} \right) \Big/ R^{0.4}$$

He gives additional relationships for cross flow and
parallel flow spray coolers; but over the range of con-
ditions for coolers considered within this program, this
analysis provides sufficient accuracy. Figure 6 presents
typical performance curves using the above relationships
for various F-factors.

From figure 6, it may be seen that given an F-factor
(for a specific cooler design) and the operating con-
ditions (which define R), the water efficiency is uniquely
defined. Thus, the cooling power is predicted for the
operating conditions by:

$$P = E_w LC(t_{ai} - t_{wi})$$

(Btu/hr in conventional units,
kW in SI units)
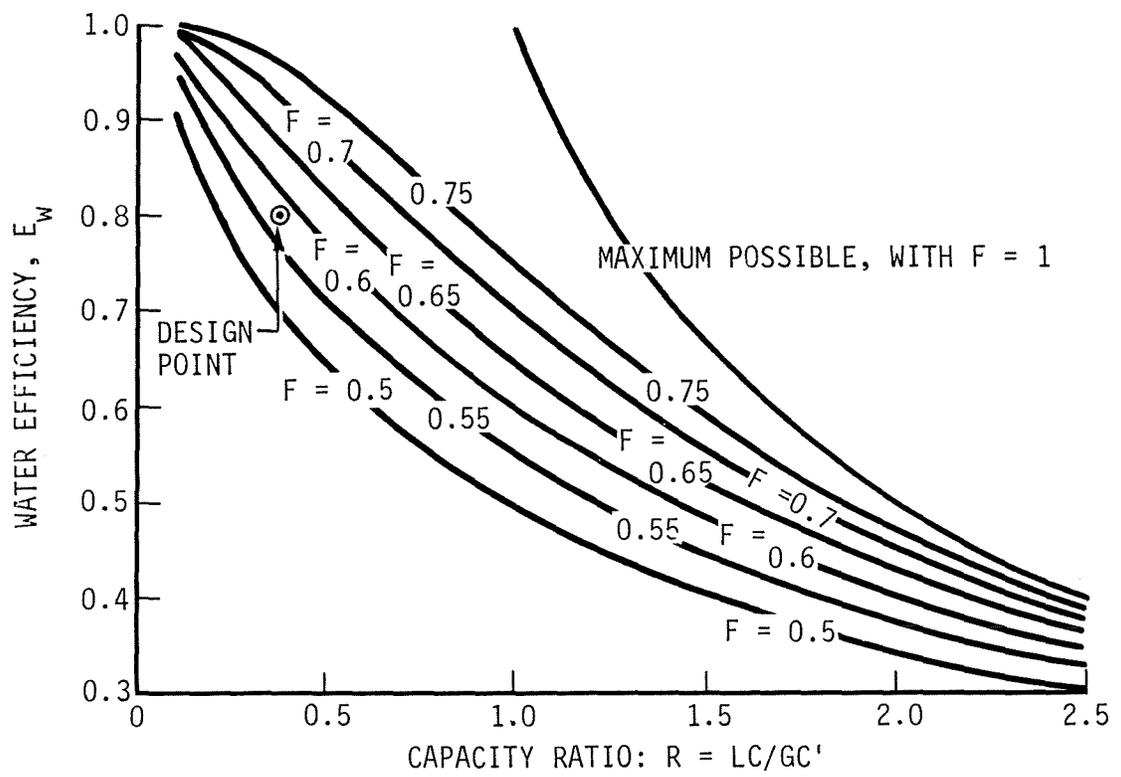
FIGURE 6. - Performance curves of counterflow spray
cooler heat exchangers.

Conversely, the F-factor is determined from water efficiency (E_w) and capacity ratio (R) analytically through the previous equations or graphically by location on figure 6. Performance F-factors in this program were determined analytically through the aid of a programmable calculator. This allowed rapid onsite analysis during both laboratory and in-mine testing.

Comments on C' are warranted. This term is the average equivalent specific heat of humid air when a change of state of water is involved. In this report, it specifically refers to the average equivalent specific heat between outlet air temperature and inlet water temperature. Alternative ways exist for calculating this term, namely:

$$C' = \frac{S_{ai} - S_{wi}}{t_{ai} - t_{wi}} \quad \text{(by sigma heat)}$$

or

$$C' = \frac{h_{ai} - h_{wi}}{t_{ai} - t_{wi}} \quad \text{(by enthalpy)}$$

For detailed psychrometric reasons, both methods have small residual errors. This report has used the first relationship based on sigma heat (see Whillier (1))for analysis. Errors are limited to between 3 to 4%; insignificant for practical work (1). Furthermore, dry bulb temperature is not necessary to calculate sigma heat, unlike enthalpy which requires a dry bulb temperature measurement. Dry bulb temperature measurements are often difficult to make in the often mist laden air at the cooler outlet.

In summary, the use of the factor-of-merit methods allows a spray cooler to be tested at a single operating condition. These results (F-factors) permit prediction of performance (cooling power in R-tons or kilowatts) at all operating conditions. F-factors lie between 0 and 1.0 inclusive. The larger the F-factor, the better the cooler performance, under all conditions.

2.2 NOTES ON MEASUREMENT TECHNIQUES

Required performance measurements are:

a.   Inlet air wet bulb temperature
b.   Outlet air wet bulb temperature
c.   Outlet water temperature
d.   Inlet water temperature
e.   Inlet water pressure
f.   Waterflow
g.   Airflow
h.   Barometric pressure
i   Static air pressure loss across the cooler.

All temperature measurements were made to better than 0.2^0 F. ASTM quality mercury in glass thermometers with graduations of 0.1^0 or 0.2^0 F proved satisfactory for most of the testing. Although moderately priced, rapid breakage during extensive in-mine testing demanded an alternative. A calibrated, digital, thermocouple (Bailey Instruments, Inc.)[1] was found to be a practical alterna- tive. This instrument eliminated the breakage problem and proved to be more easily read than the thermometers.

Wet bulb temperature measurements were made by cover-ing the thermometer sensing surface with wetted cotton gauze (finger bandage). Equilibrium at a constant tem-perature reading usually occurred rapidly after insertion into the airstream. The high humidity of the airstream resulted in very slow drying of the gauze, a desirable situation. Dry bulb temperatures were usually recorded but not used in calculations as explained in the previous section. Stagnation temperature effects were negligible in this program.

Outlet water temperatures were measured by inserting the thermometer bulb directly into the outlet water stream draining from the cooler. However, direct bulb insertion of mercury in glass thermometers into the inlet flow was unacceptable for measuring inlet water temperatures. The water pressure in the inlet pipe squeezed the glass bulb and caused temperature readings that could be many tenths of a degree higher than actual. Thermometer wells on the inlet water pipe, insulated from the ambient air, pre-vented this pressure effect and provided accurate tem-perature modification. Inlet water temperatures sometimes had to be corrected for inlet pressure. Joule heating of

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[1]Reference to specific equipment (or trade names or manufacturers) does not imply endorsement by the Bureau of Mines.

the water occurs when the pressure is relieved adiabati-
cally. Upon release of the pressure, the potential energy
from the pressure is converted to heat in the water. The
water temperature increases approximately 0.003^o F/psi.
A release of 100 psi, therefore, results in a temperature
increase of 0.3^o F, which is significant for F-factor
measurements. For water pressure of about 30 psi or less,
joule heating may be neglected.

For the smaller cooler, with waterflows less than
20 gpm, waterflow was measured using a Meter Equipment Co.
flow meter. Calibration was verified by stop watch
monitoring of water delivery into a bucket and determining
the delivered water gravimetrically. For the larger
coolers, with 50 and 100 gpm flows, a calibrated RCM
Industries differential pressure type flow meter was
used. Few problems exist with the waterflow measurement;
accuracy is very good.

A variety of instruments and techniques were used to
determine airflow. They are:

a. Annubar (multiport pitot probe)
b. Orifice meter - designed from handbook data
c. Pitot-static tube traverse
d. Anemometer traverse.

Accurate airflow measurements proved to be the most
difficult. Accuracies achieved ranged from very good to
very poor. For this reason, when a heat balance check
(comparison of heat gained by the water to heat lost from
the air) shows an unequality, the value of the airflow was
assumed to be in error and was adjusted to force a heat
balance before proceeding with the performance calculation.

Barometric pressure is needed for psychrometry and to
determine air density for calculating mass flow rate of
air. Any sensitive barometer should be satisfactory. In
this program, a sensitive aircraft altimeter was used.
This very robust, yet accurate instrument is well suited
to in-mine work.

Static air pressure loss across the cooler is not
required to determine the F-factor of the cooler, but is
necessary for determining the fan power necessary. Fan
power requirements may be important in overall mine
ventilation assessment.

For static pressure taps in this program, we used
small drilled holes (N1/16 in) on the inlet and outlet
air connections. To these holes, a "slack tube" water

manometer was connected by butting the end of 1/4-in soft plastic tube over the hole.

All instrumentation to test the performance of a spray cooler is easily man portable - even underground. This portability allows several coolers within a mine to be tested during a single man shift.

3. LABORATORY TEST FACILITY

3.1 PURPOSE AND REQUIREMENTS

An efficient cooler optimization program demands rapid, close control of parameters and rapid feedback on cooler performance. A laboratory test facility was designed and constructed to provide the required control and feedback. This facility was designed to provide humidified air at temperatures expected in hot mines. Basic requirements were established using the Bureau's earlier in-mine testing experience of Thimons (1). Specific minimum requirements established were:

a. Airflow: 5,000 cfm at 80° F wet bulb, 85° F dry bulb

b. Waterflow: 12 gpm at 100 psi, 50° F

c. Sufficient fan capacity to maintain the desired airflow through 16-in diam vent tubing.

3.2 DESCRIPTION OF FACILITY

The laboratory facility is depicted in figure 7. Figure 8 is a photograph with the prototype spray cooler installed for testing. The following is a description of the airflow and waterflow systems.

3.2.1 Airflow System

A centrifugal blower supplied airflows adjustable to over 5,000 cfm at pressure in excess of 11-in of water gauge. The air intake to the blower could be proportioned between various quantities of room air and the recycle loop (for moisture recovery). A short section of 16-in vent tubing connected the blower to the conditioning chamber. Inside the chamber the air was heated with gas burners to raise the dry bulb temperature and injected with steam to increase the wet bulb temperature. Steam was supplied from a gas-fired boiler capable of producing 8.3-lb of steam per min. After leaving the conditioning chamber, the air passed through a 15.5-in measurement section where flow rate, wet and dry bulb temperatures, and pressure were measured. After passing the measurement section, the air flowed through the test cooler. Upon leaving the cooler, the air again passed through a measurement section which monitored wet and dry bulb temperatures and pressure. Fifty feet of 16-in vent tubing followed this measurement section and led to a third measurement section where the same parameters were monitored. Air left the last measurement section and was recycled back to the blower intake through 18-in tubing.
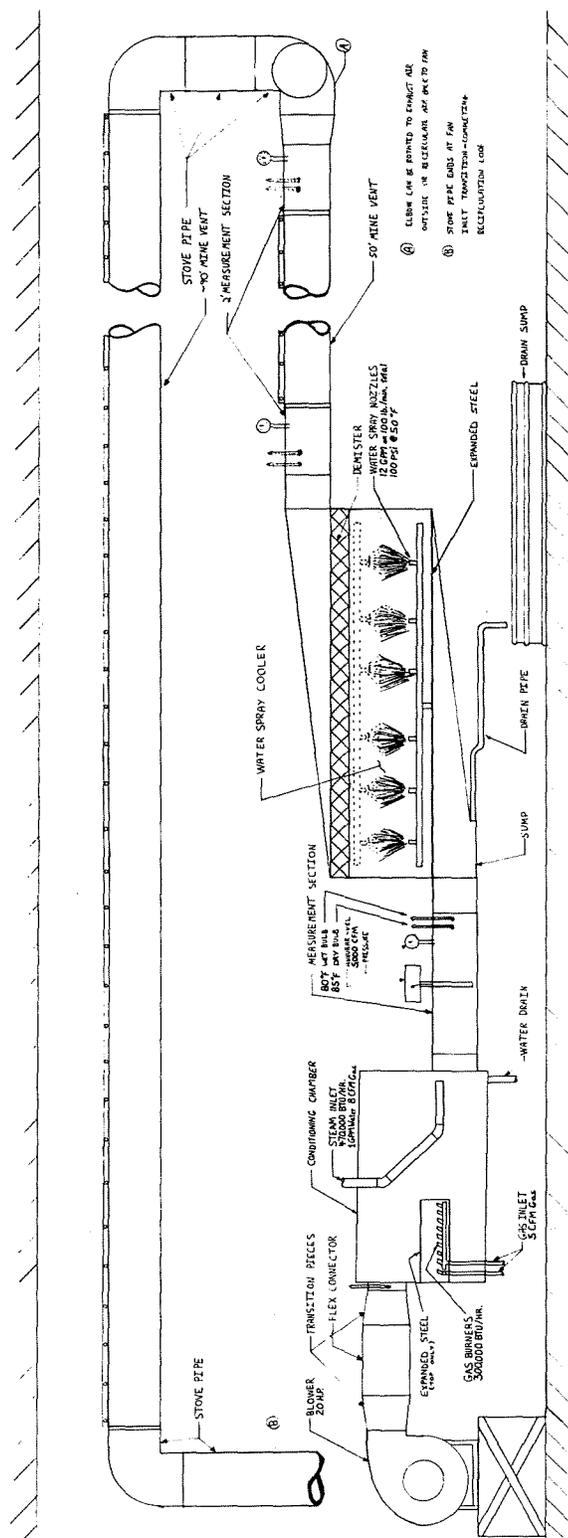
FIGURE 7 - Laboratory test facility.
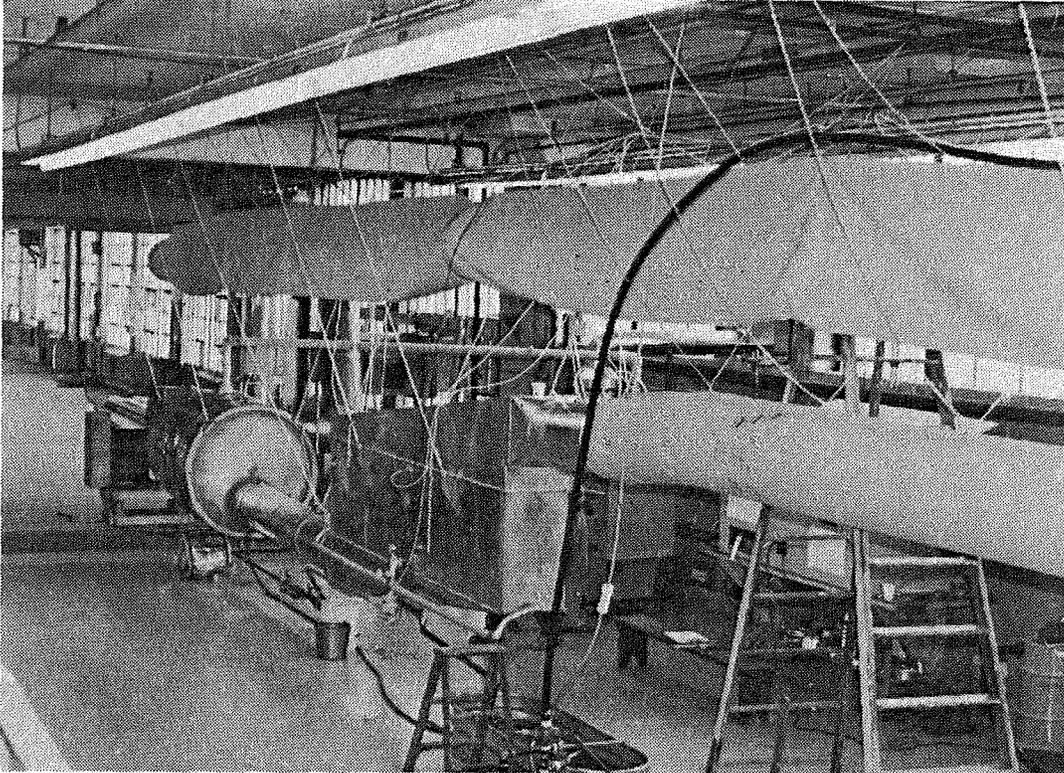
FIGURE 8 - Test facility photograph.

3.2.2 Waterflow

Original planning for the water spray system assumed that the temperature of the city water would sometimes rise above 50° F. This information, provided to us by other building occupants, led us to install a coil to cool the water to 50° F with the use of outside air. It turned out that the city water temperature always remained below 50° F, thus making the heat exchanger unnecessary. Water coming through the building's main line was directed through a 100 mesh "Y" strainer for cleaning. It then entered a "Hypro" roller pump and pressure relief valve assembly where its pressure was raised to 100 psi. Water that passed through the relief valve was recycled back to the pump. Following the relief valve, the water, running through 3/4-in pipe, passed through a Griswold constant flow control valve and a flow meter before entering the spray cooler. In the cooler, the water was sprayed from the nozzles. Spray water accumulated at the bottom of the cooler and drained out to the sump. Drained cooler water was not recirculated, but was pumped to a building drain.

4. SMALL COOLER DEVELOPMENT AND TESTING

4.1 BEGINNINGS

Initially in this program, it was assumed that cooler development would center around variations of the Bureau duct cooler (1) (see figure 5, section 1). However, application of the new Factor of Merit analysis (see section 2) indicated that the duct cooler as developed and tested by the Bureau was operated near the limit of theoretical expectations. Furthermore, engineers from the Homestake Gold Mine in Lead, SD had carried on with experiments where the Bureau efforts had stopped, and had convinced themselves that satisfactory thermal performance from the duct cooler was improbable. Practical problems had been encountered. Insufficient mist elimination had caused downstream vent tubing to tear away supports due to the weight of collected water. These practical limitations of the duct configuration indicated that new approaches were necessary.

During the second Mine Ventilation Congress at Reno, NV in November of 1979, all interested parties, the Bureau, Homestake Mine engineers, and FMI engineers and consultants met informally and resolved, for design purposes, a set of initial specifications. These are shown in table 1.

The size was determined by the need for maneuverability and situation along the rib of the Homestake Mine where testing was to be performed. Thermal conditions were set to meet or exceed existing fin-tube, coil-type coolers installed in the mine.

To satisfy the specified thermal performance, the cooler must operate at the "design point" shown in figure 6 of section 2. To do this, it was expected that considerable counterflow effect would be required. Also, the practical requirement of mist elimination had to be addressed. The magnitude of this problem can be illustrated best by the following statement. If only 10% of the 12 gpm waterflow to the cooler passes through the mist eliminator and settles in downstream vent tubing, then 600 lb/hr is deposited within the tubing. Excellent mist elimination is therefore needed.

Conceptually, our approach was to select a high efficiency pad of knitted stainless steel to be certain of total mist elimination. To be effective, these pads must be oriented horizontally, and the air velocity through

TABLE 1. - Initial specifications of spray cooler

Maximum dimensions	406-mm (16-in) wide
	1,680-mm (66-in) high
	3,050-mm (120-in) long
Inlet airflow	2.36 m^3/s (5000 fpm)
Inlet air temperature	26.67° C (80.0° F) wet bulb
	29.44° C (85.0° F) dry bulb
Inlet waterflow	0.76 l/s (12 US gpm)
Inlet water pressure	689 kPa (100 psi)
Inlet water temperature	10° C (50° F)
Minimum water efficiency	80% (this implies a cooling duty of 42.2 kw, (or 12R-ton)
Other	No significant mist shall be present in the downstream air flow since the weight of water deposited in ventilation ducting can lead to tubing support failure.

them reduced to about one-tenth the velocity in the 16-in duct. This requirement suggests turning upward and expanding the airflow. Gravity could then aid in producing counterflow heat exchange with a downward waterflow.

4.2 PROTOTYPE COOLER

4.2.1 Description of Prototype

The prototype cooler developed in accordance with the above specifications is depicted in figure 9. In this prototype, air is brought horizontally into the bottom of the cooler, where it is slowed and turned upward. The

FIGURE 9. - Original prototype spray cooler.
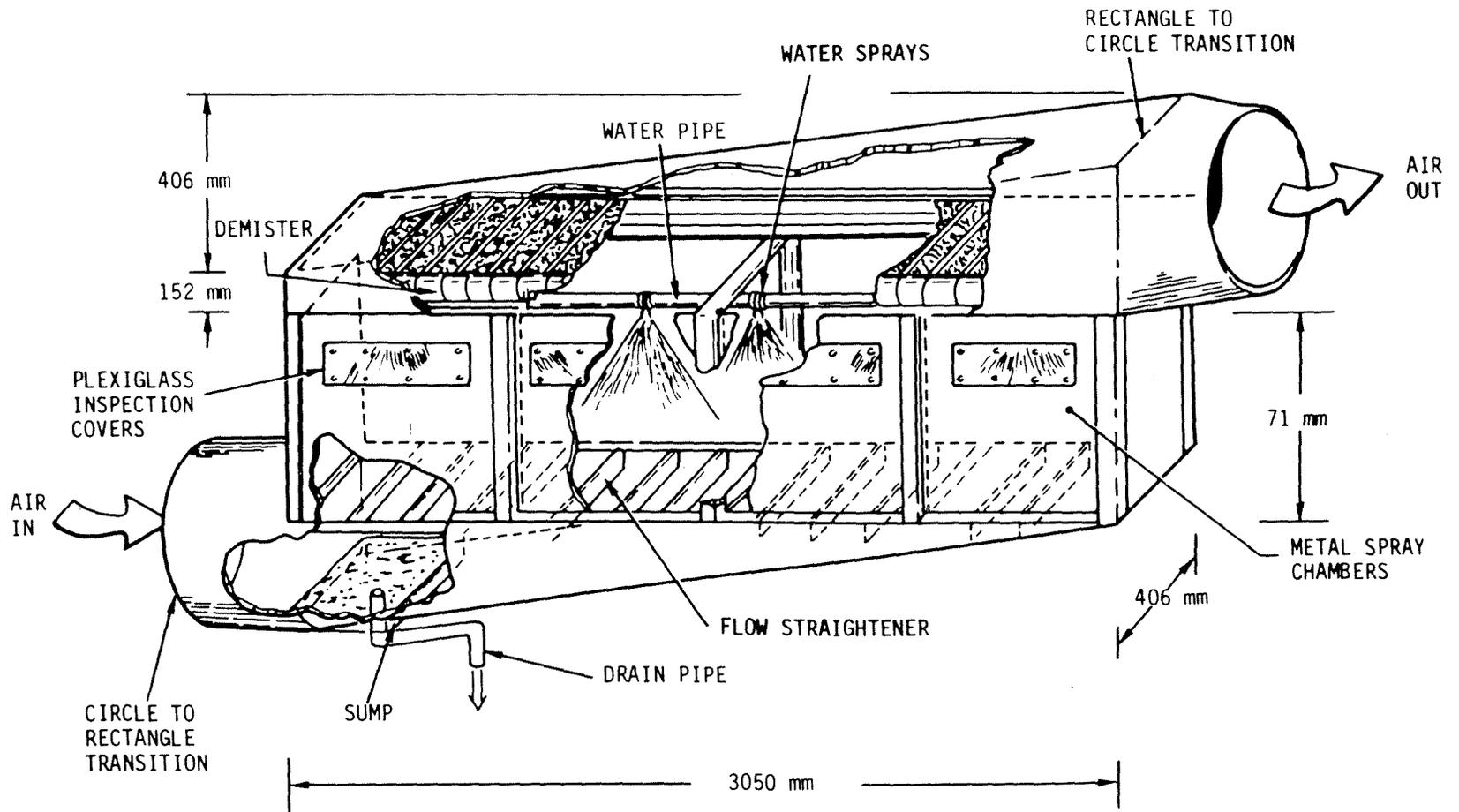
ratio of duct air velocity to chamber velocity is approximately 10:1. Water was sprayed downward against the airflow from seven, 46°, solid cone nozzles. Above the manifold, a mist elimination pad of knitted stainless steel is installed to remove droplets from the airstream. Above the mist eliminator, air is turned from vertical flow to horizontal flow and accelerated to duct velocity.

A prototype cooler based on the original concept was fabricated from galvanized sheet steel. This unit was installed in the test facility which is described in section 3.

4.2.2 Laboratory Testing

With the prototype cooler installed in the facility, a tuning, testing, and modification program was undertaken to optimize the performance.

Elements evaluated experimentally were:

a. Techniques for turning the airflow to provide constant velocity upward within the chamber to aid in promoting a uniform mix of air and water

b. Spraying water downward or upward (relocating the manifold at the bottom of the chamber for upward spray)

c. Various types and numbers of water spray nozzles consistent with the waterflow and pressure requirements

d. The use of sidewall-mounted fins to aid in entraining water deposited on the walls.

Through this experimental development effort, considerably improved design and performance was achieved. The resulting cooler is illustrated in figure 10. It has seven upward-spraying nozzles with solid-cone sprays of about 46°. This was the smallest standard angle, solid-cone spray that uniformly impinged on the mist eliminators. Upward-spraying nozzles showed a slightly more efficient performance than downward-spraying nozzles. This effect is suspected to be due to more uniform water loading throughout the chamber volume, and, possibly more importantly, the involvement of the wetted mist eliminator pad in promoting good water-to-air thermal contact. It was found that drip fins installed on the sidewalls improved performance slightly. For this particular prototype, reasonably uniform upward air velocity distribution can be achieved using "egg crate" material
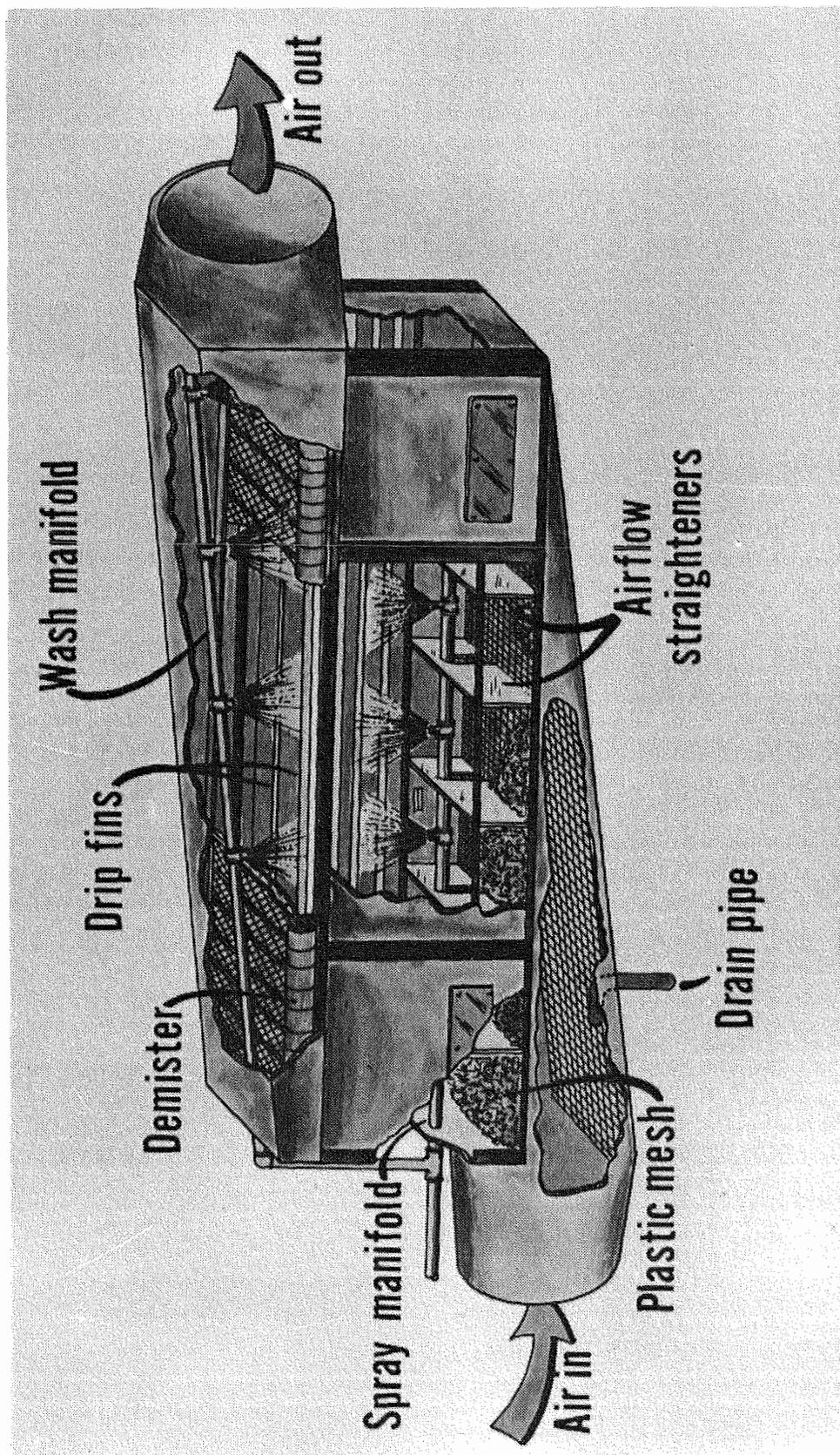
FIGURE 10. – Final configuration of prototype cooler.

(approximately 1/2-in square by 3/8-in thick plastic
material, such as is used in lighting fixtures) and the
flow-straighteners located mid-way between spray nozzles.
Not installed for the laboratory testing was the plastic
mesh shown between the flow-straighteners under the spray
manifold. This material was installed during the in-mine
testing which is discussed later.

At the end of the laboratory testing, the prototype
cooler was prepared for mine testing by incorporating all
features in a mechanically robust manner. Also, a wash-
down manifold of sprays was installed above the mist
eliminator as a method of countering any dust accumulation
on the pad. On the water side, a Griswold constant-flow
valve was installed to limit water consumption to about
12 gpm as the nozzle orifices eroded to larger diameters.
As a precaution against nozzles clogging with particulate
matter in the water, a 100-mesh "Y" strainer with back
flush capability was installed in the water feed line.

Before shipment to the mine, the cooler was tested at
specification conditions (table 1). The test results gave
a water efficiency of 81.3 % and an F-factor of 0.62 with a
cooling duty of about 45 kW or 12.8R-ton. This perform-
ance exceeded the initial specification of a Factor of
Merit of 0.58. At this operating point the air pressure
loss across the cooler from inlet duct to outlet duct was
measured as 2.2 in of water gauge.

A summary of the features of the prototype cooler at
the time of final testing is as follows:

a. Average air velocity through this 15.5-in diam
   inlet and outlet connection: 64 ft/s

b. Average air velocity through the spray chamber and
   mist eliminator: 6.2 ft/s

c. Average water flux through mist elimination pad:
   0.91 gpm/ft^2

d. Average waterflow per unit chamber volume: 0.41
   gpm/ft^3

e. Ratio of nozzle spacing to chamber width: 1.1:1,
   for a width of 16 in

f. Nominal thickness of mist eliminator pad: 4-in
   knitted stainless steel wire.

With the encouraging results presented above, the
cooler was shipped out for in-mine testing.

4.2.3 In-Mine Testing

The prototype cooler was installed at the 6,200-ft level of the Homestake Gold Mine in Lead, South Dakota. Figure 11 shows a photograph of the cooler installed. Inlet air flows, air temperatures, and water temperatures were found to be considerably different from the initial specifications, and also from those which were used to test the cooler in the laboratory. Because these conditions differed from the design and test specifications, the Factor of Merit method for comparison was invaluable. During these tests, the wet bulb temperature of the air entering the cooler ranged from 80° to 83° F. (The dry bulb temperature was raised about 9° F over ambient due to heating from the forced draft blower). Inlet water temperature ranged from 54.1° to 63.8° F. For 24 tests, the F-factor averaged 0.61 with a standard deviation of 0.015. This matched the laboratory performance and considerably exceeded the original specification of 0.58.

Had cooling power alone been used as a performance measure, instead of F-factor, it would not have been obvious that laboratory performance had been matched. During laboratory testing, 12.8 R-tons had been realized while the maximum cooling achieved during in-mine testing was 10.5 R-ton. This decrease in cooling was due to the
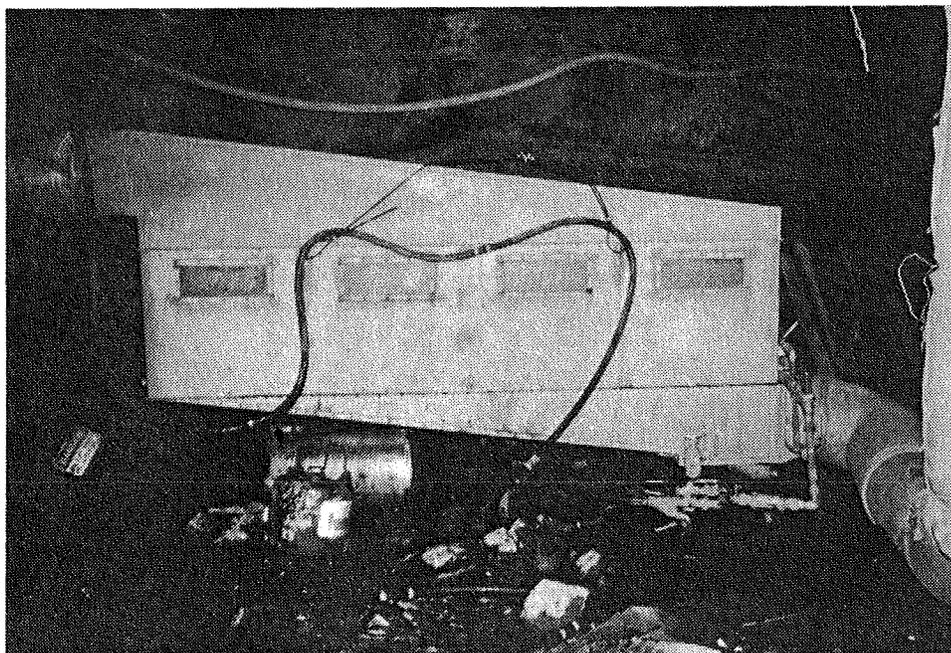

FIGURE 11. - Prototype cooler installed in mine.

hotter inlet water temperature at the mine, and did not indicate poorer performance of the cooler itself.

During the in-mine testing period, it was recognized, from both laboratory and mine test data, that the mist eliminator was likely to play a major role in the water-to-air contact and, hence, heat transfer. Realizing this, an additional eight layers of loosely packed, knitted mesh mist eliminator material was installed underneath the spray manifold (see figure 12). It was felt that this material enhanced the contact between the falling water and the rising air. The increase in F-factor achieved due to this mesh was about 0.02 or 3%.

During the 4-week underground testing period, the ambient temperature of the working stope cooled by this prototype cooler dropped from 81.5/82.5° F (wet/dry bulb) to 77/80° F. During the extended test period, no problems were encountered with water collecting in the downstream 350-ft of vent bag tubing.

These tests were run in May of 1980. At the writing of this report (June 1984), this cooler is still operating at the Homestake Gold Mine. Mine maintenance personnel continue to monitor the performance of this unit. The cooling performance is still considered good and the level of required maintenance is low.

4.3 DEVELOPMENT OF THE DESIGN

4.3.1 Redesign

The performance of the prototype cooler and its acceptance by mine personnel was overwhelming. A patent was granted (U.S. Patent 4,394,142) which is assigned to the United States of America as represented by the Secretary of the Interior. A successful basic cooler design had been found. However, before building and testing the additional coolers required contractually, some redesign was desirable. This redesign incorporated:

a. Lower cost fabrication features

b. Simpler access to internal parts

c. Better external shape for mine geometries

d. Addition of a lower mesh for additional heat exchange.

Figure 12 shows the resulting redesigned cooler. Fifty coolers have already been built to this improved design. Eleven were for the subject program and the balance were built by FMI for commercial sales.
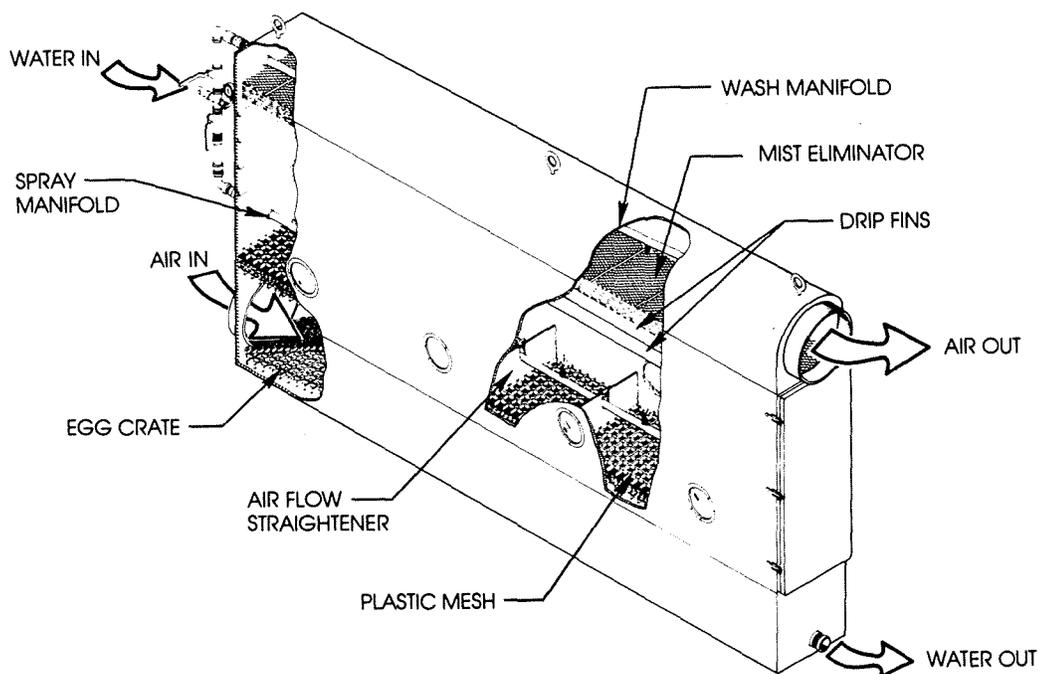
FIGURE 12. - Redesigned spray cooler-galvanized steel (120- by 16- by 66-in).

4.3.2 Laboratory Testing of the Redesigned Coolers

4.3.2.1 Performance Verification

To ensure that redesign did not adversely affect performance, several of the redesigned coolers were tested in the laboratory test facility (see section 3). Table 2 shows an average F-factor of 0.626 for the redesigned cooler. This represents a modest improvement over the prototype cooler. Data from the verification tests are plotted in figure 13. Note that the prediction curve of Bluhm (see section 2.1) for a value of F=0.626 fits the data except for some experimental scattering. Also shown is a point indicating the performance originally specified. The redesigned cooler shows a water efficiency that is five percentage points higher than required by the original specification.

4.3.2.2 Dust Removal Testing

The design of the spray cooler suggests that dust scrubbing from the cooled air is probable. To check this, a series of tests were run in which fine Arizona Road Dust was injected into a cooler's inlet duct, about 20 ft upstream. Personal sampler pumps with various preselection

Test No.	Air[1] flow, cfm	Water- flow, gpm	T_{ai}, °F	T_{ao}, °F	T_{wi}, °F	T_{wo}, °F	R	E_w, %	F	Cooling, tons
B1	4490	12.7	80.7	73.1	55.0	76.2	0.424	82.9	0.622	11.0
B3	4968	12.8	79.4	73.4	57.2	76.0	0.382	84.5	0.619	9.9
B4	4979	12.8	82.2	75.3	55.8	78.4	0.377	85.5	0.626	11.9
B2	5208	12.8	92.4	84.9	57.4	88.9	0.320	89.9	0.646	16.6
B5	5643	12.8	82.2	76.1	55.6	78.4	0.333	85.6	0.607	12.0
B7	4352	18.2	80.1	70.6	56.7	74.6	0.619	76.2	0.633	13.4
B6	4432	18.2	79.9	70.6	56.8	74.6	0.609	76.8	0.635	13.3
B8	4786	18.2	80.7	71.1	54.8	74.9	0.569	77.4	0.627	15.0
B9	4982	18.1	81.4	72.0	54.7	75.6	0.541	78.0	0.623	15.5
B10	5603	18.1	81.1	72.6	54.7	76.0	0.484	80.5	0.623	15.8
B11	4892	29.8	81.3	67.4	54.6	72.1	0.915	65.2	0.626	21.4
B12	5285	29.8	79.9	67.2	54.6	71.6	0.860	66.8	0.625	20.7

TABLE 2. – Redesigned spray coolers, laboratory test data

Average $\overline{F}$ = 0.626
Standard Deviation, σ = 0.009

[1]All data calculated by using an adjusted airflow to force a heat balance.
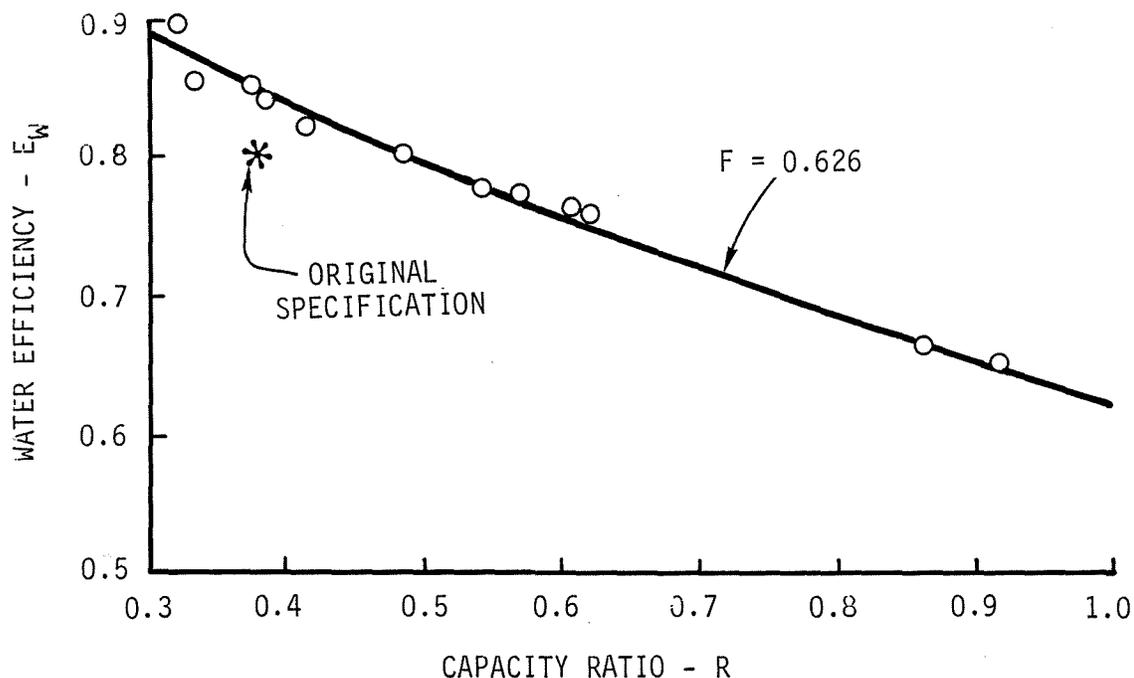
FIGURE 13. — Test performance.

impaction stages ahead of the collection filter were used
to measure dust concentration upstream and downstream of
the spray cooler. From these measurements, dust removal
efficiencies were calculated. Figure 14 shows the
results. In the dust particle size ranges tested, well
over half the dust in the air was removed. In some cases,
as much as 80% was eliminated. This dust removal effect,
particularly over an extended period, would contribute
significantly to a healthful work atmosphere.

4.3.3   In-Mine Testing

Nine of the redesigned coolers were installed at the
Homestake Gold Mine along with the original prototype. The
purpose of the extended in-mine testing was to determine
the long term performance of the coolers under actual
operating conditions.

Figures 15 through 18 show typical underground
installations. These coolers were continuously providing
cooling for miners working in the stopes. For these
miners, the coolers did not represent an experiment, but
their source of cool air. Miner acceptance has led to the
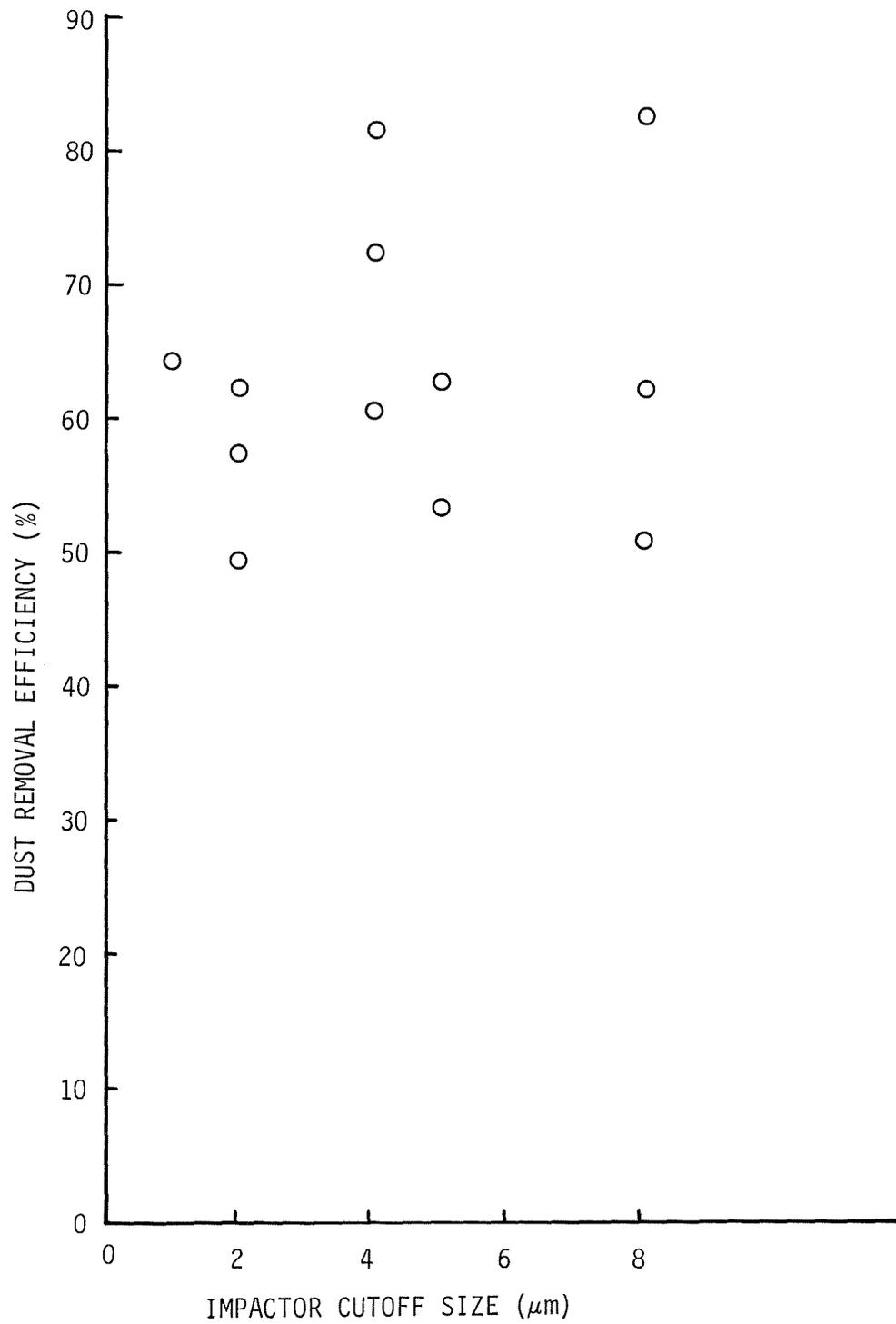Homestake Mine purchasing an additional 34 units by
mid 1984.

FIGURE 14. - Dust removal by the spray cooler.
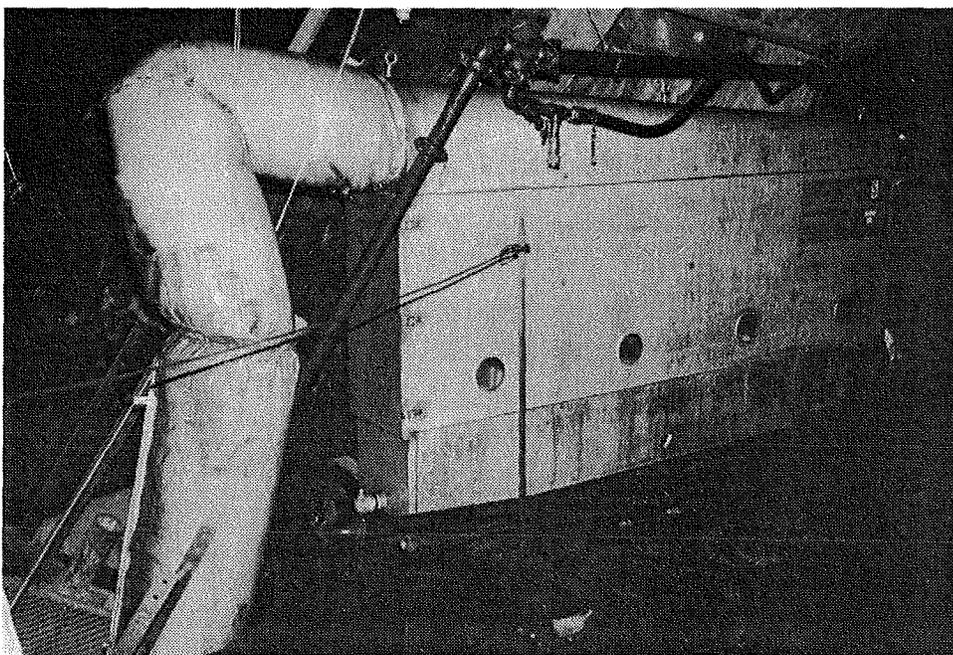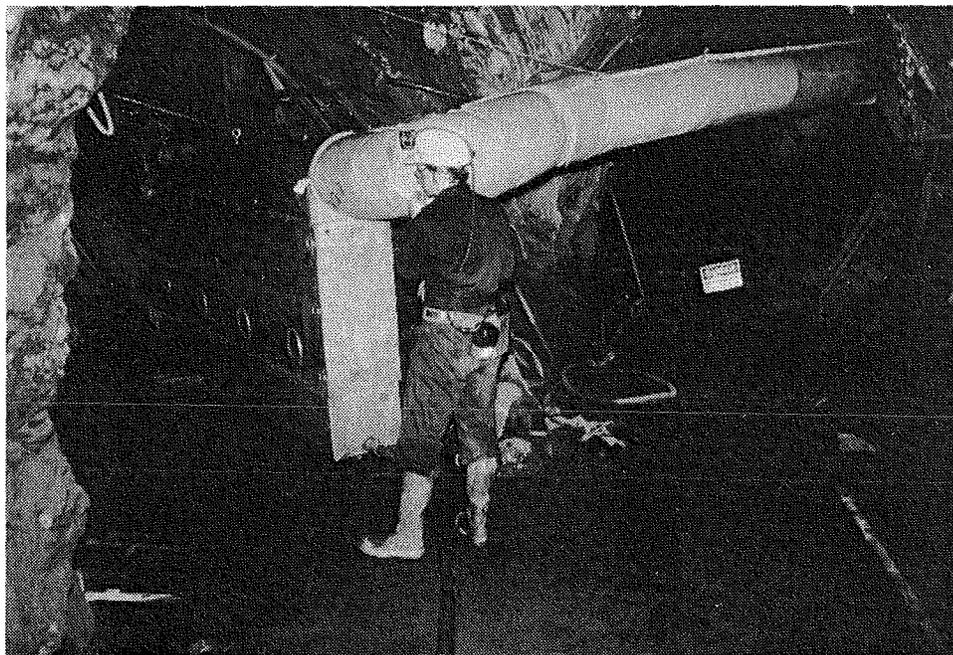
FIGURE 15. - Spray coolers - coping with
different situations.
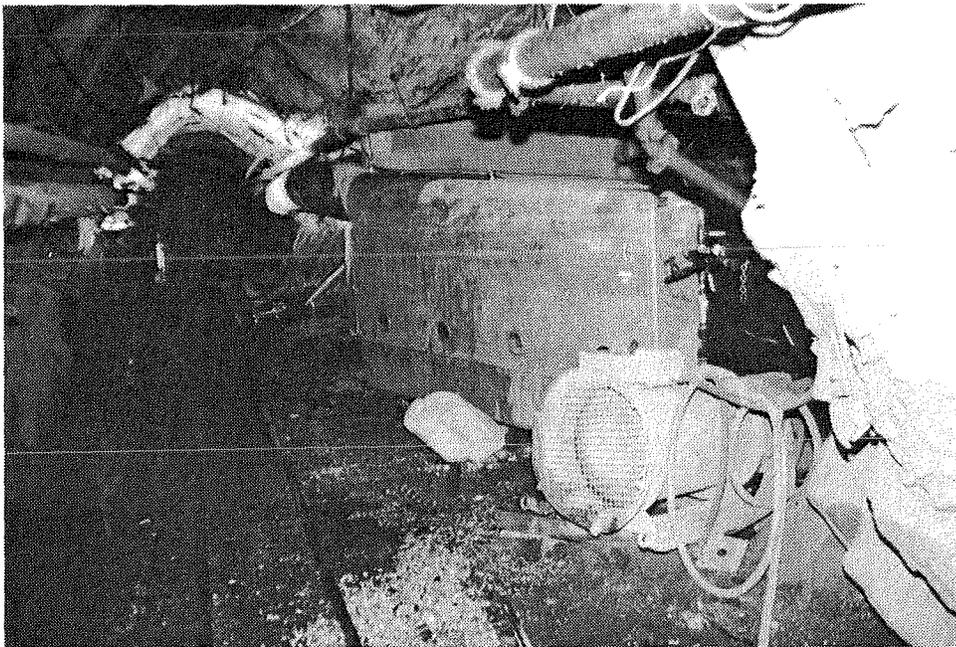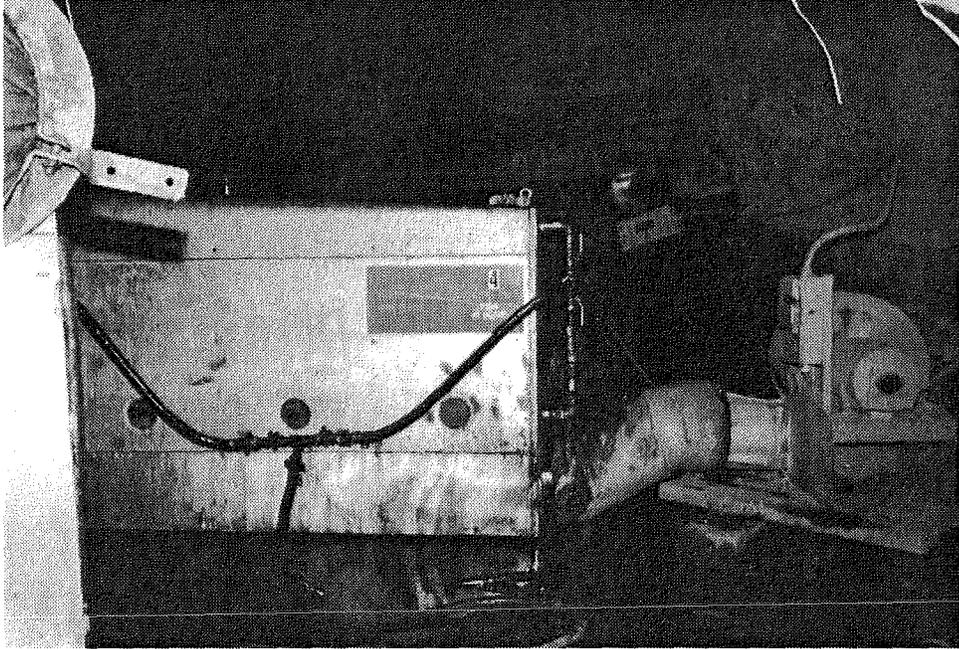
FIGURE 16. - Spray cooler with different
fan arrangements.
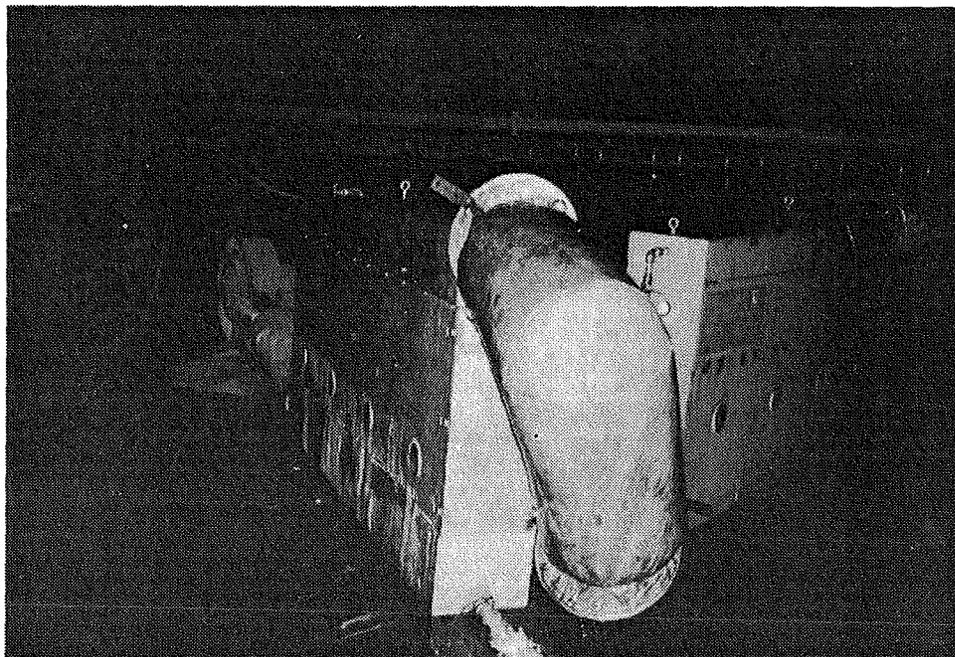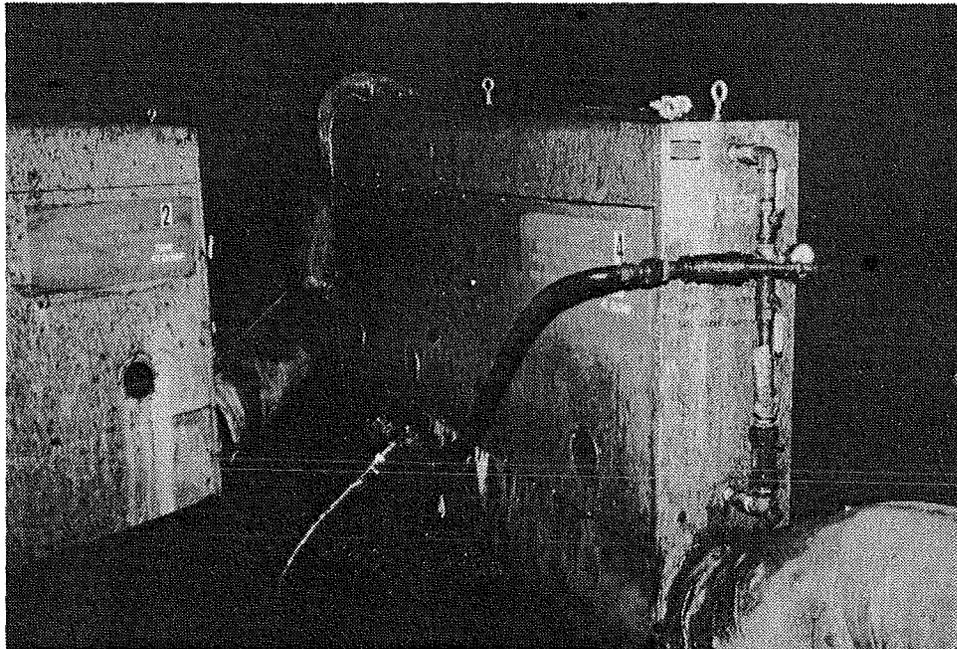
FIGURE 17. - Cooler installed in tandem to achieve
increased cooling power with lower outlet air
temperatures.
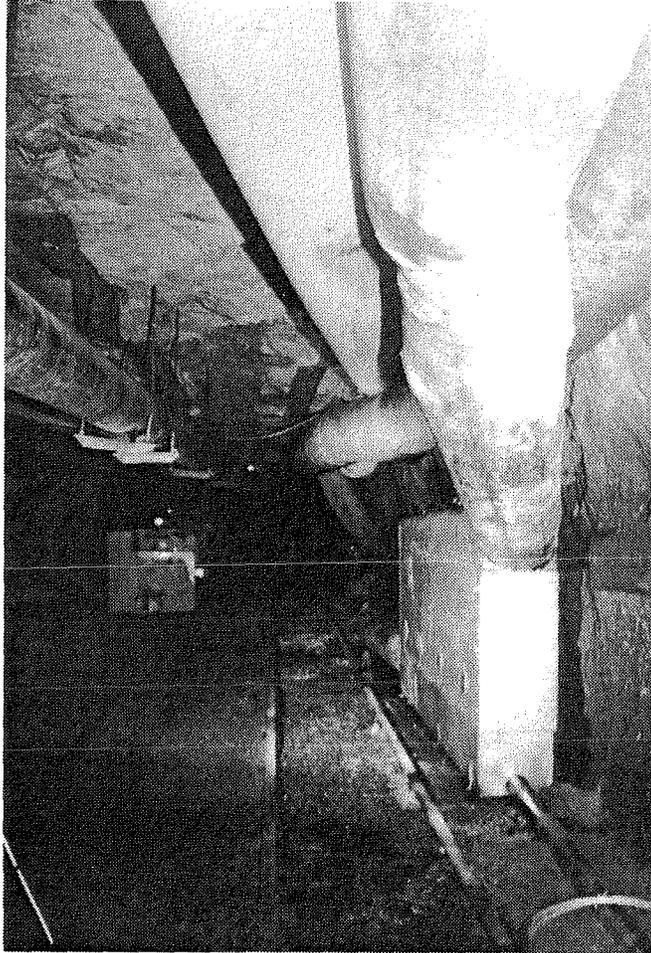
FIGURE 18. - Spray cooler - out of the way of traffic.

The first long term test was performed after the units
had accrued over 4 months of continuous, maintenance-free
operation. Figure 19 graphically shows the effects of
underground use. The laboratory tested F=0.626 is
presented for reference. A few of the coolers still
performed at this value, but the average F-factor had
decreased to 0.593 with the scatter of results showing a
standard deviation of 0.025. F=0.593 is very good
performance - much better than required by the original
design specifications.
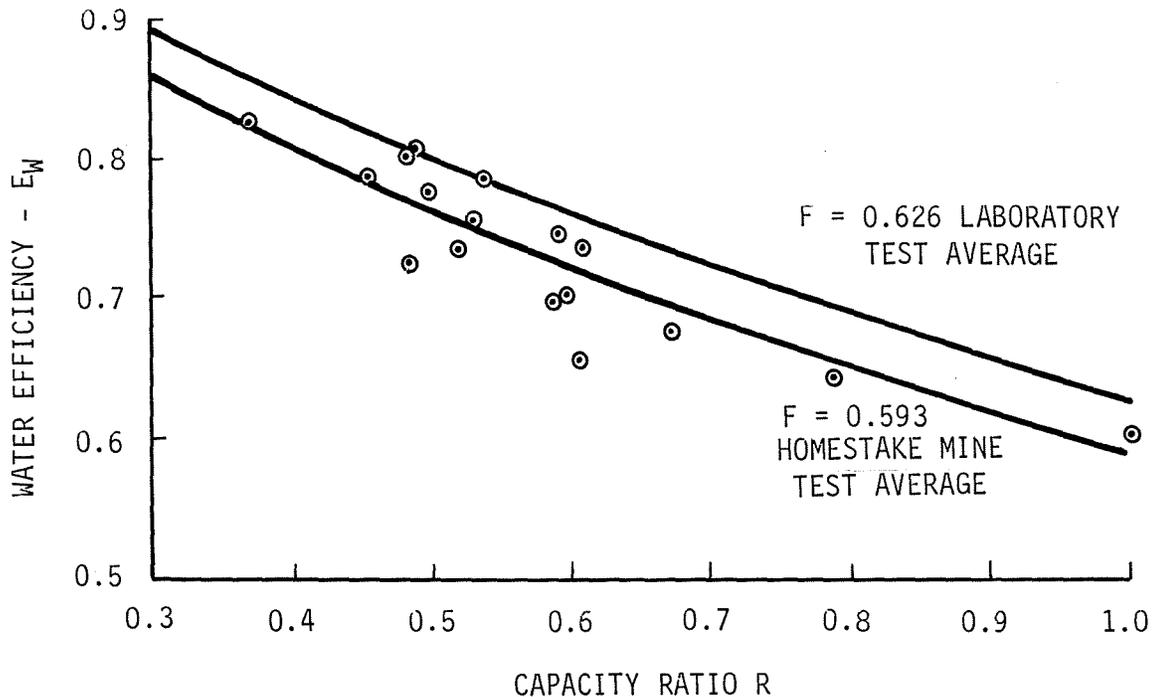
FIGURE 19. - Effect of in-mine use on F-factor.

Resistance to the wear and tear of mine use is excellent. Factors which contributed to the slightly degraded performance include:

a. Water pressure dropped due to nozzle wear, thus creating less uniform mixing of the air and water

b. Lime deposits on the bottom mesh, due to the water hardness, produced areas of uneven airflow

c. During some in-mine installations, flow diverters had been jostled out of place, creating uneven airflow

d. Airflows were very low, sometimes less than half the laboratory tested conditions, due to kinked inlet tubing, contributing to uneven air distribution.

Five months later, when the cooler had accumulated 9 or more months of continuous operation, the coolers were retested. During this period, no maintenance or service was performed, although most of the coolers had been moved to new locations. In some cases, damage was incurred during the moving operation. The procedure employed in all

Sketches of the two installations at the Sunshine Mine are shown in figures 20 and 21. Initial test results are presented in table 4. Coolers performed as expected. The cooling power is limited by the relatively high inlet water temperature. With 50° F inlet water and 5,000 cfm air, these coolers would supply in excess of 12 R-ton of the existing inlet air.

Long term experience has not been gathered on these coolers. However, the experience of the Homestake Gold Mine is assumed to be typical for these units.

4.4 MESH COOLERS OF OTHER SIZES

Performance of this spray cooler concept can be scaled. Provided the spray nozzles just fully wet the demister pad, performance is expected to be at or slightly above F=0.6. Evidence of this is a cooler developed at the Chamber of Mines of South Africa (COM) which uses the spray cooler concept developed in this program. The COM cooler was built for a nominal 8,500 cfm. The prototype unit, shown in figure 22, has 22-in duct connections. Thermal performance tests produced F=0.622. The COM work has been extended to large coolers. Commercialization in South Africa has also developed as noted by advertising in mine journals. The number of units currently in operation is unknown to the authors.
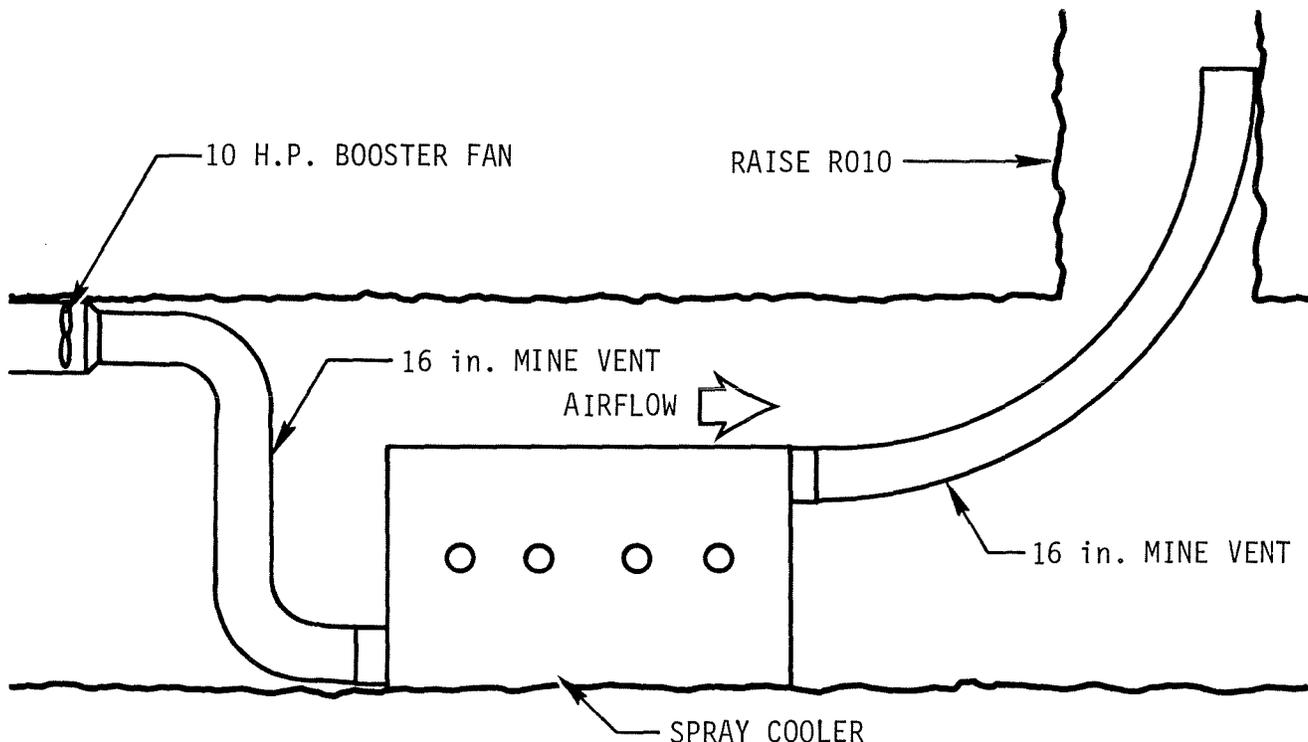
FIGURE 20. – Sunshine Mine, 3,700-level cooler location.
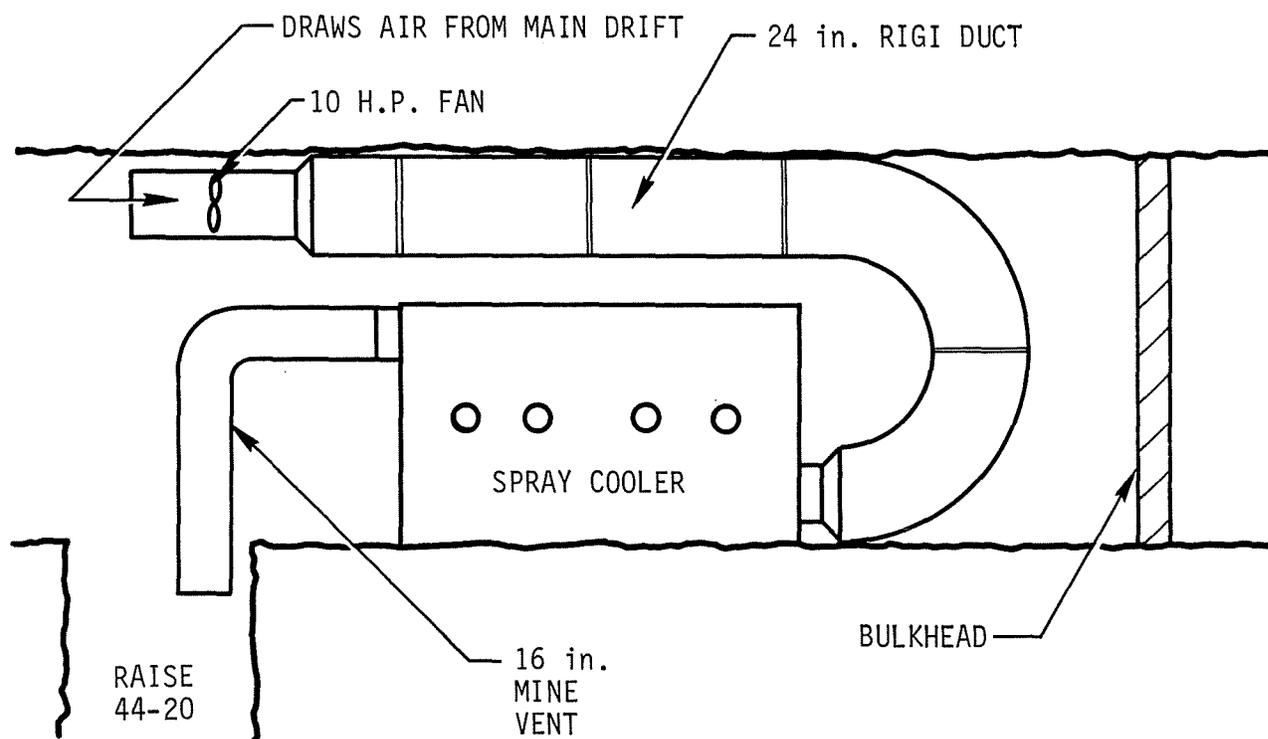
FIGURE 21. – Sunshine Mine, 4,200-level cooler location.

A single, smaller cooler was built for the Bureau on this program. This is a clear plastic, table-top, scale model to demonstrate the operating principles to the mining community. Figure 23 is a photograph of this model. This cooler operates and functions like the 5000 cfm units at 1/4 geometric scale.

TABLE 4. - Sunshine Mine spray coolers performance test data

Test No.	Cooler Level	Water-flow, gpm/ pressure, psi	Factor of merit	T_{wi}, °F	T_{wo}, °F	Dry Bulb in °F	T_{ai} °F	T_{ao} °F	E_w	R	R-ton	Air-flow,[1] cfm
1	3700	10.3/55	0.630	71.1	79.5	91.4	81.0	77.9	0.844	0.410	3.5	3187
2	3700	10.3/55	0.612	70.9	79.4	91.7	81.4	77.9	0.804	0.458	3.5	2847
3	3700	10.3/55	0.604	70.0	78.6	91.4	80.8	77.2	0.791	0.466	3.6	2839
4[2]	4200	12.8/100	0.733	86.5	88.3	89.2	89.2	87.2	0.627	1.33	0.8	947[2]
5	4200	12.8/100	0.606	71.6	81.6	95.3	88.0	78.9	0.603	1.01	5.2	1490
6	4200	12.8/100	0.615	71.1	81.5	95.7	87.6	78.5	0.624	0.971	5.4	1566

Mean "F" number: 0.613
Exclude test 4

[1]All airflows adjusted to obtain a heat balance.
[2]Water temperatures on test 4 were changing too quickly to get an accurate test. Results are shown to give an indication of initial tempertures. Results are not included in the average.
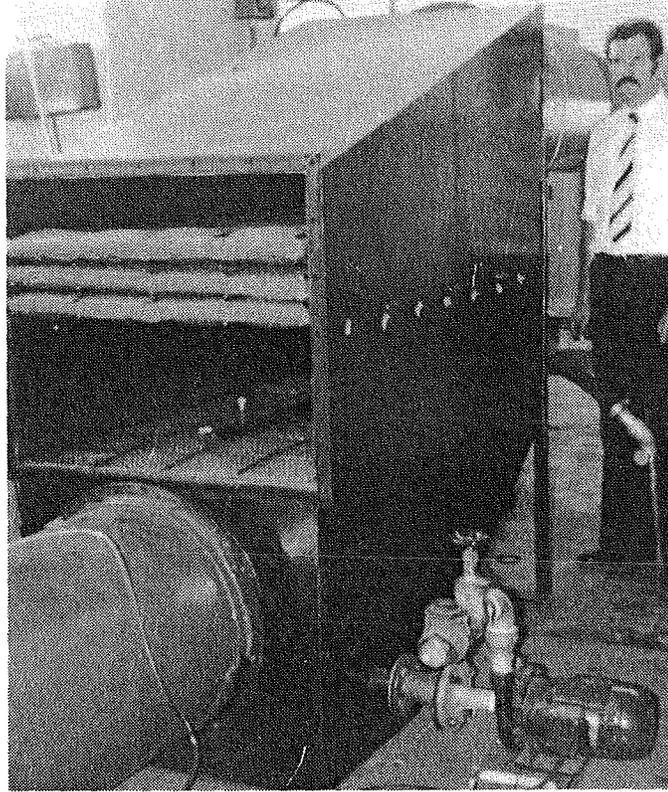
FIGURE 22. – Chamber of Mines of South Africa prototype 8,500 cfm cooler.
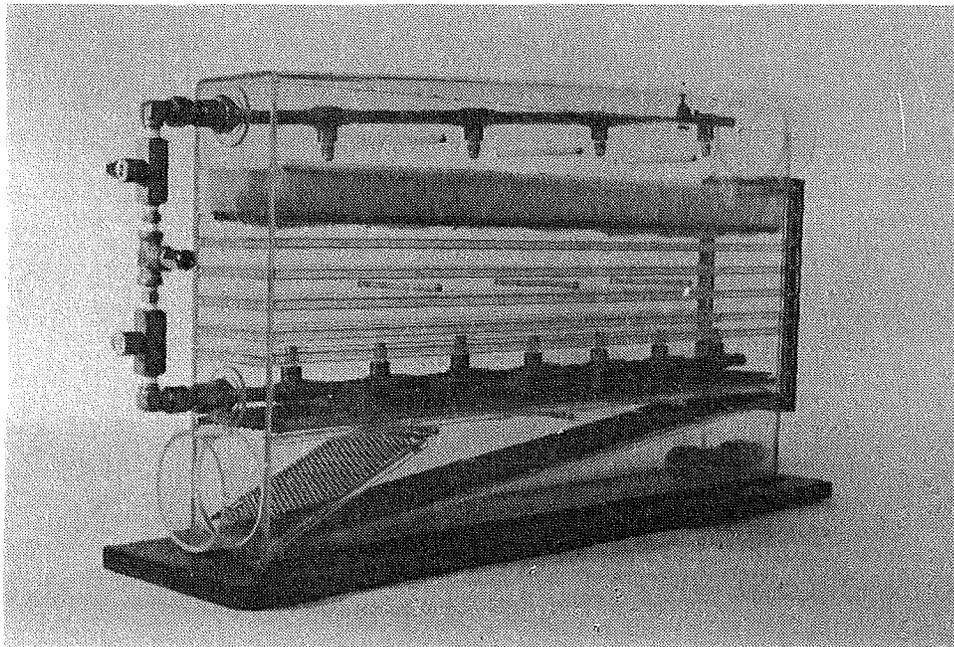
FIGURE 23. – Table-top demonstration cooler.

5. LARGE NOVEL COOLERS

5.1 OVERVIEW

Following satisfactory development of the small spray coolers, this program was expanded to develop, build, and test four coolers with nominal airflows of 25,000 cfm. The Bureau had entered into a cooperative agreement with the Gulf Mineral Resources Co. to test these coolers at their Mt. Taylor Mine. Engineers at the Mt. Taylor Mine had developed a series of physical and performance specifications which are presented in table 5. The physical specifications resulted from the practice of mounting the coolers to the roof of 16-ft high haulageways, which required free clearance underneath for vehicular traffic. Also frequent relocation of coolers was expected due to rapid mine development. Air pressure loss across the cooler was specified at a low value to minimize fan power requirements.

TABLE 5. - Spray cooler specification for the
Mt. Taylor Mine

Size:	4- high by 6- wide by 11.25-ft long (spray chamber with mist eliminator 10 ft), 31.5-in inside diameter flanges each end
Design performance:	100 R-ton 22,500 cfm at 4200 ft above sea level $t_{ai} = 90^\circ$ F (saturated) $t_{ao} = 77^\circ$ F (saturated) $t_{wi} = 55^\circ$ F, 130 gpm at 25 to 30 psi $t_{wo} = 73.5^\circ$ F Air pressure loss $\leq$ 1.5-in water gauge
Other:	Sturdy galvanized or stainless steel case with skids, Spray orifice diameter $\geq$ 5/32-in

These specifications, primarily those related to size, demanded that different approaches be considered to achieve the desired water-air heat exchange. The meshtype spray cooler concept described in the previous section was rejected for two reasons:

a.   The 4-ft vertical height limit did not allow
     sufficient space for turning the airflow from
     horizontal to vertical to horizontal with room
     left for the required vertical spray chamber.

b.   The allowable air pressure drop within the
     required length would not permit diverting the
     airflow from a generally horizontal direction.
     Indeed, meeting the air pressure loss requirement
     with the very minimal allowances for transition
     length is a major challange.

Development of a cooler of this size was also con-
strained by the absence of a test facility with adequate
capacity. The cost of a test facility for units of this
size was prohibitive. Therefore, all thermal performance
measurements had to be done during in-mine testing. Since
feedback on performance was not available during develop-
ment, it was decided jointly by the Bureau, Gulf Minerals,
and FMI that several novel concepts be developed into
prototype hardware and comparatively evaluated through an
in-mine testing program.

The thermal specifications of table 5 do not reflect a
heat balance. However, the 100 R-ton at the specified
operating conditions was to be taken as the primary
specification. Flush with success from developing the
mesh-type smaller cooler, FMI felt that 100 R-ton could be
met with waterflows of only 100 gpm instead of the allowed
130 gpm. Better thermal performance (higher F-factor) was
counted on to reduce the required waterflow. Specifi-
cally, an F-factor of 0.6 was the informal target.
Descriptions of the several novel coolers are presented in
the next section.

Economic hard times within the mining industry
impacted this program. Only one of the four coolers
designed and built was installed and tested in the
Mt. Taylor Mine. The mine was closed for economic reasons
before the three remaining coolers could be installed.
The Bureau TPO approached several hot mines seeking an
interested cooperating mine. The Homestake Gold Mine
viewed these larger coolers as an opportunity to explore
alternate approaches to their mine cooling program. They
were the only mine to positively respond to the offer.
(Economic conditions diminished the interest of other hot
mines.) However, considerable mine development was
necessary before each cooler could be installed at Home-
stake. This slowed the program progress to the point
where only two of the remaining coolers have been tested
at program end. The fourth and remaining cooler is
scheduled to be installed early in 1985. Mine engineers
will then test this cooler and provide the results to the

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Bureau. The following sections of this report cover the testing of only three of the four large novel coolers.

5.2 COOLER DESCRIPTIONS

During this program, four large spray coolers were designed and built. The designs included coolers with one, two, and three stages of cooling. Two variations of the three-stage coolers were built. All four cooler designs are described in a following section.

Because of the severe air pressure drop and size constraints discussed in the previous section, the general approach was to vary the spray configurations within the allowed chamber envelope. Air enters the cooler through a 31.5-in diam flanged inlet with a nominal velocity of about 69 ft/s. Inside the cooler, the flow area expands to an approximately 4- by 6-ft rectangular cross section. The air velocity drops to about 16 ft/s at the locations where it is exposed to one, two, or three stages of water sprays. Near the end of the 4- by 6-ft section, mist is removed with a wave plate eliminator. The air is then accelerated to about 69 ft/s as it exits through the 31.5-in diam outlet flange.

5.2.1 Single-Stage Cooler Description

The single-stage cooler is depicted in figure 24. In this skid-mounted unit, an inlet transition and diffuser plate aids in spreading the inlet air jet to achieve more uniform flow. The air then passes through a stack of corrugated plates. These plates are constructed from corrugated fiberglass roofing material and are stacked as shown in figure 25 to produce channels. Air leaving these channels passes through the cooling water spray pattern shown in figure 24. Mist is then removed from the cooled air in the wave plate eliminator before exiting through the transition section.

Cooling water is sprayed at a rate of 100 gpm from a manifold at 30 psi through 40, 80 degree angle, solid-cone spray nozzles. The spray pattern produces a uniform water flux across the downwind end of the corrugated plates. Water entering the channels runs along the bottoms of the channels, draining out of the upwind end of the cooler. This is achieved by installing the cooler in a manner that pitches it slightly downward at the upwind end. The water spray direction and channel flow direction are opposite to the airflow. This is intended to produce a counter-current cooling effect. Water is drained out from the low end onto the floor.
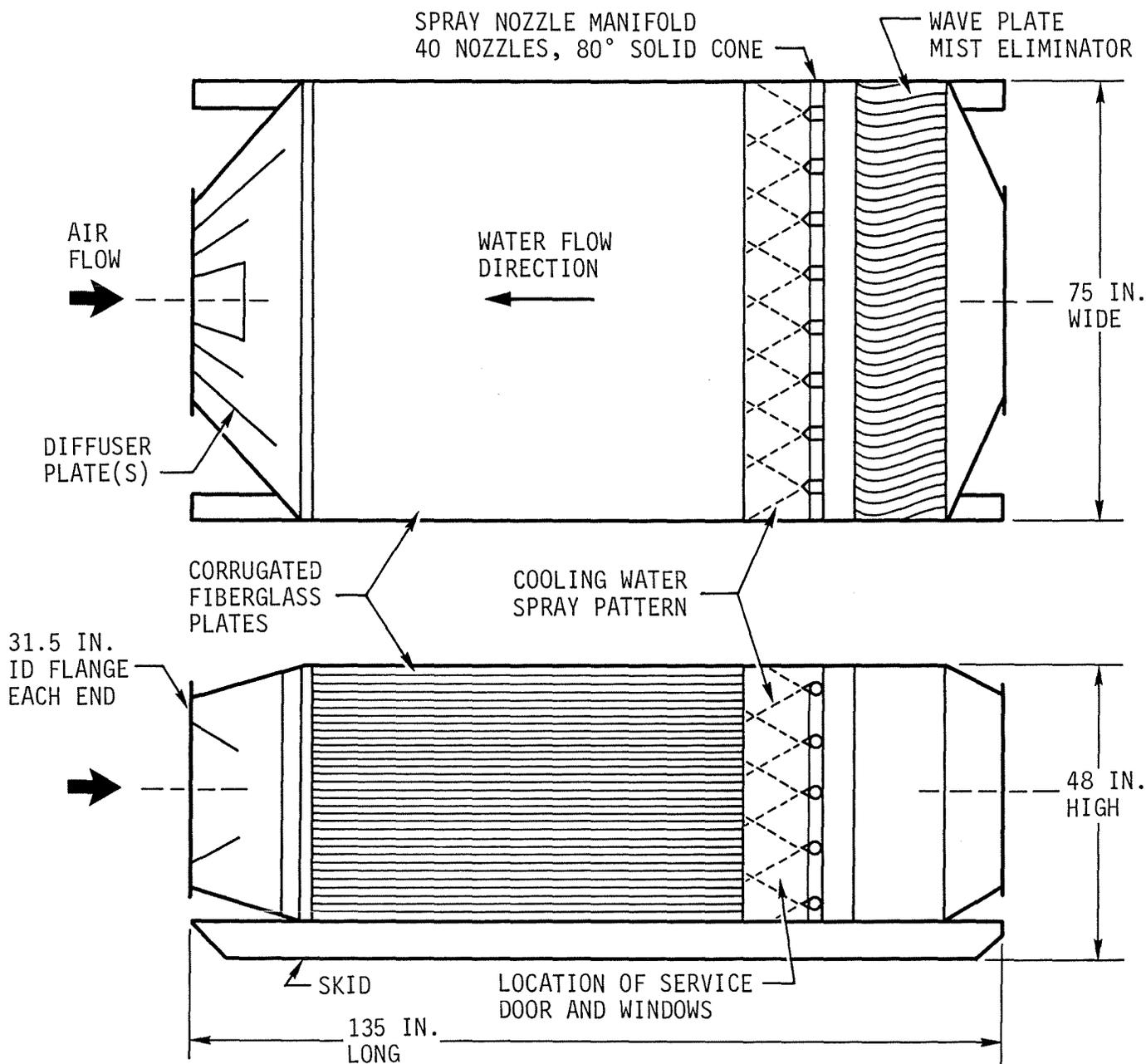
FIGURE 24. – Single-stage cooler – tested at Mt. Taylor.
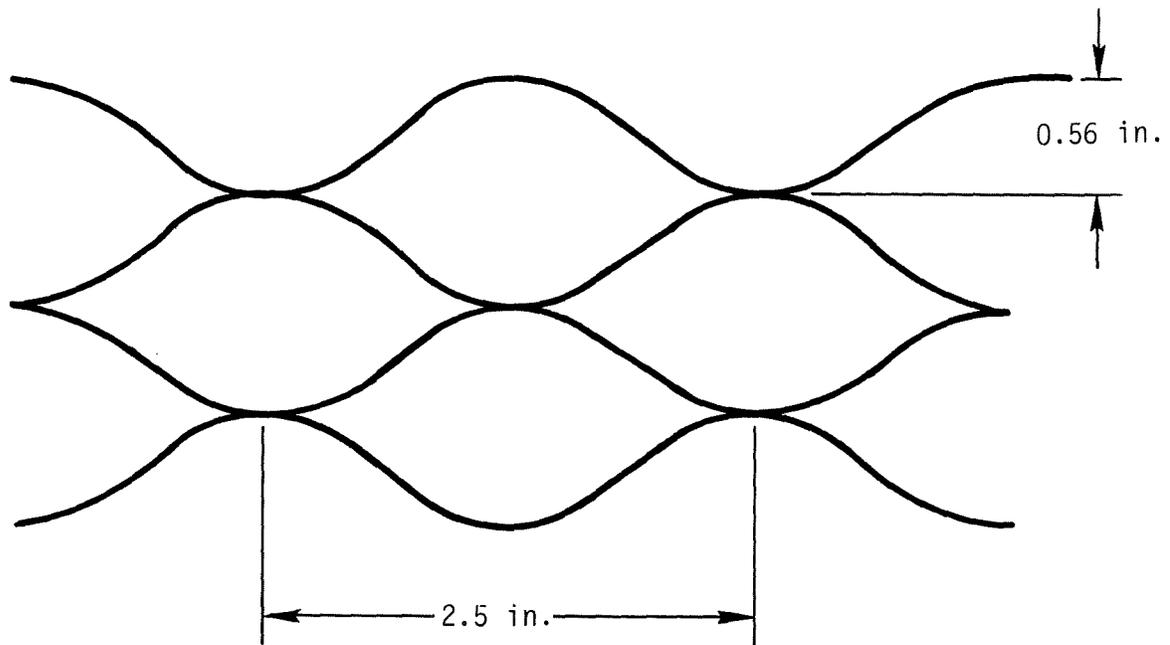
FIGURE 25. - Typical section of corrugated fiberglass plates.

This cooler was tested in the Mt. Taylor mine. The installation and testing results are discussed in subsection 5.3.

5.2.2 Two-Stage Cooler Description

The Mt. Taylor Mine had several nominal 300 R-ton, bulk, single-stage spray coolers installed which performed to the satisfaction of the mine engineers. These units had been built by the Pneumafil Corporation of Charlotte, NC. The Pneumafil Corporation, upon learning of this program, volunteered to adapt their concept to a two-stage design that met the required specifications of this program. Their offer was accepted. Design of the cooler is depicted in figure 26.

This cooler had an overall length of 151.5 in, somewhat longer than specification. However, the downwind transition section could be removed during in-mine transportation. The increased cooler length permitted the use of a somewhat longer inlet transition that required no diffuser plates to achieve uniform flow.

Hot air entering the cooler encountered the second water stage first. This was to promote the desirable
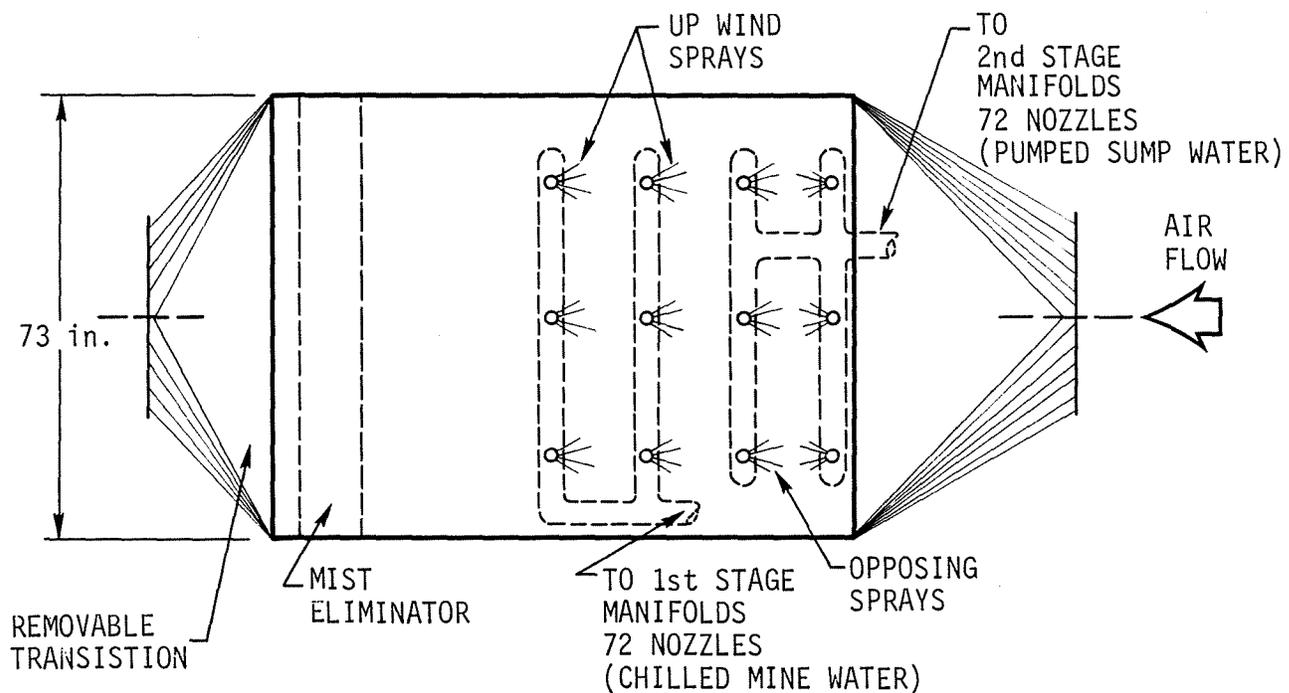
(a) Top View
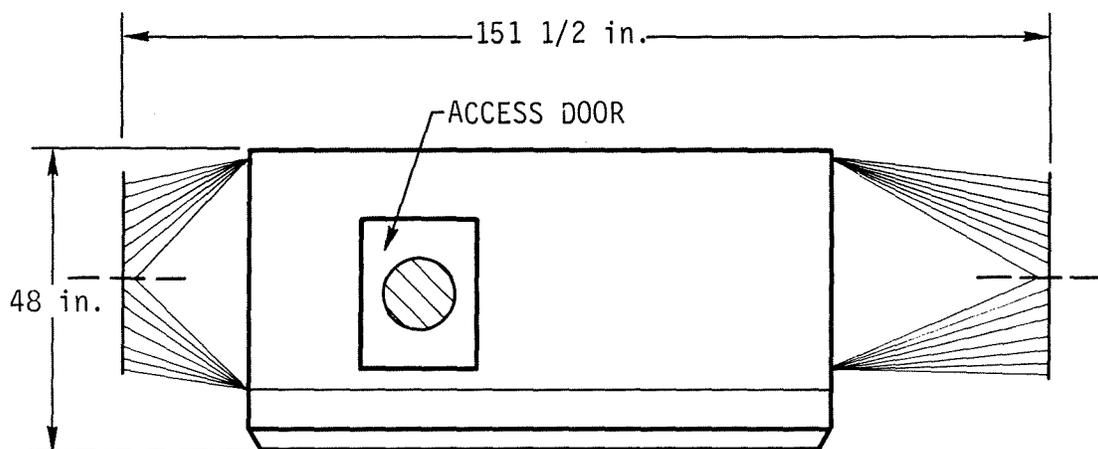
(b) Side View

FIGURE 26. – Two stage Pneumafil Corporation cooler
tested at the Homestake Gold Mine.

counterflow effect. The second water stage consisted of a pair of spray nozzle manifolds holding 36 nozzles each. These manifolds sprayed in direct opposition to each other as shown in Figure 26. Design flow rate for the 72 nozzles was 100 gpm at 20 psi. Each spray nozzle produced a 72 degree angle, hollow-cone spray. Water was supplied to the second stage by a remotely located, 3 hp pump drawing water from the sump of the first water stage.

Air leaving the second water stage passed through the spray pattern of the first water stage. The first water stage was similar to the second water stage except all 72 sprays were directed against the airflow. Cold service water was supplied at 100 gpm to the first stage at 20 psi pressure.

Air was demisted by a wave plate eliminator before exiting the cooler. The water removed from the exiting airstream drains to a sump which collects water from the first stage (which is supplied to the second water stage as described previously). Water from the second water stage falls to a second water stage sump where it is drained out onto the floor.

Testing of this cooler was performed at the Homestake Gold Mine. Some modifications were made to adapt to the Homestake requirements. These modifications and the results of the testing are described in subsection 5.3.

5.2.3 Three-Stage Cooler Descriptions

Both three-stage cooler designs can be understood by reference to figure 27. These units incorporated dual 7-1/2-hp pumps on skids that are separable from the skids of the cooler for in-mine transport. The air inlet transition is identical in design to that used on the single-stage cooler. All three water spray manifolds are identical. A uniform array of 40, 80 degree angle, full-cone spray nozzles are installed on each manifold. The 40 nozzles deliver 100 gpm at 30 psi.

In operation, hot air passes through the third and second water stages in that order. In one design, a wave plate mist eliminator demists the air before it passes through the first water stage. The other design does not have this demist stage. The intent here was to evaluate any effect on thermal performance of the mist eliminator. Some effect was presumed likely. Downwind of the first water stage, the air is demisted and exits the cooler.

The three-stage cooler with two mist eliminators has been modified and tested at the Homestake Gold Mine. These tests are described in the following section. The

INLET DIFFUSION PLATES

WAVE PLATE ELIMINATOR (ONE DESIGN ONLY)

WAVE PLATE ELIMINATOR

75 in.

REMOVABLE PUMP SKIDS (2 UNITS)

3rd STAGE MANIFOLD, 40 NOZZLES

2nd STAGE MANIFOLD, 40 NOZZLES

1st STAGE MANIFOLD, 40 NOZZLES

48 in.

ACCESS DOOR (2)

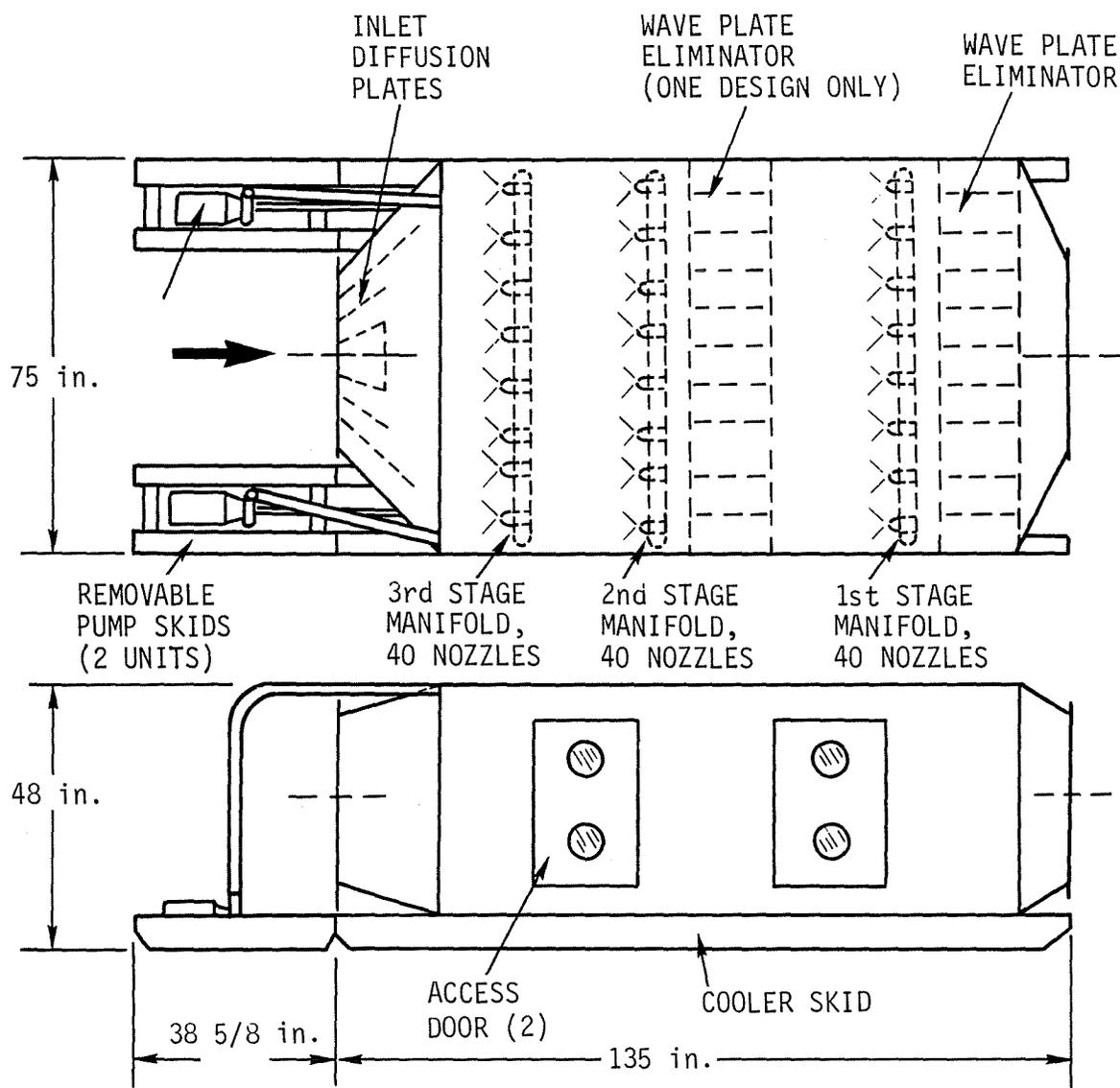COOLER SKID

38 5/8 in.

135 in.

FIGURE 27. - Three stage coolers.

three-stage cooler with only one mist eliminator had not been installed by the end of this program.

Assembly drawings of all coolers have been sent to the TPO during this program.

5.3 TESTING AND RESULTS

5.3.1 Single-Stage Cooler Tests

The single-stage cooler was installed and tested in the Mt. Taylor Mine during June 1982. Installation of the cooler presented no particular mechanical problems, but 105° F wet bulb temperatures in the drift required crew rotation. The cooler was hung by two chains from the roof as shown in figures 28 and 29. Figure 30 depicts the test site at the mine. The cooler was pitched with the air inlet (water outlet) end 6 in lower than the air outlet end for proper water drainage through the channels in the corrugated plates.

Table 6 presents the measured performance data. The 100 R-ton cooling specification was met easily due to somewhat unexpected lower inlet water temperatures and higher inlet air temperature. An average F-factor of 0.525 was achieved, which is considered excellent for a single-stage spray cooler with only horizontal airflow. This was achieved in spite of evidence of temperature stratification which would result from nonuniform air-water contact throughout the cooler. Nonuniform air-water contact could be expected due to the limited cooler length, which prevented the installation of an inlet transition to produce a flat velocity profile within the cooler. At the outlet of the mist eliminator, wet bulb temperatures ranged from 7-1/2 to 12-1/2° F higher near the corners than near the center. Elimination of this stratification would improve cooler performance.

Air pressure drop across the cooler was only slightly higher than specification. A loss of 1.6-in water gauge was measured compared to the 1.5-in value specified. The pressure loss across the channels formed by the corrugated plates contributed only about 0.2 in of water gauge. The loss of air pressure against the oppositely directed water sprays was 0.6 in. Loss of pressure through the mist eliminator should be only about 0.16 in based on manufacturer's data. Loss through the combined mist eliminator and outlet transition was measured at 0.9 in, indicating about a 0.7-in loss in the outlet transition. Loss in the inlet transition was much less than 0.1 in.
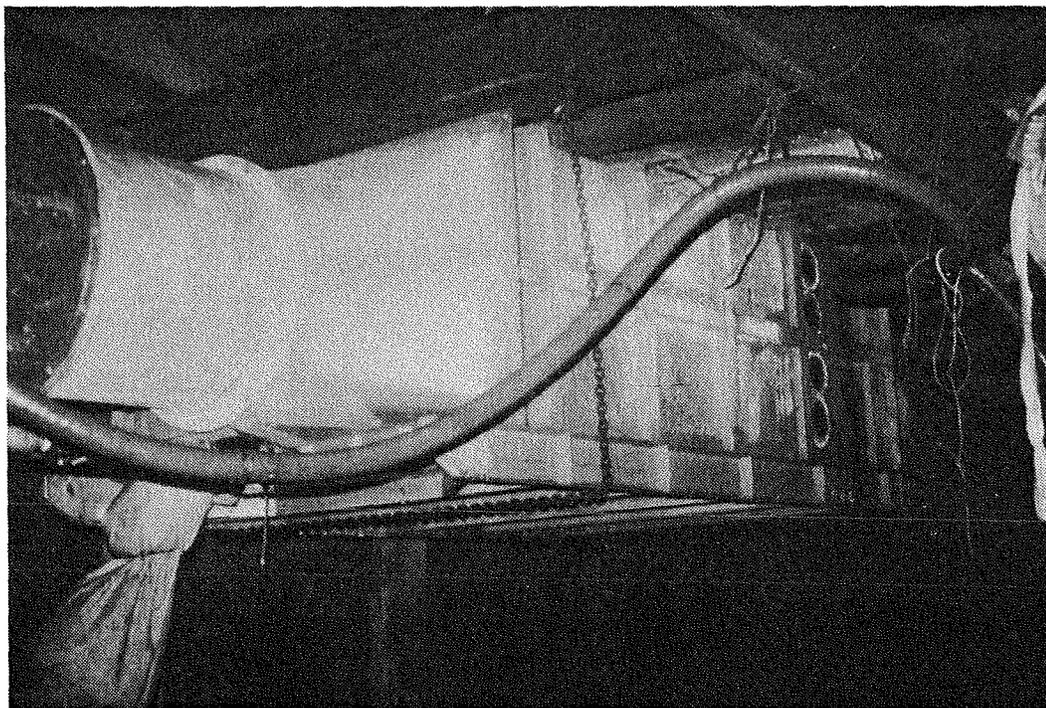
FIGURE 28. - Upwind view of single-stage cooler.
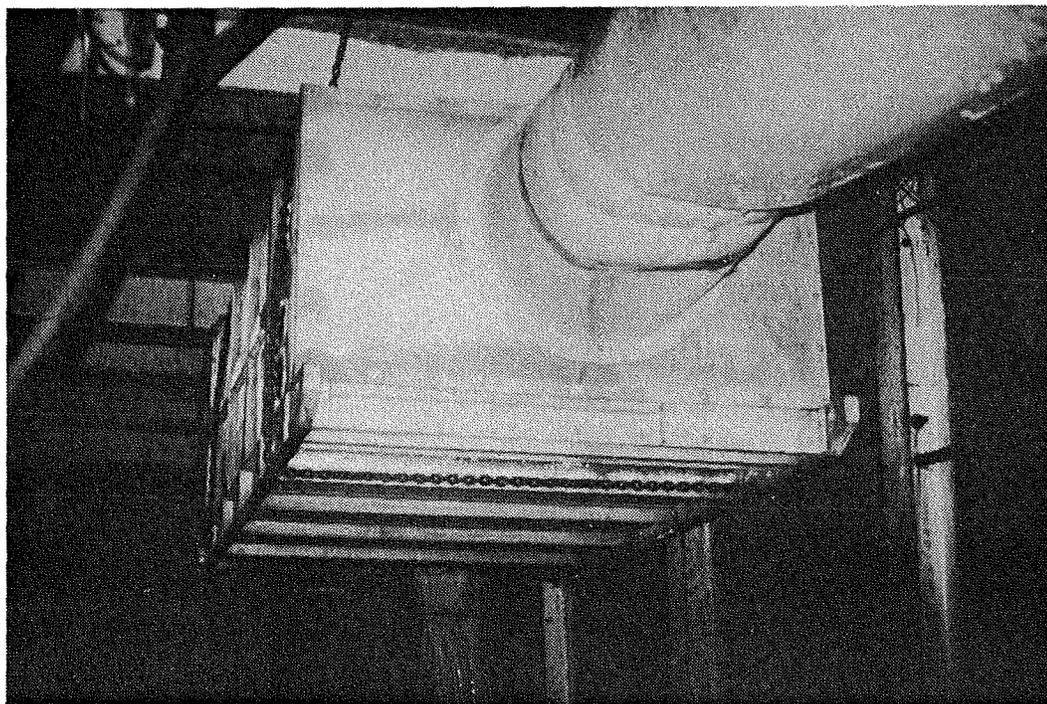
FIGURE 29. - Downwind view of single-stage cooler.
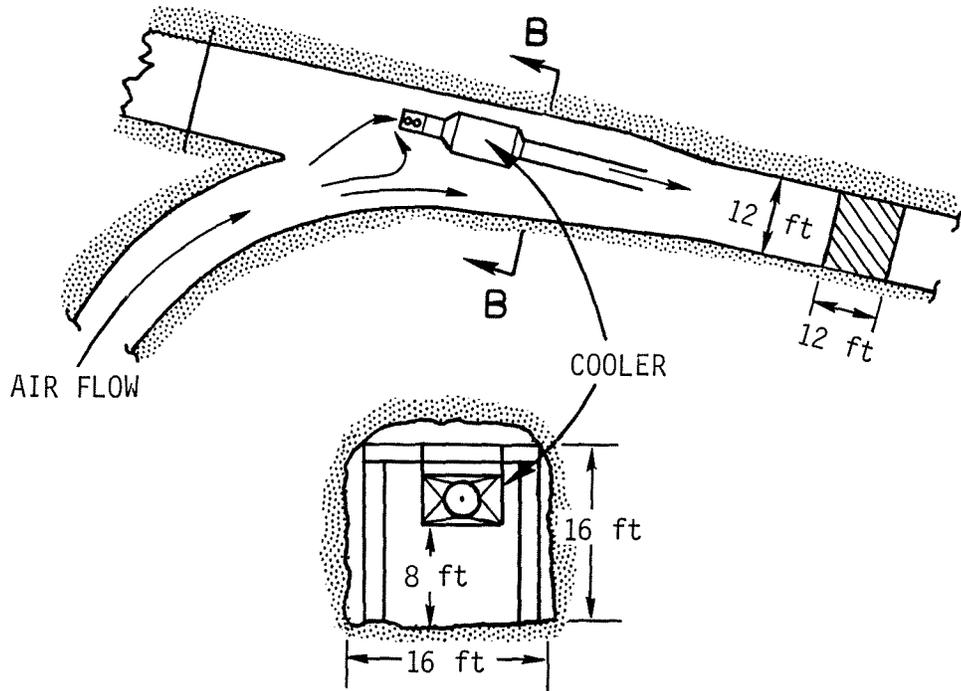
SECTION B-B

FIGURE 30. - Single-stage cooler at the Mt. Taylor Mine test site.

Mine engineers were pleased with the performance and operation of the cooler. The cooler remained in constant operation without maintenance for several months until the mine was closed.

5.3.2 Two-Stage Cooler Tests

As explained previously, all coolers could not be installed in the Mt. Taylor Mine. The two-stage cooler designed and built by the Pneumafil Corporation was installed in the Homestake Gold Mine, Lead, SD. Due to differences between mines, some modifications were warranted. At the Homestake Mine, it was not necessary to duct the cooled air; therefore, the outlet transition section was removed to eliminate air pressure drop. Also, as a larger diameter fan was to be used, the mine fabricated a new inlet transition. Significantly, the Homestake Mine did not need 100 gpm of chilled water for each cooler. For this cooler, approximately 55 gpm at 10 psi of chilled water was supplied to the first water stage. Water to the second water stage was pumped out of the sump at 72 gpm. This sump water was a mixture of water exhausted from the first water stage and a recycling of water which had been already sprayed previously from the second stage. Excess water in the sump was drained from the sump onto the floor at the 55 gpm inlet rate. This

TABLE 6. - Single-stage cooler performance data at the Mt. Taylor Mine

Test no.	Nozzle pressure, psi	Water flow, gpm	Factor of merit	T_{wi}, °F	T_{wo}, °F	Temperature dry bulb, °F	T_{aiwb}, °F	T_{aowb}, °F	E_w	R	R-ton	Air-flow[1], cfm
1	35	102	0.534	49.7	74.0	95.9	90.8	76.2	0.589	0.798	102	19,072
2	35	102	0.533	50.4	73.8	96.6	93.3	76.2	0.542	0.964	98	15,256
3	36	102	0.541	50.3	74.2	97.4	93.8	76.2	0.549	0.968	101	15,141
4	36	102	0.542	50.4	74.3	96.4	93.7	76.2	0.552	0.957	101	15,280
5	36	100	0.502	46.3	71.9	96.2	90.8	77.3	0.573	0.732	105	21,061
6	41	108	0.529	54.1	76.5	98.4	92.9	78.8	0.574	0.826	99	18,331
7	35	101	0.512	52.5	77.0	100.4	93.9	81.1	0.589	0.713	102	19,977
8	31	93	0.534	51.0	78.5	102.3	95.9	81.5	0.610	0.726	105	17,941
9	36	100	0.506	49.7	77.5	101.9	95.7	82.3	0.602	0.653	115	21,921
10	36	101	0.513	47.2	76.1	102.2	96.2	81.7	0.587	0.722	120	20,226

Mean "F" Number: 0.525
S Dev: 0.015
[1]All airflows adjusted to obtain a heat balance.

water is subsequently recovered and pumped through a con-
denser coil. The intent of using recycled water in the
second water stage was to promote a higher water-to-air
loading for better heat absorption from the hot entering air.

Table 7 presents the test results. Average F-factor
is about 0.5. Temperature stratification of about
6-1/2° F was observed in the outlet air and may explain
the scatter of results. This cooler was not operated at
the conditions or in the manner intended by the Pneumafil
Corporation; performance somewhat below the predicted
F = 0.6 can therefore be expected.

Only 4 months of operation have been accumulated on
this unit at the time of writing this report. Engineers
at the mine are pleased with the maintenance-free opera-
tion to date. They think a lower horsepower pump would be
more appropriate because of the reduced flow rate and
pressure required at the second stage.

5.3.3 Three-Stage Cooler Tests

Of the two three-stage coolers that were built, only
the unit with two mist eliminating sections has been
tested during this program. The remaining three-stage
cooler will be installed and tested by the Homestake Mine
at a later date.

The cooler that was tested was modified for the
Homestake Mine application. Operation was with the outlet
transition removed. In this configuration, air pressure
loss at about 26,000 cfm was measured at 1.8-in water
gauge with the water sprays on and 1.0-in water gauge with
the sprays off. The water supply to the stages was:

a. First stage - chilled mine water; 50 gpm at 30 psi

b. Second stage - mixed first and second stage sump
    water; 82 gpm at 35 psi

c. Third stage - mixed second and third stage sump
    water; 87 gpm at 40 psi.

Figures 31 and 32 are pictures of the cooler installed at
Homestake. A problem encountered with the diffuser plates
breaking loose in the inlet transition caused them to be
removed. This may be expected to degrade the performance
somewhat.

Table 8 shows the results of performance measure-
ments. The average F = 0.606 is very good. As can be
seen, each of the three stages is important to achieving
this performance.

Type of unit	Test no.	Stages tested	Water flow gpm	Factor of merit	T_{wi}, °F	T_{wo}, °F	Dry bulb inlet, °F	T_{ai}, °F	T_{ao}, °F	E_w	R	R-ton	Air-flow[1], cfm
	1	2	55	0.500	50.0	67.2	82.8	76.8	69.3	0.638	0.533	39	16,387
	2	2	55	0.489	50.0	66.9	82.4	76.6	69.5	0.631	0.515	38	16,991
Pneumafil 2-stage 7400 ft	3	1	55	0.433	50.1	65.6	81.8	76.9	70.9	0.574	0.482	35	18,073
	4	2	55	0.510	51.0	67.9	83.6	77.7	69.5	0.629	0.589	38	14,580

Average F = 0.500
Exclude Test No. 3
[1]All airflows adjusted to obtain heat balance.

TABLE 8. — Performance data of the three-stage cooler with two mist eliminators at the Homestake Gold Mine

Test no.	Stages tested	Water flow, gpm	Factor of merit	T_{wi}, °F	T_{wo}[1], °F	Dry bulb inlet, °F	T_{ai}, °F	T_{ao}, °F	E_w	R	R-ton	Air-flow[2] cfm
1	3	50	0.571	49.6	70.9	85.3	76.7	69.8	0.784	0.398	44	20,169[3]
2	3	50	0.612	50.4	72.2	86.4	76.1	69.6	0.847	0.363	45	22,128[3]
3	3	50	0.645	48.7	77.4	87.7	80.1	73.5	0.913	0.299	59	26,200
4	First and second stage only	50	0.572	48.6	75.6	88.8	80.4	74.5	0.840	0.287	56	27,300
5	First stage only	50	0.436	48.6	65.6	84.6	75.6	70.5	0.626	0.375	35	21,800[4]
6	3	50	0.596	48.7	71.5	84.0	75.5	69.6	0.850	0.319	47	25,600
7	3	50	0.606	48.4	71.8	84.0	75.8	69.6	0.859	0.326	49	25,050

Mean "F" Number: 0.606, Standard deviation α = 0.024
Excluding Tests No. 4 and 5
[1]0.6° F subtracted from two temperatures to account for 0.6° rise caused by pumps.
[2]All airflows adjusted to obtain heat balance.
[3]Tests 1 and 2 were done with outlet transition on, producing lower flows.
[4]Test 5 had a large energy imbalance due to large variation in inlet conditions during the test.

To date, this cooler has accumulated over 16 months of operation. Mine engineers are generally pleased with performance. Some unexplained "black gunk" has, however, clogged the third-stage spray nozzles causing the unit to operate as a two-stage cooler. FMI has retested the cooler after 12 months operation - with clogged third-stage nozzles and some windows missing due to damage from nearby blasting. An $F = 0.53$ was measured which is less than the $F = 0.57$ measured for two stages on installation (see test No. 4 of table 8). The lower value is attributed to the blasting damage. Mine personnel are working on a prevention or cure for the clogging problem.
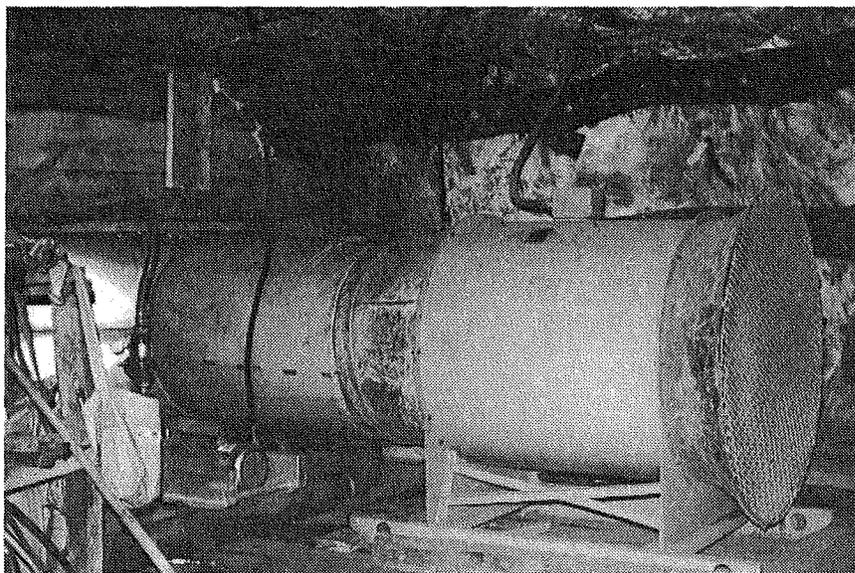

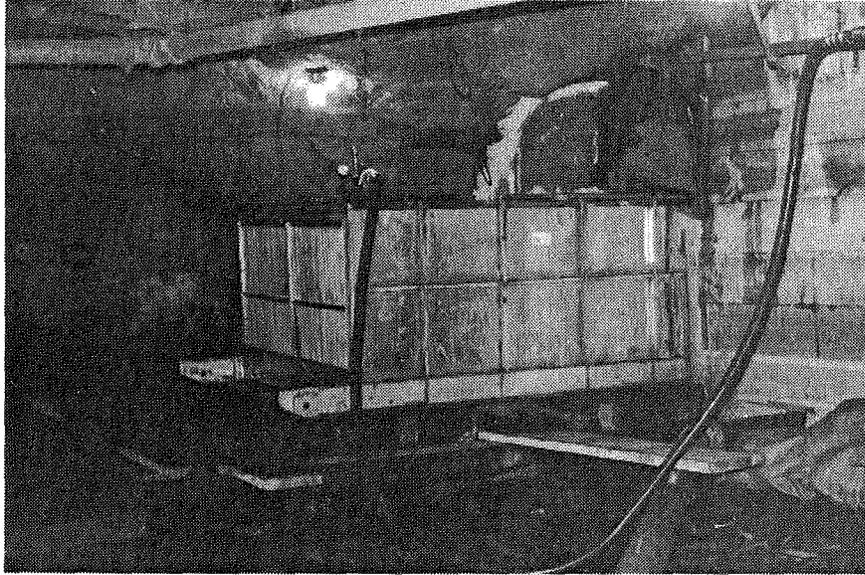FIGURE 31. - Inlet of three-stage cooler installed at the Homestake Mine.

FIGURE 32. - Outlet of three-stage cooler with transition removed.

6. CONCLUSIONS AND REMARKS

The success of the small, mesh type coolers (section 4) is verified by the 50 units now in service in the United States. Twelve of these units were produced as part of the fundamental research of the program; the remaining 38 were the result of commercial sales. The commercial units are incorporating evolutionary improvements as mine use indicates need. The concept upon which these units are based has also been adopted overseas and has been adapted to the local conditions. Acceptance of these coolers underground is due to a satisfactory combination of thermal performance and mechanical reliability. Both are essential. Good thermal performance ensures the efficient use of costly chilled water to cool workspace air. Reliability and low levels of required maintenance are critical features of underground equipment. Finned-tube coolers can provide thermal performance similar to that of the spray coolers, but the maintenance effort required to clean the accumulated dust off the fins has lead to their replacement by spray coolers.

For mines which use a closed circuit circulating loop of chilled water, the spray coolers do present practical problems. Since spray coolers release water to atmospheric pressure, this water must be recaptured and pumped back to the refrigeration plant for recooling, unless the plant is at a lower level and gravity can be used for returning water. In this program, using some of the power from the incoming water pressure and flow to drive a pump for returning water to the refrigeration plant was considered. However, it was found that low mechanical efficiency or very high cost made water-powered motors in the small size range unattractive. This leaves an auxiliary electric or compressed air motor-driven pump as the only choice.

While dust scrubbing is a virtue of spray coolers for health reasons, the dust in the drain water must be removed if the water is recycled through the refrigeration plant. Dust deposition in the heat exchanger tubes is likely to be unacceptable. In the opinion of the authors, however, open chilled water systems are likely to become more common as deep, hot mining becomes more extensive. This opinion is based on observing South African practice, where mining in hot conditions is far more widespread than here.

Experience with the large spray coolers (section 5) indicates the difficulty of achieving high performance (F 0.6) with only horizontal airflow. It took three stages of sprays with two electric pumps to reach this

level. In any spray cooler, good performance requires uniform distribution and thorough mixing of the air and water. This usually demands both a uniform air velocity profile and a uniform water loading throughout the contact region. None of the large coolers achieved good, flat air velocity profile. Temperature stratification existed, therefore, resulting in loss of performance. Restriction on cooler size is the major problem in trying to achieve flat air velocity profiles with minimal loss of air pressure. Size restriction is a fact of life in mining equipment and solutions must be found within that constraint.

The single-stage large cooler with corregated plates produced a very respectable F=0.525 even with evidence of large temperature stratification. Experience with the F=0.626 of the small mesh cooler led to an unwarranted condescending attitude toward an F=0.525 Factor of Merit. Upon relfection, F=0.525 should be considered very good for the first attempt at a single-stage, horizontal flow spray cooler. Bluhm (4) gives F=0.55 as the upper limit of expected (not theoretical) performance for a single-stage spray cooler. Considerably better performance is believed possible with development of this concept. This would make a good beginning for any further research on mine spray coolers. Engineers at the Mt. Taylor Mine were satisfied with the existing level of performance and pleased with the low level of maintenance indicated during the six plus months of operation.

A concluding note on the "Factor of Merit" is in order. At the beginning of this program, almost no one in North America was familiar with this method of rating direct water-to-air heat exchanges. Five years later, many engineers responsible for mine cooling are either "comfortable" with the concept or rely on it exclusively for rating spray coolers. As a minimum, F-factors provided considerable efficiency to this program and indeed it may have been necessary for successful cooler development. In a single number, the performance of a cooler can be summed up without regard for particular temperature and flow rates. Required laboratory testing is dramatically reduced and can be quickly correlated to mine test results and vice versa. F-factors also facilitate comparisons between different coolers in different mines. As a unitless number, there is no confusion in international work where different conventions of units are employed; one number suits all. One quantitative gauge of this program is that Thimons, et al. (1) had achieved an F-factor of almost 0.5 as the beginning baseline. Now an F-factor of about 0.63 is the new baseline to measure further development.

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