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Certified that the attached dissertation on " Air-washers and their performance in Air- Conditioning"

was submitted by

Sri Shatrughan Mehrotra

and accepted for the award of Degree of Master of Engineering in Applied Thermodynamics (Refrigeration & Air- Conditioning)"

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AIRWASHERS AND THEIR PERFORMANCE IN AIR CONDITIONING

By
S. MEHROTRA, B. E. (HONS)

THESIS SUBMITTED IN PARTIAL FULFILMENT
OF
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DEPARTMENT OF MECHANICAL ENGINEERING
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ROORKEE (India)

MARCH 1963

CERTIFICATE

Certified that the thesis entitled "AIRWASHERS AND THEIR PERFORMANCE IN AIRCONDITIONING" which is being submitted by Shri S. Mehrotra, in partial fulfilment for the award of theDegree of Master of Engineering in Applied Thermodynamics (Refrigeration and Airconditioning) of the University of Roorkee, is a record of student's own work carried out by him under my supervision and guidance. The matter embodied in this thesis has not been submitted for the award of any other Degree or Diploma.

This is further to certify that he has worked for a period of 4½ months for preparing this thesis for Master of Engineering Degree at the University.

Rajendra Prakash

Roorkee.

Dated March 23 ,1963

(Rajendra Prakash)
Reader in Mechanical Engineering,
University of Roorkee,
Roorkee.

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CHAPTER - I.

INTRODUCTION.

CHAPTER - I.

I N T R O D U C T I O N.

Evaporative water cooling was well known to ancient people. In India, evaporation was used to make ice. Shallow beds dug in the earth were filled with 1 ft of straw upon which earthern pans were placed. On still frosty nights even when the temperature fell no less than 43° F or 11° ~~xxxx~~ above freezing ice would be formed. Evaporation plus radiation into the night sky provided the refrigeration. This method is reported to be in use in Modern Iran also.

In Iran homes are frequently semi-underground to escape solar heat and in some there is a pool of running water above which a ventilation tower opens. The tower catches the breeze and diverts it occross the water pool and then into the room. An ancient Persian practice was to cover tents with felt which was kept wet, a practice used by American Indians also.

In India, people developed the cocotatti . Doors are replaced in summer by Tattis or frameworks similar to screen doors, covered with dry grass and kept watered by hand. In some forms there is a trough provided at the top from which water tickles down gradually.

Another Indian evaporative air cooler is the Thermantidote. Here the door is replaced by a wheel like frame work covered with grass which is revolved and kept wet.

The cooling effect of fallingwater has also been used. It is reported that Peter the great in one of his Leningrad gardens had a tree piped so that the curtain of water fell from its outer most branches but did not wet the benches near the trunk.

American evaporative cooling developed chiefly in Arizona and ~~sother~~ Southern California during the late 1920's and early 1930's. However, actual manufacture began from about 1936 and 1937. As time passed, ~~the~~ various methods developed and these are listed as follows :-

1. The common desert cooler.
2. The Air-washer cooler
3. The unit humidifier cooling system.
4. The exhaust type window pad cooler.
5. The rotating pad cooler.
6. The portable desert cooler.
7. The indirect evaporative cooling systems.

As the name suggests the primary purpose of air-washer in the initial stages was the cleaning of air. Later on, however, it is used in airconditioning work not only for cleaning but also for either humidifying or dehumidifying and cooling the air used for ventilation and for industrial airconditioning. It is generally observed that after a rainstrom the outdoor air is much fresher and cleaner than it was before the storm, and this fact was probably the basic idea that led to the application of airwasher for this service.

Basically airwasher consists of a casing with a tank, a single or multiple bank of spray nozzles for producing fine spray in the form of mist, an eliminator section at the leaving end to remove entrained moisture, and inlet louvers to prevent water from splashing out of the unit at the entering side. Water sprayed through the nozzles is collected in a tank where it passes through a strainer, into the pump and back up to the nozzles. Thus if the water is neither heated nor chilled and is recirculated the air is evaporatively cooled by intimate contact with the water.

In applications with extremely high sensible heat loads, cooling by means of refrigeration is out of question because of the very high initial and operativecosts. At present the use of airwasher~~s~~ evaporative cooler for such applications has been gaining more and more favour because they have proved able to provide satisfactory comfort conditions while being economical to purchase, maintain and operate.

In laundries and dry cleaning plants and also in a large number of industrial applications these airwasher cooling systems are delivering air though unrefrigerated at diffuse temperature from 75° F to 80° F, even with high outdoor conditions prevailing. It has now become possible to maintain comfort conditions ~~and~~ using 100 percent outdoor air which upto this time were considered unsuitable for workers to remain in during summer conditions.

An airwasher evaporative cooling system properly designed requires the use of all outside air without recirculation from within the cooled space. In some cases direct or 'spot cooling' is the most practical method of air distribution. This is especially true in building in which the number of workers is relatively few compared to the size of the building, or in which the buildings have extremely high ceilings, in which case it will be uneconomical to attempt cooling the entire space. It is also advantageous when there are hot spots in the building in which case it will be imperative to bring cooling just to those areas. In such cases ducts can discharge the air to each individual thereby cutting down the capacity of the system required and assuring cooling only at these critical spots.

Airwashers are also used to a great extent in textile mills where accurate and rather high humidity conditions are required. Heat relief for workmen around furnances, steam presses, molten metal etc., is a common task for this unit. The cooling of large electric motors and generators is accomplished by supplying a large amount of air through the airwasher to the windings to carry the heat that is generated.

Humidification with airwasher can be accomplished in three ways :-

1. Use of recirculated spray water without prior treatment of the air.

2. Preheating the air and washing it with recirculated spray water.

3. Using heated spray water in which case the air is heated and humidified, the wet bulb and dry bulb temperatures both increased.

Cooling and dehumidification can also be accomplished by the airwasher. Dehumidification will result if the leaving water temperature is below the entering dew point temperature of air.

The performance of air washer under different conditions of spray water temperature, number of banks, direction of spray, and air velocity have been investigated.

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CHAPTER - II

AIRWASHER - TYPES AND CONSTRUCTIONAL DETAILS

CHAPTER - II.

AIRWASHER - TYPES AND CONSTRUCTIONAL DETAILS

2.1 SPRAY TYPE WASHERS :

2.1.1 SUMMARY :-

A spray type air washer consists essentially of a chamber or casing containing a spray nozzle system, a tank for collecting the spray water as it falls, and a series of vertical scrubber and eliminator plates at the discharge end, for removal of dust and entrained drops of water from the air. In the operation of an air washer the air to be cleaned and cooled or heated flows successively.

1. Through a water spray from a group of small nozzles and

2. over the wetted surface of a series of vertical scrubber and eliminator plates. A pump recirculates water at a rate greatly in excess of the evaporation rate. The water spray from the nozzles must be sufficiently large in amount so that it may be mixed thoroughly with the air. Intimate contact between the spray water and flowing air causes heat and mass transfer between the air and the water. The scrubber and eliminator plates are provided with deflectors to direct the flow of air through them. Dust and other solid particles of matter that are carried by the air to be cleaned when coming in contact with the wetted surface of the scrubber and eliminator plates are caught up by the water on the surface of the plates and carried away to the

settling tank. The water discharges over the scrubber and eliminator plates from a series of nozzles.

The nozzles for spraying water are placed in a row across the direction of flow of air so as to distribute the water in a fine spray or mist to thoroughly saturate the air. In some designs of air washers more than one row of nozzles is used. Two or more spray banks are usually used where a very high degree of saturation is required, and for cooling and dehumidification applications requiring chilled water. Two stage washers are used for dehumidification when the quantity of chilled wateris limited, or when the temperature of water is above that necessary for single stage design. Arranging the two stages for counterflow of water permits the use of smaller quantity of water with a higher temperature rise.

The most effective cleaning of the air is obtained by the scrubber plates of the air washer which are designed to change the direction of the flow of the air so that the dustis removed from the air by its inertia and by its contact with the wet surface of the scrubber plates. The eliminator plates are provided to remove the drops of water from the leaving air. The accumulated dirt must be removed from the setting tank at frequent intervals. Sometimes the temperature of the outside air entering the air washer is lower than the freezing temperature of the water. A tempering heater must then be placed at the air inlet to protect the water pipes and nozzles of the air washer from freezing.

It may appear apparently that most of the dirt mixed with the air should be removed while the air is in the spray chamber, this is not the case, practically all of the air borne dirt ~~is~~ being removed by the eliminators. Air washer will not always remove greasy particles such as soot. Tobacco smoke will ordinarily pass through an air washer.

Most odor can be removed from air in an air washer. An odor is usually due to the vapor of some compound mixed with the air. Many of these vapors will dissolve in water. In fact as the water becomes saturated with the soluble vapors, it becomes less and less able to remove adors from the air. Furthermore, the water itself acquires an odor due to the material in solution, therefore, it must be changed frequently under these conditions and the tank filled with fresh water.

Generally, the water in an air washer is recirculated. When the air is to be humidified either the air or the water or sometimes both are heated before entering the washer. If the air is to be cooled and dehumidified in the washer, the water must first be cooled.

Occassionally, water that is cold enough ~~so that no further cooling is needed,~~ can be obtained from a city main or well. In such a case, fresh water is supplied continuously to the sprays and is then wasted to the sewer. When water is used in a washer in this way, it should contain no substances in solution that may give unpleasant odor to the air. This must be checked very carefully because of the initimate contact of the water and air in the washer. Well water occassionally contains sulphur compounds in solution which impart an odor to the air.

The figure 2.1 shows the air washer consisting primarily of a spray chamber in which a number of spray nozzles and water supply risers are installed.

2.1.2 WATER SUPPLY TANK :-

The air washer tank which is provided for holding the water supply to the spray nozles is from 16" to 20" deep and is built of galvanized sheet iron. It is thoroughly braced on the outisde by 1½ by 1½" by 1/4" galvanized steel angles, and is provided with a frame around the top for connection to the air washer casing. All seams of the tank should be rivetted and soldered and all rivet heads should be soldered to make the tank tight.

The water supply tank is usually divided into two compartments by a strainer of 12 mesh copper wire cloth rigidly held on a removable galvanized iron frame. All of the water flowing to the pump must first pass through this screen. The openings of the screen are made smaller than the nozzle openings. Although this minimizes difficulties with nozzle clogging it does not entirely eliminate clogging. Strainers should be readily removable for inspection and cleaning. Belt type, automatic and other specially designed strainers are available for use in textile mills and other industries where lint and heavy concen-

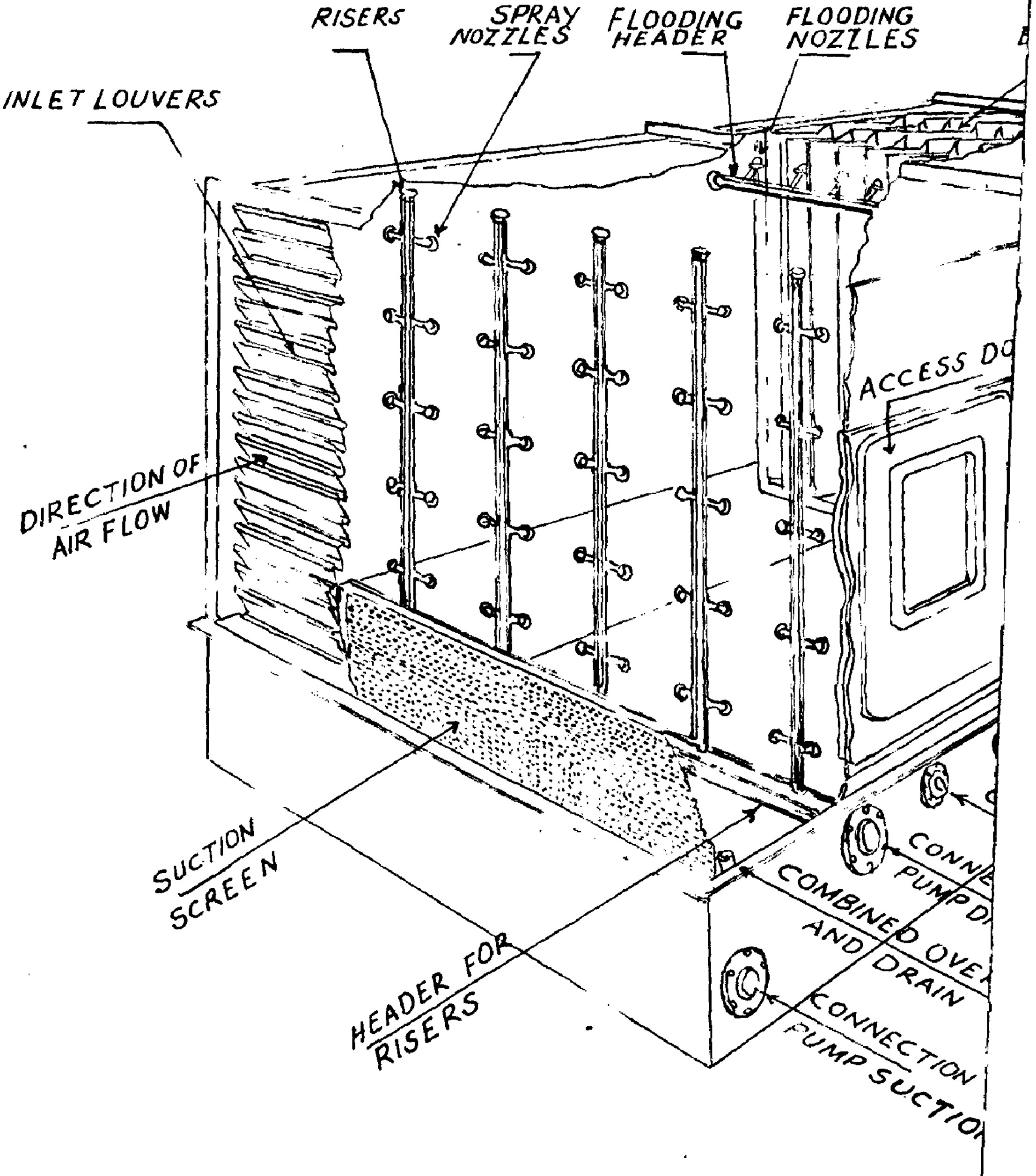

FIG-2·1 AIR WASHER

trations of dust are present.

Over the top of the compartment formed between the strainer and the end of the tank to which the suction of the pump is connected, a hinged cover should be provided to prevent any dust falling in as the air passes over it.

A float valve is provided in washwers to automatically admit make up water to replace any water lost through entrainment or evaporation. There is a sealed overflow connectionto the sewer to prevent flooding the tank in case the float valve sticks or the float becomes water logged. A drain valve is needed at the bottom of the water supply tank so that all the water may be drawn off for cleaning the tank or to prevent freezing if exposed to very cold air when not in use. The overflow consists of a removable length of the pipe, the top of which is slightly below the upper edge of the tank. In order to fill the tank quickly after it is emptied a quick fill connection is provided. This is nothing but a second water connection to the tank. This connection is supplied with water from the same line feeding the float valve. A hand operated valve is installed in the quick fill line.

2.1.3 AIR WASHER CASING :-

The casing of an air washer may be constructed of various metals but it is generally made of 18 gage glavanized sheet iron thoroughly stiffened and braced on the outside by 1½ by 1½ by 1/4 inch galvanized steel angles, spaced not more than 3 feet apart. The top and sides of the casing are made separately and ~~is~~ are bolted together. The casing should be attached by similar means and with rubber gaskets to the top of the water supply tank. All seams,exposed edges of sheets and rivet heads should be soldered.

The length of the air washer is between 4 feet to 10 feet depending upon the number of banks provided. The usual air washers are made of five seven and either eight or nine feet length. The five and seven foot air washer have usually one bank of sprays and the nine foot washer has two or three banks of sprays. Two banks of sprays can be installed in seven foot washer but the they are considered to be less effective than the long washer. The five foot washer is least expensive, but seven foot washer is commonly used. However, where there is any substantial air conditioning load two banks of sprays are generally used in order to provide sufficient water for conditioning purposes. In special cases it is necessary to install three or four banks of sprays. They can therefore, be made having as many banks as are required. Such special washers have sometimes length as long as 15 feet. Cooling coils for the direct expansion refrigerant are occassionally installed in the spray chamber of such washers. This practice was quite common a few years ago but is rarely adopted today.

If a set of eliminator plates is installed after each group of two banks of sprays, the washer is spoken as a two stage washer.

Sometimes, a two stage washer may have three banks of sprays in each stage.

In the smaller sized air washer one inspection door is provided and in the long type two doors are installed. All doors should be hinged and preferably of cast iron mounted on a cast iron frame and closed by means of four cams against a solid rubber gaskets to make the door water and air tight. A glass panel of not less than 9 by 12 inches should be provided inthe doors for observations of sprays. The door frame should be provided with a lip on the outside to catch any drip and return it to the washer for any reason, the door is not firmly clamped. A water proof marine type light fixture to take a standard type electric light bulb should be provided at the top of the washer for inspection and observations.

2.1.4 DIFFUSER PLATES :-

The diffuser plate consisting of a series of galvanized sheet iron louvers at the front of the air washer are often though not always, installed. These are so designed that the atmoized water will be entirely retained within the spray chamber, even if the fan is not in operation, and it serves also for distributing the air uniformly and maintaining equal velocities through the spray chamber. They also prevent spray water from wetting the floor and walls ahead of the washer.

2.1.5 SPRAY NOZZLES AND SPRAY PIPING :-

Spray nozzles are designed to produce a finely atmoized spray and are ~~such~~ So spaced as to give uniform coverage through which the air must pass and thus completely filling the spray chamber. Spacing of the spray nozzles varies from about .75 to 2.5 nozzles per square foot per bank. For poor spray densities it is recommended that the small orifice should be used for avoiding bypassing of air because of a poor spray coverage. Nozzles used for air washers ordinarily have a capacity of approximately 1 to 2 gpm per nozzle. The pressure required to force this quantity of water through these is of the order of 20 to 40 psig. Small orifices at pressures upto 40 psig produce a fine spray necessary for high humidifying efficiencies while the larger orifices at pressures around 25 psig are common for dehumidification. Nozzles of the self cleaning type have recently become available. In a single bank spray washer about 5 gpm of water per 1000 cfm is usually provided. In a two bank air washer twice as much water can be supplied in approximately 10 gpm per 1000 cfm and 15 gpm per 1500 cfm in a three bank washer. The quantity of water delivered by each bank can be varied by installing a larger or smaller number of nozzles.

The method used for breaking up the water into very fine spray varies with the make of the air washer, but a common method is to provide a small chamber in the nozzle casing which gives the water a whirling or centrifugal action which increases as the water approaches the point of discharge, and produces a very finely divided spray as the water leaves the nozzles. The nozzle should be designed to make it free from clogging with foreign material and the smallest opening should be larger than the openings in the suction strainer. The nozzles are constructed of brass and are evenly spaced over the cross section of the washer to give a uniform distribution of water and are set to discharge water along or against the direction of flow of air. About 3 nozzles should be provided per 1000 cuft of air per minute passing through the air washer . At the nozzles a gage pressure not exceeding as a rule 25 lbs/square inch is required to discharge 1½ gallons per minute through each nozzle.

The spray nozzles are screwed into vertical risers of galvanized steel pipe which in turn should be screwed into a horizontal galvanized cast iron header. In the case of a washer which is of less than 8 feet height, 1 in. riser should be used, and for a washer over 8 feet in height, 1¼ in risers . The tops of the risers are sometimes provided with a special cap for securing to guide rail at the top.

2.1.6 <u>CAPACITY OF AIR WASHER</u> :-

The cross sectional area of the washer is determinied by the design velocity of the air through the spray chamber. Washers are commonly available from 2000 to 250,000 cfm. capacity, however, there is no practical limit to sizes specially constructed. The gross cross sectional area of air washers (the area perpendicular to the direction of the air flow) is usually based on an air velocity of 500 fpm. If higher velocities are used trouble may be experienced with entrained moisture being carried out of the washer. The capacity of air washer is determined by multiplying the velocity by the area of cross-section, the commonly adopted air velocity for rating of air washers being 500 fpm. Thus an air washer having a gross cross sectional area of 20 sq.ft. would have a capacity of 10,000 cfm.

Knowing the cross sectional area the height and width of the air washer can be known. Thus a air washer having a cross sectional area of 24 sq.ft. can have its height and width as 4 feet and 6 feet respectively. If the tank is 18 inches deep the total height is then equal to 5 ft. 6 inches. From the economic point of view the washer having equal height and width is preferred although these proportions are not necessary. The spacing between the two consequetive spray banks is between 2½ to 4½ ft. The distance between the first bank and the entering end of the washer is about one feet and a distance of 1½ feet is provided between the last spray bank and the leaving end of the washer. If the air washer is furnished with the heating and cooling coil within the chamber, the overall length of the washer is changed.

2.1.7 ELIMINATOR PLATES :-

The properly designed eliminator plates give several advantages :-

1. It eliminates the small particles of water that may be carried away with the air.

2. It offers the minimum resistance to the air.

3. It provides the wetting surface as large as possible to obtain the best cleaning effects.

4. Simplicity of construction so that these plates may easily be taken out for inspection, repairs and painting

The eliminator plates are so designed and installed that the direction of the air current changes abruptly in order to throw out the entrained moisture against the eliminator plates. The direction of air current changes about 6 times as it passes through them. The wider the space between the plates, the greater must be the necessary angle through which the air must be baffled. The eliminator plates spaced 1-1/8 inch apart and having an angle of deflection of 60°, removes the particles of water more effectively than a 90° angle with a 4 to 6 in. spacing of the eliminator plates.

The resistance in the eliminator plates depends more on the angle through which the air is baffled and the eddy currents that the plates produce than upon the surface friction. As a matter of fact the resistance of this surface friction is so small that it is difficult to measure it. The eliminator plates can be arranged vertically or horizontally depending upon the airwasher that is used.

In the carrier air washer the eliminator plates form an integral part with the scrubber plates. These consist of series of corrugated galvanized iron sheets spaced 1-1/8" apart, across the discharge end of the spray chamber. Each eliminator plate is made of single sheet stamped with six corrugations, the last three corners having projecting lips or gutters which remove entrained moisture from the air. The plates are assembled in an iron frame attached by clips to the sides of the air washer casing. The frame is provided with slots into which the edges of the eliminator plates are readily slipped in and out. Some designs of air washers use horizontal eliminator plates consisting of corrugated iron sheets with 4 currugations in each plate. The plates are punched to provide a series of projecting lips which cut the air into a multitude of small streams, thus bringing all of it into contact with the wet surfaces upon which any dirt remaining in the air is deposited.

In addition to removal of moisture, the eliminator plates also serve the purpose ~~as~~ of cleaning surfaces for removal of dust which impinges on them. Eliminator flooding nozzles are provided to keep a considerable portion of the eliminator plated flooded with a stream of water. These nozzles are of the rain spray type which do not break up the particles into fine mist but direct it in the form of constant shower over the plates. When comparing the amount of eliminator plate surface in different air washers, only one side of the plate should be considered and there should be about 20 sq.ft. of this surface for 1000 cu.ft. of air per minute passing through the air washer. Nozzles should be provided to supply, about 5 gal. of water per minute for each foot of width of the eliminator plates. The flooding nozzles are installed on a headeron three inch centres and have a capacity of approximately 1.0 gpm per nozzle with a pressure ranging between 3 to 5 psi. The eliminator plates are usually made up of galvanized sheet iron of 24 gage, although they are occassionally built of copper or other non corrosive metals.

2.1.8 CIRCULATING WATER PUMP :-

A centrifugal pump should be provided for circulation of water through the nozzles of the spray chamber. The pump having double suction intake is preferred. Although this type is more expensive than a pump with single suction intake connection it exerts no end thrust on the bearing of an electric motor, driving the pump. The pump should have a bronze impeller, bronze covered steel shaft and bronze bushing to ensure maximum wear and service. Provision should be made for priming the pump with water from a supply distributed by pressure as from city water mains, an air cock ~~should be~~ being provided at the top of the pump casing for the removal of air from the casing when priming. The pump should be directly connected through a flexible coupling to a suitable electric motor mounted on the same base. The base should be rigidly constructed to prevent distortion when handling. The size of the motor should be correctly determined so that there is no danger of the motor being overloaded. Since small centrifugal pumps have a low efficiency and the air washer pump must run continuously during the operation of an air washer, the service conditions are severe.

2.2 CELL TYPE WASHERS :

The intimate contact of air and water is obtained by passing the air through cells packed with glass, metal or fibre screens. The cells are arranged in tiers over which the water is distributed. They are followed by conventional blade type or glass mat eliminators. Generally the cell type washers are arranged for counter current air and water flow. Cell washers are made in a range of sizes in either insulated or uninsulated construction. Standard washers are available upto 10 cells high by 12 cells wide with a capacity of 130,000 cfm. They are also made in a unitary apparatus up to 33,000 cfm complete with fan, motor pump and external spray piping.

The atomization of the spray water is not required for cell type washers. But good water distribution over the faces of the cells is essential. The quantity of water required when the air is nearing saturation is very low. The saturation effectiveness of 90 to 97% is obtained. Water requirements vary from 2¼ to 4 gpm per 1000 cfm with resistance to air flow ranging from 0.15 to 0.65 in depending upon the water circulated and the air velocity through the cells.

Each washer consists of a number of cells normally 20 inch square arranged in tiers. A typical cell consists of a metal frame packed with glass fiber strands. The glass occupies only 3 to 5 % of volume of the cell but represents a total wetted area when sprayed of approximately 100 to 125 sq.ft. Wire mesh screens are provided at both faces of the cell frame to retain the glass pack. Each tier is independant of the other and has its own spray header.

Either a flat or sloping bottom is provided in the tank which serves as a reservoir for circulating water and for dirt flushed from the cells. The sprays consist of a pipe header with nozzles with each tier of cells. These are usually mani-folded externally with a valve and pressure gage in connection to each header. The pressure required for the nozzles is about 6 psig and delivers 2 gpm. Eliminators are provided down stream from the cells to remove entrained moisture from the air stream. They may be of metal blade type providing 30° deflection of the air stream with hooks for tapping impinged droplets or glass mat type arranged in tiers similar to the cell arrangement. The glass mat type is 2" deep and is packed with glass fibre in a metal frame.

Connections are provided for internal spray headers drain, overflow, steam injectors for cleaning the cells in place, pump suction, water quick fill and make up water controlled by a ~~flat~~ float valve. A typical cell type air washer is as shown in Fig. 2.3.

The cell type washers are efficient air cleaners. They remove from 70 to 90 percent by weight of the air borne solid matter, which includes most particles exceeding 5 microns in size and many down to 1 micron. However, they should not be

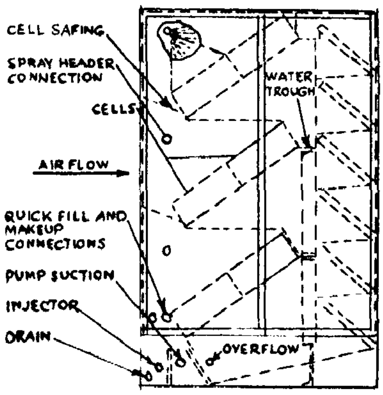

FIG. 2.3
Typical Cell-Type Air Washer

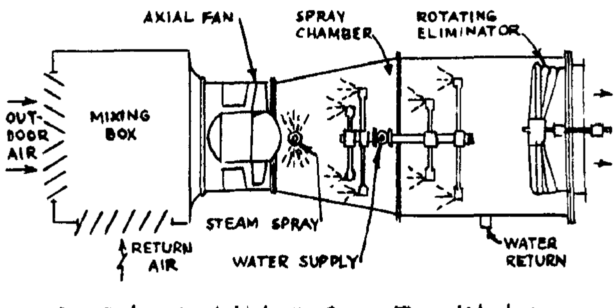

Fig. 2.4 High Velocity Spray-Type Washer

used in cotton mills and other installations where large volumes of fibrous or linty mate/rial are present in the air, unless it is filtered out completely ahead of the washer.

In these type of washer it is necessary to clean the glass media. A differential draft gage to measure the air resistance across the cells can be used to determine the frequency of this cleaning. Mat type eliminators in these washers should be removed and cleaned with suitable detergent solution. Any glass mats that have been eroded by the sprays should be replaced.

2.3 HIGH VELOCITY SPRA-Y TYPE WASHER :-

Recently a high velocity type of air washer has come into use which employs a rotating eliminator. The unit operates on the same general principles as the conventional spray type washer. However, the casing is cylinderical and the eliminator consists of a revolving assembly of sheet metal blades extending radially from a cylinderical hub as shown in Fig. 2.4.

The higher air velocities permissible with the rotating design reduces the cross sectional area of the unit while a high degree of efficiency is obtained by use of more spray banks and a deeper spray path. Velocities as high as 2400 fpm have been employed.

High velocity washers with axial flow fans designed for 1" external pressure are presently built in sizes from 10,000 to 45,000 cfm. They are furnished with 1 to 6 spray banks, making possible the humidifying efficiency as large as 98%. The pressure that is commonly recommended is between 30 and 40 psig, with a nominal water flow ranging from 80 to 400 gpm, depending upon the size of air washer. The high velocity air drives the rotating eliminator which discharges moisture by centrifugal force so that the water and its entrained material drain from the unit into a central storage tank. Return and outdoor air dampers are built into this unitary design of washer. The use of high velocity of the order of 4 to 5 times those of conventional washers substantially reduces space and weight requirements.

2.4 SPECIAL WASHERS :-

The problem of air washing are many and varied, particularly in the industrial field and the experience of air washer manufacturers has shown that there are many places where variations from the standard designs of air washers can be used to advantage. In some cases where the main purpose of the washer is the elimination and recovery of valuable fine dust, the spray nozzles have been omitted and a second and sometimes a third set of eliminators spaced on 3/4 inch centres have been used. Each group of eliminator plates are provided with flooding nozzles in order to flood the surface thoroughly. A gage pressure of only 5 lbs/square inch is required on the flooding headers, a considerable saving power for pumping is affected.

In some cases where the dust content of the air is small, and where it is desirable to humidify the air as little as possible, a single row of closely spaced properly flooded scrubber plates is satisfactory and the investigations show that this arrangement will do 90 percent of the cleaning that can be done with a completely equipped air washer. The increase in humidity under ordinary conditions will amount to the equivalent of the absorption of 1/3 grain of water per cubic foot of air.

When the primary purpose is the cleaning of air ample surface of the scrubber plates is the most important consideration, and when cooling or complete saturation of the air is required, thorough spraying of the air and a sufficient length of the travel through the spray chamber are equally important factors.

2.5 AIR WASHER TESTS :-

The performance test of an airwasher include the determination of :-

1. Capacity
2. Resistance of the air washer,
3. Humidifying Efficiency,
4. Visible entrainment of free moisture,
5. Cleaning effect.

Capacity and resistance determinations of an air washer are made by tests in accordance with the standard Code of Practice of American Society of Heating and Ventilation

Engineers.

Visible entrainment is observed by means of a mirror not over 6 inches square held at right angle to the air flow and at a distance not over 6 inches from the outlet end of the eliminator plates.

The humidifying efficiency is calculated by the average readings of the dry bulb and wet bulb temperatures taken at air washer inlet and outlet.

The cleaning efficiency of an air washer is generally based on factory tests where the nature of the dust and average size and density of dust particles is under the control of the testers and where the air washer can be operated at exactly its rated capacity. Apart from the factory tests, other tests must be made at the rated capacity of the air washer and the dust density of 4 to 10 grains per 1000 cft of air uniformly distributed over the entire area of the inlet of the washer. It is desirable that a standard size dust is used preferably screened and are floated vacuum cleaner dust. The dust should be distributed in the room in a cloud by an air blast and carried by a current of air of which the velocity should not exceed 25 ft. per minute.

It should be observed that the temperature of air leaving the humidifier are taken very accurately compared to the inlet temperature of air. This is specially important when the air is near the saturated state. The mirror should be exposed to the test air for considerable time before the readings are taken. Conditions may not be uniform owing to clogged nozzles, air bound tempering coils or unequal room temperatures.

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CHAPTER - III.

EVAPORATIVE COOLING.

3.1 Principle of Evaporative Cooling.

3.2 Evaporation from surface.

3.3 Thermodynamic changes between air and water.

3.4 Efficiency of an evaporative cooler.

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CHAPTER - III.

EVAPORATIVE COOLING:

3.1 PRINCIPLE OF EVAPORATIVE COOLING :-

The process of cooling air by evaporation of water can be considered as adiabatic since the construction of the cooler is such that no appreciable amount of heat is transferred to or from the surroundings. Heat energy is required to evaporate the liquid and in the case of adiabatic cooling sensible heat flows from air to water, lowering the dry bulb temperature, thereby supplying the latent heat of evaporation which in turn increases the humidity. Thus a transfer from sensible to latent heat has taken place with no change of enthalpy of air vapour mixture. Being adiabatic the process follows a contact enthalpy line on the psychrometric chart, but there is no serious error involved in charting it along a constant wet bulb line. Since then the evaporative cooling process follows the constant wet bulb line on the psychrometric chart, the entering wet bulb temperature determines the lowest dry bulb temperature at which the air can be supplied. The degree of saturation that is obtained and the reduction in dry bulb temperature depends upon the type of saturating device used. In some cases it is desired that the air may be 100 percent saturated while in other cases, the degree of saturation required may be limited to best serve the design requirements.

3.2 EVAPORATION FROM SURFACE :-

Evaporation from the surface can be best illustrated from the concept of mass transfer. When there is an exchange of mass between a solid or liquid surface and a moving fluid, the transfer is considered to take place by convection. If the fluid motion is a forced motion, the process is said to be forced convection, and it is called a free convection if the fluid motion is due to gravity.

The theory of mass transfer by convection is analogous to the theory of heat transfer by convection. Consider an example of a heat transfer from a flat surface to a moving air stream. There being a film of hot air adjacent to the hot surface, some of the air from this film is picked up and transporated by the air stream. In the case of evaporation from a wetted surface the process is similar. There is a film of saturated air at the surface and some of the saturated air is picked up by the moving air stream.

The rate of heat transferred is usually expressed as

$$q = KV^{2/3} A \Delta t$$

where q = Rate of heat transfer B.T.U./hr.

V = Air velocity,

A = Area in sq.ft.

Δt = Temp. difference in deg. F.

K = Contant.

For evaporation from a similar wetted surface the formula for mass transfer is ;

$$\text{Mass transfer per unit time} = KV^{2/3}\, A \times (\text{vapor pressure difference})$$

The two formulae are similar.

We see that both mass transfer and heat transfer vary approximately as 2/3 power of the air velocity. In the case of heat transfer difference in temperature is the driving force while in case of mass transfer the driving force is the vapour pressure difference.

There are so many other processes in which there is a simultaneous transfer of heat and mass between water and air. In such cases both the water and air change temperature during the process. This is true for evaporative condensers, cooling tower and for air washers in which either the water is heated or chilled external to the spray chamber.

3.3 THERMODYNAMIC CHANGES BETWEEN AIR AND WATER :-

1. In figure 3.1, the water is constantly re-circulated so that it soon reaches the equilibrium with the thermodynamic wet bulb temperature, of the entering air, both water temperature and the wet bulb temperature of the air being the same.

The water temperature remaining constant, lowers the drybulb temperature of the entering air so that it approaches the

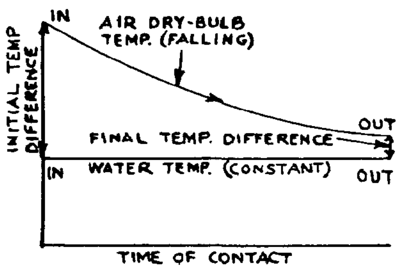

FIG. 3·1

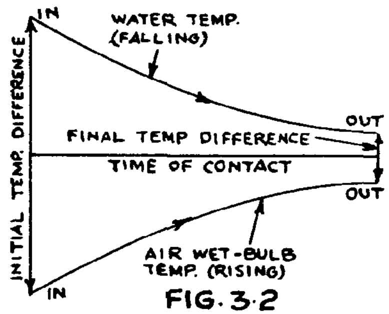

FIG. 3·2

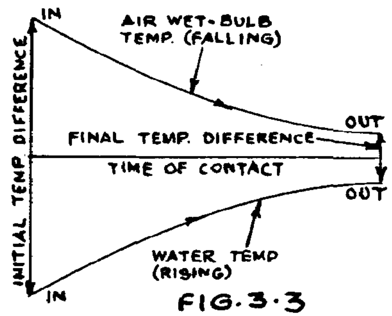

FIG. 3·3

Interaction of Air and Water in Evaporative Air Cooling water temperature. This being an adiabatic process, the wet bulb temperature remains constant but as the humidity is increased, the dbt of air falls.

2. If the heated water is supplied to the apparatus, the air will be heated. The water temperature is lowered as the wet bulb temperature of the air is increased. This is illustrated as shown in Fig. 3.2. In such a process in which heating and humidification take place, the final wetbulb temperature approaches the final water température.

3. If instead chilled water is supplied to the apparatus, the cold water will be heated up as the wet bulb temperature ofthe air is lowered as shown in Fig. 3.3. In such a cooling process the final wet bulb temperature of the air will always be higher than the final water temperature. When the spray water temperature is lower than the dew point temperature of the incoming air, the air will be both cooled and dehumidified. The lower the spray water temperature, the greater will be the condensation of moisture from the air and greater will be the latent heat removal.

3.4 EFFICIENCY OF AN EVAPORATIVE COOLER :-

To evaluate the performance of an evaporative cooler the terms saturating effective-ness, or evaporative cooling efficiency or humidifying efficiency are used. This relationship can be defined as the ratio of the actual lowering of air temperature to the wet bulb depression - the difference between the entering dry and wet bulb temperatures.

The following formula for saturating efficiency is given :-

$$E = \frac{ta - t1}{t_a - t_a'}$$

where E = Saturating efficiency,

t_a = entering air drybulb temperature.

t_1 = leaving air drybulb temperature

t_a' = entering air wetbulb temperature.

The above equation does not apply when there is cooling with dehumidification. It is only applicable to an evaporative cooling process in which the wet bulb temperature remains constant and spray water is neither heated nor chilled.

The saturating and heat transfer efficiencies of air washers are dependant upon the following main factors :-

1. Air Velocity,
2. Water to air ratio.
3. Spray nozzle design and pressure (fineness of atmoization).
4. No. of spray banks and no. of nozzles.
5. Entering air drybulb and wet bulb temperatures.
6. Entering water temperature and water temperature range.

3.5 TYPES OF EVAPORATIVE COOLING :-

.3.5.1 DIRECT EVAPORATIVE COOLING :-

This will be discussed for outdoor air entering the cooler at one dry bulb temperature 95 F but two different wet bulb temperatures (65 F and 75 Deg. F) as shown in psychrometric chart of fig. 3.4. The air is brought into intimate contact with water in a number of ways, one way being by the drip type evaporative cooler as shown in Fig. 3.5. As the process is plotted along the wet bulb line it tends to become saturated. The final condition of the air depends upon the evaporative or saturating efficiency of the device used. In this case, we have assumed the efficiency to be 80 percent.

The cooling load, the quantity of air delivered by the cooler, the dry bulb temperarure of the cooler outlet, determine the conditions of the air leaving the cooled space, or the air flow rate is determined from the sensible cooling load and allowable temperature rise of supply air in the space, if certain design conditions are to be met. If 6^{o} F temperature rise is assumed in the space for the conditions selected as in Fig.3.4, this will result in a room temperature of 85^{o} F when the cooler entering W.B.T. is 75 and a room temperature of 77 deg. F when the entering wet bulb temperature is 65 deg. F. The air flow rate per ton of sensible cooling is 1900 and 1755 cfm respectively. It is evident that air flow rates are higher when evaporative cooling is employed as compared to

DIRECT EVAPORATIVE AIR COOLING.

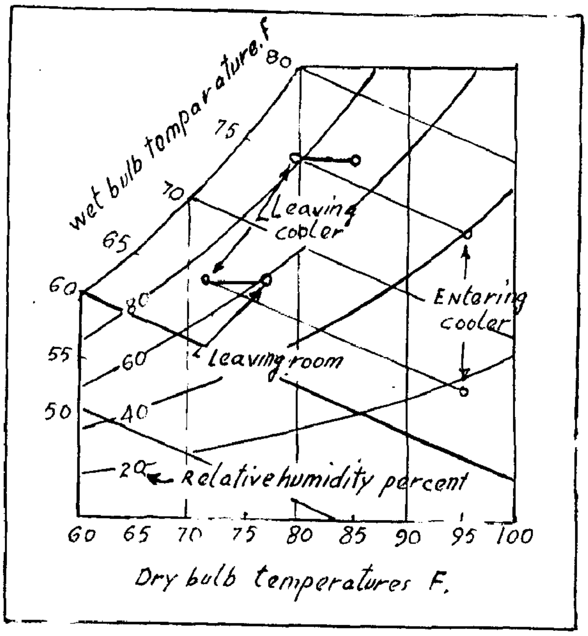

FIG- 3.4

DRIP TYPE EVAPORATIVE COOLER

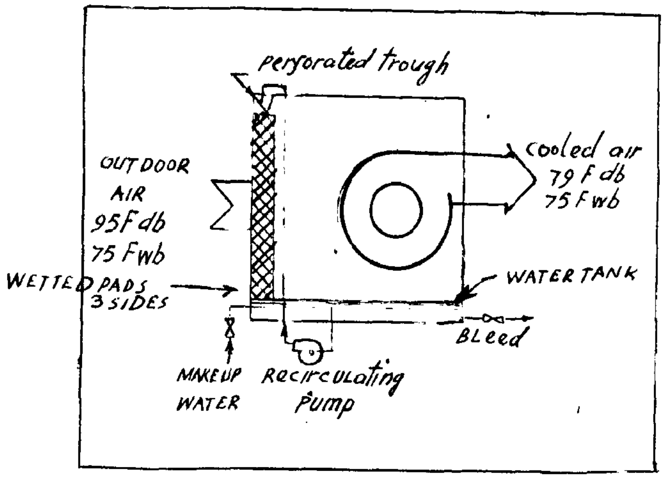

FIG- 3.5

refrigerated air conditioning. Since there is no dehumidification the internal latent load does not in any way effect the sizing of the evaporative cooling device. Moreover the evaporative cooling system uses 100 percent outside air.

3.5.2 TWO STAGE COOLING :-

For the same two entering conditions as in Fig.3.4 the condition line is plotted on the psychrometric chart for a two stage evaporative cooling process . Two stage evaporative air cooling involves sensible cooling in first stage and adiabatic saturation in second stage as shown in Fig. 3.6. Indirect use of evaporative cooling in first stage permits delivery of supply air at a wet bulb temperature below that of entering air. In the Fig. 3.7, cooling tower and water ~~tk~~ to air heat exchanger followed by a direct evaporative cooler are shown. Inthe indirect stage it is assumed that the temperature of outdoor supply air is reduced to within 8^{o} F of the outdoor wet bulb temperature by employing a cooling tower and a dry extended surface coil. The reduction will require an 8 row coil operated in counterflow. In the first stage, the evaporative cooling effect is used to cool water circulated through the coils in the supply air stream. The supply air stream at a reduced dry and wet bulb temperature is then taken through a direct evaporative cooler having a saturation efficiency of 80% in this example. Comparing the two cases, it is seen that in the indirect system, the reduction in dry and wet bulb temperatures is much more. Assuming that the temperature

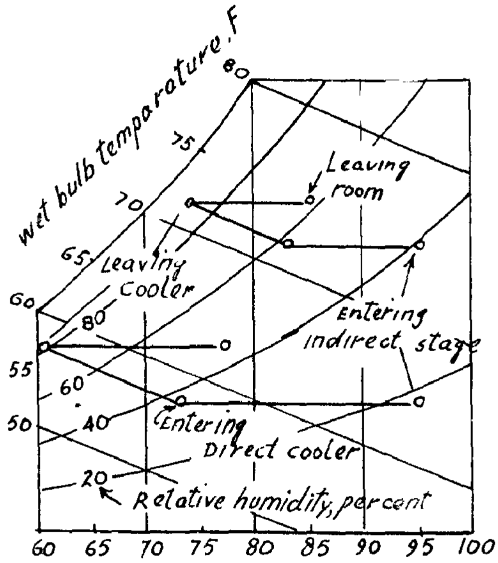

FIG-3.6 Dry bulb temperature, F

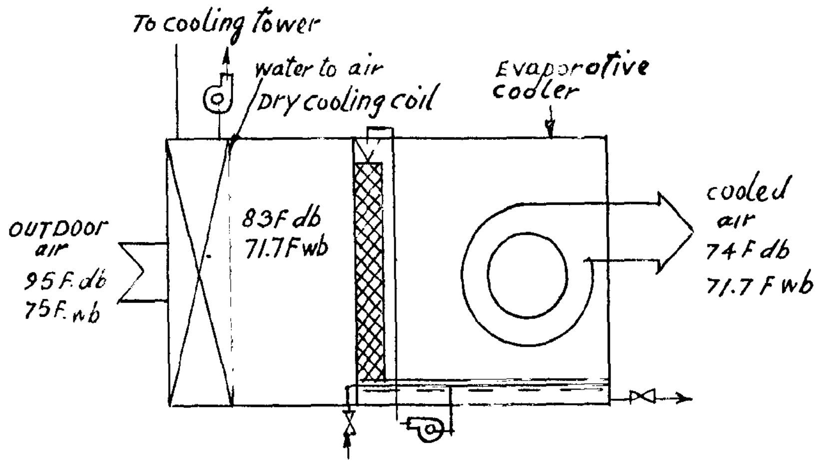

FIG- 3.7

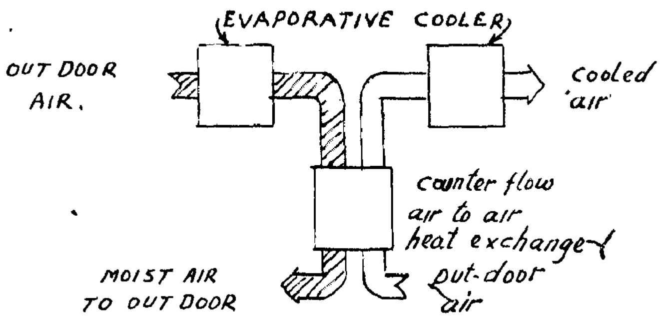

FIG- 3.8

leaving the cooled space is same i.e. 85 deg. F for W.B.T. of 75°F and 77° F for W.B.T. of 65° F, the air flow rates for each ton of sensible cooling comes out to be 1050 and 665 cfm respectively. This is in contrast with 1900 and 1755 cfm in the case of direct cooling. The dew point temperature and relative humidity in the space are lower in both the cases. The reduced air flow rates and improvements in leaving conditions are obtained at the expense of the additional equipment i.e. cooling tower, coil and piping system, required to accomplish the first stage of cooling. If the air is to be cooled approximately to the dew point temperature, other stages can be added, however, diminishing results are accomplished in the succeeding stages. From the economic point of view, one indirect and one direct stage is used.

3.5.3 MULTI-STAGE EVAPORATIVE COOLING :-

As stated above there can be more than two stages of cooling to reduce the wet bulb temperature of the air delivered from evaporative cooling system and thus increasing the overall efficiency. In the previous example, the first stage cooling was done by means of a cooling tower and water to air heat exchanger, which was followed by a direct cooling system.

Another type of stage cooling utilizes two air-washers and an air to air heat exchanger. As shown in Fig.3.8, the outside air is first drawn through an air washer which then evaporatively cools it. The moist cool air is then passed through another air to air heat exchanger which cools another stream of outside air. The moist air after picking heat is discharged to atmosphere. The dry cooled outside air is then taken through a second air washer where it is evaporatively cooled and then delivered to conditioned space. The drawback of this type of stage cooling is that since air to air heat exchanger is used instead of water to air heat exchanger, which results in a lower heat transfer rate. This requires a greater heat transfer area. Consequently, the dry cooling state is designed for 12^{0} F approach to the moist air temperature rather than 8 deg. F. approach to the entering wet bulb temperature. In areas where the wet bulb temperature is high, this method of cooling is less efficient.

Another type of multistage indirect cooling system replaces the air to air heat exchanger with a water to air heat exchanger producing the same results in the conditioned space. When one side of the surface is wet a smaller heat exchanger can be used due to better heat transfer.

3.5.4. THE REGENERATIVE DRY COOLING SYSTEM :-

This system bypasses the cooled air withdrawn from the conditioned space through the spray chamber instead ofthe outside air. This will result in lowering the spray water temperature to within 4 deg. F of the room wet bulb rather than the wet bulb temperature of the outside air. Air is drawn from the room by a fan and is discharged through a spray chamber, cooling water which is circulated by a pump through an extended surface heat exchanger. Outside air is drawn by another fan which after flowing over the coils is delivered to the room.

3.5.5 PLATE TYPE HEAT EXCHANGER SYSTEM :-

This is a small variation from the previous system described. In this system, the cool return air from the conditioned space is passed through a spray chamber and then passed through a large plate type air to air heat exchanger. In this exchanger the outside air is cooled by conduction and is then delivered to the conditioned room. The disadvantage of this system is the bulk and high cost.

3.5.6 COMBINATION OF EVAPORATIVE COOLING WITH REFRIGERATION :-

Indirect evaporative cooling may be used effectively with precooling with refrigeration.

3.6.4. MAINTENANCE OF EVAPORATIVE AIR COOLERS :-

The condition of the evaporative cooler is the main factor in effecting the efficiency of these types of coolers. If the water used is hard thereby forcing scale, it will clog the pad and consequently the efficiency will drop in proportion to the degree in evaporation and airflow. The use of a bleed off is therefore, essential if minimum maintenance and good performance are to be achieved. A portion of the sump water is continually drawn off. This reduces themineral concentration thus reducing scale deposits.

A bleed off rate for evaporative coolers may be determined by the fllowing equation :-

$$B = \frac{C_1}{C_2 - C_1} \times E$$

where B = Bleed off rate gallons per hour.

C_1 = Hardness of supply water grains per gallon.

C_2 = Hardness of sump water grains per gallon.

E = Rate of evaporation gallons, per hour.

If the bleeding off rate is equivalent to one half of the amount of water evaporated, satisfactory control of scaling is obtained. This is approximately 1 gallon per hour per 1000 cfu.

3.7 EVAPORATIVE COOLING AND COMFORT :-

3.7.1 GENERAL :-

Both dry bulb temperature and relative humidity have an effect on the comfort of an individual. There are various combination of these two properties which result in equally desirable conditions for comfort. The American Society of Heating and Ventilating Engineers have published data showing the comfort zone super-imposed on the psychrometric chart. From this chart it can be known whether the final condition that is obtained by cooling falls within the limits of physiological comfort. Evaporative cooling has been found to be much useful in maintaining conditions that fall well within these limits.

The increase in relative humidity is the major drawback which prohibits the use of evaporative cooling systems in some areas. To predict whether evaporative cooling is feasible in certain areas, actually it is more practicable to consider wet bulb design temperature instead of relative humidity. Since the wet bulb temperature governs the leaving air temperature from the evaporative cooler, a low wet bulb assures a low conditioned temperature. For example, near the Gulf of Mexico the relative humidity is low, but the wet bulb temperature are high ranging from 80° F to 85° F. Although, the dry bulb temperature near these places is about 120 to 125 deg. F, the net coolingproduced can not bring the conditioned air to the comfort zone. In contrast, some of the intermountain states where the wet bulb design temperatures are of the order of 64 to 68 deg.F. make evaporative cooling ideal.

The comfort level of the room conditions that can be maintained must be predicted in evaluating whether evaporative cooling can be used. It is, however, difficult to relate the environmental factors of dry bulb temperature, wet bulb temperature, or relative humidity, room air motion and mean radiant temperature or the average temperature of the enclosing surface on individual comfort. The American Society of Heating and Airconditioning engineers have given the effective temperature which is at present the only index available for correlating the first three of these variables to determine environments of equal comfort or equal discomfort. Dr. Bedford has modified the effective temperature chart to include the effects of mean radiant temperature as indicated by a globe thermometer. Thus this chart is very useful in comfort cooling installation as it gives all four factors of concern necessary for comfort.

Due to the changes that have occured in standard of living, dress etc., the comfort chart is under revision. Recent studies have shown that below dry bulb temperature of 80° F, effective temperature over emphasizes the effects of relative humidity, while above this temperature, effective temperature, more nearly represents the effects of relative humidity,. For this reason, also it is seen that the conditions at which most sedentary are comfortable fall within a dry bulb temperature range from 73 to 77 deg. F. and relative humidity from 25 to 60 percent, when room air velocities do not exceed 25 to 35 fpm. In a qualitative way these limits also take in to account the effect of mean radiant temperature. The upper limit i.e. drybulb temperature of 77 def. F and 60%~~xxgxxRxp~~

relative humidity correspond to an effective temperature of 73^{o}F.

3.7.2 AIR MOTION FOR COMFORT :-

If the same effective temperature is desired, the other conditions of dry bulb, wet bulb temperature and air - velocity can be obtained from the comfort chart. Selected combinations of these variables corresponding to an effective temperature of 73 deg. F are given in table 3.1.

TABLE 3.1

ROOM CONDITIONS PROVIDING EFFECTIVE TEMPERATURE OF 73^{o} F BASED UPON ASHAE EFFECTIVE TEMPERATURE CHART.

Dry bulb Temp.	Wet bulb temperature	relative humidity percent.	room air velocity F.P.M.
77	67	60	Less than 20
80	64.3	43	Less than 20
80	70	61	170
80	75	80	320
85	58	16	Less than 20
85	65	33	260
85	70	48	480

From the chart a significant thing is noted that as the wet and dry bulb temperatures are increased to maintain the effective temp. of 73 F, the room air motion increases appreciably. Thus the increased air motion can compensate if the dry and wet bulb temperatures are high. This is the main reason ~~by~~ why ~~we can use evaporative cooling effectively~~

for cooling of sedentary individuals in regions where the wet bulb temperature is moderately high. In some hot industry applications the target velocity (the velocity of the air impinging upon the worker) of more than 300 fpm has been used with success. But this is too high for the comfort cooling of sedentary individual. This shows that higher velocites can not always offset the higher temperatures. The recent investigation has shown that for higher temperatures and high humidity the air velocities of 120 fpm produce the best results. Velocities of 200 fpm to 300 fpm have been found to give some improvement on the comfort of sedentary individual.

3.7.3 EFFECTIVE TEMPERATURES :-

The higher limit of effective temperature that produces relief from sensible perspiration has been found to be 78 F. For most sedentary individuals the environments at this effective temperature is uncomfortably warm. But moderately active ~~in~~ or standing person would feel such conditions as comfortable or comfortably warm. In such cases evaporative cooling will be considered as suitable for relief cooling measure. Lower room air velocities are allowed if a higher design effective temperature is used or the higher design effective temperature would permit the applications of evaporative cooling in areas of higher wet bulb temperature. It should be noted that the conditions maintained would not be satisfactory for all the persons, but would provide some relief, In areas of low wet bulb temperature (65 to 70 F) the same room air velocities will be desirable for evaporative cooling systems as for refrigerated air conditioning.

3.7.4 RELIEF COOLING IN HOT INDUSTRY :-

The foregoing statements are for providing satisfactory summer conditions by the use of evaporative cooling for sedentaryindividuals or slightly active individuals. In hot industrial applications, the effective temperature is not applicable since the heat productionof the individual is greater The Belding and Hatch heat stress index affords one means of evaluating the effectiveness of evaporative cooling in providing relief cooling to minimize heat strains in hot areas. The improvement that can be made with evaporative cooling is illustrated by the following example.

Assume a plant located where the design dry bulb and wet bulb temperatures are 95 and 75 respectively. It is assumed that the mean temperature of the solid surroundings is 105 F. The workers are standing and doing moderate work at a machine or bench so that their matabolism is about 1000 BTU per hour. Suppose that a ventilation system is provided so that in one case outdoor air at 95 F (assuming no temperature rise in air) is directed on the worker. The wet bulb temperature of this air is 75 F. Evaporative cooling having a saturating efficiency of 80%, is then added. The condition of the air leaving the evaporative cooler will be 79 F dry bulb and 75 F wet bulb temperatures. Assuming a 6 F temperature rise before the air impinges upon the worker, the dry and wet bulb temperature of the air will be 85 F and 76.5 respectively. The Belding and Hatch heat stress index for these ~~gm~~ temperatures conditions and for several air velocities is given in Table 3.2

TABLE 3.2

Heat Stress Indexes computed from Belding and Hatch data for an assumed industrial spot cooling problem for ventilation with outdoor air and with outdoor air evaporatively cooled, showing improvements made with evaporative cooling.

Mean Temp. of surrounding	Globe Thermometer reading	Supply air temp.		Velocity of air on worker fpm	Heat stress index	Comment.
		Dry bulb F	Wet bulb F			
106	100	95	75	100	82	Very severe strain only small %age of workers qualified for work
106	99	95	75	300	60	Severe heat strain, some decrease in the performance not suitable where mental effort is required.
106	97.5	95	75	600	50	Same as for heat stress index 60 but not as severe
106	93	85	76.5	100	65	Severe to very severe strain
106	91	85	76.5	300	37	Moderate to severe strain. Some decrease in the performance in physical jobs expected.
106	89.5	85	76.5	600	28	Mild to moderate strain. Some decrease in performance in jobs involving higher intellectual functions.

Comparing the heat stress indexes at the highest velocity considered (600 fpm) shows the improvement made by evaporative cooling. When there is no cooling and the air velocity is 600 fpm the heat stress index is 50 which is defined as the severe heat strain. By employing evaporative cooling device the heat stress index is reduced to 28 which shows a mild strain only causing a slight decrease in performance. This shows the improvement done by employing evaporative cooling.

3.8 DESIGN OF EVAPORATIVE COOLING SYSTEMS :-

In the preceeding chapter evaporative cooling for the comfort of sedentary individual and relief cooling in hot industries have been discussed. For designing of evaporative cooling system, the system should be well engineered as is done in mechanically, refrigerated systems. The most common method of calculating the evaporative cooling system was to find the cubical content of the room and then arbitrarily assuming a 2, $2\frac{1}{2}$ or 3 minute air change. Equipment was selected on the basis of this air change.

3.8.1 EVAPORATIVE COOLING OF SPACE :-

Knowledge of the local wet bulb and dry bulb temperature is very important in the application of evaporative cooling systems. Conditioning of stores, small offices, churches, etc., by means of evaporative cooling requires a thorough knowledge of the history of wet and dry bulb temperatures of the locality. From this history, a design wet bulb temperature is selected. Once the design wet bulb temperature is selected, the condition ofthe air leaving the system can be known by assuming a certain value of saturating effectiveness. While selecting the temperature of the air leaving the cooled space it should be seen that the relative humidity does not exceed 60 to 70 percent, the air velocity is not excessive and that a suitable effective temperature is maintained.

The capacity of the evaporative cooling system is determined by the air flow rate and the saturating effectiveness. A central air washer system may be selected in some cases. When large air volumes are to be handled the attention for the noise generated in the duct system should be given. Inorder to reduce the temperature rise in the duct and reduce the duct size, the highest air velocities consistent with good practice is desirable. To provide uniform air motion the proper supply outlets are selected so that they discharge at the correct level.

Since 100 percent outside air is used and no recirculation of the room air is allowed, the important factor in designing the evaporative cooling system is to provide a provision for exhausting the same quantity of air as that delivered by the evaporative cooling system. This is very important otherwise the air flow rate will be reduced and the relative humidity will rise to an uncomfortable level. Exhaust can be of natural type or mechanical type.

The following table gives the air change rates that have been found to provide acceptable results with evaporative cooling for various wet bulb depressions.

TABLE - 3.3

Time required for air change, minute.	Approximate wet bulb depression
3	31.35
2	25.30
1½	20.25

These values do not take into account the variations in heat gain and thus are only useful as a guide.

3.8.2 INDUSTRIAL RELIEF COOLING :-

While evaporative cooling using all outside air finds its chief application in comfort cooling there are many uses ofit in industrial airconditioning. Evaporative cooling improves the working conditions or processes.

During the summer in the factory, even to obtain the conditions prevailing outdoors will require a huge quantity of outside air, because of a large internal load created inside the factory. Through evaporative cooling and using a reasonable air quantity the temperature can be reduced from 5 to 10 F below the temperatures prevailing outside.

In regions handling volatile and inflammable solvents evaporative cooling may be used to maintain sufficiently high humidity to prevent explosion. In printing, in which it is desirable to maintain proper humidity control to size the paper accurately, evaporative cooling can be effectively used.

3.8.3 SPOT RELIEF COOLING :-

Many industrial processes create localized hot areas which present exceeding difficult working conditions. Evaporative cooling applied at these hot spots where the workers are required to stand provides relief and improves the efficiency.

When cooling a relatively large area such as areas where casting and remelting are done, target velocities of 600 fpm are used. For spot cooling each man receives 2000 cfm or less with supply air velocities ranging from 400 to 1000 fpm. The natural air path is directed at the waist line at a point near the shoulder. This location was found to be most suitable after research. Workers objected to an overhead blast which impinges on the head or face. Some manual control is provided. By means of adjustable outlets the worker if desired directs any part of the air on his head or shoulders.

When the cooling is done at the hot areas maximum saturation of the air is recommended. The temperature rise in the ducts is minimized by the use of high duct velocities. Velocities of 3000 fpm have been used. This is permissible seeing the average noise level in the plants. In extremely hot temperatures the duct material should be such that it has low emissivity.

3.9 APPLICATIONS OF EVAPORATIVE COOLING :-

3.9.1 LAUNDRIES AND DRY CLEANING :-

Evaporative cooling of laundries and dry cleaning establishments, requires the same precautions as given in industrial relief cooling. If the ceiling height is more spot cooling is usually employed. The air flow rates vary from 500 fpm to 1000 fpm per worker. Target air velocities of 600 fpm are used. The supply velocities range from 1000 to 1200 fpm.

In areas where the ceiling height is low and the number of workers is high, general space cooling may be employed.

Mechanical exhaust is preferred to natural exhaust to prevent possible increase of humidity.

3.9.2 GREEN HOUSE COOLING :-

Proper regulation of green house temperatures during the summer is essential for developing high quality crops. Plant growth is affected by the temperature because it influences the processes which occur within the plant.

Uptill now the green house temperatures were controlled by natural ventilation and the application of shading devices to protect from the solar heat. A properly sized and installed evaporative cooling reduces the temperature 10 to 15 F below the outside temperature in summer without the use of any shading devices. Even if the natural ventilation system is very effective and the air is moving at rate equal to one air change each minute, temperature cannot be held down, to satisfactory level and will range between 5 to 15 F above the outside temperature. Evaporative cooling improves and increases the productivity and reduces material and labour costs for the following basic reasons :-

1. It lowers summer temperatures by 15 to 30 F.
2. It improves the humidity conditions.
3. It reduces the need for watering the plants.
4. It eliminates stagnant green house air.
5. It provides cooler and more comfortable working conditions.
6. It eliminates the use of shading devices except for light control.
7. It reduces insects, bacterial, dust etc.
8. It keeps plants on schedule.

Uniform air motion is important because all surfaces of the plant are to be cooled. Velocities should be held as low as possible to prevent the damage ~~of~~ to the plant. Velocities from 20 to 50 fpm are optimum. To compute the heat gain for evaporative cooling, the solar heat gain is the most important to be considered. Of the total solar heat gain 50 percent must be removed by cooling.

The evaporative system consists of aspen wood pads on one side of the house. The exhaust fan is located on the other side so that the air is drawn accross the beds of the plant. Pads are wetted by troughs at the top. Water is supplied to the pads at the rate of 1/3 gpm per linear foot of the pad. The air velocity through the pads is about 123 fpm. The pads are such that they can be either covered in winter or can be replaced by glass. With this system 90 F are maintained in some New England installations.

Other uses of evaporative cooling are the conditioning of common storage of fruits and vegetables, animal shelters and dairy houses.

3.9.3. INDUSTRIAL APPLICATIONS :-

The field of application of evaporative cooling is much more vide in relief cooling of industrial places than the comfort cooling in office spaces or other places where the persons are le ss active. This is also applicable as a process cooling device where comfort is of secondary importance.

The industrial applications include relief cooling of individuals in various metal working operations, the rubber industry around tire process and mills, laundry and dry cleaning establishments, the textile industry and other industrial operations in which the heat gain from the process is the primary heat source.

Process cooling applications would include cooling of parts leaving paint and enamelling ovens, hot extruded products and similar processes.

Generators, motors and transformers may be cooled directly by blowing cool air through the windings. The air-washer provides both dust free and cool air for this application.

There are two general methods of air-washer application for generator cooling termed as the open and closed systems.

3.10 CONTROL OF EVAPORATIVE COOLING SYSTEMS :-

An evaporative cooling system should be designed so that it can be used for straight ventilation only at the beginning and end of the cooling season. Thus full use of the cooler is obtained when the outside temperature is high. It gives good ventilation when no cooling is required. The most effective method of control which has proved most practical is a thermostat that turns the complete unit off when the conditioned space becomes too cool. It also controls the blower and water supply on changes in room dry bulb temperature. Modulating dampers are also preferred in some designs. In these, the fan is stopped at a predetermined minimum supply temperature. Since the evaporative cooling can be used for ventilation, automatic control can be used to shut down the water supply during mild weather and at nights when evaporative cooling is not needed. When several units are used in parallel, the control can be arranged to cut off one or more coolers to modulate air temperature.

Two stage evaporative cooling system can also be controlled. A two stage dry bulb thermostat first starts the direct evaporative cooling stage. At a preset higher temperature the dry cooling stage is included to provide maximum cooling.

Humidity control can be supplied where it is desired so that in case the humidity rises too high, the evaporator can be turned off. A humidistat is employed to keep down the humidity within acceptable limits.

3.11 MAINTENANCE OF EVAPORATIVE COOLING SYSTEMS :-

For satisfactory operation of the evaporative cooling system proper maintenance is very important. The motor and blower should be frequently inspected to prevent excessive noise. The most important aspect of maintenance is keeping the water distributing systems in perfect working order. This includes pads, nozzles etc. For systems using recirculated water, the mineral concentration increases as the water is evaporated. Scale deposits form that plug or clog the wetted surfaces and reduce the air flow. The control is effected by bleeding sufficient water to keep re-concentration of solids to a minimum. In some cases water is bled continuously.

Pads exposed to outdoor air will gradually become dirty through collection of dust, leaves etc. Frequent inspection should be made and they should be replaced when they are seen to give indication of deterioration or plugging. The pads should be replaced annually.

When spray equipment is used, periodical cleaning of cells, nozzles and baffles should be done.

-:O:-

CHAPTER - IV.

AIRWASHER THEORY.

4.1 Diffusion.

4.2 Mass Transfer Theory.

4.2.1 Heat and Mass Transfer in a pan Humidifier.

4.2.2 Molecular Diffusion.

4.3 The Diffusion Coefficient.

4.4 Analogy to Conduction Heat Transfer.

4.5 Equimolal Counter Diffusion.

4.6 Lewis Relation.

4.7 Simultaneous Heat and Mass Transfer Between Water Wetted Surface and Air .

4.8 Basic Equations for Airwashers.

4.9 Airwasher Humidification Process on Psychrometric Chart.

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4.11.1 Contact Mixture theory as applied to Humidification Process.

4.11.2 Heated Water with zero bypass factor,

4.11.3 Heated Spray water with bypass factor - Process of Humidification for any general process.

-:0:-

CHAPTER - IV.

AIR WASHER THEORY:

4.1 DIFFUSION :-

When a wet surface is dried by an airflow the vapor escaping from the surface is carried away in the air flow by convection. First, however, it must penetrate the boundary layer on the surface by diffusion. The mass flow arising by diffusion is given by Fick's law which is much the same as the heat conduction equation. We describe a simple experiment to understand this law.

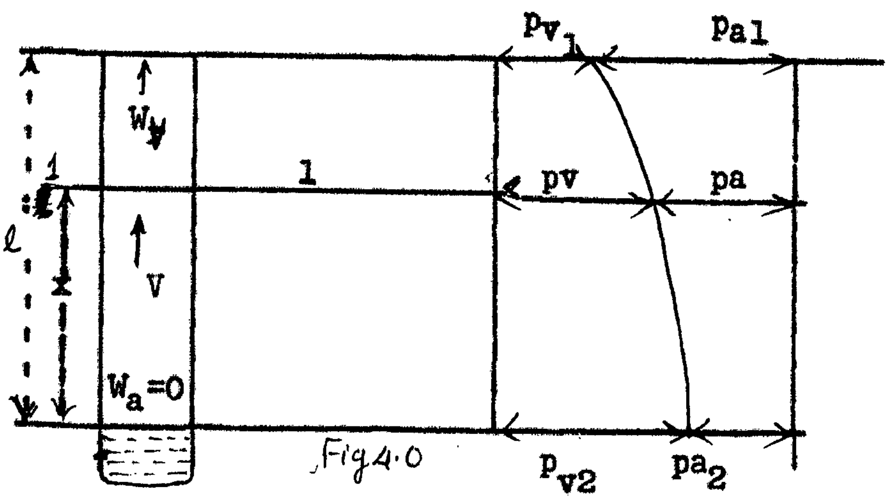


Fig.4.0

A glass tube closed on one end may be filled at the bottom with a little water. Over the open end of the tube an air flow is blown with a certain concentration of water vapour. The air velocity may be small. The total pre-ssure within the tube is then constant and equal to the outside pressure. The partial pressure of the water vapour outside the tube is genera-lly different from that over the water surface. The partial pressure of the water vapour over the water surface equals the saturation pressure at the surface temperature. The partial pressure at the upper end of the tube is fixed by the partial pressure in the airflow.Within the tube the partial pressure may be as shown in Figure 4.D. The weight of water vapour per unit of area 1-1, the weight velocity connected with the ~~differences~~ diffusion process is according to Fick's law.

$$W_v = -D \quad d_{cv}/dx$$

where cv is the concentration of the vapour in ~~the~~ lbs per cubic ft.

D = diffusion coefficient

and d_{cv}/dx = the gradientof the concentration alongthe tube axis.

The diffusion depends upon both masses which diffuse into each other.

The concentration c_v can be expressed by the partial pressures p_v -

$$c_v = W_v/V \qquad = p_v/R_v T$$

where W_v = lbs of water vapor.

With this the Fick's law becomes,

$$W_v = \frac{-D}{R_v T} \qquad \frac{dp_v}{dx} \qquad \text{-------- A}$$

Negative sign is because of the concentration gradient being negative in the direction of diffusion. In this form only the Fick's law is valid for great temperature difference At extremely high temperature differences there is added to the diffusion given by equation -(A) a second kind of diffusion which arises at constant partial pressure and which is called thermodiffusion.

Since the sum of the partial pressures p_v and p_a is const. and equal to the total pressure p, there corresponds to a gradient of vapor pressure a gradient of the partial pressure of air.

As equation -(A) must also be valid for air, a mass flow of the air must arise which is in opposite direction to the vapor flow. Since the tube is sealed from the bottom no air can leave from the bottom. Therefore, a convective flow in upward direction must be present within the tube which compensates for the diffusive air flow. The velocity of this convective flow may be v. The amount of vapor which is transported by this convective flow through the cross section 1-1 per unit area and time is vc_v. Therefore, the whole weight velocity of the water vapour in the cross section 1-1 is

$$w_v = -\frac{D}{R_v T} \frac{dp_v}{dx} + v\frac{p_v}{R_v T}$$

The same relation must be valid for the resulting air flow. Since the flow is zero the resulting equation is

$$W_a = -\frac{D}{R_a T} \frac{dp_a}{dx} + v\frac{p_a}{R_a T} = 0$$

which gives

$$v = \frac{D}{p_a} \quad \frac{dp_a}{dx}$$

Since $p_a + p_v = $ contt.,

$$p_a = p - p_v$$

Differentiation gives ; $dp_a/dx = -dp_v/dx$

Hence $v = -\dfrac{D}{(p-p_v)} \quad \dfrac{dp_v}{dx}$

and for the weight velocity of the vapor.

$$W_v = -\frac{D}{R_vT} \quad \frac{dp_v}{dx} + v \quad \frac{p_v}{R_vT}$$

or $W_v = -\dfrac{D}{R_vT} \quad \dfrac{dp_v}{dx} - \dfrac{D}{p-p_v} \quad \dfrac{dp_v}{dx} \quad \dfrac{p_v}{R_vT}$

$$= -\frac{D}{R_vT} \; \frac{p}{p-p_v} \quad \frac{dp_v}{dx} \quad \text{--------(B)}$$

This equation is called the Stefan's law.

When the cross-section of the tube is contant over x equation (B) can be integrated, since W_v is then independent of x. Also the velocity 'v' is contt.

By separating the variablex and integrating

$$\int_1^2 \frac{dp_v}{p-p_v} = -\int_{x=0}^{x=\ell} \frac{W_v}{D} \; \frac{R_vT}{p} \; dx$$

$$\log \frac{p-p_{v_1}}{p-p_{v_2}} = W_v \quad \frac{\ell}{D} \quad \frac{R_vT}{p}$$

$$\text{or} \quad W_v = \frac{D}{\ell} \; \frac{p}{R_v T} \; \log \frac{p - pv_1}{p - p_{v2}} \qquad \text{------(C)}$$

From this equation the weight velocity can be calculated from the partial pressures at both ends of the tube when the diffusion coefficient is known. Inversely such an experiment can be used to determine the diffusion coefficient by measuring the partial pressures and the weight evaporated per hour.

As long as the difference in partial pressures is small when compared with the total pressure equation (C) can be simplified to

$$W_v = \frac{D}{\ell} \quad \frac{1}{R_v T} \left(pv_2 - p_{v_1} \right)$$

This equation is exactly of the same form as the heat conduction equation in plane wall.

4.2 MASS TRANSFER THEORY :-

Mass transfer by diffusion or by convection is the transport of one component of a mixture due to a concentration gradient. Heat transfer by conduction or convection is the transport of energy due to a temperature gradient. Heating or cooling of air involves only heat transfer resulting in temperature change of the air. In a true airconditioning process however, there is a simultaneous transfer of heat and mass (water vapour) between the air stream and a wetted surface. The wetted surface may be water droplets in an air washer, wetted slats of cooling tower, condensate covering the surface of a dehumidifiying coil or wetted tubes of an evaporative condenser.

4.2.1 HEAT AND MASS TRANSFER IN A PAN HUMIDIFIER :-

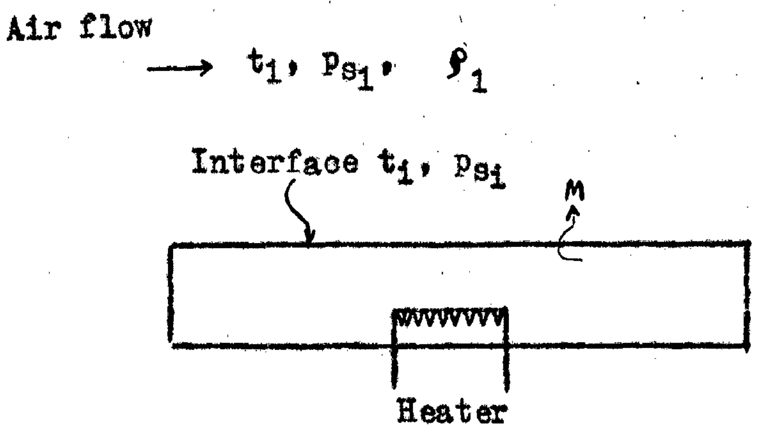


Fig. 4.1

Air at a temperature t_1, flows steadily over a pan containing water at a bulk temperature t_w. The air contains a certain mass fraction of water $W_{s_1} = \frac{m_s}{m}$

where m_s = mass of water vapor,

and m = mass of mixture of air and water vapor.

Subscript 's' indicates properties of the diffusing component (water vapor) subscript 'a' denotes the properties of the nondiffusing components (dry air) and symbols without letter subscripts indicate properties of the mixture.

The water vapor content of the air can be expressed alternately in terms of its partial pressure p_{s_1} mass density ρ_1, humidity ratio $W_1 = m_s/m_a$ or mol fraction x_{s_1}-

At the interface between the air stream and water evaporation is taking place and a saturation state exists at the interface temperature t_i. Thus the partial pressure p_{si} at the interface is the saturated vapor pressure of water corresponding to tamp. t_i, . The temperature t_i alone fixes the properties of the interface.

The heat transfer per unit of surface area between the air stream and the interface is expressed as the product of the heat transfer coefficient 'h' and the difference between the interface and air stream temperatures.

$$\frac{q_a}{A} = h\,(t_i - t_1) \qquad \text{--------- (1)}$$

where q_a = heat transfer rate between air stream and the interface BTU/hr.

A = heat transfer surface area

h = heat transfer coefficient BTU/hr/sq.ft./^oF.

The corresponding mass transfer per unit area is expressed in terms of the mass transfer coefficients h_D or h_g as

$$\frac{M}{A} = h_D\,(\rho_{s_i} - \rho_{s_1}) \qquad \text{.----------(2)}$$

but $\dfrac{\rho_{s_i}}{\rho} = W_{s_i}$

hence $M/A = \rho h_D (W_{s_i} - W_{s_1})$ ----------(3)

also, $M/A = h_G\,(p_{s_i} - p_{s_1})$ ----------(4)

where M = mass transfer rate in lbs/hr.

h_D = mass transfer coefficient -lbs/hr/sq.ft. (lbs/cu.ft.)

ρ_s = partial mass density of water lbs/c.ft.

= mass density of mixture lbs/c.ft.

W_s = mass fraction of the water vapor in mixture lbs of water vapor/lb. of mixture.

h_G = Mass transfer coefficient, lbs/(hr)(sq.ft)(lbs/sq.ft.)

p_s = partial pressure of water vapor lbs/sft.

Thus partial mass density (mass concentration) mass fraction or partial pressure may be used as the driving potential for mass transfer.

Since the water vapor transferred to the air stream must be supplied with its latent heat of vaporisation hfg, the heat flux between the liquid and interface and consequently the heat supplied by the heater is ;

$$q/A = \frac{q_a}{A} + \left(\frac{M}{A}\right) hfg$$

or $$q/A = h(t_1 - t_1) + h_D(\rho_{s_1} - \rho_{s_1})\, hfg \quad \text{----}(5)$$

Also, $$q/A = h_w(t_w - t_1) \quad \text{--------------}(6)$$

where q = heat transfer rate between liquid and interface.

hfg = enthalpy of evaporation BTU/lb.

h_w = heat transfer coefficient for the liquid side B.T.U/(hr) (sq.ft) (oF)

If the flow conditions are specified so that the heat and mass transfer coefficients can be predicted and the air stream properties and pan water temperature are given, equations (1) to (6) can be solved simultaneously to find (1) the interface saturation state (2) the mass transfer rate (3) the heat transfer to the air (4) the total heat transfer supplied to the water by the heater.

In the previous discussion the water temperature and the air temprature remained contant throughout the process. In many cases however, as in counterflow cooling towers the air and water temperature vary throughout the process with resultant variations as the interface state.

4.2.2 <u>MOLECULAR DIFFUSION</u> :-

Consider a quantity of air in a closed container where the air at one end is slightly heated to cause a temperature gradient. If the container is isolated, the temperature will soon equalize as a result of passing of the energy by a random mixing of the air molecules. This is a familiar process of heat conduction in a gas. Assume now that a small amount of water vapor is introduced at one end of the container which is again isolated. After some time the water vapor will be equally distributed throughout the air by same molecular movements. The mass transfer process is known as molecular diffusion. The equations governing this process are analogous to those for heat conduction.

The basic equation for molecular diffusion is usually referred to as Fick's law and can be derived from the principles of Kinetic theory of gases. Depending upon the parameter chosen for the driving potential, the one dimensional diffusion rate per unit area in y direction (normal to the transfer surface) is expressed as ;-

$$M/A = - D_v \left(\frac{d \rho_s}{dy}\right) \text{------------}(7)$$

or $$M/A = - \rho D_v (dW_s/dy) \text{-----------}(8)$$

or $$M/A = - (D_v/R_s T)(dp_s/dy) \text{--------}(9)$$

The negative sign is there because the concentration gradient is negative in the direction of diffusion. The quantity D_v is known as the mass diffusivity or the diffusion coefficient and is a property of the gas mixture. Since the mass flux M/A is expressed in lbs/hr.ft and the partial density ρ_s in lbs/c.ft. the units of D_v from (7) is

$$D_v = \left\{\text{lbs. per (hr)(ft}^2)\right\}\left\{\text{ft}^3 \text{ per lb.ft.}\right\} = \text{ft}^2/\text{hr}$$

4.3 <u>THE DIFFUSION COEFFICIENT</u> :-

Kinetic gas theory calculations show that D_v is inversely proportional to the pressure and with temperature it increases as T^{1+n} where n has a value between 0.5 to 1. The equations for diffusion coefficient in gases derived by several investigators from Kinetic theory may be expressed by equation;

$$D_v = b \frac{T^{1.5}}{P_T (v_{MA}^{1/3} + v_{MB}^{1/3})^2} \sqrt{\frac{1}{M_A} + \frac{1}{M_B}} \text{----------------}(10)$$

where b = contant

P_T = total pressure atmoshperes

M_A & M_B = molcular weights of the two gases in the gas mixture lbs/lb.mol.

V_{MA} & V_{MB} = molecular volumes of the two gases in the mixture cu.ft. per pound mol.

Upon noting that several theories agreed upon this form Gilliland evaluated the contt. b empirically by plotting the results of some 400 experimental determinations. The resulting value was

$$b = 0.0069$$

The principal weakness of equation (10) is the temperature function $T^{1.5}$ since D_V is more nearly proportional to T^2. Where experimental data are available for D_V they should be preferred to the semitheoretical equation (10) A few experimental values for diffusion of some common gases in air are as given below :-

Mass diffusivity for gases in air.

Gas	D_V in sq.ft./hr. (gases at 1 atm & 77 F)
Ammonia	1.08
Benzene	0.34
Carbon dioxide	0.64
Ethanol	0.46
Hydrogen	1.60
Oxygen	0.80
Water vapor	0.90

Of particular interest to airconditioning is the mass diffusivity of water vapor in air. Analysing the available data, Spalding found that upto a temperature of 2000 °F the mass diffusivity of water vapor in air may be expressed as

$$D_v = \frac{.000146}{P_T} \quad \frac{T^{2.5}}{T + 441} \qquad \text{------(11)}$$

4.4 <u>ANALOGY TO CONDUCTION HEAT TRANSFER</u> :-

Molecular diffusion is directly analogous to conduction heat transfer, both phenomenon are a result of random molecular mixing in a stagnent fluid or in a fluid in laminar flow. In each case the equation govering the process can be expressed in the form ;

Flux = diffusivity x concentration gradient.

For a two component gaseous system (eq. water vapor and air) in which the density of the gas ρ and the specific heat c_p are contant, the following equation can be written;

Fick's law for constant ρ

$$M/A = - D_v \frac{d}{dy} (\rho_s) \qquad \text{---------(12)}$$

Fourier law for contt. cp.

$$q/A = - K \frac{d}{dy}(T) = - \alpha \frac{d}{dy} (\rho c_p T) \qquad \text{--------(13)}$$

where K = thermal conductivity,
B.T.U./hr. (sq.ft.) oF ft).

α = thermal diffusivity sq.ft./hr.

c_p = sp. heat at contt. pressure.
B.T.U./lb. (oF),.

The comparison of heat and mass transfer is shown in Table 4.1

TABLE - 4.1

Transfer	Driving potential	Law	Equation	Diffusivity	Flux
Heat	Temperature or energy concentration	Fouriers	$\frac{q}{A} = -K\frac{dT}{dy}$	$\alpha = \frac{K}{\rho c_p}$	q/A
Mass	Mass concentration.	Ficks	$\frac{M}{A} = -D_v\frac{d\rho_s}{dy}$	D_v	M/A

Equation (12) states thatmass transfer occurs because there is a gradient in Mass Concentration and the mass flux equals the mass diffusivity D_v times the mass concentration gradient, Equation (13) states that the heat transfer ~~takes~~ occurs because there is a gradient in energy concentration and the heat flux equals the thermal diffusivity α $(\alpha = K/\rho\, c_p)$ times the energy concentration gradient.

The equation for the viscous shear force in one dimensional flow is

$$\tau_{yx} = -\frac{\mu}{g}\frac{d}{dy}(V_x)$$

$$\text{or}\quad g\tau_{yx} = -\nu\frac{d}{dy}(\rho V_x) \quad \text{------------(14)}$$

Newton's law for Contt. ρ

where τ_{yx} = viscous force in the yx plane, pound force per square foot.

g = gravitational acceleration = 4.17×10^8

μ = absolute viscosity lbs per ft.hr.

V_x = local velocity of gas in the x direction parallel to the transfer surface ft./hr.

= Kinematic viscosity or momentum diffusivity square ft./hr.

Equation (14) states that momentum transfer occurs because thereis a gradient in momentum concentration and momentum flux ($g\,\tau_{yx}$) equals the momentum diffusivity (Kinematic viscosity) γ times the momentum concentration gradient.

It will be noted that the mass, thermal, and momentum diffusivities all have the same units ft sq. per hr., and that the concentration gradients are linear for a uniform medium of steady state. If these diffusivities all have the same value the concentration gradients would be identical. That is if a fluid were in steady one dimensional laminar flow over a surface such that $D_v = \alpha = \gamma$ the partial pressure, temperature and velocity profiles would all be identical in form. This is true if D_v and cp are essentially contt. through the fluid field and if the diffusivities are identical.

Stated in terms of dimensionless parameters the condition is

$$\gamma/\alpha = \left(\frac{\mu}{\rho}\right)\left(\frac{\rho\,cp}{K}\right) = \frac{cp\,\mu}{K} = NP_r = 1$$

$$\gamma/D_v = \mu/\rho D_v = N_{Sc} = 1$$

where N_{Pr} = Prandtl number = $cp\,\mu/K$ dimensionless

N_{Sc} = Schmidt number = $\mu/\rho D_v$ dimensionless,

where the Prandtl number ($N_{Pr} = c_p \mu/K$) equals the Schmidt number ($N_{Sc} = \mu/\rho D_v$) the temperature and partial pressure profiles are identical and if in addition both are unity these profiles are the same as the velocity profile. For air and water vapour at normal room conditions, the Prandtl number is .71 and the Schmidt number is .60, a condition close to that stated above.

4.5 EQUIMOLAL COUNTER DIFFUSION :-

Consider a mixture of air and water vapor in a closed container the total pressure being P_t (in lbs/sq.ft.) and the absolute temperature T. A concentration gradient exists through the mixture so that the partial pressure p_s of the water vapor is as shown in Fig. 4.2 below.

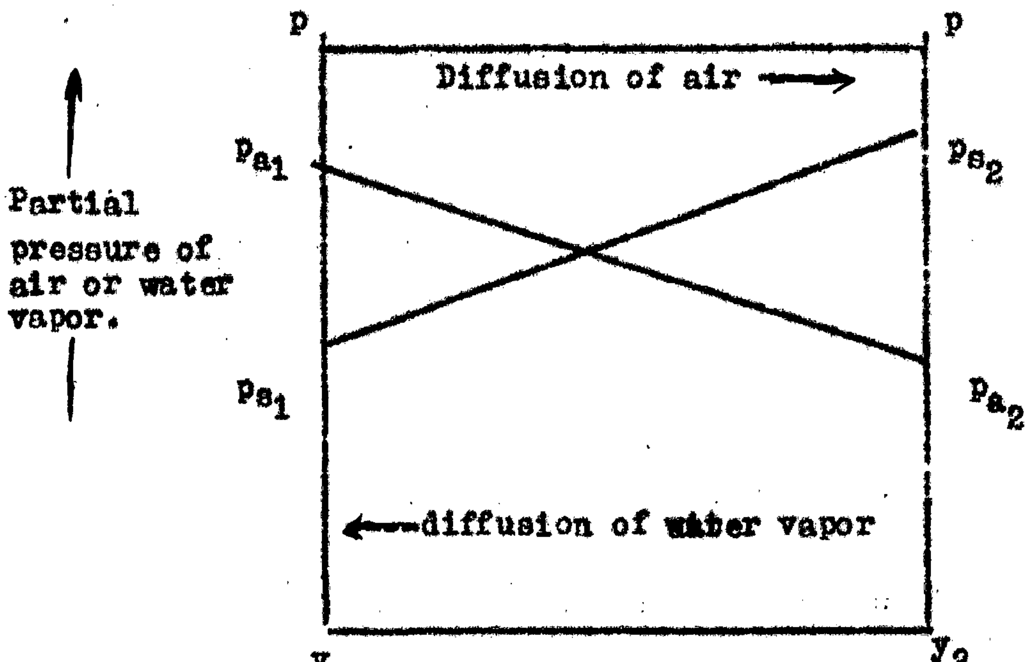


Fig. 4.2

It is assumed here that both components behave as a perfect gas and that the Gibbs Daltons law is valid hence,

$$p_s + p_a = P_t = \text{contt.} \quad \text{-------(A)}$$

where P_t = total pressure lb. force /sq.ft.

Thus the partial pressure of water vapor must decrease if the partial pressure of air increases so that the sum of both remains constant and equal to P_t.

Dividing both sides of the equation (9) by M_s at the molecular weight of water

$$\frac{N_s}{A} = - (D_V/R_gT)(dp_s/dy) \quad \text{-------(15a)}$$

Similalry, $N_a/A = - (D_V/R_gT)(dp_a/dy)$ --------(15b)

where, N_s = metal diffusion rate of water vapor pound mols per hour.

R_g = universal gas contant

N_a = metal diffusion rate of dry air lbs mols per hour.

y = distance measured to the transfer surface ft.

D_V is the same as for diffusion of water vapor through air as for air through water vapor. Equation (A) requires that,

$$(dp_s/dy) = - (dp_a/dy)$$

By integration between plane 1, & 2 of Fig. 4.2,

$$N_s/A = - N_a/A = \frac{- D_V (p_{s_2} - p_{s_1})}{R_g T (y_2 - y_1)} \quad \text{----------(16)}$$

This process is known as equimolal counterdiffusion ~~diffusion.~~ The rates of diffusion of air and water vapor are equal but in opposite direction.

4.6 LEWIS RELATION :-

For air and water vapor mixture W.K. Lewis derived the following relationship :

$$N_{Le} = \frac{h}{h_D \rho c_p} \approx 1$$

i.e. the ratio of heat transfer coefficient to the mass transfer coefficient is equal to the specific heat per unit volume at constant pressure of the mixture. This relation was first derived by Lewis and is designated by N_{Le}

It is important to note that this relation is very nearly true for air and water vapor at low mass transfer rates. It is not in general true for other gas mixture because the ratio of thermal to vapor diffusivity (α / D_V) is usually different from unity.

4.7 SIMULTANEOUS HEAT AND MASSTRANSFER BETWEEN WATER WETTED SURFACES AND AIR.

For solving problems involving simultaneous heat and mass transfer Lewis relation gives satisfactory results for most air conditioning processes.

The quantity usually used to express the water vapor concentration in the air is the humidity ratio W defined as :

$$W = m_s/m_a = \rho s / \rho a$$

It is, therefore, convenient to define a mass transfer coefficient using W as the driving potential, hence,

$$M/A = KD (W_i - W_1)$$

where the coefficient K_D has the units lbs per (hr) (sq.ft)

Using the definition of W and noting that for dilute mixtures $\rho a_i \simeq \rho a_1$ i.e. the partial mass density of dry air changes very slightly between interface and free stream conditions. We can write,

$$M/A = K_D / \rho a_m \; (\rho si - \rho s_1)$$

where ρ am is the mean density of dry air lbs/cft.

Comparing this with the previous equation (2)

$$M/A = h_D(\rho s_1 - \rho_1)$$ we see that

$$h_D = K_D / \rho am$$

The humid specific heat cp_m of the air stream is by definition $cp_m = (1 + W_1)$ cp BTU/°F (lb.of dry air)

$$\text{or } cp_m = (\rho / \rho a_1) \; cp. \quad \therefore \; cp = cp_m \frac{\rho a_1}{\rho}$$

Substituting in the Lewis relation,

$$N_{Le} = \frac{h}{h_D \rho \; cp} = \frac{h \; \rho a_m}{K_D \; cp_m \rho a_1} = \frac{h}{K_D \rho \; cp_m}$$

$$1 \text{ --------(17)}$$

Since ρ am $\simeq \rho a_1$ due to small change in dry air density.

Using a mass transfer coefficient with humidity ratio as the driving force, the Lewis relation becomes,

Ratio of heat to Mass transfer coefficient equals humid specific heat.

For the pan humidifier illustrated in fig. 4.1, it has been shown that the total heat transfer from the liquid to the interface is ;-

$$q/A = q_a/A + (M/A)\ h_{fg}$$

Utilizing the definitions of mass transfer coefficients,

$$q/A = h(t_1 - t_1) + K_D\ (W_1 - W_1)\ h_{fg}$$

Assuming the Lewis relation equation (17) to be valid gives,

$$q/A = K_D\left[c_{pm}(t_1 - t_1) + (W_1 - W_1)\ h_{fg}\right] \quad \text{---(18)}$$

The enthalpy of the air is by definition ;

$$h = cp_a t + W h_s \quad \text{(BTU per lb dry air)}$$

where $h =$ enthalpy of air BTU/lb.

$h_s =$ enthalpy of water vapor BTU/lb.

The enthalpy of water vapor h_s may by the perfect gas law be expressed as ;-

$$h_s = cp_s\ (t - t_o) + h_{fgo}$$

where the base of the enthalpy is taken as saturated water at temperature t_o. choosing $t_o = 0F$ to correspond with the base of the dry air enthalpy, gives

$$h = cp_a t + Wh_s$$

$$= cp_a t + Wcp_s t - Wcp_s t_o \; Whfgo$$

$$= (cp_a + Wcp_s)\, t + Whfgo$$

$$= cp_m t + Whf_{go} \quad \text{------------------} (19)$$

Comparing equation (18) and (19) if the small changes in the latent heat of vaporization of water with temperature are neglected, the total heat transfer can be written ;

$$q/A = K_D(h_i - h_1)$$

Thus where the driving potential for heat transfer is temperature difference and the driving potential for mass transfer is mass concentration or partial pressure, the driving potential for simultaneous heat and mass transfer in an air water mixture is enthalpy.

4.8 BASIC EQUATIONS FOR AIR WASHERS :-

For the direct contact spray chamber as shown in the figure 4.3 of cross sectional area A_C and length 'l' the steady flow rate of dry air is $W_a/A_C = S_a$ and the mass flow rate of water flowing parallel with the air $W_L/A_C = S_L$

where W_a is mass flow rate of air lbs/hr.

S_a is mass velocity or flow rate per unit cross sectional area for air lbs/per hr(sqft)

W_L is mass flow rate of liquid ,lbs/hr.

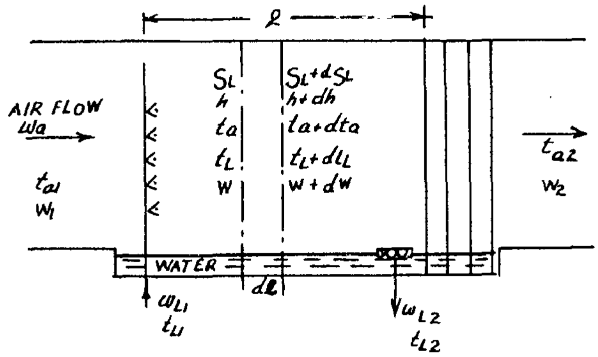


FIG 4.3 . . . AIR WASHER SPRAYCHAMBER

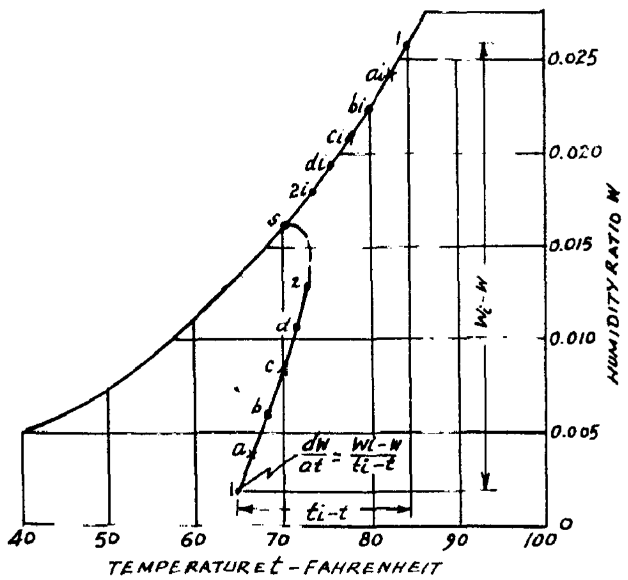


FIG_4.4 AIR WASHER HUMIDIFICATION PROCESS ON PSYCHROMETRIC CHART

S_L = mass velocity or flow rate per unit cross sectional area, for liquid lbs per (hr) (square ft).

Since water is evaporating or condensing S_L changes by an amount dS_L in a differential length dl of the chamber. Similar changes occur in temperature, humidity ratio, enthalpy and other properties as shown in figure 4.3

Since in a direct contact equipment like airwasher it is difficult to evaluate true surface area it is convenient to work on unit volume basis. If a_H and a_M represent square foot of the heat transfer and mass transfer surface per cubic foot of chamber volume respectively, the total surface areas for heat and mass transfer are then ;

$$A_H = a_H A_C \ell$$

and $$A_M = a_M A_C \ell$$

The basic equations for the process occuring in the differential length dl may be written for

1. <u>Mass Transfer</u>

$$- dS_L = S_a dW = K_D a_M (W_i - W) dl \text{ ---------- } (20)$$

that is the water evaporated, the moisture increase of air and the mass transfer rates are all equal.

2. <u>Heat Transfer to the Air :-</u>

$$S_a cp_m dt_a = h_a a_H (t_i - t_a) dl \text{ ---------- } (21)$$

3. Total energy transfer to the Air :-

Assume heat and mass transfer areas are identical ($a_H = a_M$) which is quite valid in case of spray chambers. Also $N_{L_e} = 1$, then

$$S_a dh = K_D \; a_M (h_i - h) \, dl \qquad \text{------------(22)}$$

4. Energy Balance :-

$$S_a dh = \pm S_L C_L \; dt_L \qquad \text{---------------(23)}$$

A minus sign refers to parallel flow of air and water vapor and the plus sign refers to counterflow. The water flow rate changes between inlet and outlet due to mass transfer . For exact energy balance therefore, the term ($C_L t_L \, dS_L$) should be added to the right hand side of equation (23) . The percentage change in S_L is quite small in all normal applications of airconditing and therefore, may be neglected.

5. Heat Transfer to the Water :-

$$S_L \, C_L \, dt_L = h_L \; a_H \; (t_L - t_i) \; dl \qquad \text{-------(24)}$$

Equations (20 to (24) are the basic relations necessary to the solution of simultaneous heat and mass transfer processes in direct contact airconditioning equipment.

To facilitate the use of these relations in equipment design or performance three other equations may be obtained from the above set. Combination of equations (22), (23) and (24), gives :-

$$\frac{h_i - h}{t_L - t_i} = \frac{h_L a_H}{K_D\, a_M} = \frac{K_L\, U}{K_D}$$

or $$\frac{h - h_i}{t_L - t_i} = -\frac{h_L}{K_D} \qquad \text{------------(25)}$$

Equation (25) relates the enthalpy potential for the total heat transfer through the gas film to the temperate potential for this same transfer through the liquid film.

Also from equation (21), and (22) and the Lewis relation,

$$N_{Le} = \frac{h}{h_D\, \rho\, cp} \simeq 1 \quad \text{we get,}$$

$$\frac{dh}{dta} = \frac{K_D\, a_M}{h_a\, a_H}\, cp_m\, \frac{(h_i - h)}{(t_i - t_a)}$$

since $h/K_D\, cp_m \simeq 1$ from (17), we get,

$$dh/dt_a = h - h_i \,/\, t_a - t_i \qquad \text{--------------(26)}$$

Similarly combination of equations (20, (21) and the Lewis relation we get,

$$\frac{dW}{dt_a} = \frac{W - W_i}{t_a - t_i} \qquad \text{------------------(27)}$$

Equation (27) states that at any cross section in the spray chamber, the instantaneous slope of the air path dW/dt_a on psychrometric chart is determined by a straight line connecting the air state with the interface saturation state at that cross section.

4.9 AIRWASHER HUMIDIFICATION PROCESS ON PSYCHROMETERIC CHART:

In the Figure 4.4 state 1 represents the state of air entering the parallel flow air washer. The washer is operating as a heating and humidifing appratus so that the interface saturation state of the water at air inlet is the state designated by 1_i. The initial slope of the air path is then along a line directed from state 1 to state 1_i. As the air is heated, water is cooled and the interface temperature drops. Corresponding air states and interface saturation states are indicated by the letter a, b, c, d in the Fig. In each case the air path is directed towards the associated interface state. The interface states are found from equations (24) and (25). Equation (24) describes how the air enthalpy changes with water temperature and equation (25) how the interface saturation state changes to accommodate this change in air and water conditions. The solution for the interface state on the normal psychrometric chart of the Fig. 4·4 shown must be either by trial and error from equations (24) to (25) or by a complex graphical solution. This method utilizes a psychrometric chart with enthalpy and temperature as coordinates. It will be best illustrated by an example.

4.9.1 ILLUSTRATION :-

A parallel flow air washer has to be designed using a spray chamber. The design conditions are as follows.

Water temperature at inlet $t_{w_1} = 95^\circ$ F

Water temperature at outlet $t_{w_2} = 75^\circ$ F

Air temperature at inlet $t_{a_1} = 65^\circ$ F

Air wet bulb at inlet $t_{a_1}' = 45^\circ$ F

Air mass flow rate/unit area S_a = 1200 lbs/(hr)(sq.ft)

Spray Ratio S_L / S_a = 0.7

Air heat transfer coefficient

per cubic ft. of chamber volume $h_a aH$ = 72 BTU/(hr)(F$^\circ$)(cu.ft.)

Liquid heat transfer coefficient

per cu.ft. of chamber volume $h_L a_H$ = 900 BTU/(hr)fdeg)(Cu.ft.)

Air volume flow rate = 6500 cfm.

Solution :-

The air mass flow rate W_a = 6500/13.25 = 490 lbs./ min. and the required spray chamber cross sectional area is then

$$A_C = \frac{W_a}{S_a} = 490\text{x}60/\ 1200 = 24.5 \text{ sq.ft.}$$

The mass transfer rate is given by the Lewis relation,

$$K_D\ a_M = \frac{h_a aH}{cp_m} = 72/.24 = 300 \text{ lbs/(hr)(c.ft)}$$

Fig. 4.5 shows the enthalpy temp. psychrometric chart with the graphical solution for the interface states and the air path through the washer spray chamber. The solution proceeds as follows :-

1. Enter bottom of the chart with $t_a' = 45^oF$. Follow upto saturation curve to establish air enthalpy $h_1 = 17.65$ BTU/lb. Extend this enthalpy line to intersect initial air temperature $t_{a1} = 65^o$ F (state 1 of air) and initial water temperature t_{L1} of 95^o F at pt A (We utilize the temperature scale both for air and water temperature).

2. Through point A, construct the energy balance line AB with a slope of

$$dh/dt_L = -\frac{S_L}{S_a} = -0.70$$

Point B is determined by intersection with the leaving water temp. $t_{L_2} = 75^o$ F. The negative slope is because of the parallel flow, which results in the air water approaching but not reaching the common saturation state S. The line AB has no physical significance as far as representing any air state on the psychrometric chart. It is merely used as a construction line in the graphical solution.

3. Through point A construct the tie line. $A1_i$ having a slope of

$$\frac{h - h_i}{t_L - t_i} = -\frac{h_L a_H}{K_D a_M} = -\frac{900}{300} = -3$$

The intersection of this line with the saturation curve gives the initial interface state 1_i at the chamber inlet. The energy balance line and tie line representing equation (24 and (25) are combined to give a simple graphical solution on the h - t chart for the interface state.

4. The initial slope of the air path may now be

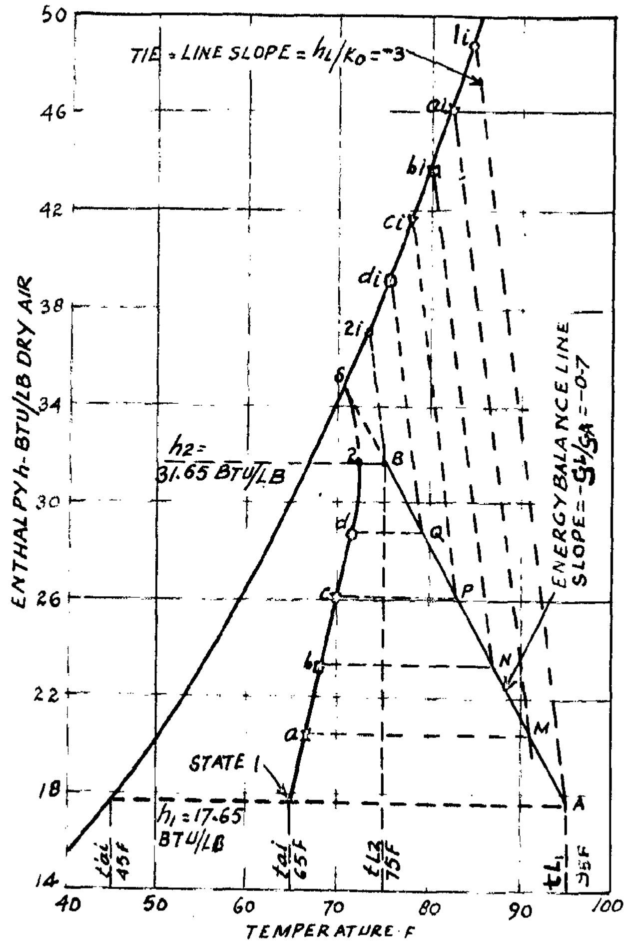


FIG-4.5.... GRAPHICAL SOLUTION FOR THE AIR-STATE PATH IN WASHER.

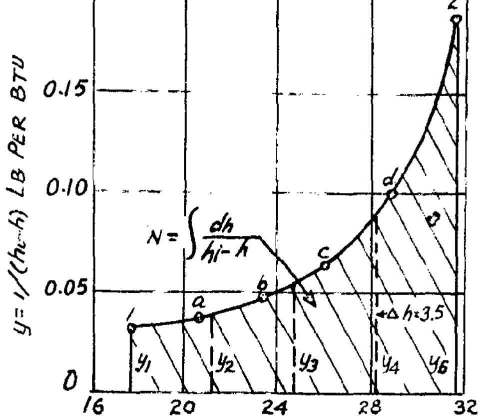


FIG-4.6 GRAHICAL EVALUATION OF $\int dh/(h_i - h)$

constructed according to equation (26) by drawing line 1_a toward the initial interface 1_i. The length of the line 1_a depends upon the degree of accuracy required in the solution.

5. Construct the horizontal line a_M locating the point M in the energy balance line. Draw a new tie line with the slope of -3. 0 as before from ~~at~~ point M to a_1 locating interface state a_1, continue the air path from a to b by directing it toward the new interface state a_1.

6. Continue in the manner of step 5 until pt. 2 the final state of the air leaving the chamber is reached, and hence the temperature t_{a_2} and corresponding enthalpy h_2 can be obtained.

7. The final step is to calculate the required length of the spray chamber,

From Eq. (22),

$$\ell = \frac{S_a}{K_D a_M} \int_1^2 \frac{dh}{h_i - h} \qquad \text{------(28)}$$

The integral is evaluated graphically by plotting $1/h_i - h$ vs h as shown in the figure 4.6. By Simpson's rule if h is the equal increment and if there are 4 equal increments, then,

$$A = \int_1^2 \frac{dh}{(h_i - h)}$$

$$= \frac{4h}{3} \left\{ y_1 + 4y_2 + 2y_3 + \cancel{4y_3} + 4y_4 + y_5 \right\}$$

and hence the length can be obtained by means of equation (28).

4.10 HUMIDIFICATION OF AIR :-

By spraying air with a finely divided water stream the state of that air may be greatly changed. Whether the air is heated, cooled, humidified or dehumidified depends upon the temp. of spray water in relation to the initial state of air.

For cross flow of air and water as shown in Fig. 4.7 in which

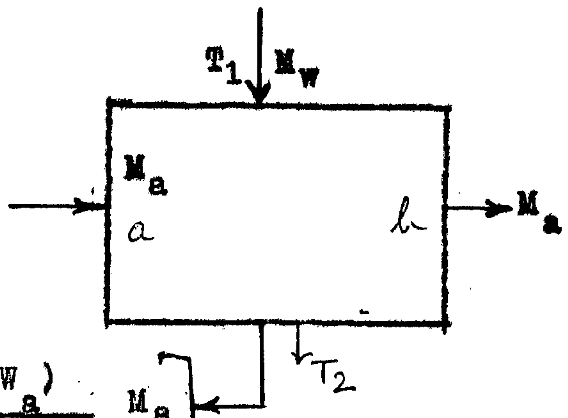
M_W = lbs. of water /hr.

M_a = lbs. of air/hr.

h_b = enthalpy of leaving air.

T_1 = entering temp. of water.

T_2 = leaving temp. of water.



$$M = \left[M_W - \frac{(W_b - W_a)}{7000} M_a \right]$$

Fig. 4.7

the energy balance gives,

$$M_a h_a + M_W (T_1 - 32) = M_a hb + \left\{ M_W - \frac{(Wb - W_a) M_a}{7000} \right\} (T_2 - 32)$$

$$\text{or } M_a (hb - h_a) = M_W (T_1 - T_2) + \frac{Wb - W_a}{7000} M_a (T_2 - 32) \qquad \text{------(1)}$$

Now $h_b = \Sigma\, b + W_b (T_2 - 32)$ where Σ denotes sigma heat function.

$$h_a = \Sigma\, a + W_a (T_2 - 32)$$

Eqn. (1) is $M_a (\Sigma\, b - \Sigma\, a) + M_a \dfrac{Wb - W_a}{7000} (T_2 - 32)$

$$= M_W (T_1 - T_2) + \frac{W_b - W_a}{7000} (T_2 - 32) M_a$$

Hence the final equation is,

$$M_a (\Sigma b - \Sigma a) = M_W (T_1 - T_2) \text{ ----------(2)}$$

For parallel flow $M_a \, d\Sigma = - M_W \, dT$

Since there is no appreciable numerical difference between sigma heat function value and enthalpy for all practical purposes enthalpy is used in equa. (2).

4.11 <u>CONTACT MIXTURE THEORY</u> :-

Assumptions :-

1. Air stream particles come in to contact with a hypothetical surface.

2. The contacted particles assume saturated state at the temperature of the hypothetical surface.

3. These are equivalent numbers of air particles which contact the surface and other particles do not contact.

4. The condition of air after passing over the surface is that of homogeneous mixture of contacted and uncontacted particles.

If A = Area of contact (surface area)

m = no. of particles contacting /unit area/ unit of time.

Z = no. of particles /unit time passing over any section of the surface.

y = no. of already contacted particles at that section in unit time.

Then the above theory states :- The increase in number of contacted particles per unit time to total number of contacts per unit time is equal to the number of uncontacted particles in the mixture passing that section divided by Total No. of particles / unit time.

$$\text{i.e.} \quad \frac{dy}{mX\, dA} = \frac{(Z - y)}{Z} = X = \text{Bypass factor} \qquad \text{------(3)}$$

4.11.1 CONTACT MIXTURE THEORY AS APPLIED TO HUMIDIFICATION PROCESS :-

If T = hypothetical surface temp.

W_T = specific humidity at surface temp.

t = dry bulb temp. of air.

dt = change in dry bulb temp. of air.

Then,

$$\frac{dt}{mdA} = \frac{T - t}{Z} \quad \text{(by anology to contact mixture theory)} \qquad \text{------(4a)}$$

$$\text{Similarly,} \quad \frac{dW}{mdA} = \frac{W_T - W}{Z}$$

$$\text{and} \quad \frac{d\Sigma}{mdA} = \frac{\Sigma T - \Sigma}{Z} \qquad \text{------(4b)}$$

From Equ. (3) $\frac{dy}{mdA} = \frac{Z - y}{Z}$ or $\frac{m}{Z} dA = \frac{dy}{Z - y}$

Integrating,

$$\int_0^A \frac{m}{Z} dA = \int \frac{dy}{Z-y}$$

m and Z are constants.

$$\frac{m}{Z} A = - \log (Z-y) \quad \log Z$$

or $$\frac{m}{Z} A = - \log \frac{Z-y}{Z} = - \log X$$

$$\therefore \quad X = e^{-\frac{m}{Z}A}$$

where $X = \frac{Z-y}{Z}$ is called the by pass factor.

Equation (4a) then gives,

$$\int_0^A \frac{m}{Z} dA = \int_{t_a}^{t_b} \frac{dt}{T-t}$$

or $$\frac{m}{Z} A = \int_{t_a}^{t_b} \frac{dt}{T-t} = -\log \frac{t_b - T}{t_a - T} = -\log X$$

$$\therefore \quad X = \frac{t_b - T}{t_a - T}$$

where t_b = leaving air dbt

t_a = entering air dbt

Efficiency of Washer $$E = \frac{t_a - t_b}{t_a - T}$$

and $$X = \frac{t_b - T}{t_a - T}$$

$$\therefore \quad X = 1 - \frac{t_a - t_b}{t_a - T} = \frac{t_b - T}{t_a - T} = 1-E$$

4.11.2 <u>HEATED WATER WITH ZERO BYPASS FACTOR :-</u>

$$M_a (\Sigma b - \Sigma a) = M_W (T_1 - T_2)$$ from equation (2)

or $$(\Sigma b - \Sigma a) = \frac{M_W}{M_a} (T_1 - T_2)$$

Let T_1 = Temp. of spray water entering the washer.

T_o = Temp. of exit spray water sigma function

Σo = Sigma function at a W.B.T. of T_o

i.e. when W.B.T. of leaving air = leaving water temp.

Therefore, $(\Sigma o - \Sigma a) = \frac{M_W}{M_a}(T_1 - T_o)$

Solving by Trial and error T_o and thus Σo are known.

4.11.3 <u>HEATED SPRAY WATER WITH BYPASS FACTOR PROCESS OF HUMIDIFICATION FOR ANY GENERAL PROCESS :-</u>

For parallel flow

$$M_a d\Sigma = - M_W \, dT$$

and $\frac{m}{Z} dA = \frac{d\Sigma}{\Sigma T - \Sigma}$ from eq. (4(b))

For the total area,

$$\frac{m}{Z}\int dA = \int \frac{d\Sigma}{\Sigma T - \Sigma} = - \frac{M_W}{M_a}\int \frac{dT}{\Sigma T - \Sigma}$$

$$\frac{m}{Z}\int dA = - \frac{M_W}{M_a}\int \frac{dT}{\Sigma T - \Sigma}$$

For solution of the intergration assume,

$$(\Sigma T - \Sigma) = a(T - T_o)$$

where T_o = temp. obtained when there is zero bypass factor.

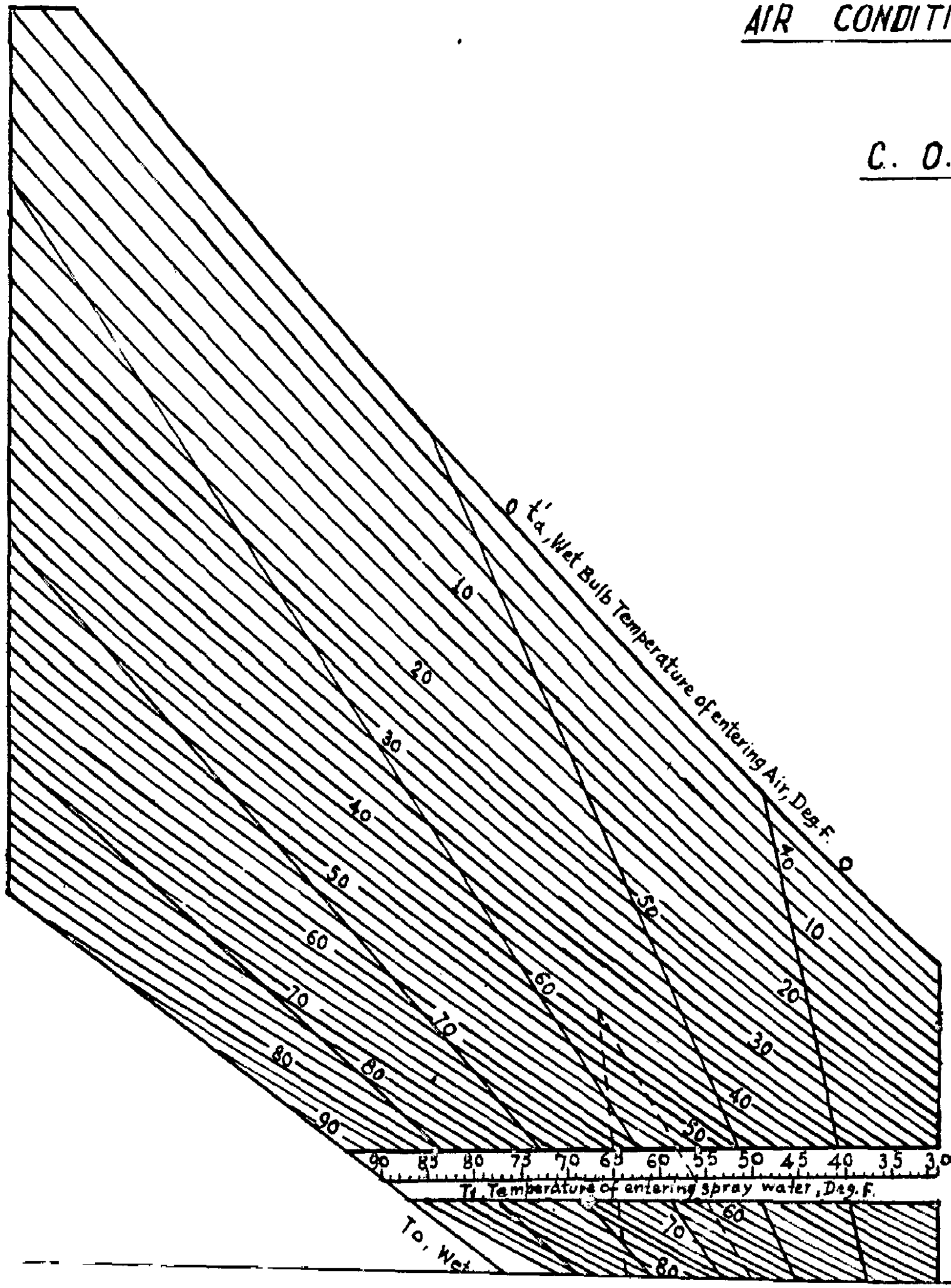
a = constant of air washer.

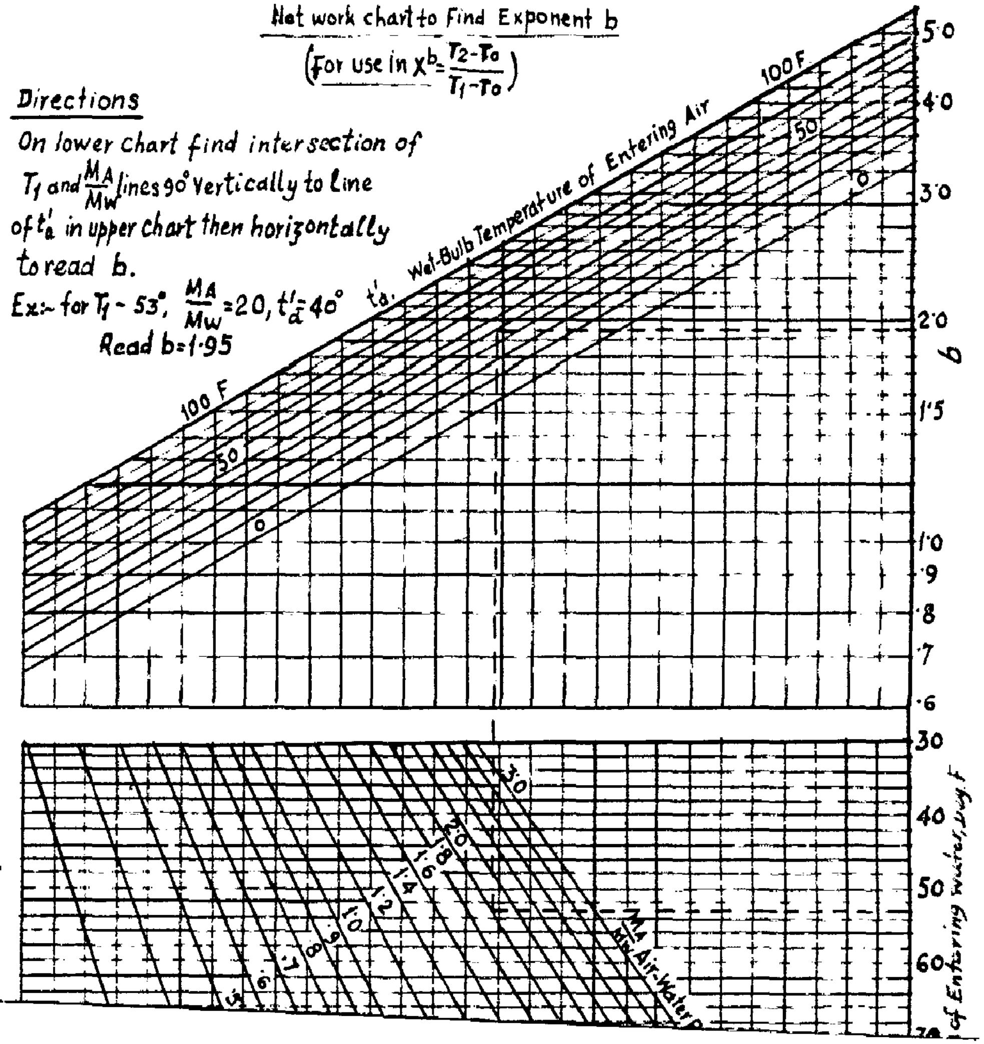
$$\frac{m}{Z} A = - \log X = - \frac{M_W}{aM_a}\int_{T_1}^{T_2} \frac{dT}{T - T_o}$$

AIR CONDITIONING PRINCIPLES

BY

C. O. MACKEY





$$= - a \frac{M_W}{M_a} \log \frac{T_2 - T_o}{T_1 - T_o} = - \frac{1}{b} \log \frac{T_2 - T_o}{T_1 - T_o}$$

$$\text{where } b = a \frac{M_a}{M_W}$$

$$\text{Hence, } b \log X = \log \frac{T_2 - T_o}{T_1 - T_o}$$

$$\therefore \quad X^b = \frac{T_2 - T_o}{T_1 - T_o}$$

$$\therefore \quad T_2 = X^b(T_1 - T_o) + T_o \quad \text{------}(5)$$

The values of T_o and b areobtained from the Fig. 4.8, in the manner as illustrated in the figure itself.

CHAPTER - V.

5\. a) Procedure of Experimentation.

5\. b) Observations recorded.

CHAPTER - V.

PROCEDURE OF EXPERIMENTATION

5.1 RECIRCULATED WATER :-

The observations for calculating the efficiency of the air washer are shown from table 5.1 to 5.45. In the first case the observations are recorded to obtain the relationship between the water quantity discharged by each nozzle i.e. gpm/ nozzle and the efficiency of the air washer. These observations are shown from table 5.1 to 5.36. In the second case to find the relationship between the spray nozzle pressure and the efficiency of the air washer observations from table 5.37 to 5.45, have been recorded. In both cases the water is recirculated so that its temperature remains constant and equal to the thermodynamic wet bulb temperature of the entering air .

The direction of spray water and the number of banks is kept constant for a particular air velocity. Then the pressure is gradually increased by the hand operated valves and read by the pressure gauges as shown in Fig. 5.1. This increases the rate of flow of water. The water quantity per nozzle is measured by means of a stop valve and a vessel. The water is collected in a vessel for 3 minutes and then the gpm per nozzle is calculated.

The inlet dry and wet bulb temperatures are recorded by the sling psychrometer. The thermometer is revolved about

30 times to get the correct equilibrium wet bulb temperature of the ~~incoming~~ air ~~in mm~~. The velocity of the incoming air is measured by the velocity meter .

For a particular air velocity the inlet dry and wet bulb and the leaving dry bulb temperatures are recorded after running the equipment for 5 minutes so that the process may attain the equilibrium state. Pressures maximum upto 48 lbs per square inch gauge are obtainable.

To get the second set of observation, the velocity of the fan is changed . Again the pressure is gradually increased, water quantity of gpm/nozzle measured and the temperatures recorded. This procedure is repeated for different air velocities keeping the number of banks and the direction of spray water constant.

Next, the number of banks is kept same but the direction of the spray water is changed by reversing the nozzles. Thus the direction of spray can be either upstream or downstream when only one bank of spray is employed. If the number of banks are two, the different combinations of spray water direction are made. Thus nozzles of both the banks can operate in the same direction as the direction of air, called the downstream direction, or they can both operate against the flow of air called the upstream direction. The third way of operating the nozzles is that they can be arranged so that they spray the water opposing each other i.e. the operation is 1 bank upstream and 1 bank downstream.

If the number of banks is 3, four different ways are employed to change the direction of spray. They are :-

1. All the banks operating downstream i.e. along the direction of flow of air.
2. 1 Bank operates on upstream and the remaining 2 banks downstream.
3. 1 bank operating downstream and 2 banks operating upstream and finally,
4. All the banks operating upstreams i.e. all banks are against the flow of air.

Thus different combinations of direction of air and water are encountered by the mannual change of direction of spray banks. This changes the air to water ratio, and hence the efficiencies of air washer at different combinations as given in the observation tables from 5.1 to 5.45 are found.

It should be seen that the greater accuracy is observed in taking the dry bulb temperature of the leaving air which would otherwise effect the efficiency of the washer. The wick of the wet bulb thermometer should always be kept wet by clean water.

5.2 HEATED WATER :-

For heating and humidification of the air, the heated water is supplied external to the spray chamber and is not recirculated. The tanks for heating the water are as shown in fig. 5.4. The water is stirred frequently to keep the temperature of hot water uniform.

For a particular arrangement of bank and the direction of spray water, the velocity is kept contant. The apparatus is started after taking the initial temperature of hot water. The equipment is allowed to run for three minutes to stablize and then the leaving temperature of the water and the leaving dry and wet bulb temperatures of the air are recorded. The pressure is gradually increased and the procedure repeated.

The above procedure is repeated for different combinations of number of spray banks and the direction of spray water as given in tables from 5.61 to 5.66.

5.3 CHILLED WATER :-

For cooling and dehumidification chilled water is supplied. The entering air dry and wet bulb temperatures are taken and from these the dew point temperature of the entering air is seen from the psychrometeric chart. For cooling and dehumidification process the leaving water temperature should be less than the entering dew point temperature of the air. Keeping this in view the water temperature is sufficiently lowered than the dew point temperature of the entering air.

The procedure for taking observations is the same as in recirculated and heated water. The observations using chilled water are recorded from Table 5.46 to 5.60 and the arrangements of spray banks and air velocity are given in each table.

Figure 5.2 and Fig. 5.3 show the details of the air washer.

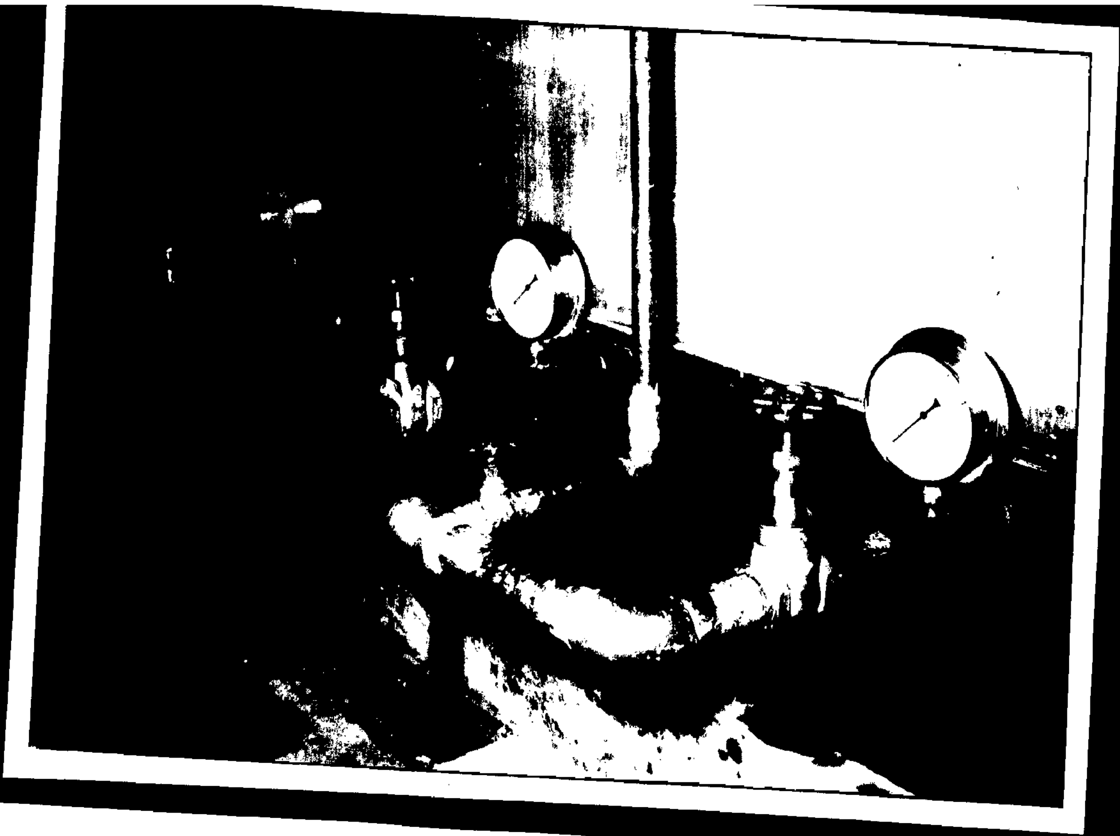


Fig. 5.1 Showing pressures gauges and piping of the air washer .

TABLE - 5.1

RECIRCULATED WATER.

No. of Banks = 1, Direction of Spray = Upstream, Air Velocity=700 fpm

S. No.	Water Quantity gpm per nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency as calculated
1	0.2	82	80	72	20
2	0.5	82	78.5	72	35
3	0.65	82	77.8	72	42
4	0.85	82	77.0	72	50
5	1.0	82	76.5	72	55
6	1.2	82	75.7	72	63
7	1.3	82	75.4	72	66

TABLE - 5.2

No. of banks = 1, Direction of spray = Upstream
Velocity of air = 600 fpm

S. No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency percent as calculated
1	0.2	82	79.7	72	23
2	0.5	82	78.0	72	40
3	0.65	82	77.2	72	48
4	0.85	82	76.3	72	57
5	1.0	82	75.8	72	62
6	1.2	82	75.0	72	70
7	1.3	82	74.6	72	74

TABLE - 5.3

No. of Banks = 1 Direction of Spray = Upstream
Air Velocity = 500 fpm

RECIRCULATED WATER.

S.No.	Water quantity gpm /nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency as calculated percent
1	0.2	80.5	77.4	70.5	26
2	0.5	"	75.9	"	46
3	0.65	"	75.1	"	54
4	0.85	"	74.3	"	62
5	1.0	"	73.6	"	69
6	1.2	"	72.9	"	76
7	1.3	"	72.7	"	79

TABLE - 5.4

RECIRCULATED WATER.

No. of banks. = 1 Direction of spray = Upstream
Air velocity - 400 fpm

S.No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.2	81.0	77.9	71.0	31
2	0.5	"	75.8	"	52
3	0.65	"	75.0	"	60
4	0.85	"	74.1	"	69
5	1.0	"	73.5	"	75
6	1.2	"	72.8	"	82
7	1.3	"	72.5	"	85

TABLE - 6.5

RECIRCULATED WATER

No. of banks = 1 Direction of spray = Downstream
Air velocity = 700 fpm

S.No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated.
1	0.2	81	79.1	71	19
2	0.5	"	77.7	71	33
3	0.65	"	77.0	71	40
4	0.85	"	76.2	71	48
5	1.0	"	75.7	71	53.5
6	1.2	"	74.9	71	61
7	1.3	"	74.6	71	64

TABLE - 6.6

RECIRCULATED WATER.

No. of banks = 1 Direction of spray = Downstream
Air velocity = 600 fpm

S.No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.2	81	78.8	71	22
2	0.5	81	77.1	71	39
3	0.65	81	76.4	71	46
4	0.85	81	75.5	71	55
5	1.0	81	74.9	71	61
6	1.2	81	74.2	71	68
7	1.3	81	73.8	71	72

TABLE - 5.7

RECIRCULATED WATER

No. of banks = 1 Direction of spray = Downstream
Air velocity - 500 fpm

S.No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency % as calculated.
1	0.2	80.8	78.3	70.8	25
2	0.5	"	76.4	"	44
3	0.65	"	75.6	"	52
4	0.85	"	74.7	"	61
5	1.0	"	74.1	"	67
6	1.2	"	73.4	"	74
7	1.3	"	73.0	"	77.5

TABLE - 5.8

RECIRCULATEDWATER.

No. of banks = 1 Direction of spray - Downstream

Air velocity - 400 fpm

S.No.	Water quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.2	80.9	77.8	70.5	30
2	0.5	"	75.8	"	50
3	0.65	"	74.9	"	58
4	0.85	"	73.8	"	68
5	1.0	"	73.2	"	74
6	1.2	"	72.5	"	80.5
7	1.3	"	72.2	"	83.8

TABLE - 5.9

RECIRCULATED WATER.

No. of banks = 2 Direction of spray - Both downstream
Air velocity - 700 fpm.

S. No.	Water quantity gpm/ nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency % as calulated
1	0.2	81	78.9	71	21
2	0.5	"	77.3	"	37
3	0.65	"	76.4	71	46
4	0.85	"	75.9	71	51
5	1.0	"	75.2	71	58
6	1.2	"	74.6	71	64

TABLE - 5.10

Recirculated Water.

No. of banks - 2, Direction of spray - Both downstream

Air velocity - 600 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t1	Entering air wtb twb	Efficiency % as calculated
1	0.2	81.1	78.9	71.3	22.5
2	0.5	"	76.8	"	43.8
3	0.65	"	75.9	"	52
4	0.85	"	75.4	"	58.2
5	1.0	"	74.7	"	65.3
6	1.2	"	74.1	"	71.5

TABLE - 5.11

RECIRCULATED WATER.

No. of banks - 2   Direction of spray - Both downstream

Air velocity - 500 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency as caluculated %
1	0.2	80.6	78.6	70.6	26
2	0.5	"	75.6	"	50
3	0.65	"	74.8	"	58
4	0.85	"	74.0	"	66
5	1.0	"	73.3	"	73
6	1.2	"	72.8	"	78

TABLE No. 5.12

RECIRCULATED WATER.

No. of banks. - 2   Direction of spray - Both downstream

Air velocity - 400 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency % as calculated
1	0.2	80.5	77.4	70.3	30.4
2	0.5	"	74.7	"	57
3	0.65	"	73.8	"	66
4	0.85	"	73.1	"	72.5
5	1.2	"	72.4	"	79.5
6	1.3	"	71.9	"	84

TABLE - 5.13

RECIRCULATED WATER.

No. of banks - 2 Direction of spray - Both oppsoing each other.

Air velocity - 700 fpm.

S. No.	Water quantity gpm/ nozzle.	Enterimg air dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency as calculated %
1	0.2	80.8	78.4	70.3	23
2	0.5	"	76.7	"	39
3	0.65	"	76.0	"	46
4	0.80	"	75.3	"	52
5	1.0	"	74.5	"	60
6	1.2	"	73.8	"	66

TABLE-5.14

Recirculated Water.

No. of banks -2 Direction of Spray - Both opposing eachother

Air velocity - 600 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering airt dbt te	Leaving air dbt t_1	Entering air wtb twb	Efficiency % as calculated
1	0.2	81	78.5	70.9	24.8
2	0.5	"	76.4	"	45.5
3	0.65	"	75.6	"	53.5
4	0.80	"	74.9	"	60
5	1.0	"	74.2	"	67.3
6	1.2	"	73.6	"	73

TABLE - 5.15

Recirculated Water.

No. of banks - 2 Direction of spray - Both opposing each other.

Air velocity - 500 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t1	Entering air wtb twb.	Efficiency % as calculated U
1	0.2	80.6	77.8	70.6	28
2	0.5	"	75.4	"	52
3	0.65	"	74.5	"	61
4	0.80	"	73.9	"	67
5	1.0	"	73.2	"	74
6	1.2	"	72.6	"	80

TABLE - 5.16

Recirculated Water.

No. of banks. 2 Direction of spray - Both opposing eachother

Air velocity - 400 fpm.

S. No.	Water Quantity gpm/ nozzle.	Entering air dbt te	Leaving air dbt t_1	Entering air twb twb	Efficiency % as calculated
1	0.2	80.5	77.2	70.3	32.4
2	0.5	"	74.5	"	58.9
3	0.65	"	73.7	"	66.8
4	0.80	"	73.0	"	73.5
5	1.0	"	72.3	"	80.5
6	1.2	"	71.7	"	86

TABLE - 5.17

RECIRCULATED WABER.

No. of banks - 2 Direction of spray - Both upstream

Air velocity - 700 fpm

S. No.	Water quantity gpm/ nozzle.	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.35	81	78.0	71	30
2	0.50	"	77.0	"	40
3	0.65	"	76.2	"	48.5
4	0.80	"	75.5	"	55
5	1.0	"	74.75	"	62.5
6	1.2	"	74.1	"	69

TABLE - 5.18

Recirculated Water.

No. of banks - 2 Direction of spray - both upstream

Air velocity - 600 fpm.

S. No.	Water quantity gpm/ nozzle.	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.35	80.8	77.5	71	33.8
2	0.5	"	76.2	"	47
3	0.65	"	75.3	"	56
4	0.80	"	74.6	"	63.2
5	1.0	"	73.9	"	70.5
6	1.2	"	73.3	"	76.5

TABLE – 5.19

RECIRCULATED WATER.

No. of banks – 2 Direction of spray – Both upstream

Air velocity – 500 fpm

S. No.	Water quantity gpm/ nozzle.	Entering air dbt t_e	Leaving air dbt t_1	Entering art wbt twb	Efficiency % as calculated
1	0.35	80.6	76.6	70.6	40
2	0.5	"	75.3	"	53
3	0.65	"	74.3	"	63
4	0.80	"	73.6	"	70
5	1.0	"	72.9	"	77
6	1.2	"	72.4	"	82

TABLE – 5.20

Recirculated Water.

No. of banks – 2 Direction of Spray – Both upstream

Air velocity – 400 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering wbt twb	Efficiency % as calculated
1	0.35	80.5	75.5	70.3	49
2	0.50	"	74.1	"	62.7
3	0.65	"	73.1	"	72.5
4	0.80	"	72.5	"	78.3
5	1.0	"	71.9	"	84.5
6	1.2	"	71.4	"	89

TABLE - 5.21

RECIRCULATED WATER.

No. banks - 3 Direction of spray - All downstream

Air velocity - 700 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	81	78.4	71	27
2	0.50	"	77.0	"	40
3	0.65	"	76.4	"	46
4	0.80	"	75.8	"	52
5	1.0	"	75.1	"	59
6	1.1	"	74.8	"	62

TABLE - 5.22

Recirculated Water.

No. of banks - 3 Direction of spray -All downstream

Air velocity - 600 fpm

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t1	Entering air wbt twb	Efficiency % as calculated
1	0.25	80.9	77.9	70.9	30
2	0.50	"	76.2	"	47
3	0.65	"	75.5	"	54
4	0.80	"	74.9	"	60
5	1.0	"	74.2	"	67
6	1.1	"	73.8	"	71

TABLE - 5.23

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - All downstream

Air velocity - 500 fpm

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	81.2	77.5	70.3	34
2	0.50	81.2	75.2	"	55
3	0.65	"	74.4	"	62.4
4	0.80	"	73.7	"	69
5	1.0	"	73.0	"	75.2
6	1.2	"	72.7	"	78

TABLE - 5.24

RECIRCULATED WATER

No. of banks - 3 Direction of spray = All downstream

Air velocity - 400 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	80.7	76.6	70.4	40
2	0.50	"	74.3	"	62.2
3	0.65	"	73.4	"	71
4	0.80	"	72.7	"	77.5
5	1.0	"	72.2	"	82.5
6	1.1	"	71.9	"	85.5

TABLE - 5.25

RECIRCULATED WATER.

No. of banks - 3 Direction of spray = 1,upstream,2 downstrea

Air velocity - 700 fpm

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated.
1	0.25	81	78.2	71	28
2	0.50	"	77.0	"	40
3	0.65	"	76.3	"	47
4	0.80	"	75.7	"	53
5	1.0	"	74.9	"	61
6	1.1	"	74.6	"	64

TABLE - 5.26

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 1 upstream, 2 downstream

Air velocity - 600 fpm

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	80.6	77.5	70.6	32.4
2	0.50	"	76.0	"	47
3	0.65	"	75.2	"	55
4	0.80	"	74.5	"	61.8
5	1.0	"	73.7	"	69.5
6	1.1	"	73.4	"	72.5

TABLE - 5.27

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 1 upstream, 2 downstream.

Air velocity - 500 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt t_e	Leaving air dbt t_1	Entering air twb twb	Efficiency % as calculated
1	0.25	80.7	76.9	70.7	38
2	0.50	"	75.3	"	54
3	0.65	"	74.5	"	62
4	0.80	"	73.7	"	70
5	1.0	"	72.9	"	78
6	1.1	"	72.50	"	81.5

TABLE - 5.28

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 1 upstream, 2 downstream.

Air velocity - 400 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	81	76.6	71	44
2	0.50	"	74.8	"	62
3	0.65	"	74.0	"	70
4	0.80	"	73.2	"	78
5	1.0	"	72.4	"	86
6	1.1	"	72.1	"	89

TABLE - 5.29

RECIRCULATED WATER.

No. of banks - 3   Direction of spray - 2 upstream
1 downstream.

Air velocity. = 700 fpm.

S.No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entring air wbt twb	Efficiency % as calculated.
1	0.25	81	78	71	30
2	0.50	"	76.7	"	42.5
3	0.65	"	76.1	"	49
4	0.80	"	75.5	"	55
5	0.90	"	75.1	"	59
6	1.0	"	74.7	"	63
7	1.1	"	74.4	"	66

TABLE - 5.30

RECIRCULATED WATER.

No. of banks - 3   Direction of spray - 2 upstream,
1 downstream.

Air velocity - 600 fpm.

S.No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb.	Efficiency % as calculated.
1	0.25	80.9	77.5	70.9	34
2	0.50	"	76.0	"	49
3	0.65	"	75.2	"	57
4	0.80	"	74.5	"	63.5
5	0.90	"	74.0	"	68.2
6	1.0	"	73.7	"	72
7	1.1	"	73.5	"	74.5

TABLE - 5.31

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 2 upstream, 1 downstream.

Air velocity - 500 fpm.

S.No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	81.2	77.3	71.3	39.5
2	0.50	"	75.6	"	56.5
3	0.65	"	74.7	"	65.6
4	0.80	"	74.0	"	72.7
5	0.90	"	73.6	"	76.8
6	1.0	"	73.2	"	81.0
7	1.1	"	73.0	"	85.0

TABLE - 5.32

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 2 upstream, 1 downstream.

Air velocity - 400 fpm.

S.No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.25	80.5	75.8	70.4	46.5
2	0.50	"	74.0	"	64.3
3	0.65	"	73.1	"	73
4	0.80	"	72.4	"	80.5
5	0.90	"	71.9	"	85
6	1.0	"	71.6	"	88
7	1.1	"	71.4	"	90

TABLE - 5.33

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - all upstream.

Air velocity - 700 fpm.

S.No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering ait wbt twb	Efficiency % as calculated
1	0.35	81	77.5	71	35
2	0.50	"	76.6	"	44
3	0.65	"	75.9	"	51
4	0.80	"	75.3	"	57
5	1.0	"	74.5	"	64.5
6	1.1	"	74.2	"	68

TABLE - 5.34

Recirculated Water.

No. of banks - 3 Direction of spray - All upstream.

Air velocity - 600 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving airdbt t_1	Entering air wbt twb	Efficiency % as calculated.
1	0.35	80.8	76.6	70.6	40.8
2	0.50	"	75.4	"	52.5
3	0.65	"	74.6	"	60.2
4	0.80	"	74.0	"	66
5	1.0	"	73.3	"	72.8
6	1.1	"	72.9	"	76

TABLE - 5.35

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - All upstream

Air velocity - 500 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.35	81.2	76.2	70.8	48
2	0.50	"	75.0	"	60
3	0.65	"	74.1	"	68.2
4	0.80	"	73.5	"	74
5	1.0	"	72.8	"	80.8
6	1.1	"	72.5	"	83.5

TABLE - 5.36

RECIRCULATED WATER.

No. of banks. - 3 Direction of spray - All upstream.

Air velocity - 400 fpm.

S. No.	Water quantity gpm/nozzle	Entering air dbt te	Leaving air dbt t_1	Entering air wbt twb	Efficiency % as calculated
1	0.35	87	75.3	71	57
2	0.50	"	74.1	"	69
3	0.65	"	73.2	"	78
4	0.80	"	72.7	"	83
5	1.0	"	72.0	"	90
6	1.1	"	71.8	"	92

TABLE - 5.37

RECIRCULATED WATER.

No. of banks - 1 Direction of spray - Downstream

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig	Entering air dbt t_e	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated
1	5	80.8	70.8	76.4	44
2	10	"	"	75.6	52
3	20	"	"	74.7	61
4	30	"	"	74.1	67
5	40	"	"	73.4	74
6	47.5	"	"	73.0	77.5

TABLE - 5.38

RECIRCULATED WATER.

No. of banks - 1 Direction of spray - Upstream

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig	Entering air dbt te	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated
1	5	80.5	70.5	75.9	46
2	10	"	"	75.1	54
3	20	"	"	74.3	62
4	30	"	"	73.6	69
5	40	"	"	72.9	76
6	47.5	"	"	72.7	79

TABLE - 5.39

RECIRCULATED WATER.

No. of banks - 2 Direction of spray - both downstream

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig	Entering air dbt t_e	Entering ait wbt twb	Leaving air dbt t_1	Efficiency % as calculated
1	5	80.6	70.6	75.6	50
2	10	"	"	74.8	58
3	20	"	"	74.0	66
4	30	"	"	73.3	73
5	40	"	"	72.8	78

TABLE - 5.40

RECIRCULATED WATER.

No. of banks - 2 Direction of spray - Both opposing eachother.

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig	Entering air dbt te	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated
1	5	80.6	70.6	75.4	52
2	10	"	"	74.8	58
3	20	"	"	73.8	68
4	30	"	"	73.2	74
5	40	"	"	72.6	80

TABLE - 5.41

RECIRCULATED WATER.

No. of banks - 2 Direction of spray - Both upstream.

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig.	Entering air dbt te	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated.
1	5	80.6	70.6	74.9	57
2	10	"	"	74.5	61
3	20	"	"	73.6	70
4	30	"	"	72.9	77
5	40	"	"	72.4	82

TABLE - 5.42

RECIRCULATED WATER

No. of banks - 3 Direction of spray - All upstream

Air velocity - 500 fpm.

S.No.	Spray nozzle pressure psig	Entering air dbt te	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated.
1	7.5	81.2	70.3	74.4	62.4
2	15	"	"	73.7	69
3	23	"	"	73.0	75.2
4	30	"	"	72.4	81.0

TABLE - 5.43

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 1 upstream, 2 downstream.

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig.	Entering air dbt t_e	Entering air wbt t_{wb}	Leaving air dbt t_1	Efficiency % as calculated.
1	5	80.7	70.7	74.5	62
2	13	"	"	73.7	70
3	23	"	"	73.1	76
4	30	"	"	72.3	84

TABLE - 5.44

RECIRCULATED WATER.

No. of banks - 3 Direction of spray - 2 upstream. 1 downstream.

Air velocity - 500 fpm.

S. No.	Spray nozzle pressure psig	Entering air dbt te	Entering air wbt twb	Leaving air dbt t_1	Efficiency % as calculated.
1	5	80.5	70.2	73.8	65.5
2	13	"	"	73.0	73
3	23	"	"	72.2	80
4	30	"	"	71.7	85.5

TABLE - 5.45

RECIRCULATED WATER

No. of Banks =3 Air velocity = 500 fpm

Direction of spray = All upstream

S.No.	Spray nozzle pressure psig	Entering air dbt t_e	Entering air wbt twb	Leaving air dbt t_1	Efficiency percent as calculated
1.	5	80.4	70.1	73.4	68.5
2.	10	"	8"	72.7	75
3.	15	"	"	72.5	76.5
4.	20	"	"	72.1	82
5.	30	"	"	71.1	90

TABLE A.46

CHILLED WATER

Direction of Spray = Downstream No. of Banks = 1

Air velocity = 700 fpm

S.No.	Spray nozzle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	10	69	80	78.8	73.1	0.48
2.	20	"	"	78.2	72.7	0.51
3.	30	"	"	77.5	72.4	0.535
4.	40	"	"	76.9	72.1	0.57
5.	47.5	"	"	76.2	71.7	0.595

TABLE 5.47

No. of Banks =1 Air velocity =600 fpm.

Direction of spray = Downstream

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	10	70	80	79.1	74.2	0.515
2.	20	"	80	78.2	73.8	0.56
3.	30	"	"	77.5	73.3	0.58
4.	40	"	"	76.9	72.9	0.60
5.	47.5	"	"	75.5	72.6	0.615

TABLE -5.48

CHILLED WATER

No. of Banks = 1

Direction of spray = Downstream Air velocity 500 fpm.

S.No.	Spray nozle pressure psig	Entering water temp.	Entering air wbt twb F	Leaving air wbt twb' F	Leaving water temp. tw' F	Performance factor as calculated
1.	10	70.5	80.5	79.3	74.9	0.56
2.	20	"	"	78.6	74.4	0.58
3.	30	"	"	77.9	74.0	0.63
4.	40	"	"	77.4	73.7	0.63
5.	47.5	"	"	77.4	73.7	0.63

TABLE -5.51

CHILLED WATER

No. of Banks = 1 Air velocity = 600 fpm

Direction of spray = Upstream.

S.No.	Spray nozle pressure psig	Entering water temp. tw oF	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1	15	70	80	78.8	74.3	0.55
2	25	"	"	78.0	73.8	0.58
3	35	"	"	77.3	73.3	0.595
4	45	"	"	76.4	72.55	0.614

TABLE -5.52

CHILLED WATER

No. of Banks =1

Direction of Spray = Upstream Air velocity = 500 fpm

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb F	Leaving air wbt twb' F	Leaving water temp. tw' F	Performance factor as calculated
1.	15	70	80	78.7	74.42	0.572
2.	25	"	"	76.4	72.9	0.60
3.	35	"	"	76.3	72.55	0.625
4.	45	"	"	75.8	72.15	0.635

TABLE - 5.54

CHILLED WATER

No. Banks =2

Direction of spray - Both Upstream Air Velocity = 700 fpm

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	10	69.5	80	78.7	75.97	0.74
2.	20	"	"	78.0	75.7	0.78
3.	30	"	"	77.1	75.3	0.825
4.	40	"	"	76.4	74.9	0.86

TABLE -5.63

CHILLED WATER

No. of Banks =1

Direction of spray = Upstream Air velocity =400 fpm

S.No.	Spray nozle pressure psig	Entering water temp.	Entering air wbt twb F	Leaving air wbt twb' F	Leaving water temp tw' F	Performance factor as calculated
1.	15	70	80.4	79.5	75.55	0.62
2.	25	"	8"	79.0	75.15	0.63
3.	35	"	"	78.3	74.65	0.65
4.	45	"	"	79.7	74.15	0.66

TABLE -5.55

CHILLED WATER

No. of Banks = 2

Direction of spray = Both upstream. Air velocity =600 fpm.

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	10	70	80	78.9	76.7	0.78
2.	20	"	"	78.0	76.3	0.825
3.	25	"	"	77.6	76.0	0.835
4.	35	"	"	76.9	75.7	0.875
5.	40	"	"	76.5	75.4	0.89

TABLE - 5.56

CHILLED WATER

No. of Banks = 2 Air velocity=500 fpm.

Direction of spray= Both upstream

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	10	68	80.2	78.3	76.1	0.815
2.	15	"	"	77.7	75.7	0.83
3.	20	"	"	77.2	75.4	0.85
4.	30	"	"	76.5	75.1	0.885
5.	40	"	"	76.0	74.7	0.905

TABLE - 5.57

CHILLED WATER

No. of Banks = 3 Air velocity = 700 fpm

Direction of spray = 2 Upstream , 1 Downstream.

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	5	67	80.3	76.5	73.9	0.80
2.	12	"	"	75.7	73.5	0.83
3.	20	"	"	75.2	73.2	0.85
4.	25	"	"	74.9	73.0	0.855
5.	33	"	"	74.3	72.7	0.88

TABLE - 5.58

CHILLED WATER

No. of Banks = 3 Air velocity = 600 fpm.

Direction of spray = 2 Upstream, 1 Down stream

S.No-	Spray nozle pressure psig	Entering water temp. tw. °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Performance factor as calculated
1.	5.	67	80.3	76.2	74.1	0.84
2.	12	"	"	75.5	73.7	0.865
3.	20	"	"	75.2	73.5	0.87
4.	35	"	"	74.7	73.2	0.887
5.	33	"	"	74.3	72.9	0.905

TABLE - 5.59

CHILLED WATER

No. of Banks = 3 Air velocity = 500 fpm

Direction of spray = 2 Upstream , 1 Downstream.

S.No.	Spray hozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Leaving air wbt twb' °F	Leaving water temp. tw' °F	Perofrmance factor as calculated
1.	7.5	66	80.3	74.7	73.0	0.88
2.	15	"	"	74.3	72.7	0.89
3.	20	"	"	73.9	72.5	0.90
4.	25	"	"	73.6	72.3	0.91
5.	32	"	"	73.2	72 .0	0.92

TABLE - 5.60

CHILLED WATER

No. of Banks = 3 Air velocity = 400 fpm

Direction of spray = 2 Upstream , 1 Downstream

S.No.	Spray nozle pressure psig	Entering water temp. tw °F	Entering air wbt twb °F	Eeaving air wbt twb' °F	Leaving water temp. tw' °F	Peroformance factor as calculated
1.	7.5	66.5	80.5	74.8	73.3	0.905
2.	15	"	"	74.4	73.1	0.91
3.	20	"	"	73.7	72.7	0.93
4.	25	"	"	73.7	72.7	0.93
5.	30	"	"	73.4	72.5	0.935

TABLE No. - 5.61

HEATED WATER

No. of Banks = 2
Both Upstream

Entering dbt = 81.6
Air Velocity = 500

S.No.	Pressure	GPM/nozzle	Entering water temp. tw	Ent. wbt twb	Leaving W.B.T twb'	Leaving water tw'	Leaving air dbt	1 - $\frac{tw' - twb'}{tw - twb}$
1	7.5	0.35	84	71	72.8	77.0	81.8	.79
2	15	0.50	84	71	74.9	79.1	82.1	.81
3	25	0.65	84	71	76.9	80.8	82.5	.83
4	32	0.80	84	71	78.5	81.9	82.9	.855
5	40	1.0	84	71	80.2	83.5	83.7	.865

TABLE No.5.62

HEATED WATER

No. of Banks = 2

Direction of spray = Both Upstream

Air Velocit = 600 fpm
Entering air dbt. = 80.7

S.No.	Pressure	GPM/nozzle	Entering water tw	Entering air wbt twb	Leaving air wbt twb'	Leaving water temp. tw'	Leaving air dbt	1 - $\frac{tw' - twb'}{tw - twb}$
1	7.5	0.35	93	70.3	71.7	77.2	81.2	.756
2	15	0.50	93	"	73.2	78.4	81.8	.77
3	25	0.65	93	"	75.1	79.4	82.2	.81
4	32	0.80		"	77.4	81.3	82.6	.83
5	40	1.0	93	"	78.9	82.4	82.9	.845

TABLE NO.- 5.63

HEATED WATER

No. of Banks = 2 — Air Velocity =700 fpm

Direction of spray = Both Upstream — Entering air dbt. =80.6

S.No.	Pressure	gpm / nozzle	Entering water temp. tw	Entering wbt twb	Leaving wbt twb'	Leaving water tw'	leaving air dbt	$1 - \frac{tw'-twb'}{tw-twb}$
1	7.5	0.35	92	70.5	71.1	77.15	80.7	.718
2	15	0.50	92	70.5	72.2	77.6	81.0	.75
3	25	0.65	92	70.5	74.1	78.6	81.3	.79
4	32	0.80	92	70.5	76.1	80.2	81.6	.81
5	40	1.0	92	70.5	77.8	81.5	82.4	.83

HEATED WATER

TABLE - 5.64

S. No.	No. of banks	Direction of spray.	Air velocity fpm	Water quantity lb/min/nozzle	Entering air dbt	Leaving air dbt when water is re circulated.	Entering air wbt	η as calculated.	Entering water Temp.	Leaving water Temp.	Leaving air wbt	Leaving air dbt
1	2	Both D/S	600	12	82	75.5	73	72.3	90	81.9	79.8	82.9
2	2	Both apposing	600	10	80	73.2	70	68	95	82.9	80.7	82.6
3	1	Upstream	500	12	82	75	72	70	98	81.3	77.8	82.7
4	3	All Down stream	600	8	80	73.6	70	64	96	83.2	80.9	82.9
5	3	2 Upstream 1 D.stream	500	8.5	81.2	73.8	71.3	75	93.5	82.9	82.1	84.7
6	3	1 upstream 2 D.stream	500	10	81	72.42	70	78	91	81.9	80.8	83.6
7	1	Downstream	500	12	80.8	73.4	70.8	74	96	82.2	78.3	81.7
8	3	All upstream	700	11	80.5	73.3	70	68.5	92	81.8	80.6	83.3

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TABLE - 5.65.

HEATED WATER

Serial No.	No. of banks.	Direction of spray.	Air velocity fpm	Water quantity lbs/min/nozzle.	Entering air dbt	Leaving air dbt when water is re-circulated.	Entering air wbt	η as calculated.	Entering water temperature.	Leaving water temperature.	Leaving air wbt	Leaving air dbt
1	2	Both upstream	500	8	80.6	73.6	70.6	70	93.5	81.2	78.1	81.9
2	2	Both down stream	500	12	80.6	72.8	70.6	78	92	82.1	79.8	81.9
3	2	Both opposing each other.	500	10	80.6	73.2	70.6	74	96	82.3	80.1	~~84.5~~ 84.5
4	3	All down stream	400	10	80.7	72.2	70.4	82.5	90	82.6	81.8	83.1
5	3	1 upstream 2 D.stream	500	10	80.7	72.9	70.7	78	91	81.2	80.3	82.9
6	3	2 upstream 1D.stream	600	11	80.9	73.5	70.9	74.5	92.5	82.60	81.50	83.6
7	3	All up-stream	400	10	81	72	71	90	90	82.7	82.1	83.2
8	3	All down stream	600	11	80.9	73.8	70.9	71	92	82.7	81.5	83.7

TABLE - 5.66

HEATED WATER

Serial Number	No. of banks.	Direction of spray	Air velocity fpm	Water quantity lbs/min/nozzle	Entering air dbt	Leaving air dbt when water is recirculated	Entering air wbt	η as calculated.	Entering water temperature	Leaving water temperature	Leaving air wbt	Leaving air dbt.
1	3	1 Upstream 1 downstream	600	11	80.8	73.4	70.6	72.5	93	82.5	81.4	83.8
2	3	2 upstream 1 downstream	500	8	81.2	74	71.3	72.7	91.5	82.1	80.9	82.4
3	3	All upstream	500	10	81.2	72.8	70.8	80.8	94	83.3	82.3	84.1
4	2	Both down-stream	500	8	80.6	74	70.6	66	95	81.3	78.5	81.9
5	2	Both opposing	600	12	81	73.6	70.9	73	94	81.1	78.3	81.7
6	2	Both upstream	500	10	80.6	73.0	70.8	77	95	82.4	79.7	82.8

CHAPTER - VI

CALCULATIONS, RESULTS AND GRAPHS

CHAPTER - VI.

CALCULATIONS, RESULTS AND GRAPHS :

6.1 EFFICIENCY OF AIR WASHER :-

When water is recirculated, the efficiencies of air washer calculated under different arrangements of spray banks, direction of spray water and air velocity, are shown from tables 5.1 to 5.45.

These efficiencies have been calculated by the formula given in 3.4, i.e.

$$E = \frac{\text{Entering air dbt} - \text{Leaving air dbt}}{\text{Entering air dbt} - \text{Entering air wbt}}$$

Sample Calculation :-

From table 5.37, at a pressure of 40 lbs./sq.inch.

t_e = entering air dbt = 80.8 F

twb = entering air wbt = 70.8 F

t_1 = leaving air dbt = 73.4

Hence,

$$= \frac{80.8 - 73.4}{80.8 - 70.8} = 74\%$$

6.2 PERFORMANCE FACTOR OF AN AIR WASHER :-

The performance factor of an air washer when cooling and dehumidification process takes place is defined as;

$$P.F. = (1 - \frac{t_{Wb}' - t_W'}{t_{Wb} - t_W})$$

WHERE, t_{Wb} = entering air wbt F

$t_{Wb'}$ = leaving air wbt F

t_W = entering water temperature F

$t_{W'}$ = leaving water temperature F.

Sample Calculation :-

From table 5.55, at a pressure of 35 psig,

t_{Wb} = 80

t_W = 70

t_{Wb}' = 76.9

$t_{W'}$ = 75.7

$$P.F. = 1 - \frac{76.9 - 75.7}{80 - 70} = 0.88$$

The performance factor for various combinations of number of spray banks and direction of spray, have been calculated likewise, and given in the observation tables from 5.46 to 5.60 for cooling and dehumidification process.

Similarly, for heating and humidification the factor $\left(1 - \frac{t_{W'} - t_{Wb}'}{t_W - t_{Wb}}\right)$ has been calculated for upstream direction of spray, the number of banks being 2 and with different air velocities. These observations have been shown from Table 5.61 to 5.63.

6.3 GRAPHS :-

6.3.1 GPM per NOZZLE VS EFFICIENCY :-

Hows ? To predict the performance of air washer, its efficiencies at different combinations of spray banks, direction of spray water and air velocity have been plotted against gpm per nozzle as shown from Fig. 6.16 to 6.20. The arrangement of flow has been shown on each graph.

6.3.2 PRESSURE VS EFFICIENCY AT CONTT. VELOCITY :-

Next the relation between spray nozzle pressure with air washer efficiency has been plotted at different combinations of number of banks, direction of spray and at constant velocity of 500 fpm. This is as shown in Fig. 6.21.

6.3.3 SPRAY NOZZLE PRESSURE VS PERFORMANCE FACTOR :-

When cooling and dehumidification, the performance factor for different combinations of spray banks and directions of spray at different air velocities, has been plotted against spray nozzle pressure. These are shown from Fig. 6.22 to 6.24

6.3.4 SPRAY NOZZLE PRESSURE VS FACTOR $1 - t_W'-t_{Wb'}/t_W-t_{Wb}$:-

When heating and humidification, number of spray banks 2, direction of spray both upstream and at different air velocities, the spray nozzle pressure has been plotted against the factor $\left(1 - \frac{t_{W'} - t_{Wb}'}{t_W - t_{Wb}}\right)$ This has been shown in Fig.6.24.

6.4 PATH OF HEATING AND HUMIDIFICATION PROCESS :-

The heating and humidification process, when hot water is supplied to the nozzles from a source external to the spray chamber, has been plotted on psychrometric charts at different combinations of no. of spray banks, and direction of spray at various air velocities, with different hot water temperatures,. These have been shown from Fig. 6.8 to Fig. ~~six~~ 6.15.

6.4.1 METHOD OF PLOTTING THE CURVE :-

From the law of conservation of energy :-

Heat absorbed by the air = heat given by water

i.e. enthalpy change of air stream = heat given by hot fluid.

$$\text{or } M_a(h_b - h_a) = M_w (T_1 - T_2)$$

where M_a = weight rate of flow of air stream lbs./min.

M_w = weight rate of flow of water per minute.

T_1 = entering temperature of water o F

T_2 = leaving temperature of water ^oF

h_b = enthalpy of leaving air BTU/lb.

h_a = enthalpy of entering air BTU/lb.

Since in this process the surface temperature is continuously and progressively changing as the air stream passes, the variable in the above formula is the leaving temperature of water. Denoting the varying surface temperature as T and corresponding enthalpy as h we write the formula as

$$(h - h_a) = \frac{M_w}{M_a}(T_1 - T)$$

We shall assume different wet bulb temperatures and corresponding surface temperatures will be obtained.

6.4.2 ILLUSTRATIVE EXAMPLE :-

For drawing the heating and humidification curve as shown in Fig. 6.11, the procedure is as follows :-

No. of banks = 2 both upstream.

Air velocity = 500 fpm

Entering air dbt = 81.5

Entering air wbt = 71

Entering water temp. = 94

Water quantity per nozzle = 8 lbs/ min.

Specific volume at entering condition of air = 13.92

Area of cross section of airwasher = 12.25 sq.ft.

$$\therefore M_a = \frac{\text{Velocity x area}}{\text{specific volume}} = \frac{500 \times 12.25}{13.12}$$

$$= 440 \text{ lbs/min.}$$

No. of nozzles = 16 per bank.

$$M_w = 16 \times 2 \times 8 = 256$$

$$\therefore M_a/M_w = 440/256 = 1.715$$

$$h_a = \text{enthalpy of entering air} = 34.92$$

$$(h-h_a) = \frac{M_w}{M_a} (94 - T)$$

Taking wet bulb temp. = 71 ^{o}F, h = 34.92,

Hence T = 94.

When W.B.T. = 73, h = 36.75,

$$(36.75 - 34.92) = \frac{1}{1.715} (94 - T)$$

Hence T = 90.86

When Wbt $= 75$, $h = 38.6$,

$$(38.6 - 34.92) = \frac{1}{1.715}(94 - T)$$

hence $T = 87.7$

When wbt $= 77$ F, $h = 40.6$,

$$(40.6 - 34.92) = \frac{1}{1.715}(94 - T)$$

Hence $T = 84.25$

When WBT $= 79$, $h = 42.6$

$$(42.6 - 34.92) = \frac{1}{1.715}(94 - T)$$

Hence $T = 80.8$

When W.B.T. $= 80$, $h = 43.7$,

$$43.7 - 34.92) = \frac{1}{1.715}(94 - T)$$

hence $T = 79.0$

Now the entering state of the air is located on the psychrometric chart as shown in Fgi. 6.11, denoted by 'a'. At condition 'a' the wet bulb temperature is 71 and the corresponding surface temperature is 94 F as given by point a' on the Fig. 6.11.

Points a and a' are connected by a straight line, Next the wet bulb temperature of 73 F, is taken along this line as located by point '1' and the corresponding surface temperature is 90.86 as shown at point 1'. Points 1 and 1' are joined by a straight line. Proceeding like wise the next wet bulb temp. of 75 is located on the line 11' as given by point 2. The corresponding surface temperature is located by the point 2' on the saturation line. Then the points 2 and 2' are joined and the next point taken along the line joining 22'. Thus the procedure is repeated until the point b is obtained which gives the condition of the leaving air. When the wet bulb temperature of 80 F is taken i.e. point '5' the corresponding surface temperature is 79 as located by the point 'c'. The curve joining the points a, 1,2,3,4,5 and c gives the path of humidification process when the water temperature is heated more than the dry bulb temperature of the entering air. The line joining the points a and b and extended cuts the saturation line at point S The ratio of Sb to Sa gives the bypass factor. In other way, if the efficiency of the washer and the path of humidification are known, the condition of the leaving air can be obtained. The method is that the line 'aS' is adjusted in such a way that it cuts the humidification curve at b such that the ratio of 'ab' to 'aS' gives the efficiency or the ratio of Sb to Sa gives the bypass factor.

The condition of the leaving air is obtained in the way described above when the efficiency of air washer is known, and compared with the experimental and theoretical results (as described below). The comparison is shown in Table 6.2.

6.4.3 THEORETICAL ANALYSIS :-

This method of obtaining the leaving water temperature and the condition of leaving air is described with an illustration.

For the same conditions as shown in Fig. 6.11, the calculations are made as follows :-

Data :- No. of banks = 2

Direction of spray = both upstream

Air velocity = 500 fpm

Entering water temp = 94 F

For the above arrangement first the efficiency of the air washer is obtained. Thus,

entering air dbt = 81.5 F

entering air wbt = 71 F

leaving air dbt = 74.15

This gives the efficiency ;

= 81.5 - 74.15 / 81.5 - 71 = 70%

Hence the bypass factor = 1 - η = 0.3 = X

From the Fig. 4.8 (a) assuming zero bypass factor the temp. of leaving water. T_o is obtained by the method shown on the graph itself. Then the value 'b' is obtained from Fig. 4.8 (b) we get,

$$T_o = 79.9, \quad b = 2.85$$

The value of leaving water temp. at bypass factor = 0.3 is then obtained by

$$T_2 = T_o + (X)^b \, (T_1 - T_o)$$

from eqn.5 of 4.11.3, assuming enthalpy instead of sigma f_n

$$= 79.9 + (0.3)^{2.85} \, (94 - 79.9)$$

$$= 80.35$$

Next the leaving wet bulb temperature is obtained by

$$(h_b - h_a) = \frac{M_W}{M_a} (T_1 - T_2)$$

hence $h_b = h_a + (T_1 - T_2) \; M_W/M_a$

$$= 34.92 \frac{(94 - 80.35)}{1.715}$$

$$= 42.87$$

Corresponding leaving wet bulb temp. from the chart comes out to be 79.18.

The exist wet bulb temperature is the same as if a constant surface temp. t_s has been assumed with an equivalent bypass factor X, where t_s is defined by the following relation;

$$X = \frac{h_s - h_b}{h_s - h_a}$$

$$h_s = \frac{h_b - Xh_a}{(I - X)}$$

$$= \frac{42.87 - .3 \times 34.92}{1 - .3} = 46.4$$

Corresponding surface temp. from charts $t_s = 82.4$

The leaving air dry bulb temp. is given by

$$t_b = Xt_a + (1-X)\ t_s \quad \text{since } X = t_s - t_b / t_s - t_a$$

$$= .3 \times 81.5 + .7 \times 82.4$$

$$= 82.1$$

Comparing the experimental, graphical and theoretical values we see ;

S.No.	Entering water temp. F T_1	Leaving water T_2	Leaving air wbt tb'	Leaving air dbt tb	Method of result obtained	Ref.
1	94	81.9	79.5	82.9	Experimental	Table 5.6 Obs. 4
2	94	-	80	82.2	Graphical	Fig. 6.11
3	94	80.35	79.18	82.1	Theoretical.	shown above

TABLE - 6.1

CALCULATIONS CHART FOR HEATING AND HUMIDIFICATIO

	No.of bank	Direction of spray	Air velocity fpm	Entering air dbt.	Leaving air dbt when water recirculated	Entering air wbt	η %age	$X = 1 - \eta$	M_a/M_w	Entering water temp T_1	T_o from graph. 4.86	b from graph 5.82	Leaving water temp $T_2 = T_o + (X)(T_1 - T_2)$
	1	Upstream	600	82	75	72	70	0.3	2.75	98	77.5	4	77.668
	1	Downstream	500	80.8	73.4	70.8	74	0.26	2.29	96	79	3.4	79.17
	2	Both D.stream	600	82	75.5	73	72.3	0.277	1.37	90	79.5	3.55	79.9
	2	Both oppo-sing.	600	80	73.2	70	68	0.32	1.65	95	79.8	2.9	80.348
	2	Both Upstream	500	81.5	74.15	71	70	0.3	1.715	94	79.9	2.85	80.35
	3	All D.stream	600	80	73.6	70	64	0.36	1.38	96	81	2.6	82.05
	3	1 upstream 2 downstream	500	81	72.42	70	78	0.25	.92	91	80.2	2.1	80.64
	3	2 upstream 1 downstream	500	81.2	73.8	71.3	75	0.21	1.07	93.5	81.2	2.3	81.69
	3	All upstream	700	80.5	73.3	70.0	68.5	0.315	1.17	92.	80	2.3	80.84
)	2	Both D/stream	500	80.6	72.8	70.6	73	0.22	1.15	92	81.2	2.3	81.53
1	2	Both opposing	500	80.6	73.2	70.6	74	0.26	1.375	96	81.3	2.6	81.74
2	2	Both Upstream	500	80.6	73.6	70.6	70	0.30	1.72	93.5	80	2.9	80.40
3	2	Both Down-stream	500	80.6	74.0	70.6	66	0.34	1.72	95	79.9	2.95	80.56
4	2	Both opposing	600	81	73.6	70.9	73	0.27	1.75	94	79.7	2.9	80.02
5	2	Both Upstream	500	80.6	73.0	70.8	77	0.23	1.38	95	81.2	2.6	81.50
6	3	All down-stream	400	80.7	72.2	70.4	82.5	0.175	.738	90	81.8	1.85	82.12
7	3	1 Upstream 2 D.stream	500	80.7	72.9	70.7	78	0.22	.916	91	81.8	2.1	81.61
8	3	2 Upstream 1 d.stream	600	80.9	73.5	70.9	74.5	0.255	1	92.5	81.7	2.3	82.26
9	3	All Upstream	400	81	72.0	71	90	0.1	.732	90	82	1.86	82.11
0	3	All Down-stream	600	80.9	73.8	70.9	71	0.29	1.0	92	81.6	2.15	82.33
1	3	1 upstream 2 downstream	600	80.8	73.4	70.6	72.5	0.275	1.0	93	82.3	2.2	82.87
2	3	2 Upstream 1 downstream	500	81.2	74	71.3	72.7	0.273	1.45	91.5	80.9	2.4	81.40
3	3	All Up-stream	500	81.2	72.8	70.8	80.8	0.192	.915	94	82.8	2.3	83.05

TABLE - 6.2

COMPARISON OF RESULTS FOR HEATING AND HUMIDIFICATION.

S. No.	Condition as shown in table 6.1	Method of results obtained.	Entering air temp. T1	Leaving water temp T2	Leaving air wbt	Leaving air dbt	Reference of results obtained.
1	2	As calculated	96	79.17	78.6	80.9	As per sample calculation 6.4.3
		Graphical	96	-	79.5	81.9	Fig. 6.8
		Experimental	96	82.2	78.3	81.7	Observation No. 7 of Table No. 5.6.4
2	3	As calculated	90	79.9	80.15	82.7	As per sample calculation 6.4.3
		Graphical	90	-	80.7	83.0	Fig. 6.9
		Experimental	90	81.9	79.8	82..	Table 5.6.4, Observation No. 1.
3	4	Calculated	95	80.348	79.2	82.1	As per sample calculation of 6.4.3.
		Graphical	95	-	80.1	82.5	Fig. 6.10
		Experimental.	95	82.9	80.7	82.6	Table 5.6.4, Observation No. 2.
4	5	Calculated	94	80.35	79.18	82.1	As per sample calculation 6.4.3
		Graphical	94	-	80	82.2	Fig. 6.11
		Experimental.	94	81.9	79.5	82.9	Table 5.61, Observation No. 4
5	6	Calculated	96	82.05	80.5	82.3	As per sample calculation 6.4.3
		Graphical	96	-	81	83.1	Fig. 6.12
		Experimental	96	83.2	80.9	82.9	Table No.5.64, Observation No. 4.

Serial No	Condition as shown in Table 6.1	Method of results obtained	Entering ~~air~~ water temp. T1	Leaving water temp T2	Leaving air ~~dbt~~ wbt	Leaving dbt	Reference of results obtained
6	7	Calculated	91	80.64	81.5	83.5	As per sample calculation 6.4.3
		Graphical	91	-	81	83.2	Fig. 6.13
		Experimental	91	81.9	80.8	83.6	Table No. 5.64, Observation No. 6.
7	8	Calculated	93.5	81.69	82.4	84.4	As per sample calculation 6.4.3
		Graphical	93.5	-	82.0	84.1	Fig. 6.14
		Experimental.	93.5	82.9	82.1	84.7	Table No. 5.64, Observation No. 5.
8	9	Calculated	92	80.84	80	82	As per sample calculation 6.4.3
		Graphical	92	-	80.1	82.9	Fig. 6.15
		Experimental.	92	81.8	80.6	83.3	Table 6.64, Observation No. 8
9	10	Calculated	92	81.53	80.1	81.75	As per sample calculation 6.4.3
		Experimental	92	82.1	79.8	81.9	Table 5.65, Observation No. 2.
10	11	Calculated	96	81.74	81.1	84	As per sample calculation 6.4.3
		Experimental.	96	82.3	80.1	84.5	Table No. 5.65, Observation No. 3
11	12	Calculated	93.5	80.505	78.6	81.38	As per sample calculation 6.4.3
		Experimental.	93.5	81.2	78.1	81.9	Table 5.65, Observation No. 1.
12	13	Calculated	95	80.56	79.2	82.2	As per sample calculation
		Experimental-	95	81.3	78.6	81.9	Table No. 5.66 Observation No. 4.

Sl. No.	Condition as shown in Table 6.1	Method of results obtained.	Entering water Temp. T_1	Leaving water temp. T_2	Leaving air ~~dbt tb~~ t_{b1}	Leaving air dbt.	Reference of results obtained
13	14	Calculated	94	80.[illegible]22	79.2	81.9	As per sample calculation 6.4.3.
		Experimental	94	81.1	78.3	81.7	Table 5.66, Observation No. 5
14	15	Calculated	95	81.504	80.6	82.5	As per sample calculation 6.4.3
		Experimental	95	82.4	79.7	82.8	Table No. 5.66, Observation No. ~~4~~ 6
15	16	Calculated	90	82.128	81.1	82.6	As per sample calculation 6.4.3.
		Experimental.	90	82.6	81.8	83.1	Table No. 5.65, Observation No.4.
16	17	Calculated	91	81.61	80.8	82.8	As per sample calculation 6.4.3
		Experimental .	91	81.2	80.3	82.9	Table No. 5.65, Observation No. 5.
17	18	Calculated	92.5	82.26	81.2	83.2	As per sample calculation 6.4.3.
		Experimental.	92.5	82.60	81.50	83.6	Table 5.65, Observation No. 6.
18	19	Calculated	90	82.112	81.8	82.8	As per sample calculation 6.4.3.
		Experimental	90	82.7	82.1	83.2	Table No. 5.65, Observation No. 7
19	20	Calculated	92	82.33	80.7	83.2	As per sample calculation 6.4.3.
		Experimental.	92	82.7	81.5	83.7	Table No. 5.65, Observation No. 8

Sl.No.	Condition as shown in Table 6.1	Method of results obtained.	Entering Water Temp. T_1	Leaving water temp. T_2	Leaving air wbt t_{h1}	Leaving air dbt.	Reference of result obtained.
20	21	Calculated	93	82.872	81	83.2	As per sample calculation.
		Experimental.	93	82.5	81.4	83.8	Table No.5.66, Observation No. 1.
21	22	Calculated	91.5	81.406	80.2	82.8	As per sample calculation. 6.4.3.
		Expèrimental.	91.5	82.1	80.9	82.4	Table No. 5.66, Observation No. 2.
22	23	Calculated	94	83.052	82.7	84.4	As per sample calculation 6.4.3.
		Experimental.	94	83.3	82.3	84.1	Table No. 5.66, Observation No. 3.
23	1	Calculated	98	77.668	79.3	82.2	As per sample calculation 6.4.3
		Experimental.	98	81.3	77.9	82.7	Table No. 5.64 Ob. No 3

The above three comparisöns are made for all the 8 figures from 6.8 to 6.15 for heating and humidification. And for the other values for which the grabhs are not drawn only the experimental and theoretically obtained values are compared as given in Table 6.2. The calculations are as shown in Table 6.1.

6.5 PATH OF COOLING AND DEHUMIDIFICATION PROCESS.

When the initial temperature of water is below the initial dew point temperature of the air, the water temperature rises as it is brought in contact with the warm air. If sufficient water is provided so that the final temperature of the water is below the dewpoint temperature of the entering air cooling of air along with dehumidification will occur.

The path of the cooling and dehumidification process is plotted in a similar way as described in 6.4.2 with the help of the energy equation.

Illustrative Example :-

Taking the curve as shown in Fig. 6.5, for the conditions stated -

No. of banks	=	2
Direction of spray	-	both oppeeing eachother.
Air velocity	=	600 fpm
Entering water temp.	=	62 F
Initial air dbt	=	85.5 F
Initial air wbt	=	80 F
Quantity of water ~~af~~ through each nozzle	=	10 lbs/min.

Area of cross section of washer $= 12.25$ sq.ft.

Specific volume at enter condition of air $= 14.2$ c.ft./lb.

There are 16 nozzles per bank,

hence $M_w = 16 \times 2 \times 10 = 320$ lbs/min.

$M_a = 600 \times 12.25/14.2 = 518$

Hence $M_a/M_w = 518/320 = 1.62$

Now by the energy equation ;

$$(h_a - h) = \frac{M_w}{M_a} (T - T_1)$$

$h_a = 43.68$ = entering air enthalpy

$$(43.68 - h) = \frac{1}{1.62} (T - 62)$$

when W.bt. $= 80$, $h = 43.68$, $T = 62$,

when wbt $= 78$, $h = 41.6$,

$$(43.68 - 41.6) = \frac{1}{1.62} (T - 62)$$

Hence $T = 65.37$

When Wbt $= 76$, $h = 39.55$

$$\therefore (43.68 - 39.55) = \frac{1}{1.62} (T - 62)$$

$T = 68.7$

when Wbt $= 74$, $h = 37.68$,

$$(43.68 - 37.68) = \frac{1}{1.62} (T - 62)$$

Hence $T = 71.7$

Procedure for drawing the cooling and dehumidification curve is the same as explained in 6.4.2, excepting that the surface temperature in this case goes an increasing as the variable wet bulb temperatures aretaken in decreasing order. In Fig. 6.5, corresponding surface temperatures are a', 1', 2', 3'.
The procedure is repeated until the final state of the air as indicated by point 'b' is obtained.

The other graphs showing the path of cooling and dehumidification process for different combinations of number of spray banks, direction of spray and velocities are shown from Fig. 6.1 to 6.7.

-:0:-

6.6 CALCULATIONS FOR DRAWING THE PATHS OF HEATING AND HUMIDIFICATION

These curves have been plotted from fig.6.8 ~~and~~ & fig.6.15
The steps are listed as follows:

TABLE No.-6.3
FIG.No.6.8

Conditions as shown in table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}(h-ha)$
	1	71	34.8	96
	2	73	36.75	91.8
2	3	75	38.6	87.6
	4	77	40.6	83
	5	79	42.6	78.4

TABLE No.6.4

Fig. No.6.9

Conditions as S.No.9 in Table 6.1	S.No.	Wet bulb temp.	Enthaply h	Surface temp. $T = T_1 - \frac{Ma}{MW}(h-ha)$
	1	73	36.7	90
	2	74	37.65	88.7
	3	76	39.55	86.1
	4	78	41.55	83.35
	5	80	43.65	80.5
	6	82	45.9	77.4

TABLE No.6.5

Fig No. 6.10

Condition as shown Table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}$ (h-ha)
	1	70	34.1	95
	2	72	35.85	92.12
	3	74	37.7	89.9
4	4	76	39.6	85.92
	5	78	41.55	82.7
	6	80	43.7	79.2

TABLE NO.6.6

Fig. No. 6.11

Condition as shown in Table 6.1	S.No.	Wet bulb Temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}$ (h-ha)
	1	71	34.92	94
	2	73	36.75	90.86
5	3	75	38.6	87.7
	4	77	40.6	84.25
	5	79	42.6	80.8
	6	80	43.7	79

TABLE No.6.7

Fig. No.6.12

Condition as shown in Table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}$ (h-ha)
	1	70	34.1	96
	2	72	35.9	93.52
	3	74	37.65	91.1
	4	76	39.6	88.4
6	5	78	41.55	85.65
	6	80	43.7	82.76
	7	81	44.8	80.2

TABLE No.6.8

Fig. No.6.13

Condition as shown in Table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}$ (h-ha)
	1	70	34.1	91
	2	72	35.9	89.35
	3	74	37.7	87.7
7	4	76	39.6	85.95
	5	78	41.6	84.1
	6	80	43.7	82.2
	7	81	44.8	81.15

TABLE No.6.9

Fig. No. 6.14

Condition as shown in Table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}(h - ha)$
	1	70.8	34.7	93.5
	2	72	35.81	92.3
8	3	74	37.65	90.33
	4	76	39.55	88.3
	5	80	43.65	83.7
	6	82	45.9	81.5

TABLE No.6.10

Fig. No.6.15

Condition as shown in Table 6.1	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 - \frac{Ma}{MW}(h-ha)$
	1	70	34.1	92
	2	72	35.9	89.9
	3	74	37.7	87.85
	4	76	39.6	85.55
9	5	78	41.6	83.22
	6	80	43.7	80.8

6.7 CALCULATIONS FOR COOLING AND DEHUMIDIFICATION PROCESS.

TABLE - 6.11

Sl.No.	No.of banks	Direction of spray	Air velocity fpm	Entering air dbt	Entering air wbt	Enthalpy of entering air	Entering water temp.	M_a / M_w
1	1	Downstream	500	85.5	80	43.68	65	2.08
2	2	Both opposing each other	600	85.5	80	43.68	62	1.62
3	1	Upstream	600	84.9	80	43.68	60	2.7
4	2	Both down-stream	700	85.7	80	43.68	61	1.72
5	2	Both upstream	500	85	80	43.68	65	1.35
6	3	All upstream	700	85.8	80	43.68	63	1.26
7	3	All down-stream.	650	85.2	80	43.68	64	1.462

6.8

CALCULATIONS FOR DRAWING THE PATH OF COOLING AND DEHUMIDIFICATION

The curves for cooling and dehumidification have plotted as shown from Fig.6.1 and Fig.6.7

The steps are tested as follows:

TABLE No.612

Fig. No. 6.2

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
	1	80	43.68	65
1	2	78	41.6	69.37
	3	76	39.55	73.6

Table No. 6..13, FIG. No. 6.5

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 \ (ha-h)\frac{Ma}{MW}$
	1	80	43.68	62
		78	41.6	65.37
2		76	39.55	68.7
		74	37.68	71.7

TABLE No.6.14

Fig. No. 61

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
3	1	80	43.68	60
	2	78	41.6	65.6
	3	76	39.55	71.1

TABLE No.6.15

Fig. No.6.3

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
4	1	80	43.68	61
	2	78	41 .6	64.58
	3	76	39.55	68.1
	4	74	37.68	71.3

TABLE No.6.16

Fig.No. 6.4

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
5	1	80	43.18	63
	2	78	41.6	67.8
	3	76	39.55	70.6
	4	74	37.68	73.1

TABLE No.6.17

Fig. No. 6.7

S. No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
	1	80	43.18	63
	2	78	41.6	65.62
6	3	76	39.55	68.23
	4	74	37.68	70.55

TABLE No.6.18

Fig. No.6.6

S.No. of Table	S.No.	Wet bulb temp.	Enthalpy h	Surface temp. $T = T_1 + (ha-h)\frac{Ma}{MW}$
	1	80	43.68	60
7	2	78	41.6	65.6
	3	76	39.55	71.1

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

NO. OF BANKS 1, UPSTREAM

AIR VELOCITY = 600 F.P.M

$\frac{Ma}{Mw} = 2.7$, $T_1 = 60°F$

PSYCHROMETRIC CHART

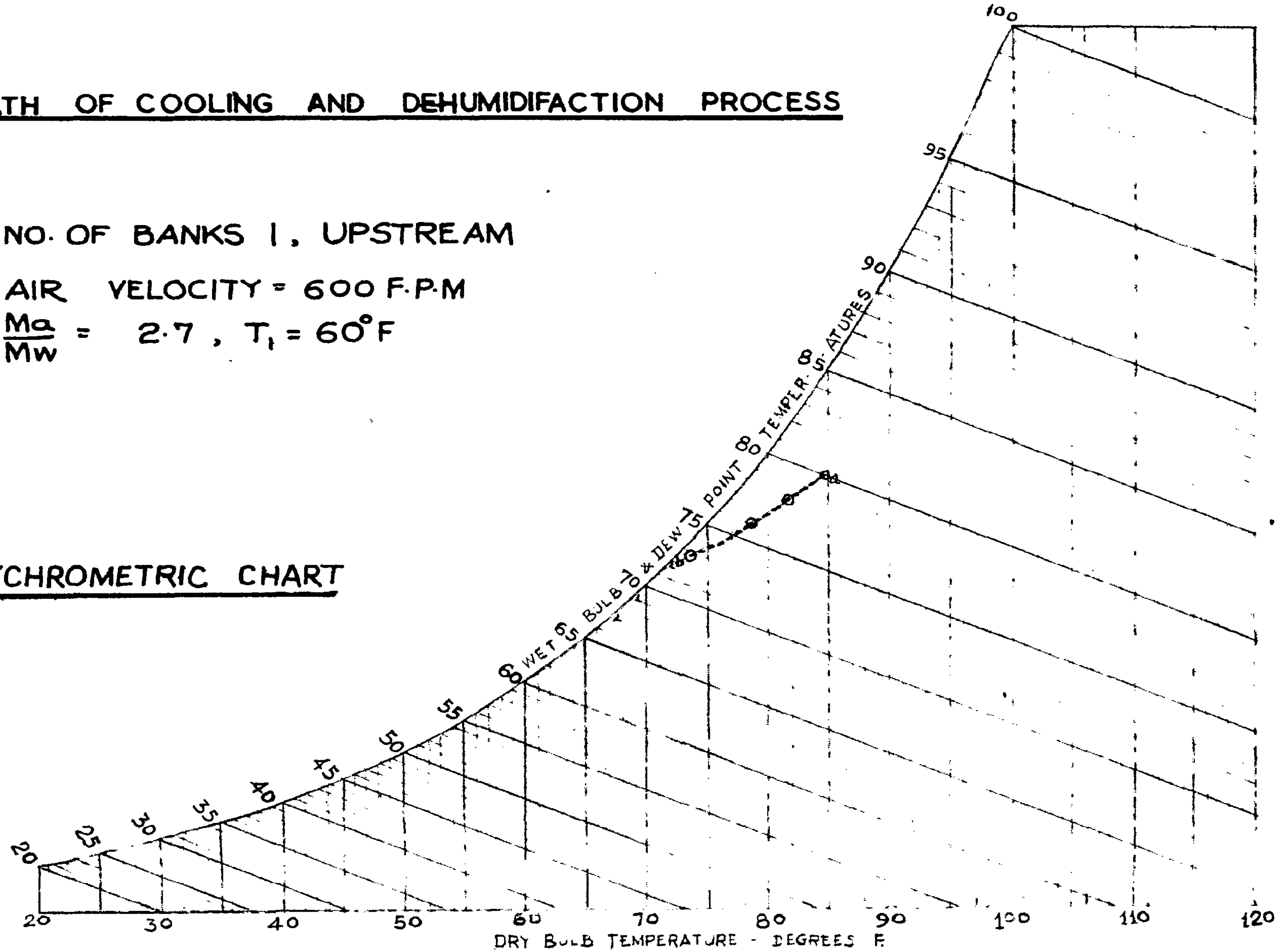


FIG 6.1

S. MEHROTRA

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

NO. OF BANKS 1, DOWN STREAM

AIR VELOCITY = 500 F.P.M

$\frac{M_a}{M_w} = 2.08$, $T_1 = 65°F$

PSYCHROMETRIC CHART

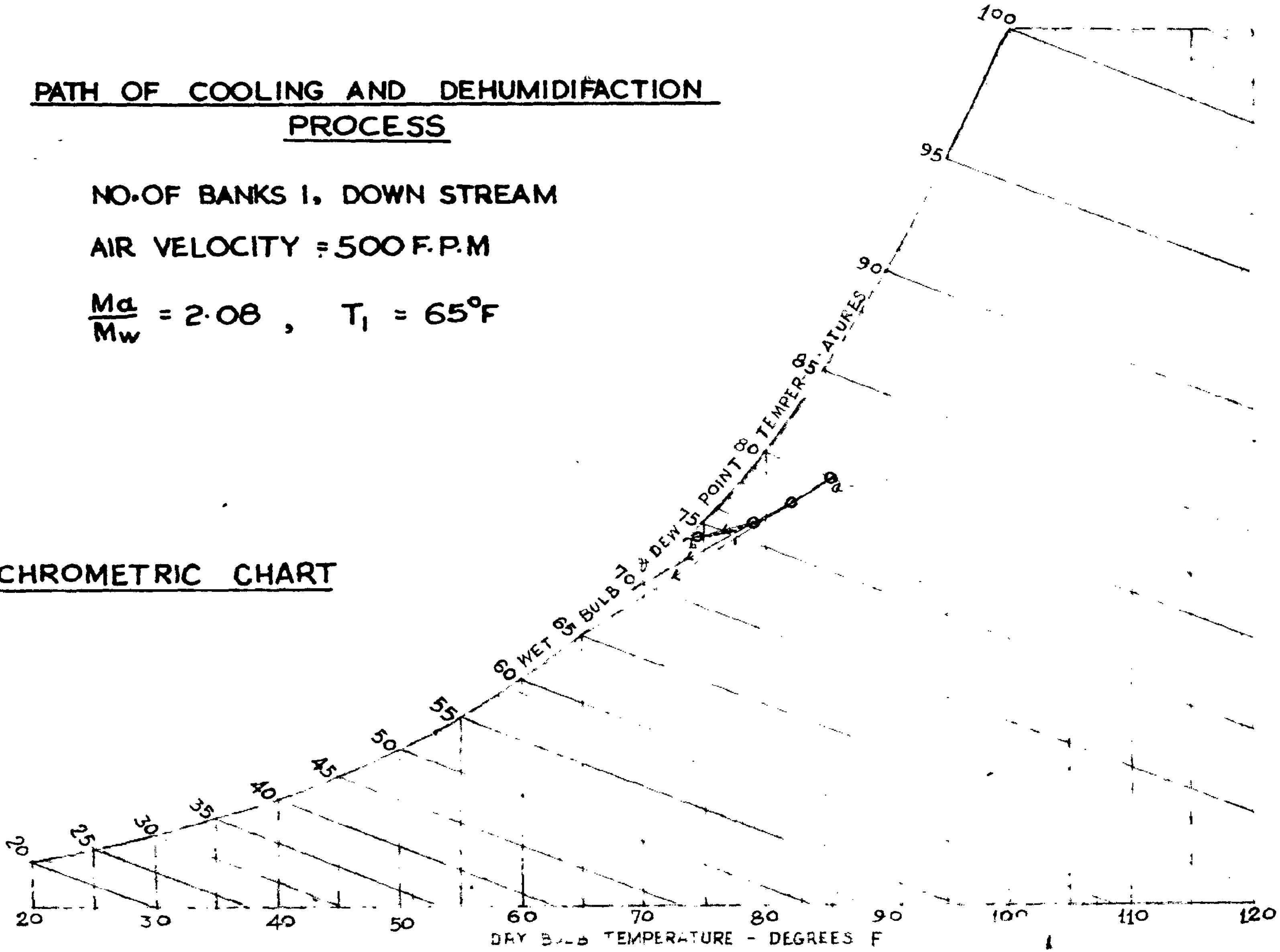


Fig 6.2

S. MEHROTRA

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

NO. OF BANKS 2

BOTH DOWN STREAM

AIR VELOCITY = 700 F.P.M

$\frac{Ma}{Mw} = 1.72, \quad T_1 = 61^\circ F$

PSYCHROMETRIC CHART

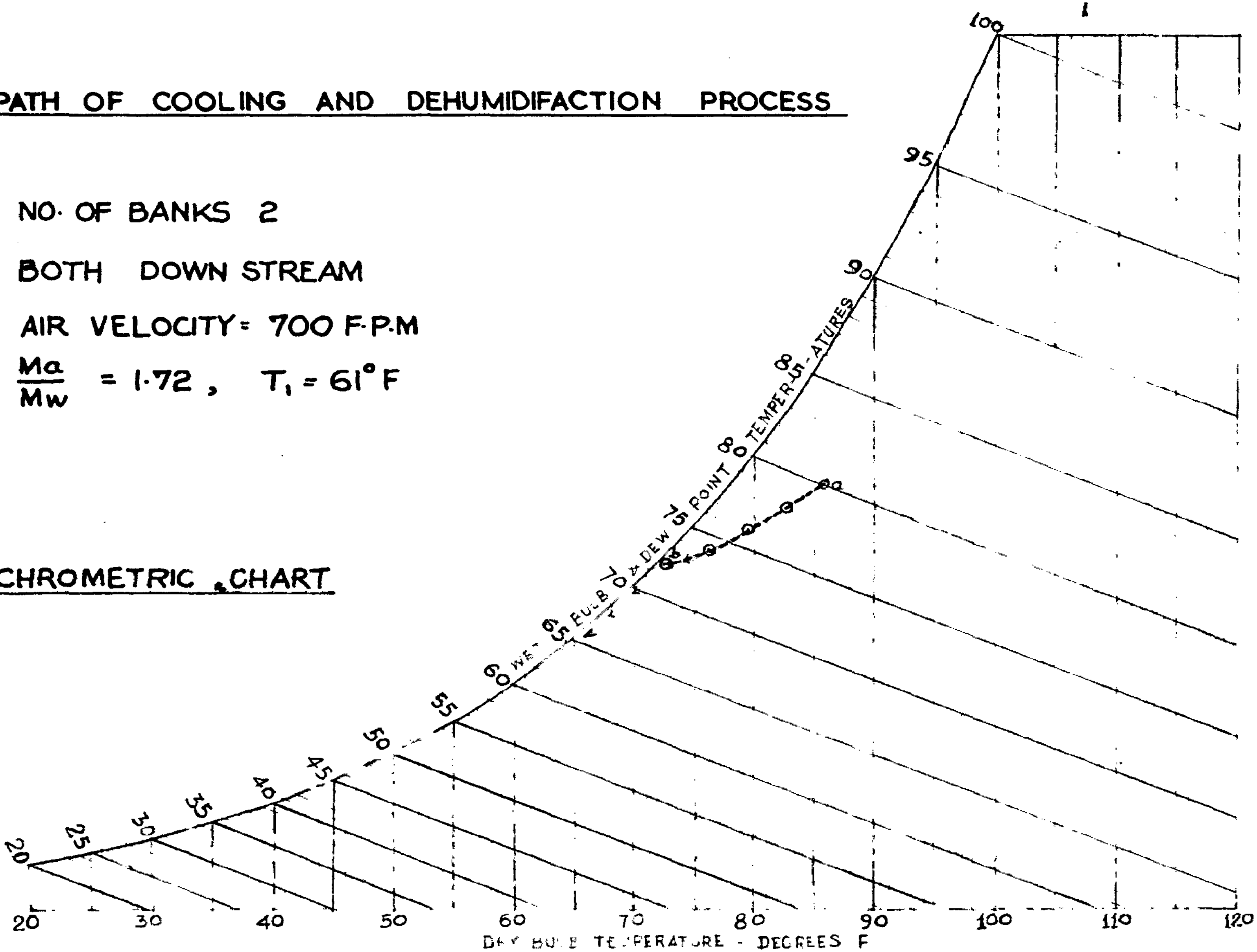


FIG 6.3

S. MEHROTRA

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

NO. OF BANKS 2

BOTH UPSTREAM

AIR VELOCITY = 500 F.P.M

$\frac{Ma}{Mw} = 1{\cdot}35, \quad T_1 = 65^{\circ}F$

PSYCHROMETRIC CHART

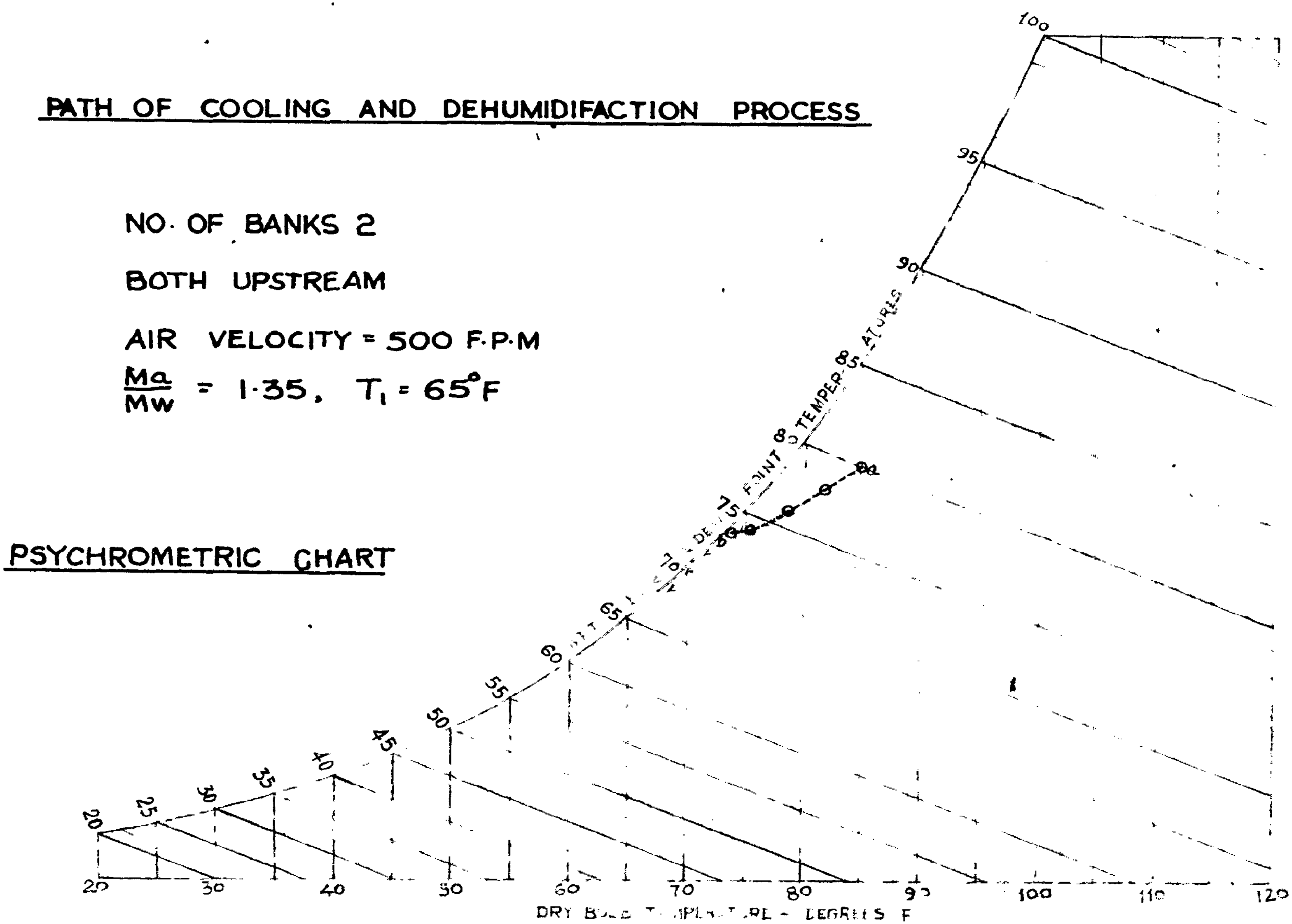


FIG 6·4

S. MEHROTRA

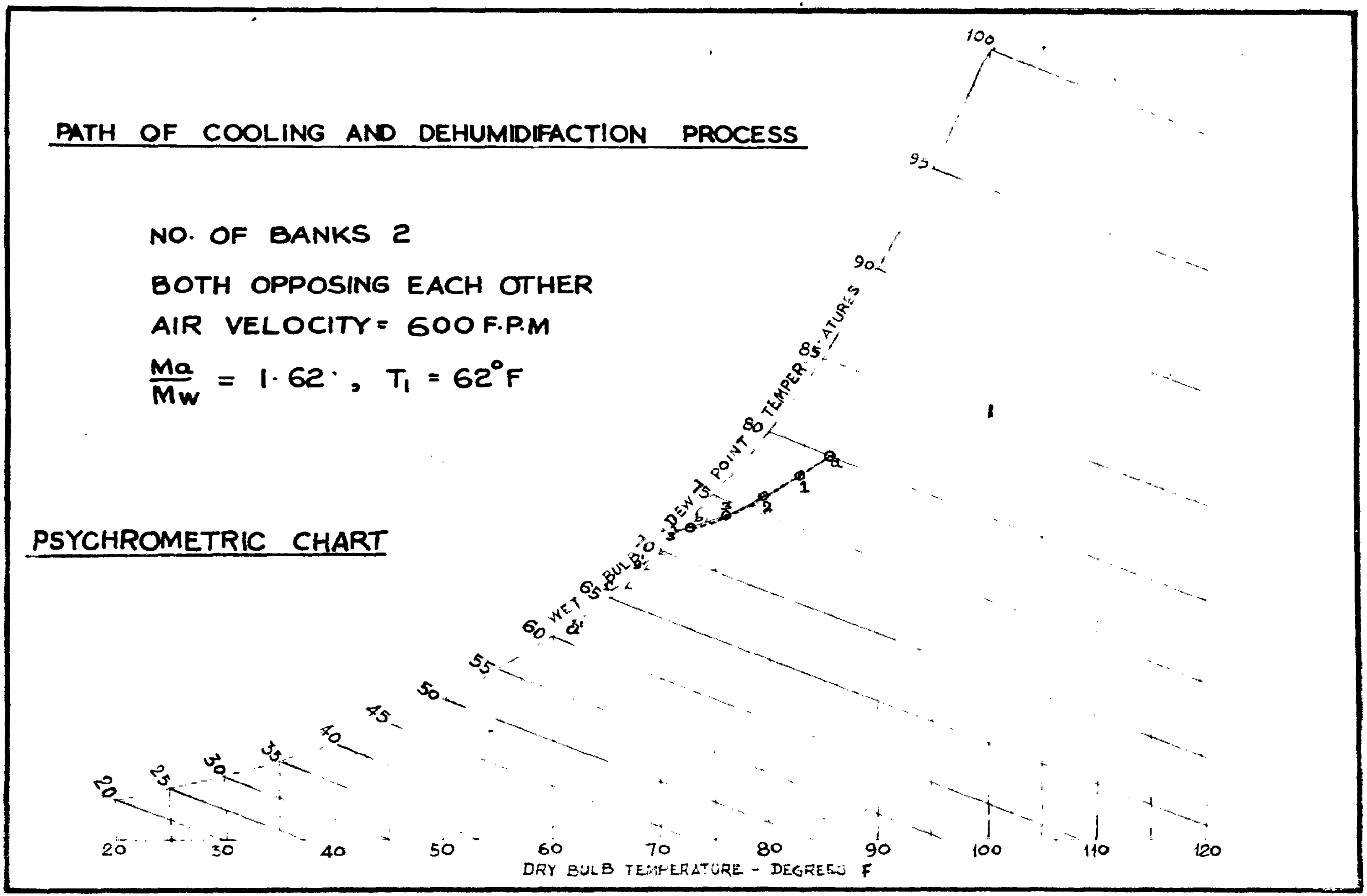


FIG. 6.5

S. MEHROTRA

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

NO. OF BANKS 3

ALL DOWN STREAM

AIR VELOCITY = 650 F.P.M

$\frac{Ma}{Mw} = 1.462, \; T_1 = 64^{\circ}F$

PSYCHROMETRIC CHART

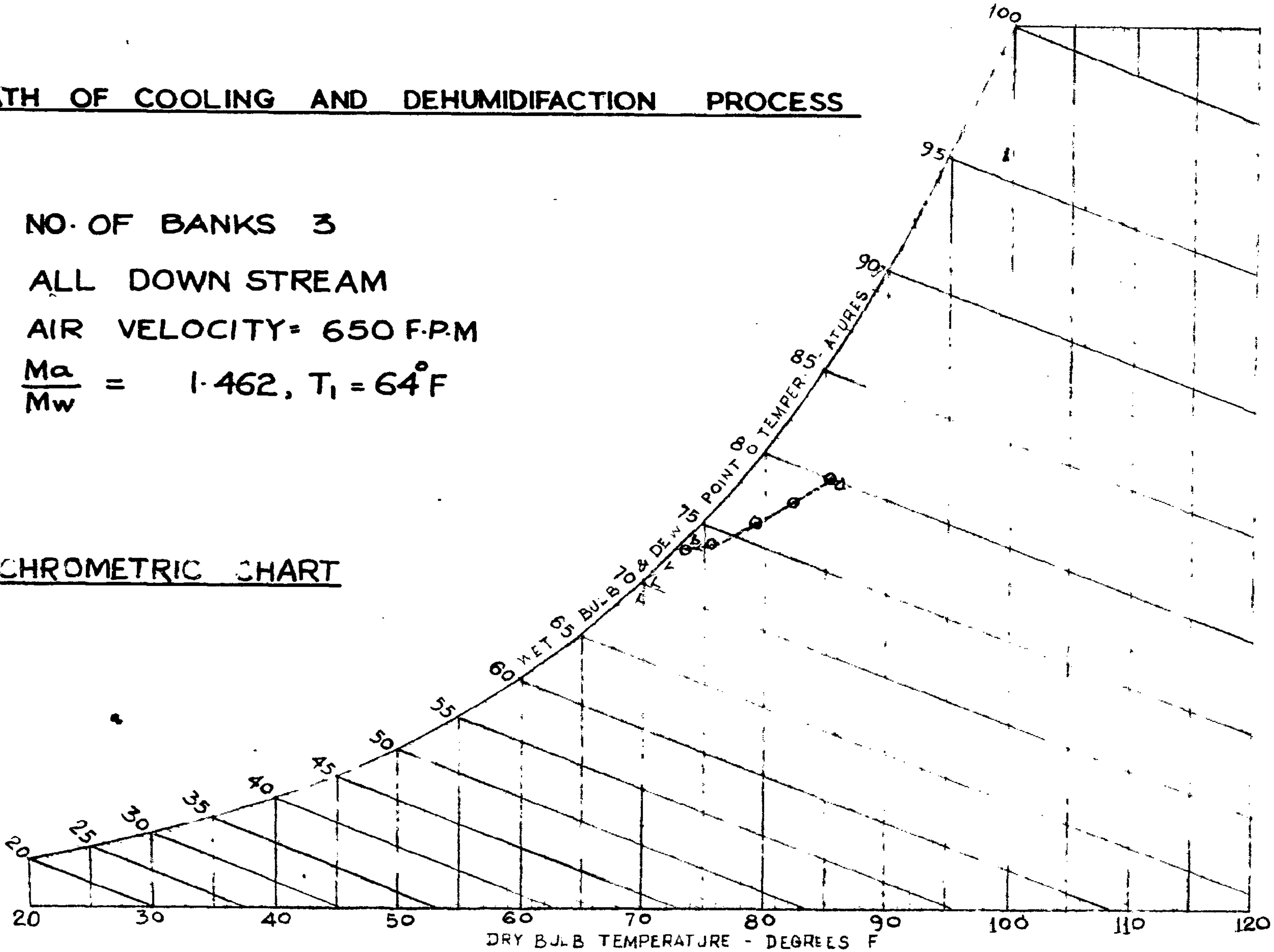


FIG. 6.6

S. MEHROTRA

PATH OF COOLING AND DEHUMIDIFACTION PROCESS

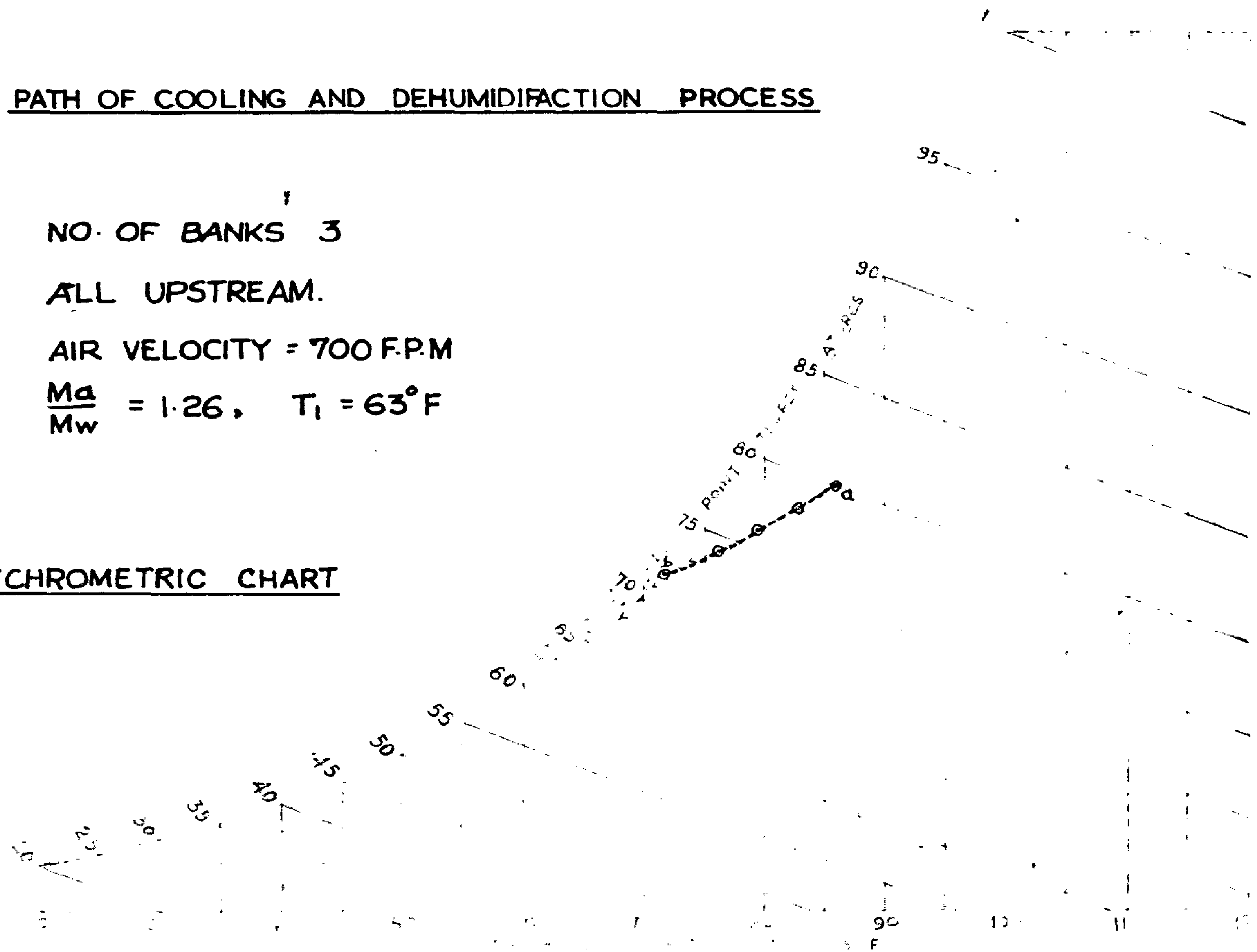
NO. OF BANKS 3

ALL UPSTREAM.

AIR VELOCITY = 700 F.P.M

$\frac{Ma}{Mw}$ = 1.26, T_1 = 63°F

PSYCHROMETRIC CHART



PATH OF HUMIDIFICATION PROCESS

HEATED WATER

NO. OF BANKS 1, DOWN STREAM

VELOCITY OF AIR = 500 F.P.M.

$\frac{Ma}{Mw} = 2\cdot29$, $T_1 = 96°F.$

PSYCHROMETRIC CHART –

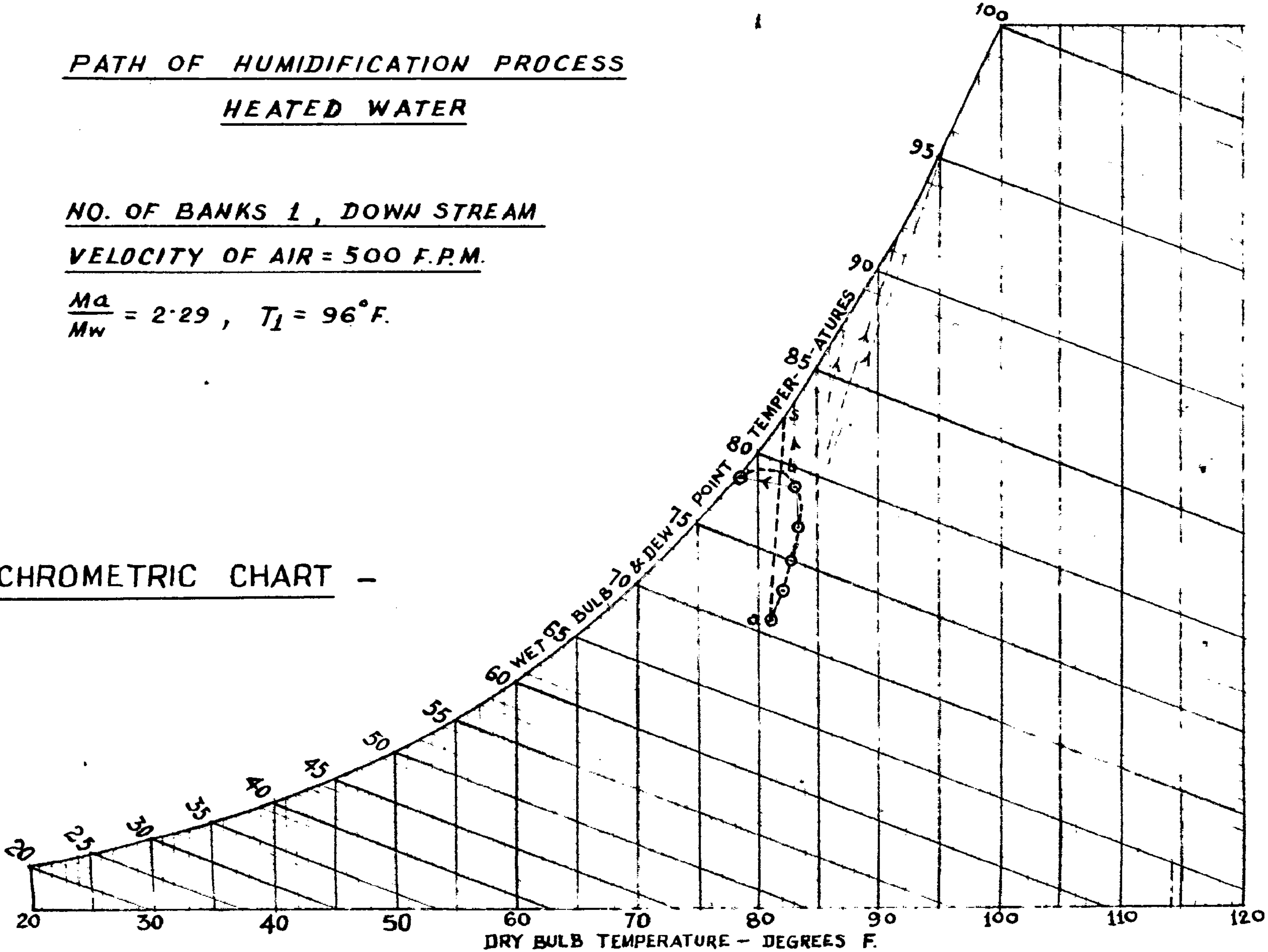


FIG 6·8

S. MEHROTRA

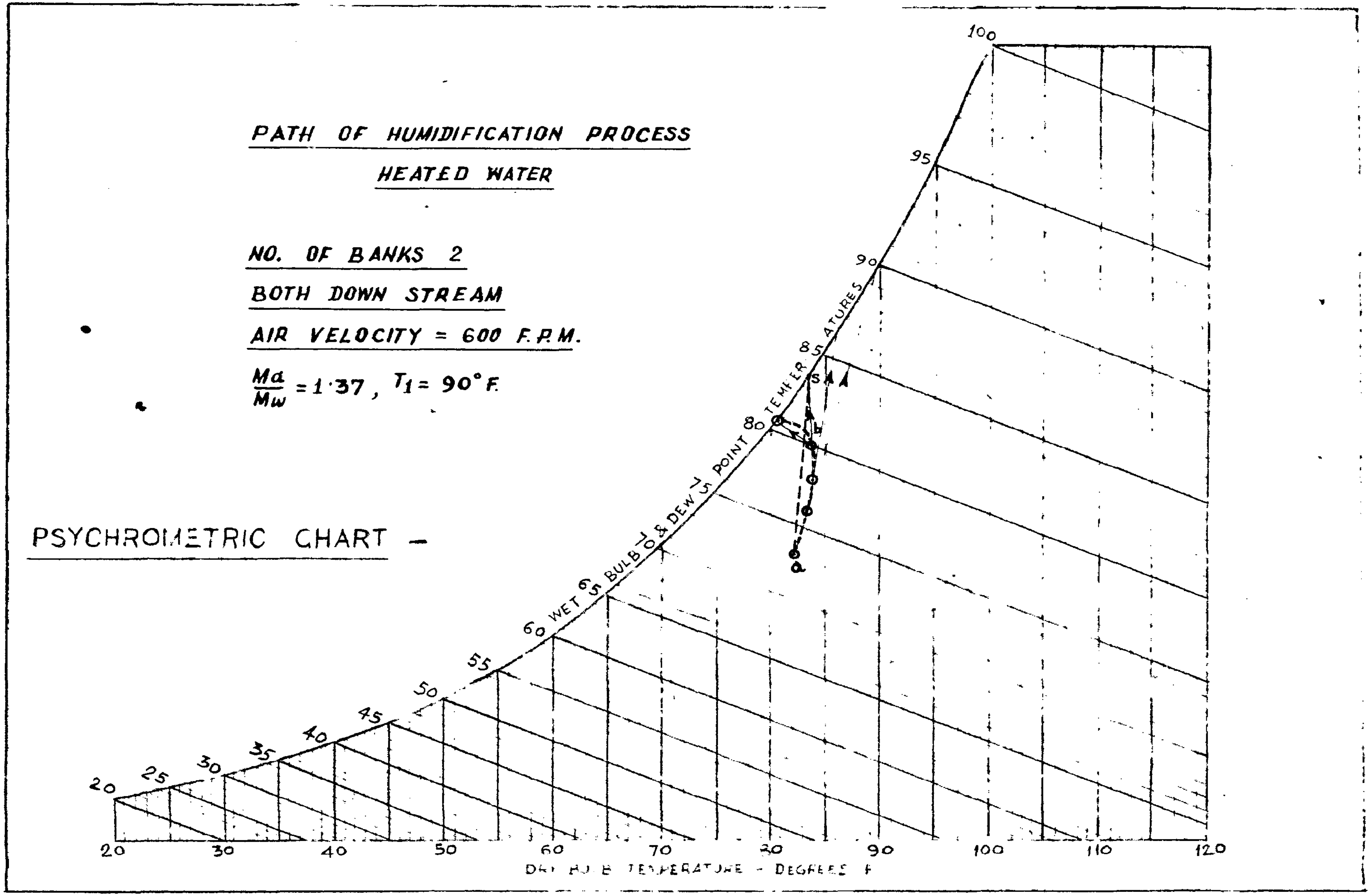

PATH OF HUMIDIFICATION PROCESS
HEATED WATER
NO. OF BANKS 2
BOTH DOWN STREAM
AIR VELOCITY = 600 F.P.M.
$\frac{Ma}{Mw} = 1{\cdot}37$, $T_1 = 90°F.$
PSYCHROMETRIC CHART -
WET BULB & DEW POINT TEMPERATURES
20
25
30
35
40
45
50
55
60
65
70
75
80
85
90
95
100
20 30 40 50 60 70 80 90 100 110 120
DEGREES F
a
b
s

FIG. 6.9

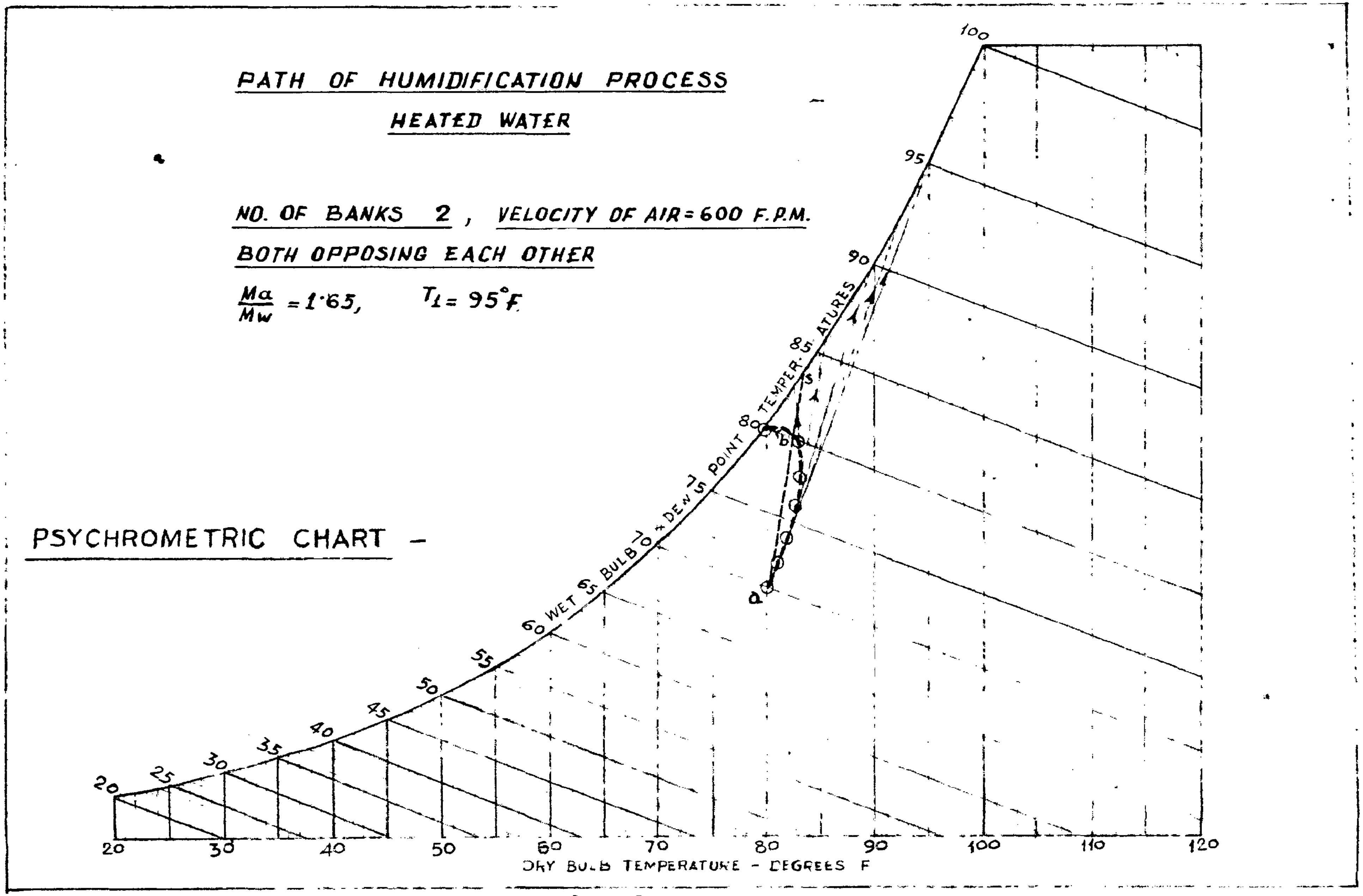


FIG 6·10

S MEHROTRA

PATH OF HUMIDIFICATION PROCESS

HEATED WATER

NO. OF BANKS 2

BOTH UP STREAM

VELOCITY OF AIR = 500 F.P.M.

$\frac{M_a}{M_w} = 1{\cdot}715$, $T_1 = 94^\circ F.$

PSYCHROMETRIC CHART –

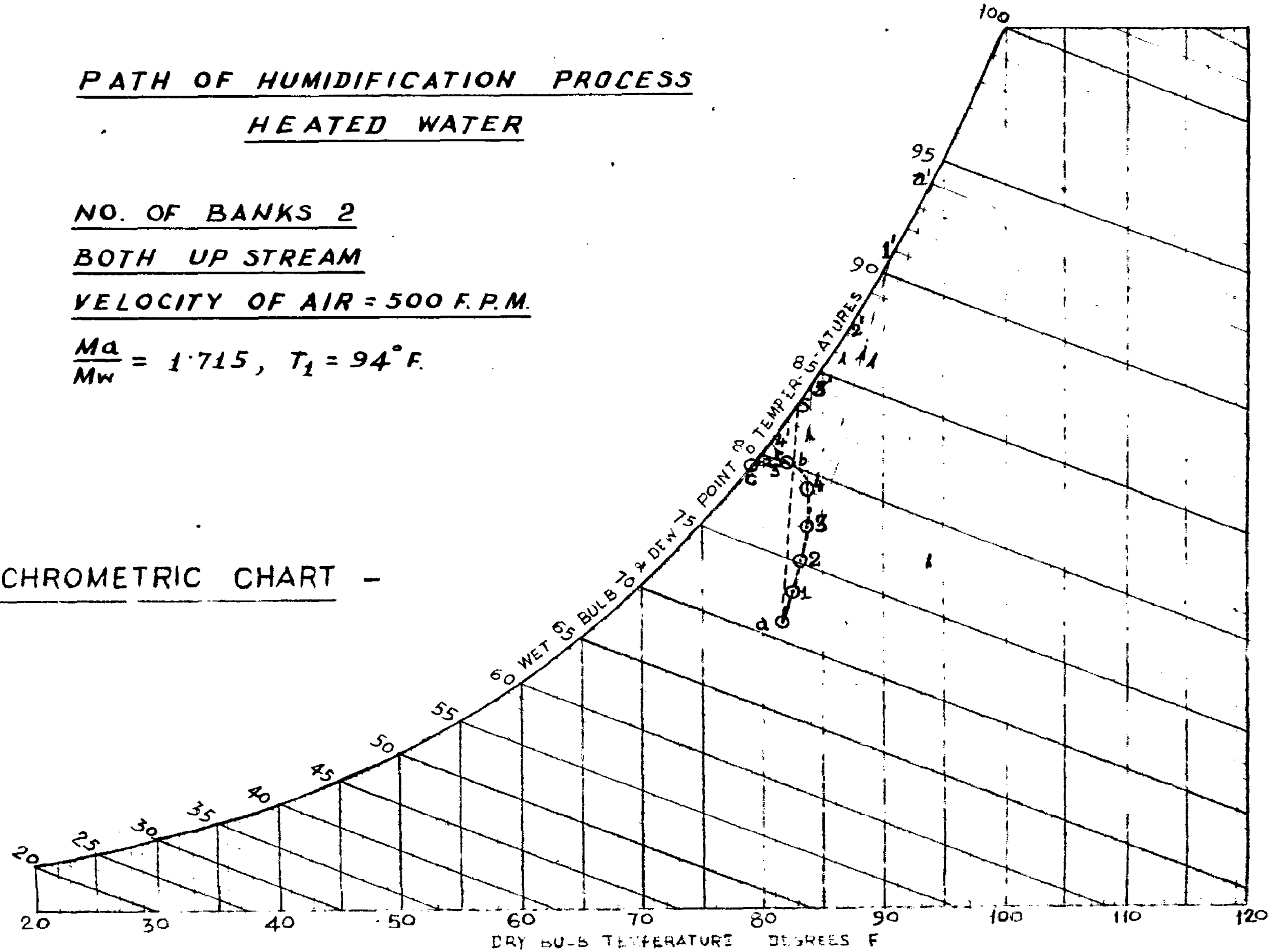


FIG 6·11

S. MEHROTRA

PATH OF HUMIDIFICATION PROCESS

HEATED WATER

NO. OF BANKS = 3.

ALL DOWN STREAM,

AIR VELOCITY = 600 F.P.M.

$\frac{Ma}{Mw} = 1{\cdot}38$, $T_1 = 96°F.$

PSYCHROMETRIC CHART –

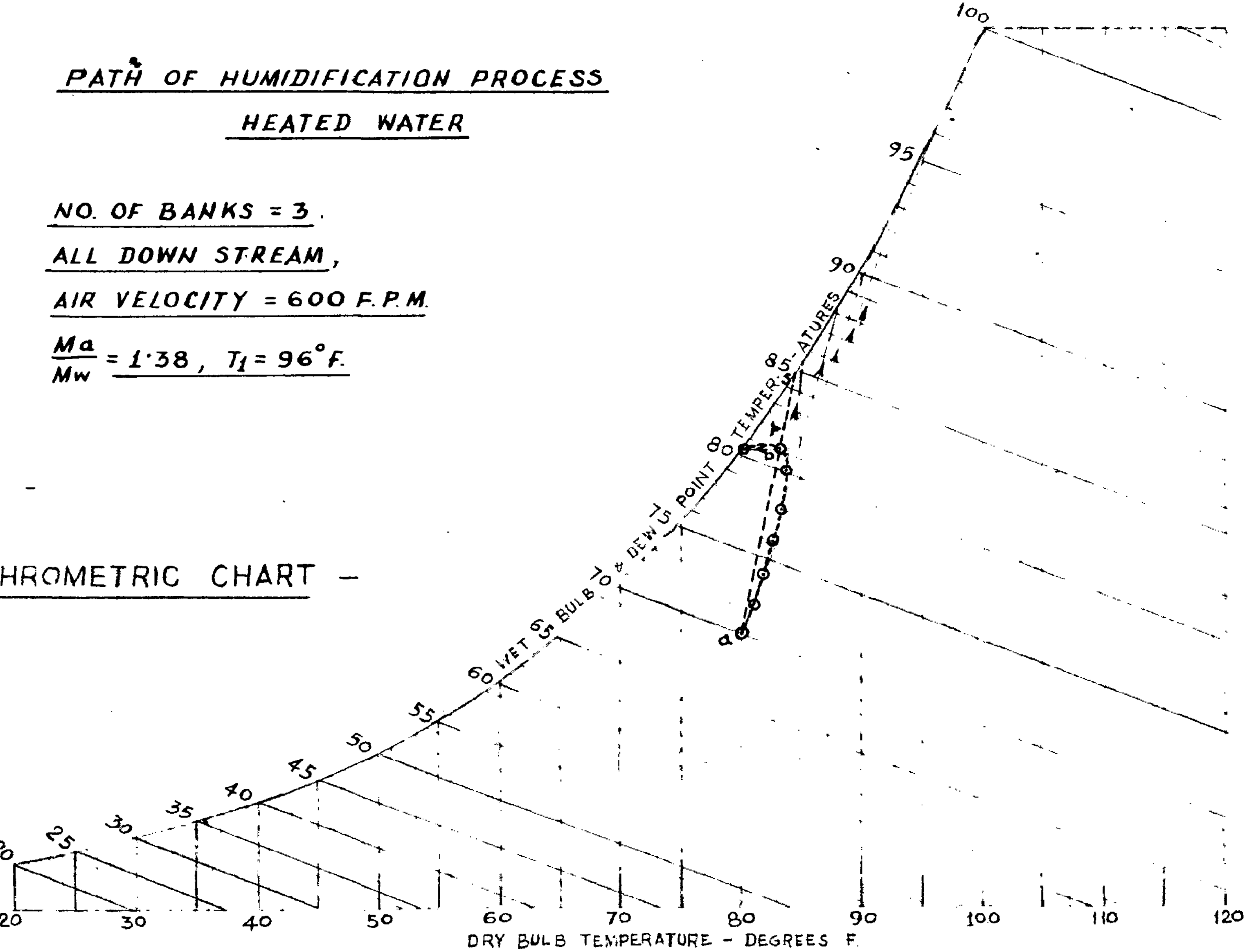


FIG 6·12

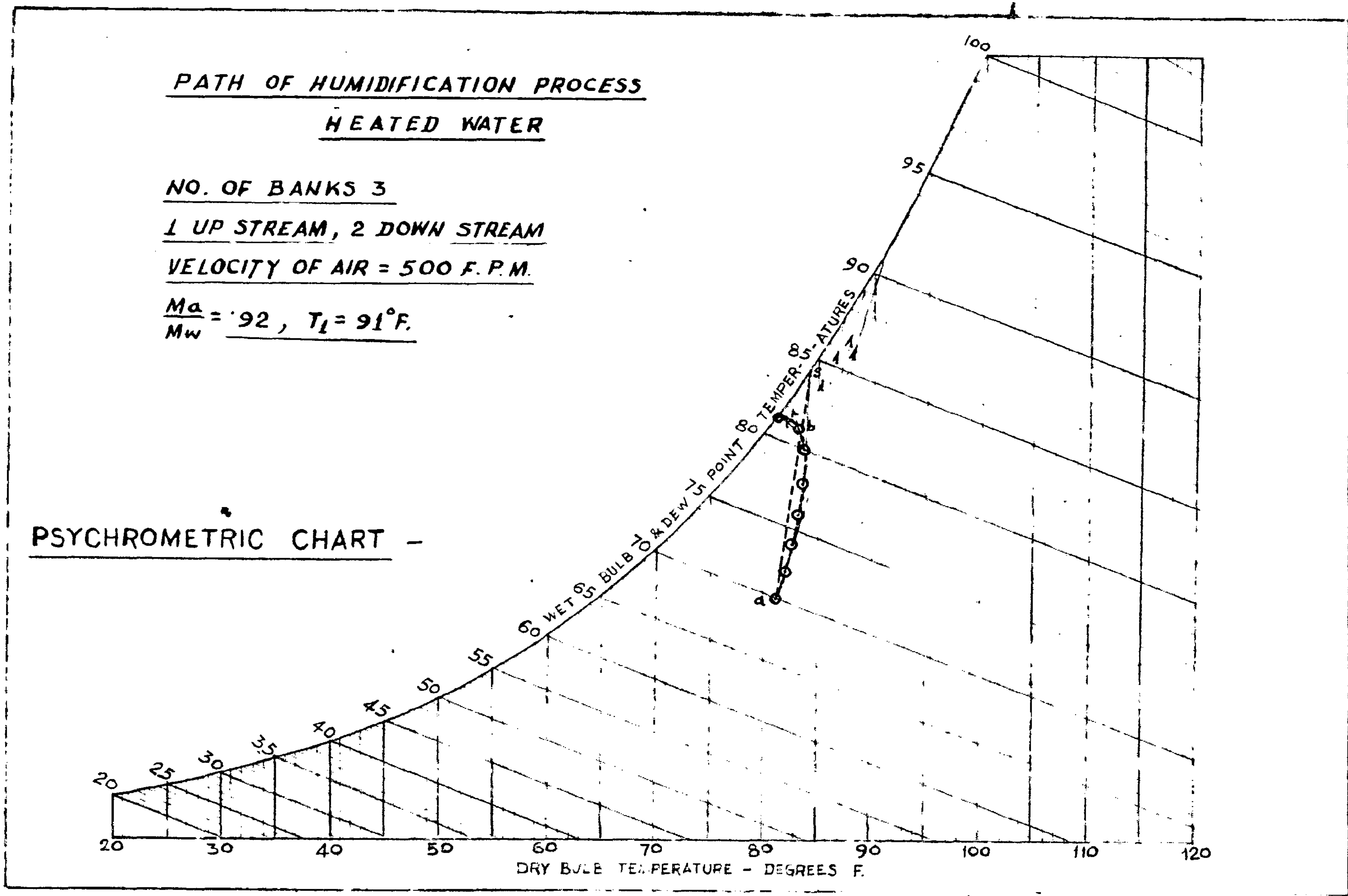


FIG 6.13

S MEHROTRA

PATH OF HUMIDIFICATION PROCESS

HEATED WATER

NO. OF BANKS 3

2 UP STREAM, 1 DOWN STREAM

VELOCITY OF AIR = 500 F.P.M.

$\frac{Ma}{Mw} = 1{\cdot}075$, $T_1 = 93{\cdot}5°F.$

PSYCHROMETRIC CHART –

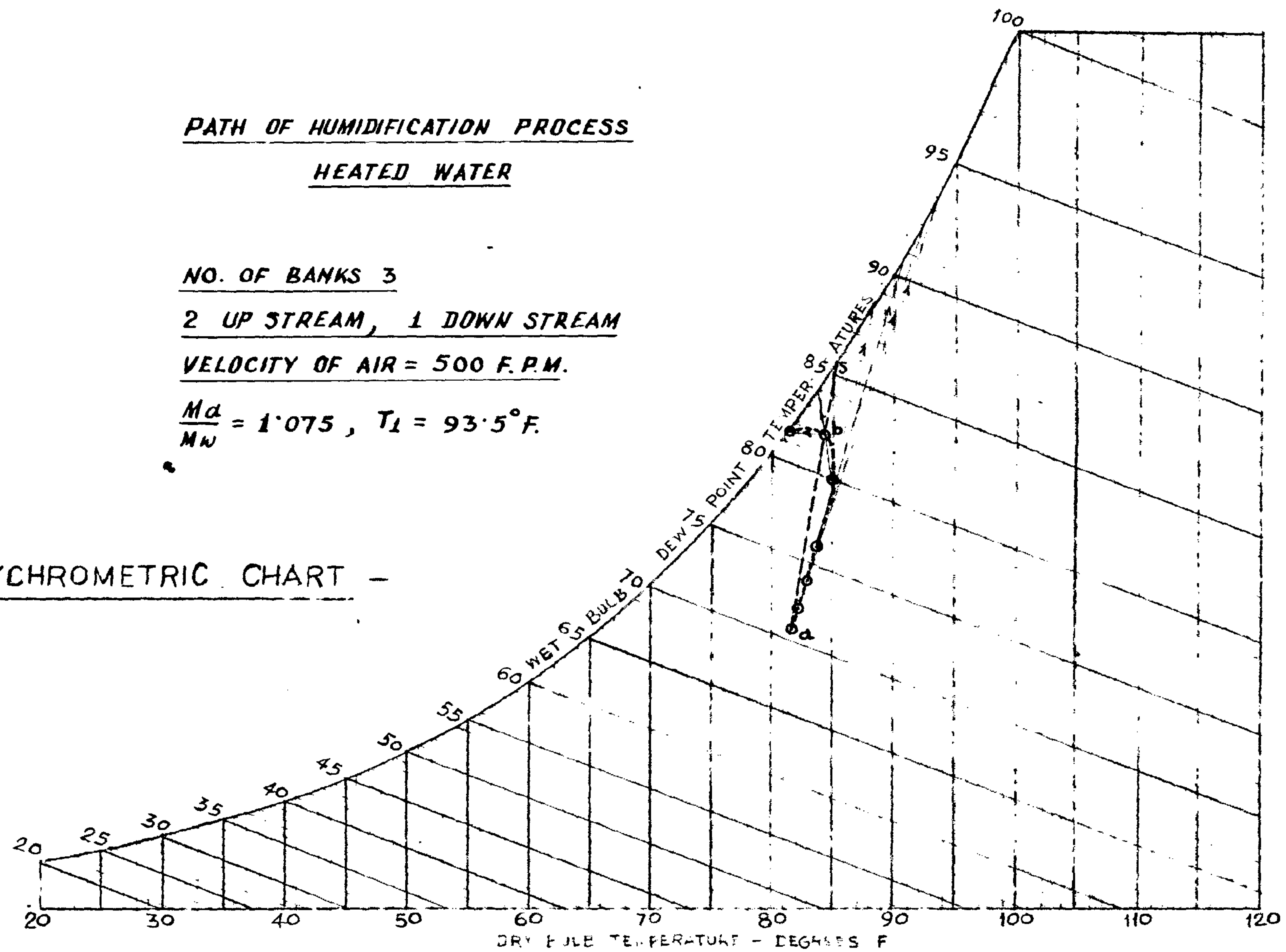


FIG 6·14

PATH OF HUMIDIFICATION PROCESS

HEATED WATER

NO. OF BANKS 3

ALL UP STREAM,

VELOCITY OF AIR = 700 F.P.M.

$\frac{Ma}{Mw} = 1{\cdot}17$, $T_1 = 92°F$.

PSYCHROMETRIC CHART –

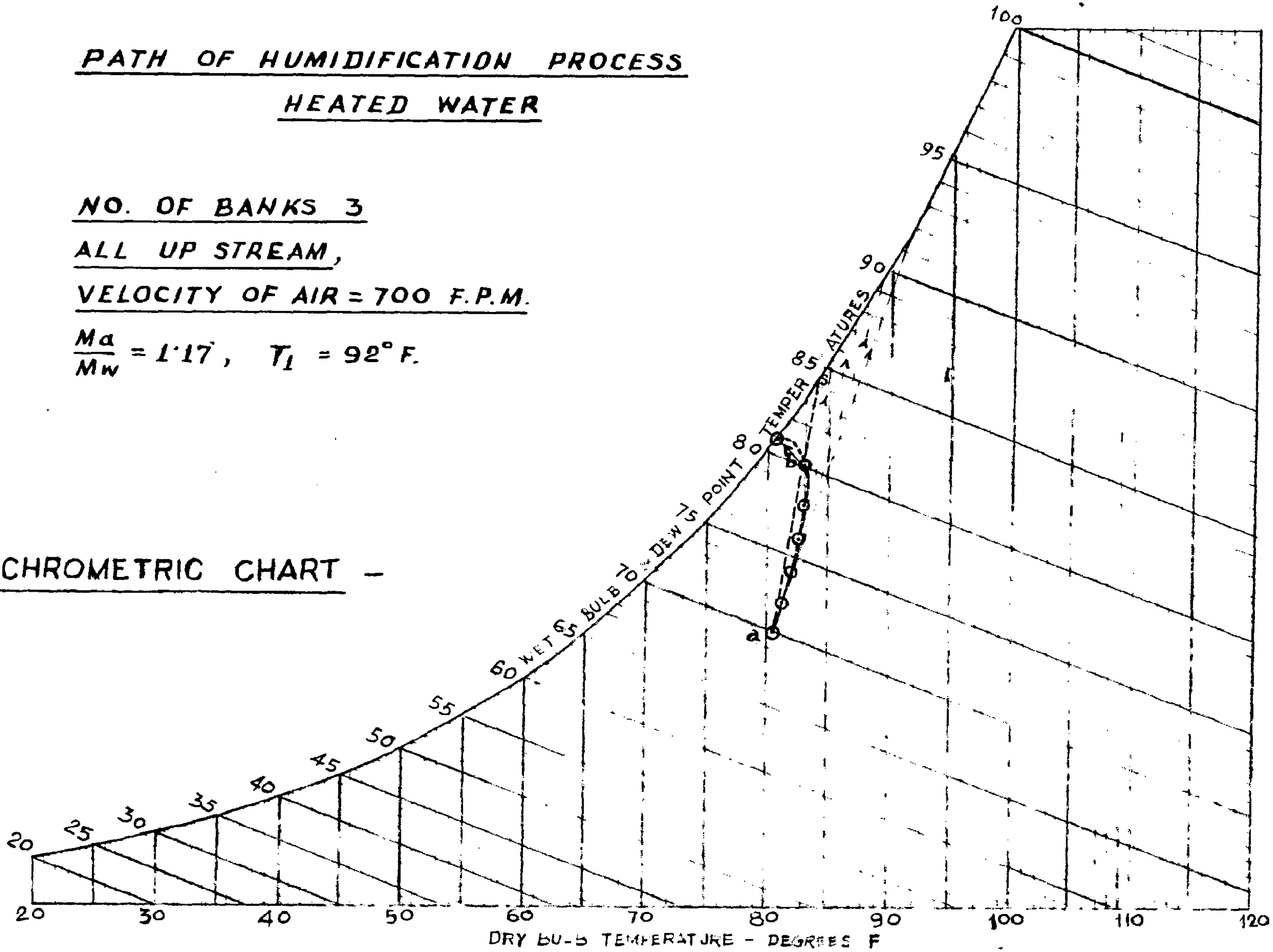
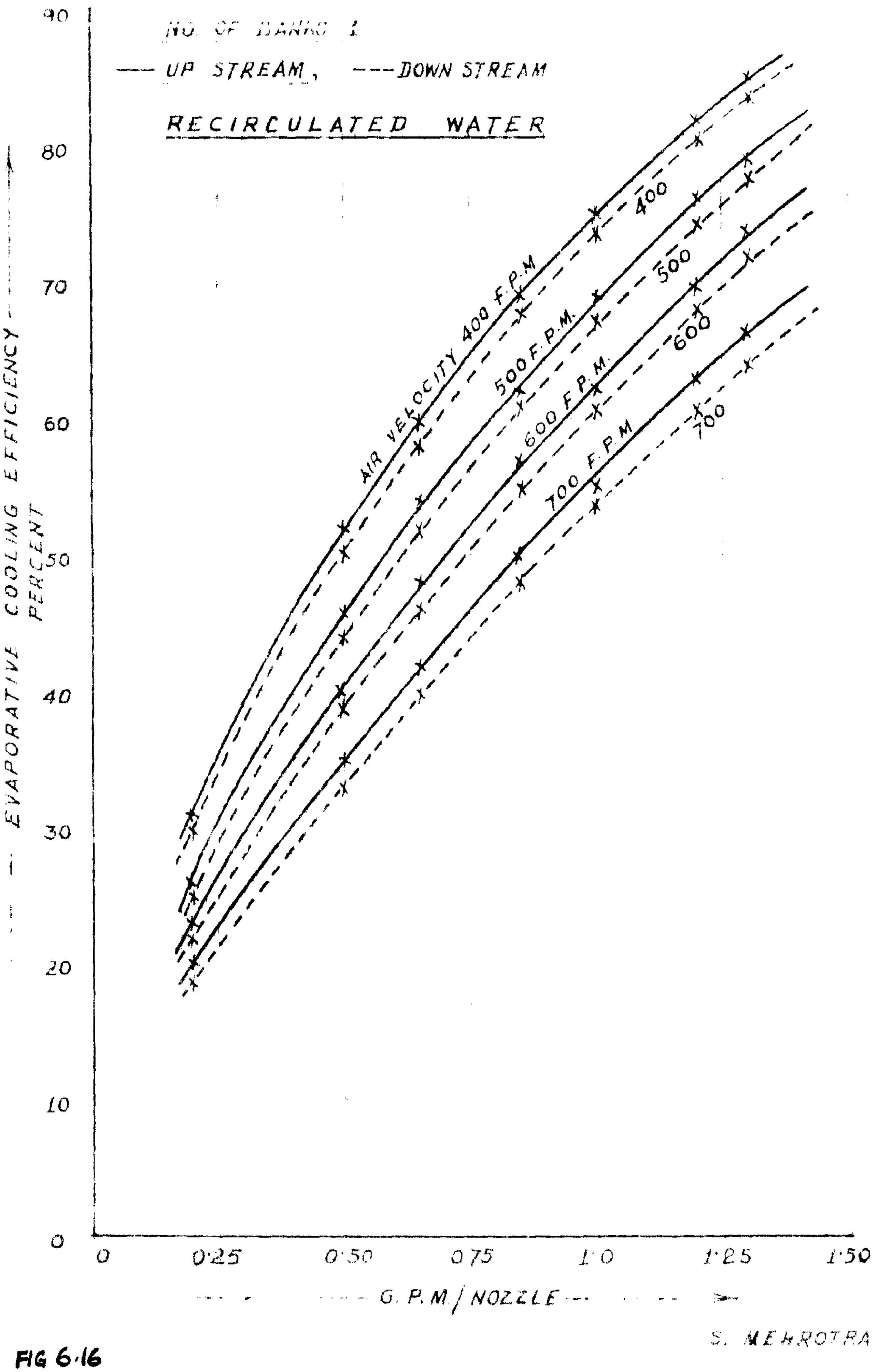
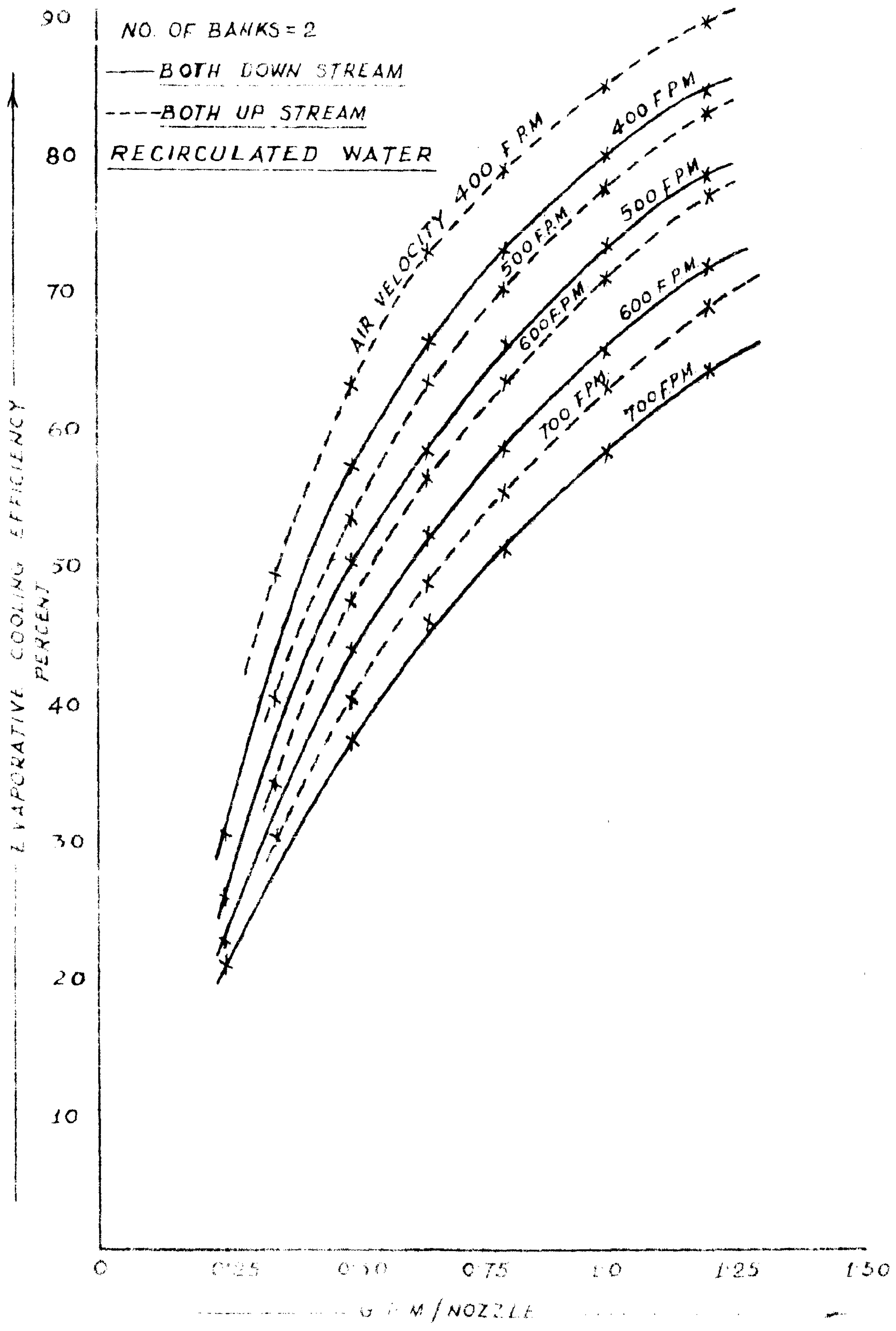


FIG. 6·15



NO. OF BANKS 1
UP STREAM, DOWN STREAM
RECIRCULATED WATER
AIR VELOCITY 400 F.P.M
500 F.P.M.
600 F.P.M.
700 F.P.M.
400
500
600
700
EVAPORATIVE COOLING EFFICIENCY PERCENT
G.P.M/NOZZLE
0
10
20
30
40
50
60
70
80
90
0
0·25
0·50
0·75
1·0
1·25
1·50
S. MEHROTRA

FIG 6·16



NO. OF BANKS = 2
—— BOTH DOWN STREAM
- - - - BOTH UP STREAM
RECIRCULATED WATER
AIR VELOCITY 400 F.P.M
400 FPM
500 F.P.M
500 FPM
600 F.P.M
600 FPM
700 FPM
700 FPM
EVAPORATIVE COOLING EFFICIENCY PERCENT
90
80
70
60
50
40
30
20
10
0
0·25
0·50
0·75
1·0
1·25
1·50
G P M / NOZZLE

FIG. 6·17

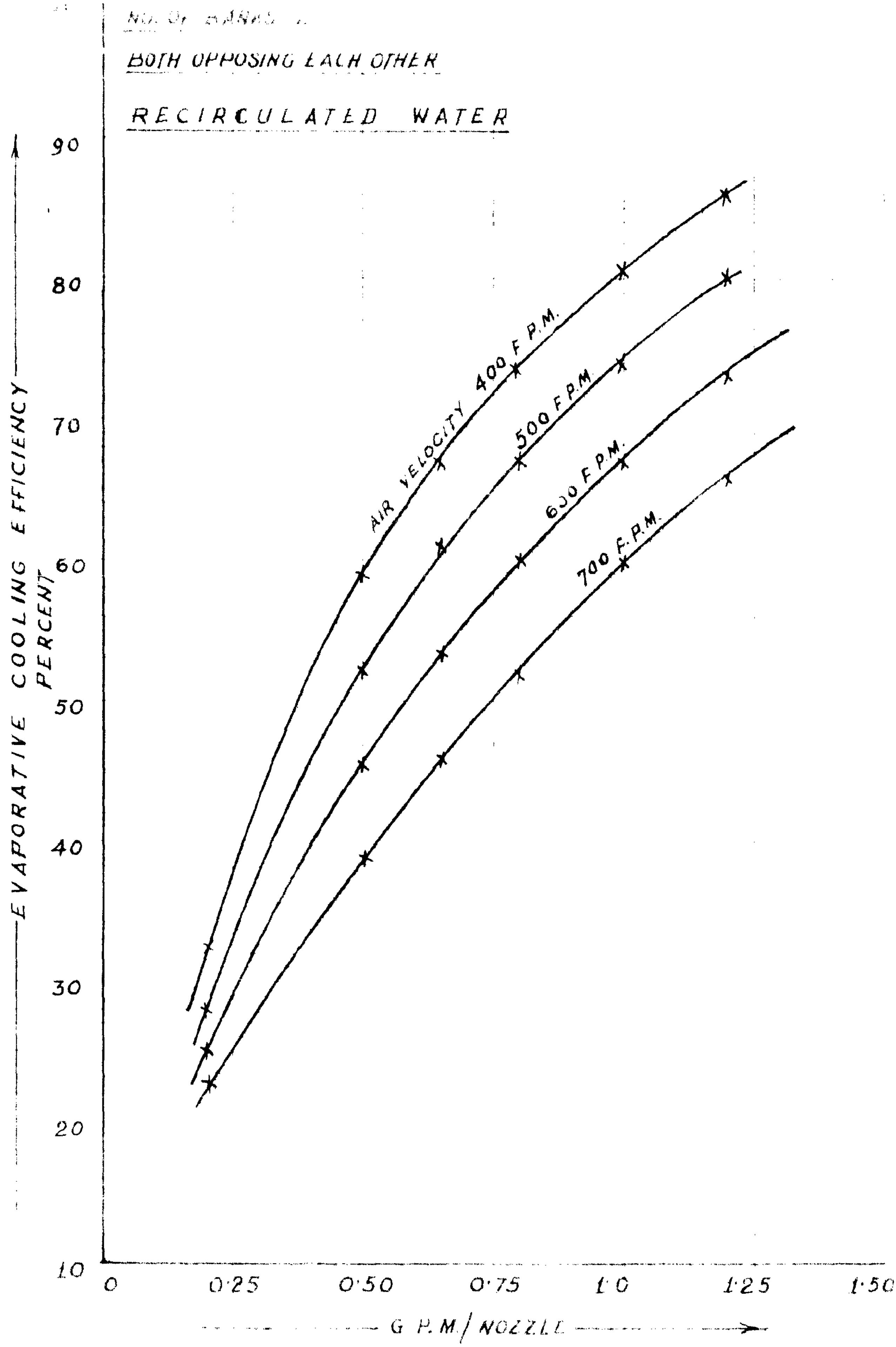
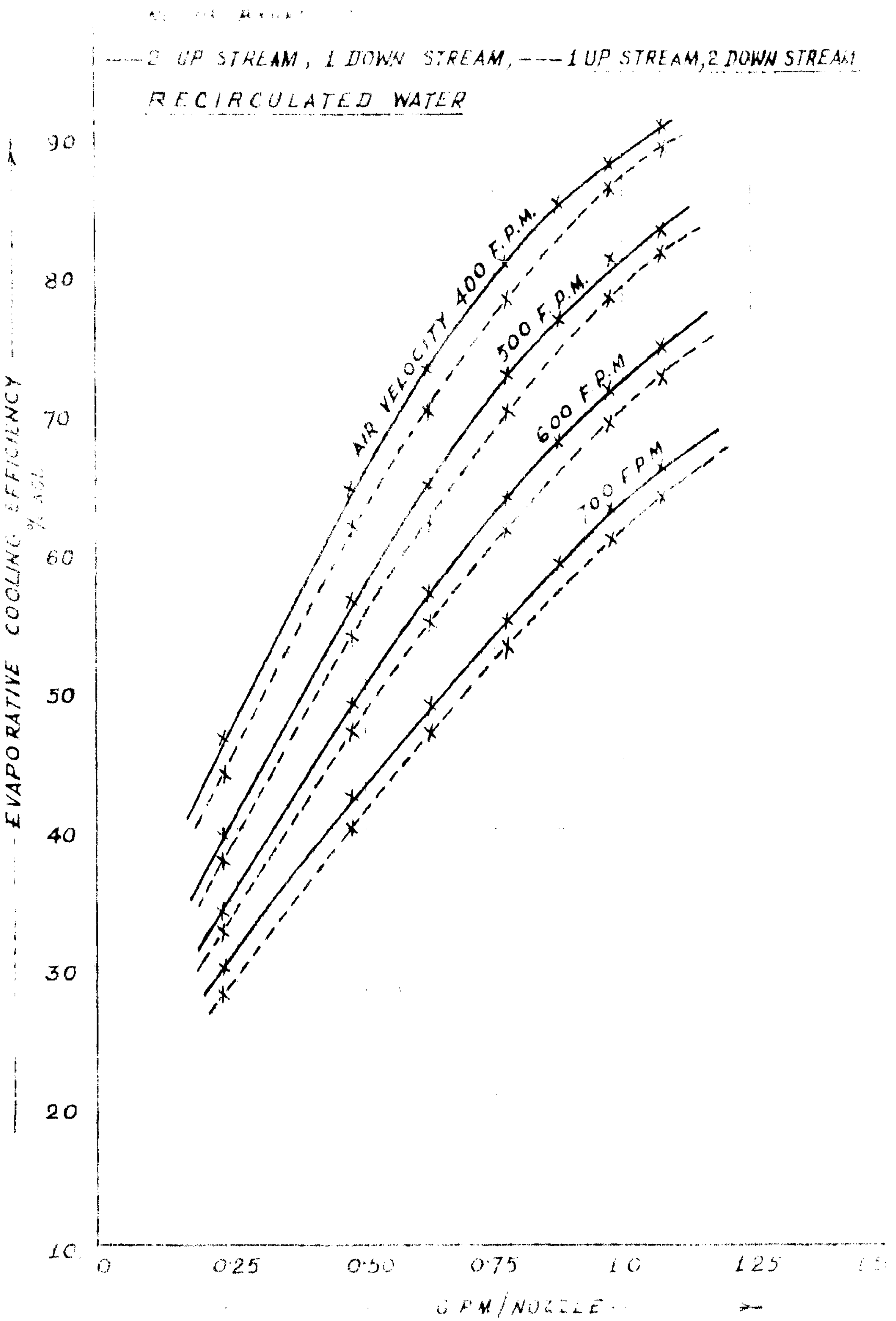

BOTH OPPOSING EACH OTHER
RECIRCULATED WATER
AIR VELOCITY 400 F.P.M.
500 F.P.M.
600 F.P.M.
700 F.P.M.
EVAPORATIVE COOLING EFFICIENCY PERCENT
G.P.M./NOZZLE
10
20
30
40
50
60
70
80
90
0
0·25
0·50
0·75
1·0
1·25
1·50

FIG 6·18



---2 UP STREAM, 1 DOWN STREAM, ---1 UP STREAM, 2 DOWN STREAM
RECIRCULATED WATER
AIR VELOCITY 400 F.P.M.
500 F.P.M.
600 F.P.M
700 FPM
EVAPORATIVE COOLING EFFICIENCY
90
80
70
60
50
40
30
20
10
0
0·25
0·50
0·75
1·0
1·25
G P M / NOZZLE

FIG 6.19

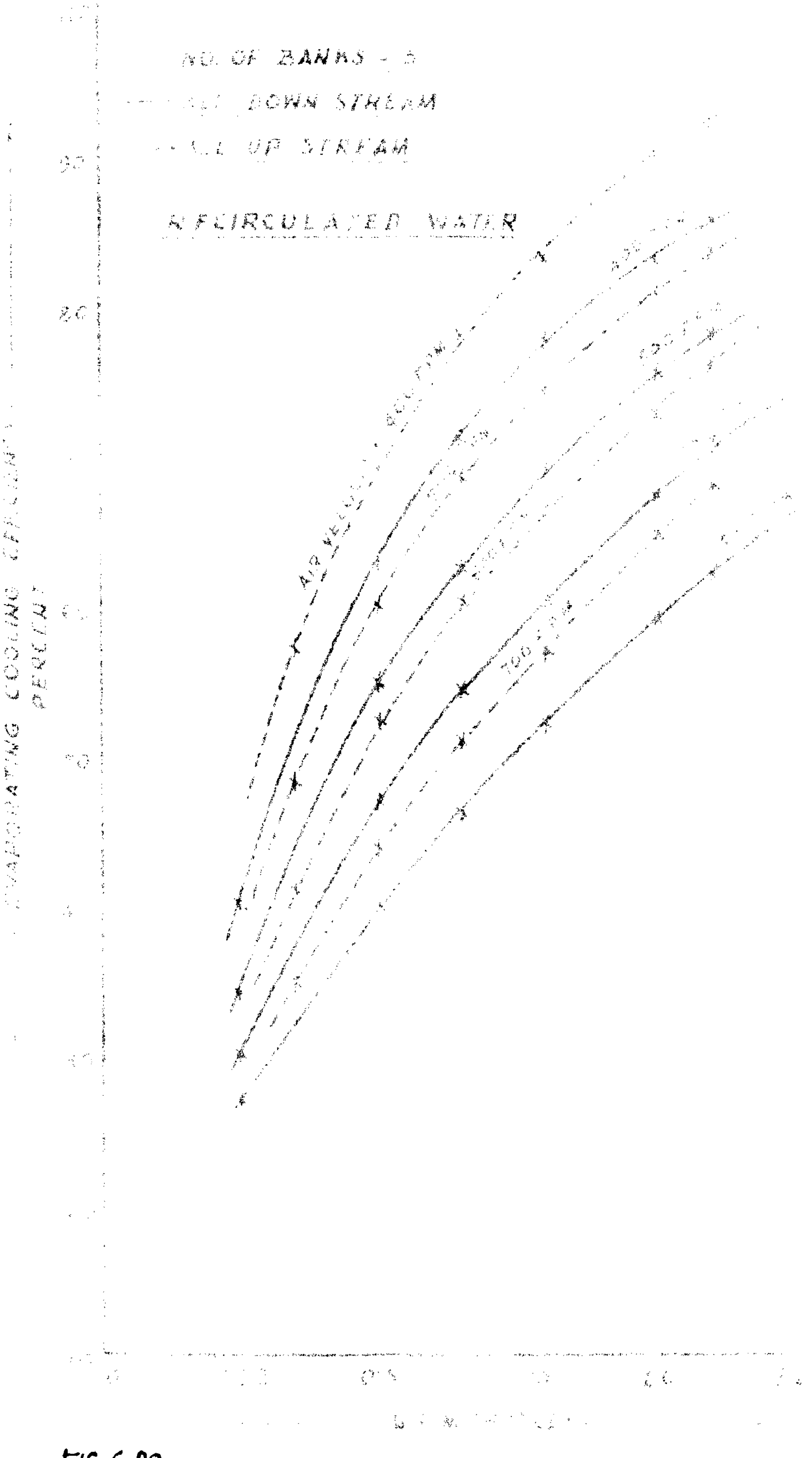

NO. OF BANKS - 5
DOWN STREAM
UP STREAM
RECIRCULATED WATER
AIR VELOCITY
EVAPORATING COOLING EFFICIENCY PERCENT

FIG 6.20

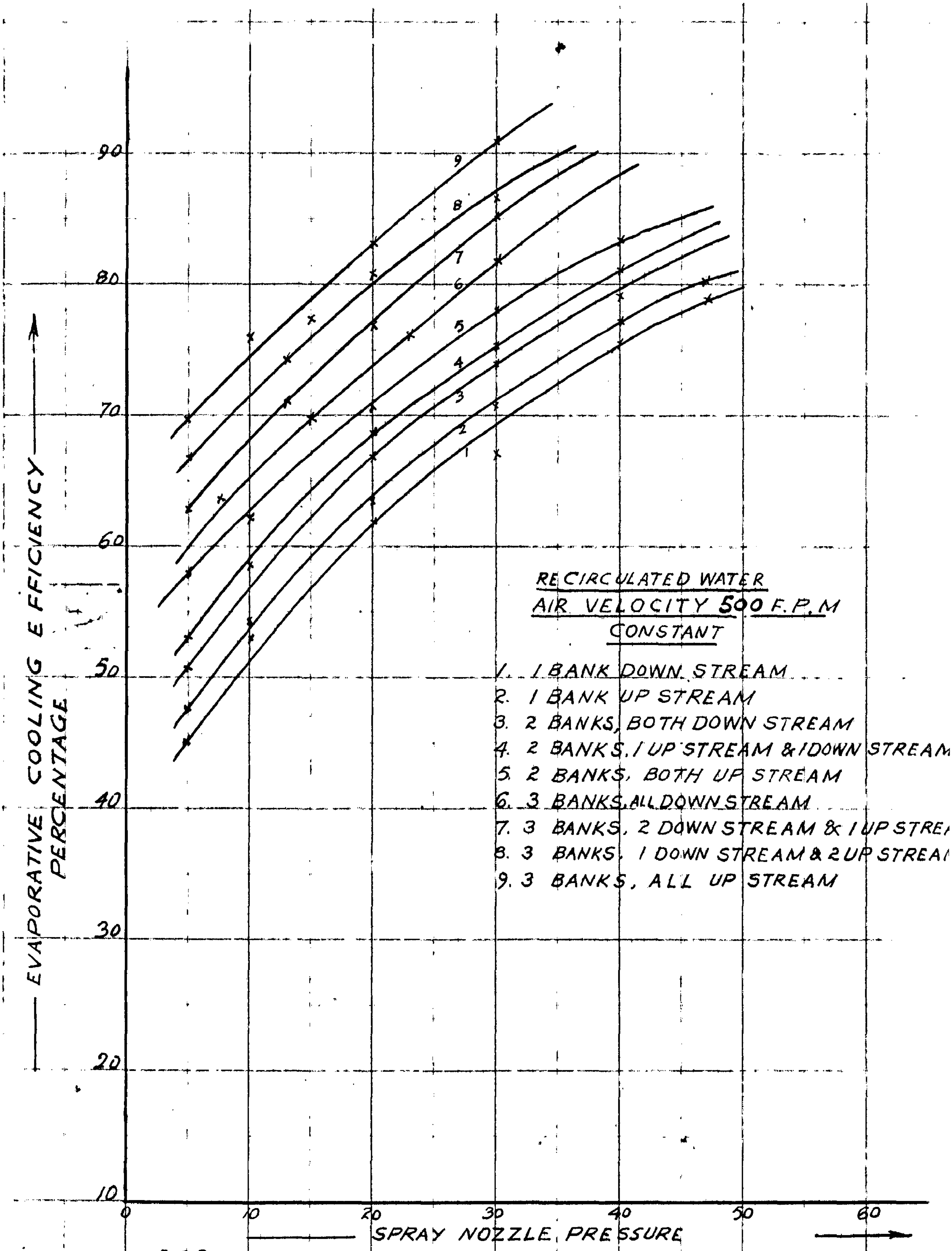
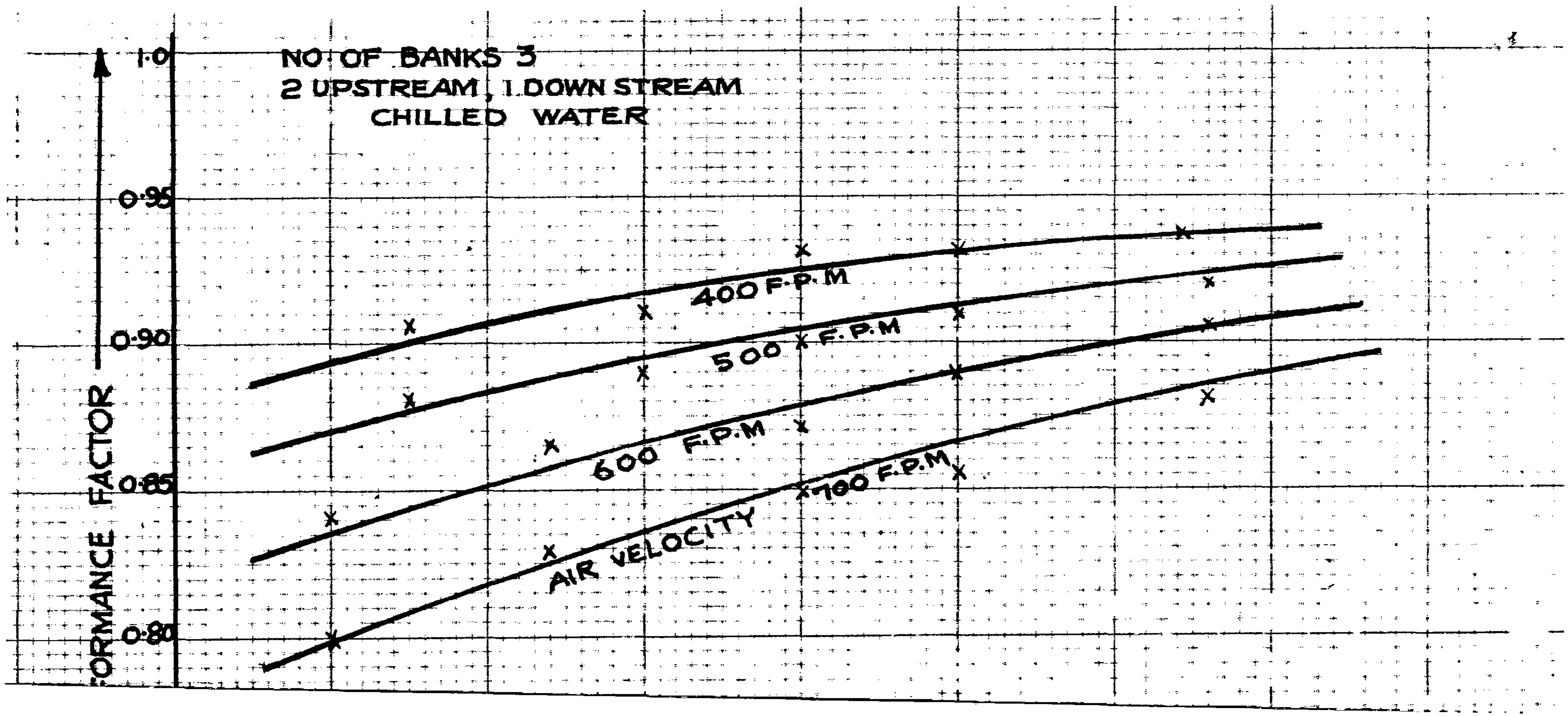
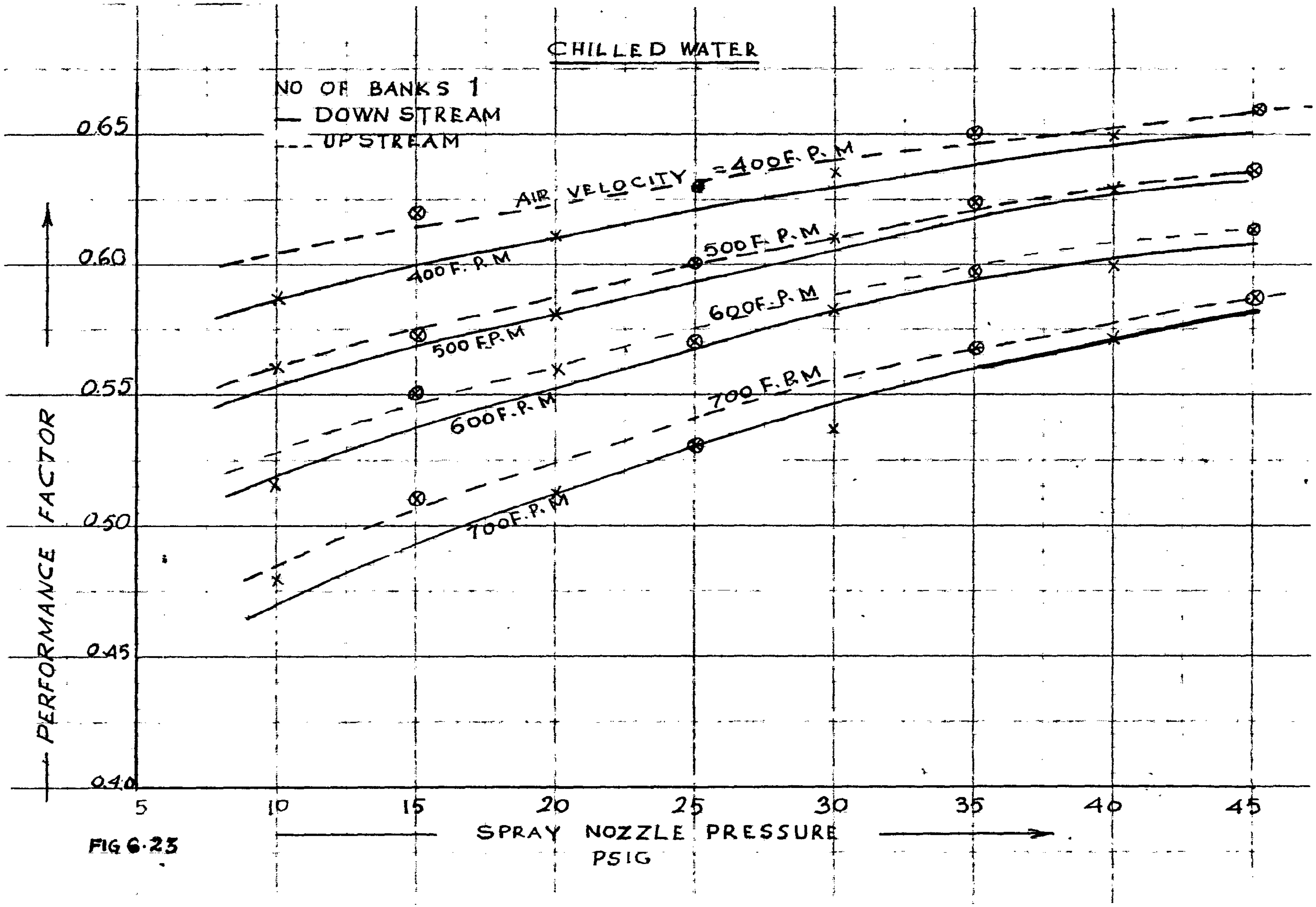


FIG 6.21

S. MEHROTRA


NO OF BANKS 3
2 UPSTREAM, 1 DOWN STREAM
CHILLED WATER
1·0
0·95
0·90
0·85
0·80
ORMANCE FACTOR
400 F.P.M
500 F.P.M
600 F.P.M
700 F.P.M
AIR VELOCITY


CHILLED WATER
NO OF BANKS 1
DOWN STREAM
UPSTREAM
AIR VELOCITY =400F.P.M
400F.P.M
500F.P.M
500 F.P.M
600F.P.M
600F.P.M
700 F.P.M
700F.P.M
0.65
0.60
0.55
0.50
0.45
0.40
5
10
15
20
25
30
35
40
45
PERFORMANCE FACTOR
SPRAY NOZZLE PRESSURE
PSIG
FIG 6.23

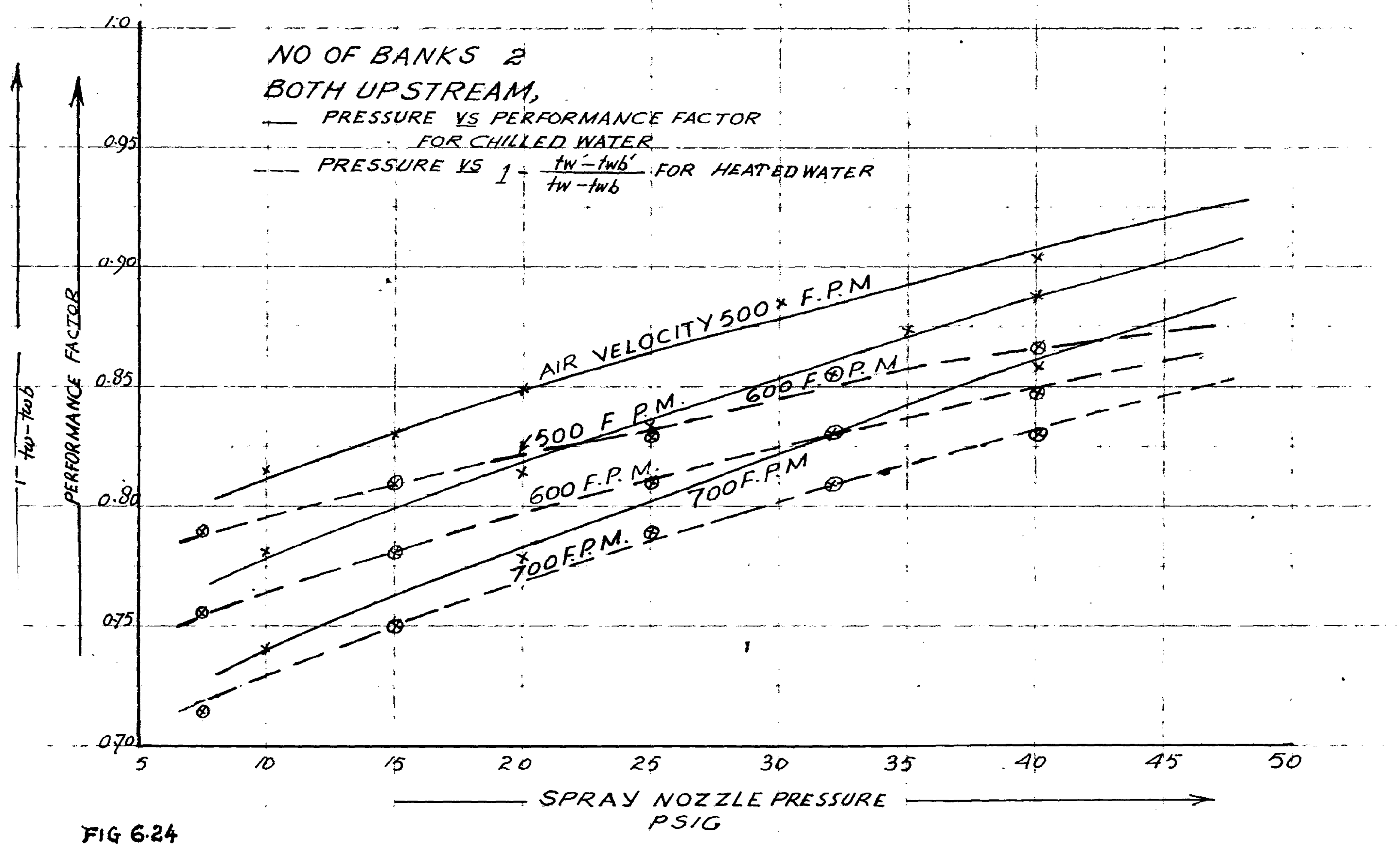

NO OF BANKS 2
BOTH UPSTREAM,
— PRESSURE VS PERFORMANCE FACTOR FOR CHILLED WATER
— — PRESSURE VS $1 - \frac{tw' - twb'}{tw - twb}$ FOR HEATED WATER
AIR VELOCITY 500 F.P.M
500 F.P.M.
600 F.P.M
600 F.P.M.
700 F.P.M
700 F.P.M.
PERFORMANCE FACTOR
$1 - \frac{}{tw - twb}$
1.0
0.95
0.90
0.85
0.80
0.75
0.70
5
10
15
20
25
30
35
40
45
50
SPRAY NOZZLE PRESSURE
PSIG

FIG 6.24

CHAPTER - VII.

DISCUSSION OF RESULTS AND CONCLUSIONS.

-:o:-

CHAPTER - VII

DISCUSSION OF RESULTS.

7.1 RECIRCULATED WATER :-

When the spray water is recirculated, it follows the path of constant enthalpy or ~~contant~~ wet bulb on the psychrometric chart.

From the thermodynamic stand point, if the injection water is supplied at the wet bulb temperature of the entering air, then each entering drop is surrounded by a film of vapour, a simultaneous transfer occurs of vapour from the drop to the air stream and of sensible heat from the air stream to the drop. The drop remains at equilibrium wet bulb temperature and the process of humidification proceeds along a constant enthalpy or wet bulb line.

7.2 HEATED WATER :-

The temperatureof water is above the dry bulb temperature of air.

As explained in Chapter - IV, the equation - 27 indicates that at any cross section of the spray chamber the instantaneous slope of the air path dW/dt_a on the psychrometric chart is determined by a straight line connecting the air state with the interface saturation state at that cross section. Thus in Fig. 6.11, state 'a' represents the state of air entering the washer chamber. The washer is operating as a heating and humidifying apparatus so that the interface saturation state is represented by a'. The initial slope of the air path is then along a line directed from a to state a'. As the air is heated the water is cooled and the interface temperature drops. Corresponding air states and interface saturation states are indicated by letters 1, 2, 3 and 4. At each instance the air path is directed towards the associated interface states. The interface states are found from the energy equation as explained in 6.4.1.

7.3 CHILLED WATER :-

When the air is supplied with sufficient water such that the final temperature of the water is below the dew point temperature of the entering air, cooling and dehumidification will occur along the complete air path. In such process there is a transfer of both heat and vapour to the drop so that the surface temperature rapidly rises as it is gaining both sensible and latent heat. Since there is a transfer of both heat and vapour , the process is one of dehumidification rather than humidification.

If the water temperature is such that it equals the dew point temprature of the entering air within the chamber then the flow of vapour ceases when the dew point is reached, but the drop continues ~~continuous~~ to gain sensible heat, its temperature rises and the vapor flow now starts in the opposite direction and hence cooling with humidification starts. In figure 6.5, the water temperature is sufficiently lowered and hence the ~~b~~ leaving temperature of water is below the dew point of entering air. In this case the curve continuously drops showing cooling and complete dehumidification.

7.4 EFFICIENCY OF AIR WASHER :-

The efficiency of the air washer has been defined earlier as the ratio of actual lowering of air temperature to the wet bulb depression.

The efficiencies at different combinations of number of spray banks, direction of spray and velocities have been plotted against the gpm per nozzle as shown from Fig. 6.16 to 6.20. Taking the Fig. 6.16 in which the efficiencies are drawn for two different direction of the spray when operating one bank only , we see that the efficiencies obtained when the direction of spray is upstream i.e. against the flow of air, are more than when the direction of spray is down stream. This is due to the fact that when the spray is against the air flow, there is a thorough mixing of air and water which, therefore, reduces the temperatureof air more effectively. In downstream spray the mixing is not so effective.

From the curves of Fig. 6.16, we see that as the pressure is increased with a subsequent increase of water quantity per nozzle, there is a sharp rise in the value of efficiency at a particular air velocity. This is beaause at lower pressures the spray mist is not sufficient to effectively reduce the leaving air dry bulb temperature.

Further, we see that as the velocity of air is reduced, the efficiency increases for a particular setting of number of banks and spray direction. This is due to the fact that decrease of air velocity reduces the mass of air and consequently, the ratio of mass of air to mass of water is reduced and hence there is a more effective saturation of the air.

Now referring to figures 6.17 to 6.20, we see that the efficiency is greatly changed by the number of banks operated, and the direction of spray. The efficiency is increased when for the same direction of spray, the number of banks are increased since then it increases the water quantity compared to the air quantity.

In figure 6.21, the efficiency is plotted with spray nozzle pressure at a constant velocity of 500 fpm at different settings of spray nozzle direction and number of banks. From these it is observed that the efficiency increases when the number of banks is increased. Also it is seen that for a particular number of banks the efficiency is more when the number of banks operating on upstream side are more than the downstream side. Thus out of the total number of banks 3, if two banks are operating upstream and 1 downstream, then the efficiency is more than when they operate with 2 banks downstream and 1 bank upstream. This is as previously stated due to the thorough mixing of air and water when operating on upstream side.

Thus we can summarise that the efficiency of an air washer depends upon the air velocity, air to water ratio, pressure, fineness of atomization, number of banks and the direction of spray and the finally the number of nozzles.

From the curves of Fig. 6.16 to 6.21, it is seen that the value of efficiencies is not too high. This is because the number of nozzles operated during the experiment was less.

7.5 PERFORMANCE FACTOR :-

7.5.1 CHILLED WATER :-

When cooling and dehumidification process occurs the performance factor has been plotted against spray nozzle pressures as shown from Fig. 6.22 to 6.24.

The performance factor for single spray bank is lower than the performance factor for 2 or 3 spray banks for a particular air velocity owing to the fact that the difference between the leaving wet bulb temperature of air and the leaving water temperature is more.

The reason for higher leaving wet bulb temperature is because of the water quantity of the chilled water being less when the number of banks is one. Thus the ratio of mass of air to water is increased.

When the number of banks is increased or the velocity is decreased, the ratio of mass of air to water decreases and hence the leaving wet bulb temperature of the air is reduced to a greater degree, which in effect gives a greater performance factor. When the direction of spray is upstream there is more thorough mixing compared to downstream with a consequent decrease in W.B. Temp. of leaving air. This increases the performance factor.

7.5.2 HEATED WATER :-

The curves between the spray nozzle pressure and the factor $\left(1 - \frac{t_W' - t_{Wb'}}{t_W - t_{Wb}}\right)$ have been drawn. As the pressure is increased water quantity increases and hence the ratio of air to water decreases with a consequent increase of the above factor. This, therefore, gives a rising characteristic of the curve.When number of banks are increased, the ratio of mass of air to water is decreased for a particular air velocity and hence the leaving wet bulb temperature increases more compared to reduction in hot water temperature. This gives a higher value of the above factor. The same thing happens when the velocity is decreased. The given factor increases with a reduction in velocity.

The curves between spray nozzle pressure vs performance factor for chilled water and between spray nozzle pressure and $\left(1 - \frac{t_W' - t_{Wb}'}{t_W - t_{Wb}}\right)$ for a particular arrangement of spray system have been superimposed as shown in Fig. 6.24. The curves for heated water have a downward characteristics compared to curves for chilled water.

On the basis of above, the following conclusions are drawn.

a) RECIRCULATED WATER :-

1. The efficiency of air washer increases with an increase of spray nozzle pressure for a particular air velocity.

2. The efficiency of air washer increases with an increase in number of banks.

3. The efficiency of an air washer increases when the direction of spray is upstream and has a lower value when the direction of spray is downstream.

4. The efficiency of an air washer increases with a reduction of air velocity.

5. The process follows a constant enthalpy or contant wet bulb lines on the psychrometric chart.

b) CHILLED WATER :-

6. When the temperature of spray water is lowered such that the leaving water temperature is below the dewpoint temperature of the entering air, cooling along with dehumidification occurs.

7. If initially the entering water temperature is below the dewpoint of the entering air temperature, cooling with dehumidification results, but if ~~the~~ after that during the process, the water temperature is more than the dewpoint of entering air cooling with humdification results.

8. The performance factor of an air washer increases

with an increase in number of banks.

9. The performance factor of an air washer increases with the decrease in air velocity.

10. The performance factor is more when the direction of spray is upstream than when it is downstream.

11. The performance factor increases with an increase of spray nozzle pressure for a particular velocity, number of banks and direction of spray.

c) HEATED WATER :-

12. When the temperature of water is increased more than the dry bulb temperature of the entering air, heating along with humidification results.

13. The factor $\left(1 - \frac{t_W' - t_{Wb}'}{t_W - t_{Wb}}\right)$ increases with an increase of nozzle pressure, decreases with an increase of velocity and increases with the number of banks.

14. The above factor increases when direction of spray is upstream than when it is downstream.

15. The performance factor for chilled water is more than the factor $(1 - t_W' - t_{Wb}' / t_W - t_{Wb})$ for a particular arrangement of spray system.

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3. Airconditioning Principles by C.O.Mackey.

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2. Heating piping and airconditioning, Vol.26, No.9, 1954, "How to calculate airwasher performance when cooling".

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.2 Showing the water pump and filter for the airwasher.

Fig. 5.3 Showing spray nozzles, risers, headers and water tank.

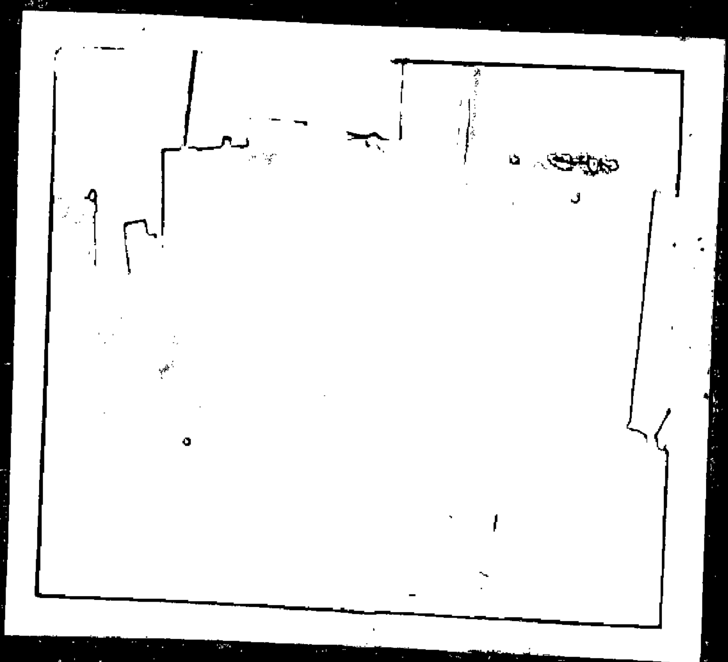

Fig. 5.4 Arrangement of tanks for supply of heated water.