The design of spray chambers for bulk cooling of the air in mines

by S. J. BLUHM *, B.Sc. (Eng.) (Visitor) and A. WHILLIER *, Pr. Eng. Sc. D. (M.I.T.) (Fellow)

SYNOPSIS

Recommendations are given for the design of multi-stage spray chambers in which air flows horizontally through water sprays. A detailed description is given of one such spray chamber at Hartebeestfontein gold mine that is used for bulk cooling of the incoming ventilation air, and that has been constructed according to these recommendations.

A method for predicting the performance parameters of horizontal multi-stage spray chambers is given, and the method is illustrated with some examples. The method is equally applicable to spray chambers in which the water, rather than the air, is cooled.

SAMEVATTING

Aanbevelings word gedoen in verband met die ontwerp van veelvoudige sproeikamers waarin lug horisontaal deur watersproeiers vloei. Daar word 'n uitvoerige beskrywing gegee van een sodanige sproeikamer by die Hartebeestfontein-goudmyn wat vir die massaverkoeling van die inkomende ventilasielug gebruik word en ooreenkomstig hierdie aanbevelings gebou is.

Daar word 'n metode vir die voorspelling van die werkverrigtingsparameters van horisontale meertrapsproeikamers aangegee en die metode word met 'n paar voorbeelde geïllustreer. Die metode is eweneens van toepassing op sproeikamers waarin water in plaas van lug verkoel word.

Introduction

Arising from the declared intention of the gold-mining industry to significantly improve the underground thermal environment in deep mines, there is likely to be a marked increase in the number of refrigeration plants installed in mines over the next few years. In general, such refrigeration plants will be used to

(i) cool all the service water that is used underground,

(ii) cool the incoming ventilation air so as to remove the heat generated by autocompression and,

(iii) cool the ventilation air in the stope.

This paper deals with one aspect of the second of these applications, namely, the bulk cooling of the incoming ventilation air in open horizontal spray chambers. This type of cooling installation is of considerable importance because of the impossibility of achieving ordinarily acceptable environmental conditions in deep mines unless the incoming ventilation air is precooled before it enters the workings.

During the summer months, the wet-bulb temperature of the downcasting air in mines increases by about 4 °C per 1000 m of depth, so that the incoming ventilation air at various depths below surface will have temperatures approximately as indicated in Table I.

TABLE I

TEMPERATURES OF INCOMING AIR AT VARIOUS DEPTHS

Depth below surface m	Typical summer wet-bulb temperatures of downcasting ventilation air, °C
1000	22
2000	26
3000	30

*Environmental Engineering Laboratory, Chamber of Mines of South Africa, Johannesburg.

If the mine ventilation-and-cooling system is to be designed so as to keep wet-bulb temperatures below 28 to 30 °C, it is clear that the incoming ventilation air in deep mines has little capacity for removing heat from the workings unless it is first cooled. Experience[1] suggests that, in hot mines (in which rock temperatures exceed about 40 °C), the air on its way to the workings should be cooled to less than 22 °C (wet-bulb) and preferably as low as 18 °C in certain cases.

Horizontal spray systems that are designed to cool the incoming ventilation air should preferably be multi-stage arrangements in which the water is pumped and resprayed a number of times so as to ensure a longer duration of contact between the water and the air. The sequence of stages should be arranged so as to produce a counterflow effect. The justification for the use of multi-staging is that a far greater amount of heat can be removed with a given flow of cold water so that, for a given cooling rate, a smaller quantity of water need be circulated from the refrigeration plant.

The performance of various existing horizontal (cross-flow) spray chambers at the Buffelsfontein and Loraine gold mines has been studied[2]. The spray chambers at the Loraine gold mine utilize the multi-stage effect, and in fact it was their good performance that stimulated the present interest in multi-stage spray chambers. Arising from these studies and the experience gained in the construction of two new spray chambers at the Hartebeestfontein gold mine, tentative recommendations have emerged for the design of horizontal multi-stage spray chambers. Detailed research into the mechanics of the heat transfer within the spray system has been initiated, and no doubt a more sophisticated approach for the optimum design of these heat-transfer systems will emerge in due course. All that can be said at present is that the design as recommended is known to work effectively, and that its performance can be predicted reliably.

This paper gives recommendations for the design of spray cooling systems and describes a method for predicting the performance of such systems. It should be noted that the method applies equally to spray chambers in which the water, rather than the air, is being cooled. A detailed specification of an existing spray chamber that was constructed in accordance with these recommendations is given in the Addendum.

Nomenclature

$\triangle P$	Air pressure drop	Pa
K	Constant in equation for calculating air pressure drop	Ns^2/m^2
Q	Flow rate of air by volume	m^3/s
A	Cross-sectional area for air flow, measured immediately upstream of the eliminator	m^2
w	Density of the air at inlet	kg/m^3
η_w	Water efficiency (equation 1)	
η_a	Air efficiency (equation 2)	
t_{wi}	Temperature of water into spray chamber	°C
t_{wo}	Temperature of water out of spray chamber	°C
t_{ai}	Wet-bulb temperature of the air entering the spray chamber	°C
S_{ai}	Sigma heat content of air entering the spray chamber	kJ/kg
S_{ao}	Sigma heat content of air leaving the spray chamber	kJ/kg
S_{wi}	Sigma heat content of air having a wet-bulb temperature equal to t_{wi}	kJ/kg
L	Flow rate of water entering the spray chamber	kg/s
G	Flow rate of air by mass	kg/s
C	Specific heat of water, 4,18	kJ/kg. °C
E	Effectiveness (equation 4)	
F	Factor of merit	
R	Capacity factor (equation 3)	

Positioning of Cooling Systems for Intake Air

Before proceeding with a detailed discussion of spray chambers, a few comments might be in order on the positioning of the facilities for bulk cooling of the intake air. Because conflicting interests are involved, compromise is necessary.

It is desirable that the cooling of the intake air should be done as close to the workings as possible in order to avoid the cooling of air that is subsequently lost through leakage without ever getting to work places, and to minimize the increase in heat pickup from the hot rock surrounding the intake airways. It is also desirable that the facilities for bulk cooling of the air on any one level should be reasonably permanent so that they will always be operational, and so that they do not entail the labour-intensive and costly requirement of having to be moved every year or so as the centres of mining move further and further away.

Mobile spray chambers that can be moved to new locations would generally be limited to relatively small cooling capacities because of space limitations. Practical cooling capacities of rail-mounted spray chambers would probably be in the range 300 to 500 kW. The design of such spray chambers is the subject of a separate study at the present time. The spray chambers that are described in this paper can have cooling capacities 5 to 10 times greater than that of mobile spray chambers.

When bulk cooling of the air is done near the shaft, the coolers should be located preferably towards the edge of the shaft pillar and not immediately adjacent to the shaft stations. There are always air leakages and large heat pickups in the many excavations within the shaft pillar — in the ore-pass system, pipe ways, travelling ways, etc. — and such places should preferably not be in the cold-air zones that are produced by bulk cooling of the air.

The additional heat pickup from the rock surrounding intake airways when the incoming air is cooled represents a relatively small increase in the total heat load for the mine, probably 5 to 10 per cent. It is necessary to emphasize that drain water must not be carried in open drains in the cooled airways but must be transported back to the shaft in closed pipes. In fact, it is always essential in mines in which heat is a problem for all drain water to be carried in closed pipes if the temperature of the water is above the wet-bulb temperature of the ventilation air. These pipes need not be insulated.

Short-circuiting of the cooled air through old workings can be reduced to acceptable proportions. It has been our experience that mining departments are always ready to co-operate in properly sealing off old workings when the reasons for such sealing and their cost significance are explained.

In regard to the location of bulk air-cooling systems, there is an analogy in rock hoisting that should not be forgotten. Ideally, all the rock should be hoisted from the level on which it is produced, but practical constraints dictate that the rock should be sent down ore passes to loading facilities that are located at the bottom of the shaft. The extra costs of the ore-pass system and of the power for the additional depth of hoisting are small in relation to the other savings and benefits. Similarly with the cooling of incoming air. A preoccupation with theoretical idealism in choosing the location of bulk cooling systems should not be allowed to obscure the fact that the object is to provide reliable systems that will cool the working places of mines so as to optimize productivity and profitability. It is better to have a system that always works, and at a known and predictable cost, than one that is a little cheaper if it works but that is often not working when most needed. Cooling systems that need to be moved from time to time often fall into the latter category.

Description of a Spray Chamber

The layout of a typical two-and-a-half-stage spray chamber is shown in Fig. 1. The significance of a 'half-stage' is clarified later.

The cold water coming from the refrigeration plant is sprayed directly in the primary stage. The flow to this primary stage (and thus to the spray chamber as a whole) is controlled by a temperature-sensitive valve that regulates the temperature of the air leaving the spray chamber to a pre-set value.

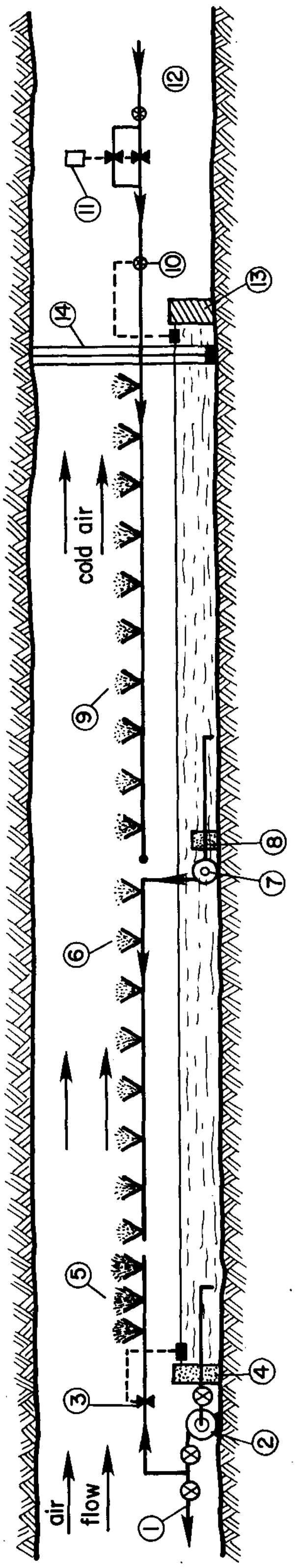

Fig. 1—Layout of typical 2½-stage spray chamber

1 return pipeline
2 pump
3 level control valve (modulating)
4 dam wall
5 final stage sprays (½ stage)
6 secondary sprays
7 submersible pump
8 low dam wall
9 primary sprays
10 float valve (high level shut off)
11 temperature control valve
12 supply pipe (cold water)
13 dam wall
14 eliminator plates (to drain into dam)

In the secondary stage, the water that has been collected in the sump of the primary stage is pumped through a second set of spray nozzles. The sumps of the primary and secondary stages are separated by a submerged dam wall that allows an overflow between these two stages, but that serves to stop the free mixing of the water in the two sumps.

The water from the sump of the secondary stage is collected and pumped back to the refrigeration plant. In order to obtain automatic control of the water level in the sumps, the rate at which water is pumped back to the plant must be controlled to match the rate of inflow of the cold water into the primary stage. This is achieved by the by-passing of some water from the delivery side of the pump to a third set of nozzles, the bypass flow being controlled by a modulating float valve that maintains a constant level of water in the dam. Thus, the quantity of water that is bypassed back to the sump is equivalent to the difference between the pumping rate and the flow of cold water into the primary sprays. In this way, the control of the water level is automatic, regardless of the setting of the temperature control valve, or of variations in the temperatures of the inlet air or water, or of variations in the flow rate of the air.

Design Recommendations

The performance of any particular spray system depends on the degree to which certain rather obvious practical conditions are satisfied. These physical factors are considered next, recommendations being given for the detailed design of spray chambers.

Distribution of air and water droplets

The distribution of air and water over the cross-sectional area of the chamber must be uniform, and the water sprays must reach all the way up to the hanging.

A uniform distribution of air can be achieved by having a smooth entry section into the chamber and by having as few bends in the spray chamber as possible.

Spray chambers have been operated successfully at air velocities as high as 6 m/s. At the present time, it is not possible to give a recommendation concerning an optimum or a maximum air speed. However, it would seem that any air velocity up to this value could be used safely; higher air velocities may indeed prove to be acceptable.

A uniform distribution of water droplets can be ensured by the correct placing and directing of nozzles. Care must be taken to ensure that at least some of the water reaches the hanging wall. The positioning of the spray nozzles may be varied from point to point down the length of the chamber to suit local irregularities in the profile of the excavation. A typical positioning of nozzles is illustrated in Fig. 2. These nozzles produce a flat, fan type of spray pattern that almost completely covers the cross-section of the spray chamber. It is recommended that spray patterns of this nature should be used, and that the nozzles should face upwards and spray at right-angles to the direction of air flow or perhaps slightly downwind. The shape of the spray pattern and the pressure at the nozzle for a given flow rate will depend on the type of nozzle that is used. For example, one type of spray nozzle (referred to in the

Addendum) requires a pressure of 200 kPa to produce a fan pattern 4 m high over an angle of 65°.

Distribution of nozzles

The distance between the nozzles along the length of the spray chamber obviously depends on the amount of space that is available and the quantity of water to be sprayed. Experimentation into the effect of the water loading is planned for the near future; it is recommended tentatively that the water loading should not exceed 0,25 (l/s)/m^3, and that the nozzles should be spaced uniformly down the length of the spray chamber.

A typical nozzle has a flow of 1 l/s at a pressure that is sufficient to spray droplets to a height of about 4 m above the nozzle. Thus, for a chamber of 10 m^2 cross-section, the nozzle spacing could be about 400 mm, or 2,5 nozzles per metre, giving a water flow rate of about 2,5 l/s per metre length of chamber. For a chamber of 12 m^2 cross-section, the nozzle spacing could be 333 mm, giving a water flow rate of about 3 l/s per metre length of chamber.

There is no need to leave a large gap between the sprays of different stages.

Staging

It is essential that the sprays are arranged so that the water moves from stage to stage in the direction opposite to that in which the air is flowing and so produce a counterflow effect. Obviously, the greater the number of stages, the stronger will be this counterflow effect and the longer the total duration of contact between the water and the air. However, a theoretical analysis[3] (which assumes that each stage has the same performance characteristic) has shown the following.

(i) Little advantage is gained by having more than four stages.

(ii) Little advantage is gained by the recycling of more water than that which is sprayed in the primary stage, provided that the individual stages are constructed according to these recommendations. Thus, each of the primary, secondary, tertiary (etc.) stages should have approximately the same water flow rate as the others.

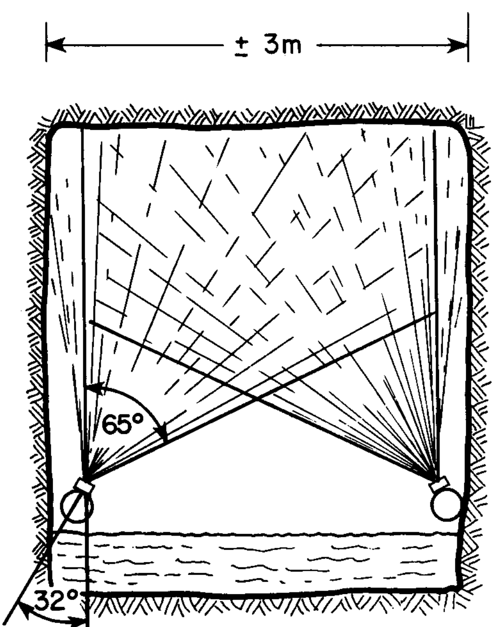

Fig. 2—Typical spray pattern

It should be noted that, in many applications, seasonal effects will result in the spray chamber operating on part load for much of the year, with correspondingly less water being used in the first (coldest) sprays, and more in the final half-stage. (The number of stages may also be reduced.)

The sumps of neighbouring stages should be separated by low dam walls (8 in Fig. 1) in order to hinder the free mixing of the water in the different sumps. Each such wall must have a vertical gap about 200 mm wide that will allow full flow of the circulating water in case the water supply to the first stage of sprays is cut off.

To recycle water from stage to stage, the water must be drawn from the sump of one stage and pumped to the nozzles of the next stage. In many cases, this would mean excessively long suction pipes to the pumps, and it is recommended that submersible pumps (7 in Fig. 1) be used.

In some situations, the main pump that returns water to the refrigeration plant for recooling may also be used to supply water to a complete stage of sprays. These sprays would be additional to the final half-stage of sprays, and a much larger pump would naturally be required.

Size of droplets

An important parameter in the performance of spray systems is the size of the droplets that are produced by the nozzles. For a given flow rate, the smaller the droplet size, the larger is the surface area available for heat transfer. However, the smaller the droplet, the lower the average relative velocity between the air stream and the droplet. This lower relative velocity tends to reduce the rate of the heat transfer. The size of the droplets is approximately inversely proportional to the nozzle pressure, so that the production of small droplets expends additional pumping power. Furthermore, the smaller the droplets, the more sophisticated must be the apparatus for eliminating droplet carry-over, and the higher the pressure loss must be across this apparatus.

Thus, depending on all these parameters, there will be an optimum size of droplet. This optimum is not known at present, but experience indicates that average drop diameters between 0,5 and 2,5 mm ensure satisfactory performance.

Eliminator plates

In most instances, no carry-over of water droplets from the sprays can be tolerated, and a droplet eliminator (14 in Fig. 1) must therefore be erected. It is advisable for the eliminator plates to be located at least 3 m downstream of the last set of nozzles so as to provide a zone for the fall-out of most of the droplets before they reach the eliminator plates.

The design shown in Fig. 3 has proved to be satisfactory. The plates can be made of aluminium, fibreglass or other rigid plastic, or asbestos cement.

The loss in air pressure caused by this configuration can be estimated from the equation

$\triangle P = K\,w\,(Q/A)^2$,

where

ΔP = pressure loss, Pa
K = 4,5 to 5,0 Ns^2/m^2
w = density of the air, kg/m^3
Q = flow rate of air, m^3/s
A = frontal face area of the eliminator, m^2.

Losses in air pressure

It has been found that the water sprays have little effect on the drop in air pressure across a spray chamber. The increase in pressure drop between a dry chamber and the same chamber with sprays in operation would generally be less than 10 per cent. This applies to sprays directed at right-angles to the air flow; if the sprays are directed against the direction of air flow, the pressure drop will be higher.

The eliminator usually accounts for about three-quarters of the total drop in pressure across the spray chamber.

Temperature control

Automatic control of the temperature of the air leaving the spray chamber is attained by regulating the flow of incoming cold water to the primary sprays (and thus to the spray chamber as a whole) by means of a temperature-sensitive valve (11 in Fig. 1). For this control to be automatic, the main return pump (2 in Fig. 1) must be controlled so as to return the same quantity of water entering the primary stage. As already described, this is done by bypassing water from the delivery of the pump to an additional set of nozzles, the bypass flow being controlled by a modulating float valve that regulates the flow automatically so as to maintain a constant level of water in the dam.

For this method of controlling water to be successful, careful consideration must be given to the flow characteristics of the final spray stage. As the float valve moves from an open to the fully closed position, the return pump will 'see' a changing delivery characteristic. The problem is illustrated graphically in Fig. 4. As the float valve opens, the operating point of the pump moves from point A to point B on the characteristic curve of the pump. The designer must ensure that the pump and motor will be able to operate satisfactorily over this range of conditions.

When the spray chamber is on maximum load, the final stage receives the minimum flow of water. Conversely, when the spray chamber is operating at a minimum cooling capacity, the final stage of sprays receives the maximum flow. Thus, when the primary stage takes a large quantity of water, the final stage receives very little water and *vice versa*. Because of this paradox, the final stage is not considered important in the thermal performance of the spray chamber since it is most effective when least needed. It is for this reason that the final stage is referred to as a 'half' stage. The sprays of this final stage need occupy little space, and large sizes of droplets can be tolerated. It has been found convenient to use Hamspray nozzles located about 1 m below roof level for this final stage of sprays.

If the system for automatic temperature control of the air leaving the spray chamber is omitted, the amount of water entering the spray chamber will vary as the pressure in the supply pipe varies, and wide fluctuations in the temperature of the air leaving the spray chamber may be experienced.

Miscellaneous items

Orifice plates and thermometer pockets should be installed at suitable positions so that the performance of the spray system can be evaluated periodically.

As a safety feature against flooding, a high-level cut-off valve (10 in Fig. 1) should be installed in the cold-water pipe feeding the primary stage. In addition, high-level and low-level alarms should be provided to warn of the possibility of flooding, of failure of the main pump, or of the pump running dry.

In order to illustrate the design of a complete spray chamber, a description is given in the Addendum of the system that has been installed at 32 level No. 4 Shaft of

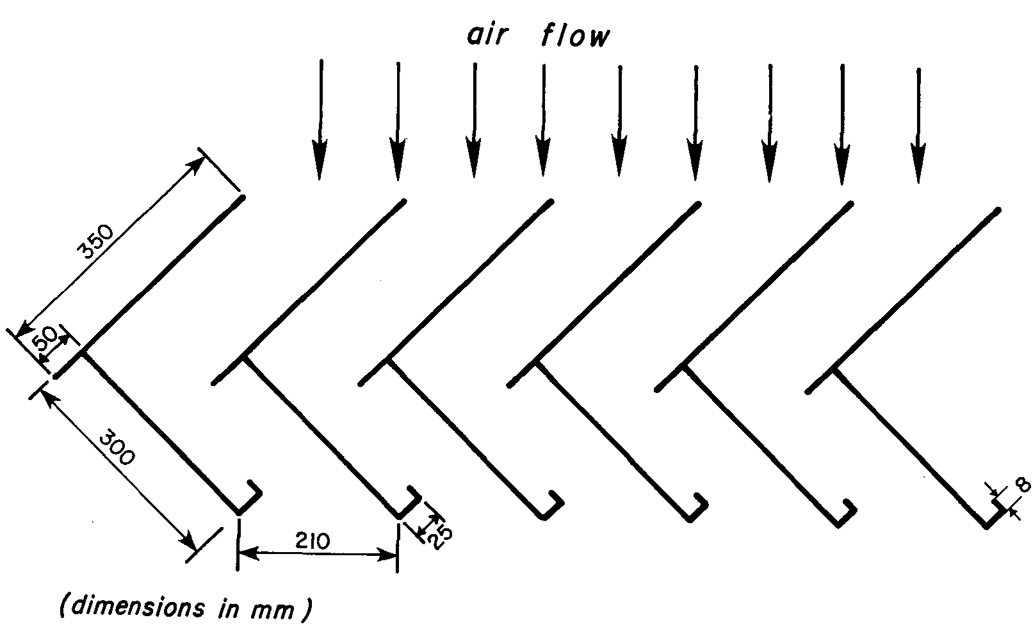

Fig. 3—Design for eliminator plates (plan view)

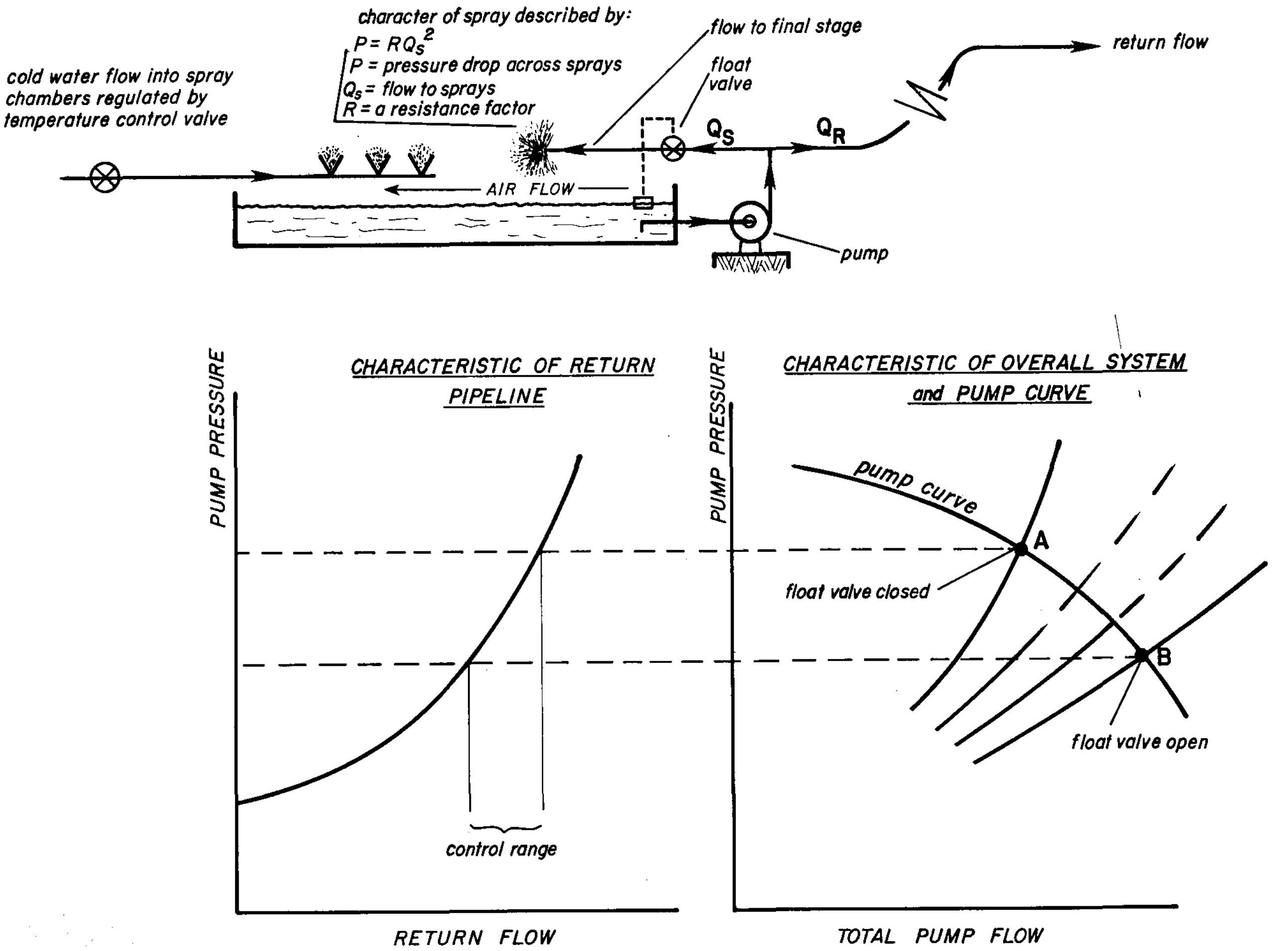

Fig. 4—Typical characteristics of an air-temperature control system

the Hartebeestfontein gold mine. This system has been designed according to the principles described here, and the spray chamber is as illustrated in Fig. 1.

Predicting the Performance of Spray Chambers

When spray chambers for the cooling of air are designed, it is necessary for predictions to be made of the performance at different operating conditions. The problem is to determine, after the physical shape and layout of the spray chamber have been specified, whether the thermal performance will be acceptable.

The method outlined below permits prediction of the performance that would be attained if the spray chamber were constructed according to the recommendations given earlier. A more detailed explanation of this method of calculation is given elsewhere[4].

The performance of a cooling tower or spray chamber can be expressed in terms of its water efficiency and its air efficiency, although in themselves these efficiencies are not sufficient to describe the performance.

The *water efficiency*, η_w, is defined as the ratio between the actual increase in temperature of the water across the spray chamber and the maximum possible increase in this temperature as imposed by the Second Law of thermodynamics:

$$\eta_w = (t_{wo} - t_{wi})/(t_{ai} - t_{wi}). \quad \quad (1)$$

The *air efficiency*, η_a, is defined in terms of sigma heats, being the ratio of the actual drop in heat content of the air to the maximum possible drop in this heat content as imposed by the Second Law of thermodynamics:

$$\eta_a = (S_{ai} - S_{ao})/(S_{ai} - S_{wi}). \quad \quad (2)$$

The limitations imposed by the Second Law of thermodynamics are that the air cannot leave the chamber at a temperature lower than that of the incoming water, and that the water cannot leave the chamber at a temperature higher than the inlet wet-bulb temperature of the air.

Another important parameter in assessing the performance of cooling towers and spray chambers is the *capacity factor*, R. The capacity factor is calculated purely from inlet-air and inlet-water conditions:

$$R = \frac{L\ C\ (t_{ai} - t_{wi})}{G\ (S_{ai} - S_{wi})}. \quad \quad (3)$$

The capacity factor has a value that is typically between 0,5 and 2,0. It depends significantly on the ratio of water to air, L/G, and in fact it can be thought of as a normalized thermal capacity ratio for the water and air streams[4].

The *effectiveness*, E, of a spray chamber (or cooling tower) is defined in an analogous way to that in the case of heat exchangers; it is the efficiency of the fluid stream having the smaller thermal capacity. Thus,

$$E = \eta_w \text{ if } R \leqslant 1, \quad \quad (4a)$$

and

$E = \eta_a$ if $R > 1$ . . . . (4b)

A *factor of merit, F,* is used to categorize the performance of spray chambers[4]. A given design of spray chamber will have a fixed factor of merit regardless of the temperatures of the air or water. This factor of merit has a value typically between 0,3 and 0,8; a factor of merit of 1,0 would correspond to a theoretically perfect spray chamber, and no spray chamber or cooling tower can have a factor of merit exceeding 1,0.

Table II gives values for the factor of merit that can be expected[2] for spray chambers with 1 to 4 stages constructed in accordance with the recommendations given earlier.

TABLE II

FACTORS OF MERIT FOR SPRAY CHAMBERS

Number of stages	Factors of merit F
1	0,40 to 0,50
$1\frac{1}{2}$	0,50 to 0,60
$2\frac{1}{2}$	0,60 to 0,70
$3\frac{1}{2}$	0,70 to 0,75

Once a spray configuration (and thus a factor of merit) has been chosen, the performance curve for that configuration can be found from Fig. 5. This diagram is a plot of effectiveness (E) against capacity ratio (R). The method of calculation using this figure can best be described through worked examples.

Example 1

Problem

Assume that a spray chamber is to cool 60 kg/s of air that enters at 28 °C(wb)/32 °C(db). Refrigeration capacity is sufficient to supply 30 kg/s of cold water that will arrive at the spray chamber at 10 °C. The ambient air pressure is 110 kPa. To what temperature will the air be cooled in a two-and-a-half-stage spray chamber?

Solution

Sigma heat of air at 28 °C (wb) =82,0 kJ/kg

Sigma heat of air at inlet-water temperature 10 °C =27,4 kJ/kg

Air flow =60 kg/s

Water flow =30 kg/s

Factor of merit of a two-and-a-half-stage spray chamber (assumed) $F = 0{,}65$

$$\text{Capacity factor } R = \frac{L\,C\,(t_{a1} - t_{w1})}{G\,(S_{a1} - S_{w1})} = \frac{30 \times 4{,}18\,(28 - 10)}{60\,(82{,}0 - 27{,}4)} = 0{,}689.$$

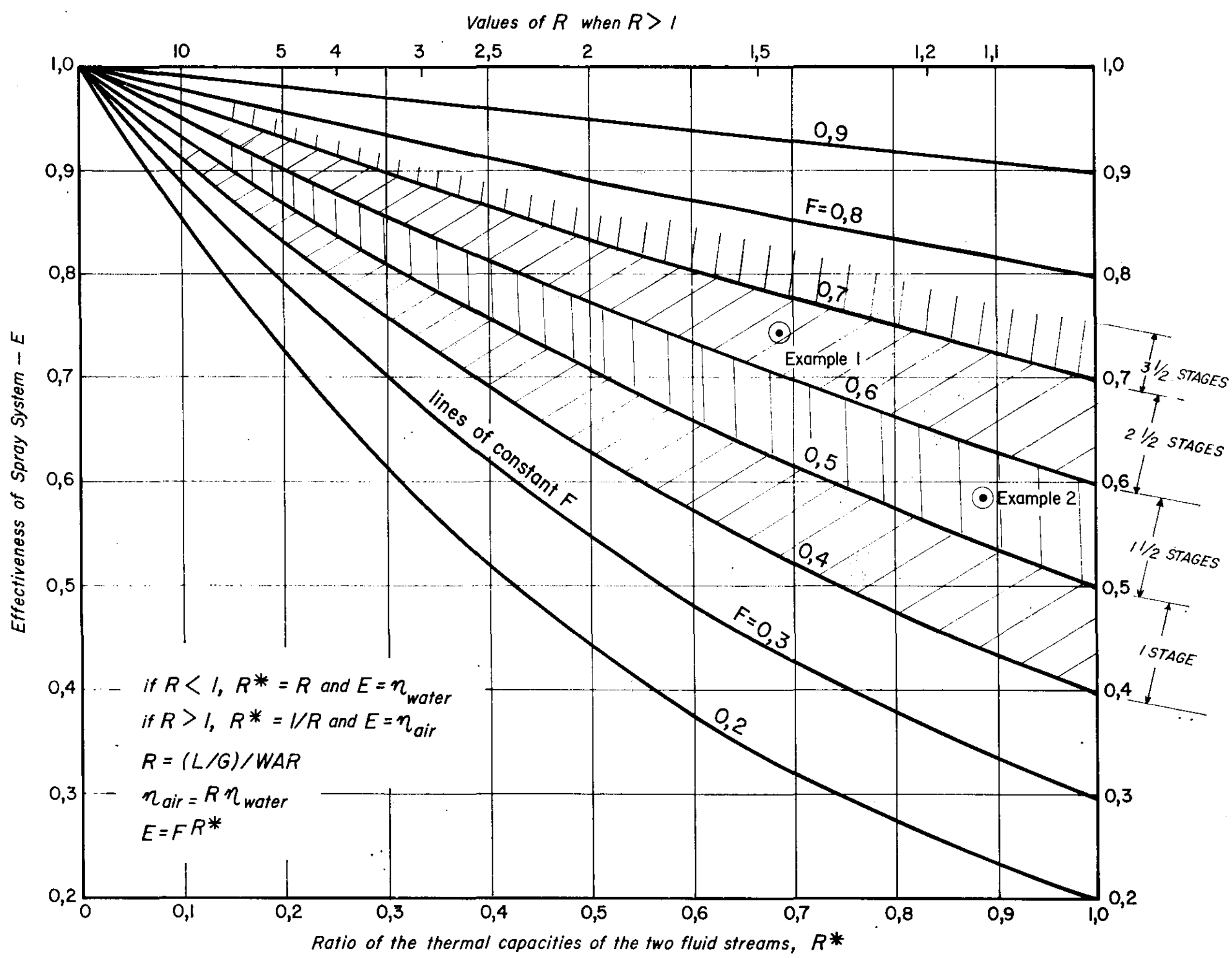

Fig. 5—Performance curves for spray heat exchangers

From Fig. 5:

A spray chamber with a factor of merit $F=0{,}650$
operating at a capacity ratio $R=0{,}689$
has an effectiveness $E=0{,}743$.

This effectiveness is equal to the water efficiency, since $R<1$.

With this information it is now possible to calculate the temperature of the exit water, the heat-transfer rate, and the temperature of the exit air.

Thus,

$$\eta_w=(t_{wo}-t_{wi})/(t_{ai}-t_{wi})$$

$$\therefore\ t_{wo}-t_{wi}=0{,}743\ (28-10)$$
$$=13{,}4\ ^\circ C.$$

The temperature of the water therefore increases by 13,4 °C, and the rate of heat transfer is

$$Q=30\times 4{,}18\times 13{,}4$$
$$=1680\ kW.$$

This is equal to the rate at which heat is removed from the air, and the corresponding decrease in sigma heat for an air flow rate of 60 kg/s

$$=1680\ kW$$
$$=28{,}0\ kJ/kg.$$

The sigma heat of the air leaving the spray chamber will thus

$$=82{,}0-28{,}0\ kJ/kg$$
$$=54{,}0\ kJ/kg.$$

Referring to sigma heat tables, the wet-bulb temperature of the air leaving the spray chamber is 20,3 °C. This air will be saturated. Thus, the air is cooled from 28 to 20,3 °C, and the cooling duty of the spray chamber is 1680 kW.

(If the factor of merit F had been 0,70 instead of 0,65, the change in water temperature would have been 14,1 °C instead of 13,4 °C, and the rate of heat transfer would have been 5,2 per cent greater.)

For the same water-consumption and inlet conditions, the performance of other systems with different numbers of stages would be approximately as shown in Table III. The higher the factor of merit, the greater the amount of refrigeration that can be distributed by the same quantity of water, and the cooler will be the air going on into the mine.

Example 2

Problem

A spray chamber is to cool 45 kg/s of air from 29 °C (wb)/35 °C(db) to 18 °C, saturated. The refrigeration capacity is such that 35 kg/s of water is available at 6 °C. The ambient pressure is 110 kPa. How many stages would be needed?

Solution

Sigma heat of inlet air 29 °C (wb) $=86{,}3$ kJ/kg
Sigma heat of exit air 18 °C (wb) $=47{,}3$ kJ/kg
Sigma heat of air at inlet-water temperature 6 °C $=19{,}3$ kJ/kg
Air flow $=45$ kg/s
$\therefore$ Heat to be transferred $=45\times(86{,}3-47{,}3)$
$=1755$ kW
Water flow $=35$ kg/s
Inlet-water temperature $=6$ °C
Exit-water temperature $=6+1755/(35\times 4{,}18)$
$=18{,}0$ °C

$$\text{Capacity ratio, } R=\frac{L\ C\ (t_{ai}-t_{wi})}{G\ (S_{ai}-S_{wi})}=\frac{35\times 4{,}18\ (29-6)}{45\ (86{,}3-19{,}3)}=1{,}12$$

$$\text{Effectiveness, } E=\eta_a \text{ (since } R>1)=\frac{S_{ai}-S_{ao}}{S_{ai}-S_{wi}}=\frac{86{,}3-47{,}3}{86{,}3-19{,}3}=0{,}582.$$

From Fig. 5: A spray chamber having an effectiveness $E=0{,}582$
operating at a capacity ratio $R=1{,}12$
has a factor of merit $F=0{,}55$.

Since a one-and-a-half-stage spray chamber would exhibit a factor of merit of between 0,50 and 0,60, it would be suitable for this duty.

If, instead, a two-and-a-half-stage spray chamber were to be used ($F=0{,}65$) for the same amount of heat transfer, the quantity of water required would be 26,2 l/s, which is a reduction of 25 per cent.

This example illustrates a very important point. The reduced water-circulation rate of 25 per cent represents a considerable saving in pumping power, and possibly also in pipe sizes if applied on a mine-wide scale, for circulating the cold water between the refrigeration plant and the spray chamber. In general, this saving would more than justify the additional cost of an extra stage in the spray chamber.

Acknowledgement

This paper arises from work carried out as part of the research programme of the Research Organisation of the Chamber of Mines of South Africa.

References

1. Van der Walt, J., and Whillier, A. The cooling experiment at Hartebeestfontein gold mine. *J. Mine Vent. Soc. S. Afr.*, vol. 31. Aug. 1978.
2. Bluhm, S. J., Ramsden, R., and Whillier, A. Performance tests on horizontal spray chambers. *Chamber of Mines Research Report* no. 42/76.

TABLE III

PERFORMANCE OF OTHER COOLING SYSTEMS

		Cooling rate kW	Temp. of exit water °C	Temp. of exit air °C
One-stage	($F=0{,}45$)	1304	20,4	22,2
One-and-a-half stages	($F=0{,}55$)	1492	21,9	21,2
Two-and-a-half stages	($F=0{,}65$)	1680	23,4	20,3
Three-and-a-half stages	($F=0{,}73$)	1818	24,5	19,5

3. BLUHM, S. J. A theoretical investigation into the performance of staged spray chambers. Report in preparation.
4. WHILLIER, A. Predicting the performance of forced-draught cooling towers. *J. Mine Vent. Soc. S. Afr.*, vol. 30. Jan. 1977. pp. 2-25.

Addendum

The following are the specifications for the two-and-a-half-stage spray chamber at 32 level, No. 4 Shaft, Hartebeestfontein gold mine. The spray chamber is as depicted in Fig. 1.

Minimum length of tunnel for the sprays	30 m
Total length of tunnel that was provided	100 m*
Cross-section dimensions (width × height)	3,1 × 4,5 m
Entire excavation gunited	
Flow rate of air 70 m^3/s	87,5 kg/s
Average air velocity	5 m/s
Barometric pressure	110 kPa
Wet-bulb temperature of air at inlet	26 °C
Temperature of air leaving the spray chamber	20 °C
Flow rate of water (winter to summer)	15 to 35 l/s
Temperature of inlet water	8 °C
Temperature of outlet water	22 °C
Water:air ratio, L/G	0,17 to 0,40
Factor of merit, F : assumed for the design	0,65
: measured after completion	0,70
Normal design rating	1 800 kW
Maximum cooling rate	2 000 kW

*This length was not all required for the spray chamber but was dictated largely by the layout of other airways in the vicinity.

Table IV specifies all the components that are itemized in Fig. 1.

TABLE IV

SPECIFICATION OF ITEMS SHOWN IN FIG. 1

Item No.	Specification	Comment
1	Return pipe line	150 mm diameter
2	Return pumps, two A.P.E. pump–motor sets, 18 kW	Pumps selected to produce 45 l/s at 250 kPa
3	Modulating float valve	100 mm, Clayton class 250 valve
4	Dam wall 0,75 m thick	Concrete
5	Final half-stage sprays; eight 25 mm Hamsprayer nozzles	Supplied by Baltana Co. Ltd, Johannesburg; maximum flow will be 30 l/s; nozzles are spread over 2,5 m
6	Secondary-stage sprays 52 Veejet, type H½U 65150 nozzles (40 l/s at 220 kPa)	Supplied by Monitor Engineering, Johannesburg; the nozzles are spread over a distance of 40 m since the space was available: 12 m would have sufficed
7	Flygt submersible pump type B2151 MT, 23 kW	Pump selected to produce 40 l/s at 300 kPa
8	Low dam wall, concrete 0,5 m thick, with vertical gap 200 mm wide	To reduce mixing of water in primary and secondary sumps, but must allow flow of 40 l/s in the event of failure of the cold-water supply
9	Primary-stage sprays 50 Veejet, type H½U 65150 nozzles	These nozzles are spread over 40 m (see comment in 6 above) and will handle up to 35 l/s at 180 kPa
10	Non-modulating float valve, 150 mm Clayton class 250	Safety feature against flooding
11	Two temperature-control valves in parallel, 75 mm Spirax Sarco type NSRA	The two valves are in parallel so as to minimize pressure losses
12	Supply pipe	
13	Dam wall 0,75 m thick and 0,5 m above footwall	
14	Eliminator plates	Made of aluminium and similar in design to that shown in Fig. 3