16
Steam Turbines

16.1 CLASSIFICATION OF TURBINES AND THEIR WORKING

The steam turbine was invented in the last decade of the nineteenth century. It has undergone several changes in design during the past eight decades. The steam turbine has been used predominantly as primemover in all thermal power stations. It is not likely to be replaced in the foreseeable future. Nowadays, steam turbines of 1000 MW capacity unit are built in many countries and units of 1500 MW capacity are planned in future power programme. Future development in materials and other area promises to achieve even better performance and holds down cost of supplying our growing energy needs.

The steam turbines are mainly divided into two groups as :

(*a*) Impulse turbines ;

(*b*) Impulse-reaction turbines (in practice are known as reaction turbines).

In both types of turbines, first the heat energy of the steam at high pressure is converted into kinetic energy passing through the nozzles. The turbines are classified as impulse or reaction according to the action of high velocity steam used to develop the power.

In impulse turbine, the steam coming out at a very high velocity through the fixed nozzies impinges on the blades fixed on the periphery of a rotor. The blades change the direction of the steam flow without changing its pressure. The resulting motive force (due to the change in momentum) gives the rotation to the turbine shaft.

The examples of impulse turbine are De-Laval, Curtis and Reteau.

In the reaction turbine, the high pressure steam from the boiler is passed through the nozzles. When the steam comes out through these nozzles, the velocity of the steam increases relative to the rotating disc. The resulting reaction force of the steam on nozzle gives the rotating motion to the disc and the shaft. The shaft rotates in opposite direction to the direction of steam jet.

In practice, we hardly find any reaction turbine as described above. The common type is impulse reaction turbine known as reaction turbine in practice.

In an impulse reaction turbine, the steam expands both in fixed and moving blades continuously as the steam passes over them. Therefore, the pressure drop occurs gradually and continuously over both moving and fixed blades. The example of such turbine is Parson's turbine.

16.2. COMPOUNDING OF STEAM TURBINES

If the entire pressure drop from boiler pressure to condenser pressure is carried out in single stage nozzle, then the velocity of the steam entering into the turbine could be very high of the order of 1500 m/sec. The turbine rotator velocity (blade velocity) will be very high, of the order of 30,000 r.m.p. as it is directly proportional to the steam entering velocity. Such high R.M.P. of the turbine rotor is not useful for practical purposes and a reduction gear is necessary between the turbine and external equipment (generator) driven by the turbine. There is also danger of structural failure of the blade due to excessive centrifugal stresses. Therefore the velocity of the blades is limited to 400 m/sec.

The velocity of the steam at the exit of the turbine is sufficiently high when single stage blades are used. This gives a considerable loss of kinetic energy (about 10 to 12%). The above-mentioned difficulties associated with the single stage turbine can be solved by compounding. The combination of the stages are known as compounding.

1. Velocity Compounding. The arrangement of this type of compounding is shown in Fig.16.1. There is only one set of nozzles and two or more rows of moving blades. There is also a row of fixed blades in between the moving blades as shown in figure. The function of fixed blades is only to direct the steam coming out from first moving row to the next moving row. The heat energy drop takes place only in the nozzle at the first stage and it converts into kinetic energy. The kinetic energy of the steam gained in the nozzles is successively used by the rows of moving blades and finally exhausted from the last row of the blades on the turbine rotor. The function of the fixed blades is merely to turn the steam into the direction required for entry into the next row of rotor blades without altering pressure and velocity of the steam. A turbine working on this principle is known as velocity compounded impulse turbine. The Curtis turbine is an example of the velocity coumpounded steam turbine.

The velocity, pressure and specific volume variation of the steam are shown on the base representing the axis of the turbine as shown in Fig. 16.1. The specific volume of the steam remains constant as the steam flows along the axis of the turbine therefore the blade height of all rows is same.

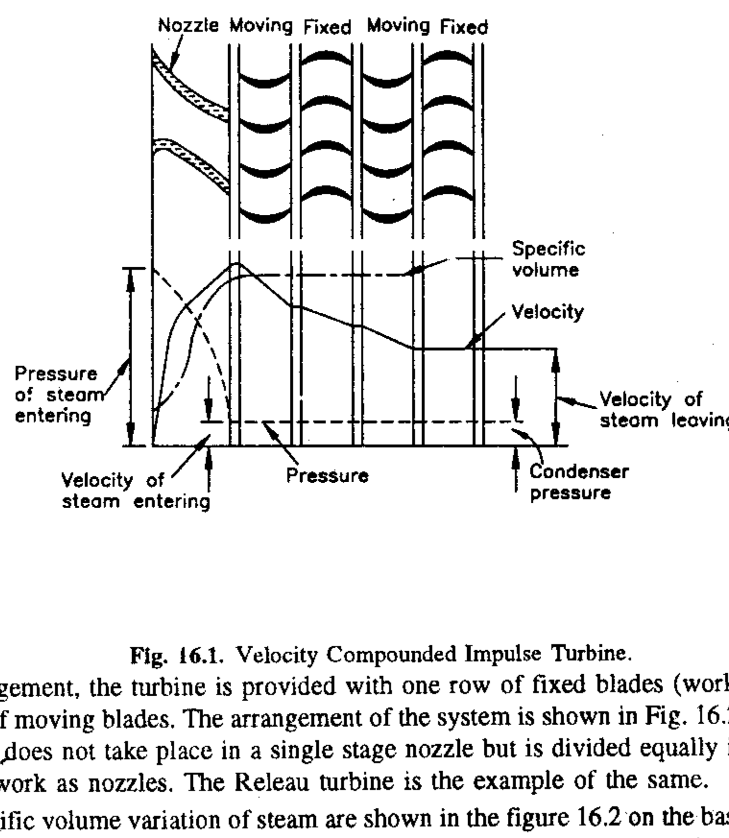


Fig. 16.1. Velocity Compounded Impulse Turbine.

2. Pressure Compounding. A number of simple impulse turbine sets arranged in series is known as pressure compounding. In this arrangement, the turbine is provided with one row of fixed blades (works as nozzles) at the entry of each row of moving blades. The arrangement of the system is shown in Fig. 16.2. The total pressure drop of the steam does not take place in a single stage nozzle but is divided equally in all the rows of fixed blades which work as nozzles. The Releau turbine is the example of the same.

The velocity, pressure and specific volume variation of steam are shown in the figure 16.2 on the base representing the axis of the turbine. As the pressure of the steam gradually decreases, the specific volume of the steam gradually increases therefore the blade height has to be increased towards the low pressure side.

3. Pressure and Velocity Compounding. This compounding is a combination of pressure and velocity compounding. The total pressure drop of the steam from boiler to condenser pressure is divided into a number of stages as done in pressure compounding and velocity obtained in each stage is also compounded as shown in Fig. 16.3.

This arrangement requires less stages and compact turbine can be designed for a given pressure drop.

The velocity, pressure and specific volume variations of steam are shown on the base representing the axis of the turbine. The blade height in the second stage must be greater than the first stage as the specific volume of steam is higher in the second stage.

This compounding has an advantage of pressure compounding to provide higher pressure drop in each stage and hence less number of stages and an advantage of velocity compounding to reduce the velocity of each blade row.

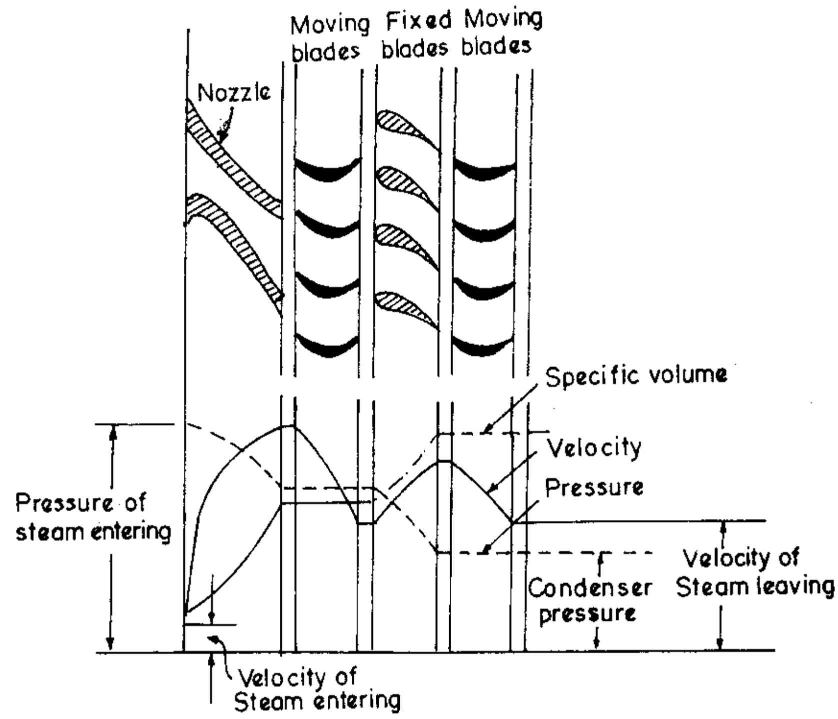


Fig. 16.2. Pressure Compounded Steam Turbine.

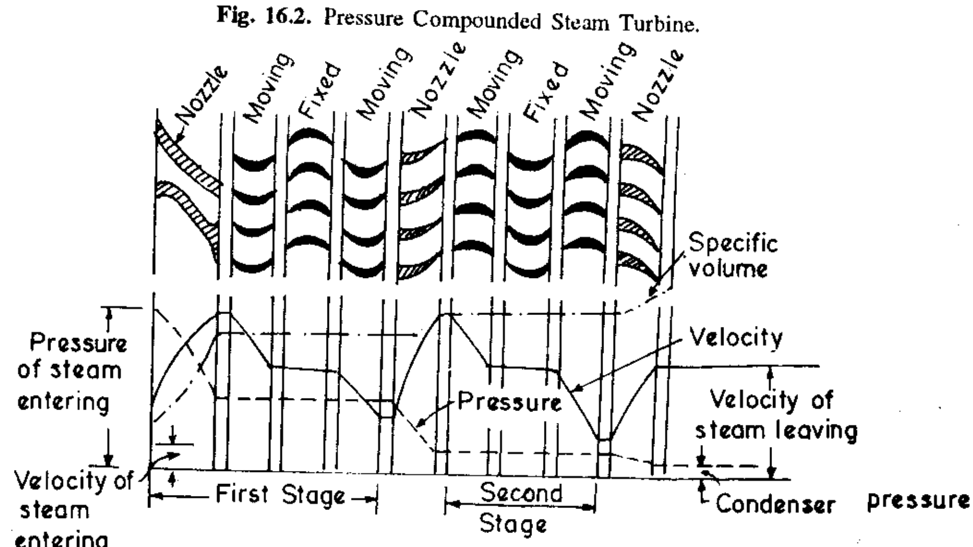


Fig. 16.3. Velocity and Pressure Compounded Steam Turbine.

16.3. ADVANTAGES AND DISADVANTAGES OF VELOCITY COMPOUNDING

Advantages :

(1) It requires less number (2 to 3 only) of stages, therefore initial cost is less.

(2) The space required is less.

(3) The system is easy to operate and more reliable.

(4) The turbine housing need not be made strong as the pressure in the housing is considerably less because the total pressure falls in the nozzle only.

Disadvantages :

(1) The friction losses are too larger due to the high velocity of steam.

(2) The maximum blade efficiency and efficiency range decrease with an increase in number of stages as shown in Fig. 16.4.

(3) The power developed in each successive blade row decreases with an increase in number of rows, even though all the rows require same space, material and initial cost. Therefore all the stages are not economically used. Velocity compounded steam turbines are generally used as drives for centrifugal compressors, centrifugal pumps, small generators and feed pumps of high capacity power plants.

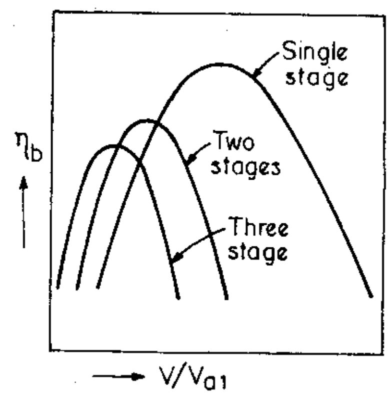


Fig. 16.4.

16.4. LOSSES IN STEAM TURBINES

The causes for the energy losses in steam turbines are listed below :

(1) Residual Velocity Loss. The steam leaves the turbine with some absolute velocity. The energy loss due to absolute exit velocity of steam is equivalent to $\frac{V_{aex}^2}{2gJ}$ kJ/kg, where V_{aex} is absolute velocity of steam leaving the turbine.

The residual velocity loss is 10 to 12% in a single stage impulse turbine. This loss is reduced by using the multistages.

(2) Loss due to Friction and Turbulence. Friction loss occurs in nozzles, turbine blades and between the steam and rotating discs. The friction loss in the nozzle is taken into account with introducing a factor 'nozzle efficiency'. The loss due to friction and turbulence is about 10%.

(3) Leakage Loss. The leakage of steam occurs at the points mentioned below :

(*a*) Between the turbine shaft and bearings.

(*b*) Between the shaft and stationary diaphragms carrying nozzles in case of reaction turbines.

(*c*) Leakage at the blade tips in case of reaction turbine.

(*d*) Leakage of steam through the glands.

The total leakage loss is about 1 to 2%.

(4) Loss due to Mechanical Friction. The loss due to friction between the shaft and bearing comes under this category. Some loss also occurs in regulating the valves.

This friction loss can be reduced with the help of an efficient lubricating system.

(5) Radiation Loss. The heat is lost from the turbine to the surroundings as its temperature is higher than atmospheric temperature. Usually the turbines are highly insulated to reduce this loss. The loss due to radiation is always negligible.

(6) Loss due to Moisture. The steam contains water particles passing through the lower stages of the turbine as it becomes wet. The velocity of the water particles is less than the steam and therefore the water particles have to be dragged along with the steam and consequently part of the K.E. of the steam is lost.

16.5. GOVERNING OF STEAM TURBINES

The main function of the governing is to maintain the speed constant irrespective of load on the turbine. The different methods which are commonly used for governing the steam turbines are listed below.

1. Throttle Governing.
2. Nozzle Control Governing.
3. By-pass Governing.
4. Combination of throttle and nozzle Governing.
5. Combination of throttle and By-pass Governing.

1. Throttle Governing. The arrangement of this governing is shown in Fig. 16.5. The quantity of steam entering into the turbine is reduced by the throttling of the steam. The throttling is achived with the help of double heat balanced valve which is operated by a centrifuagal governor through the servo-mechanism as shown in figure. The effort of the governor may not be sufficient to move the valve against the piston in big units. Therefore an oil operated relay (servo-mechanism) is incorporated in the circuit to magnify the small force produced by the governor to operate the valve.

Let the position of the governor (position of pilot piston and relay-piston) shown in the figure correspond to the full load on the turbine and running at full speed. If the load on the turbine is reduced, the turbine will start to rotate at speed greater than full load speed as the energy supplied is same. An increased speed of the turbine shaft causes the governor sleeve to move upward and this causes to move the pilot piston upwards. The upward motion of the pilot piston allows the high pressure oil to enter on the top side of the relay piston through upper part and allows the oil from the relay cylinder to come out (to oil return) through pilot cylinder. The downward motion of the relay piston partly closes the throttle valve causing the reduction in steam supply. The reverse operation takes place when the load on the turbine increases.

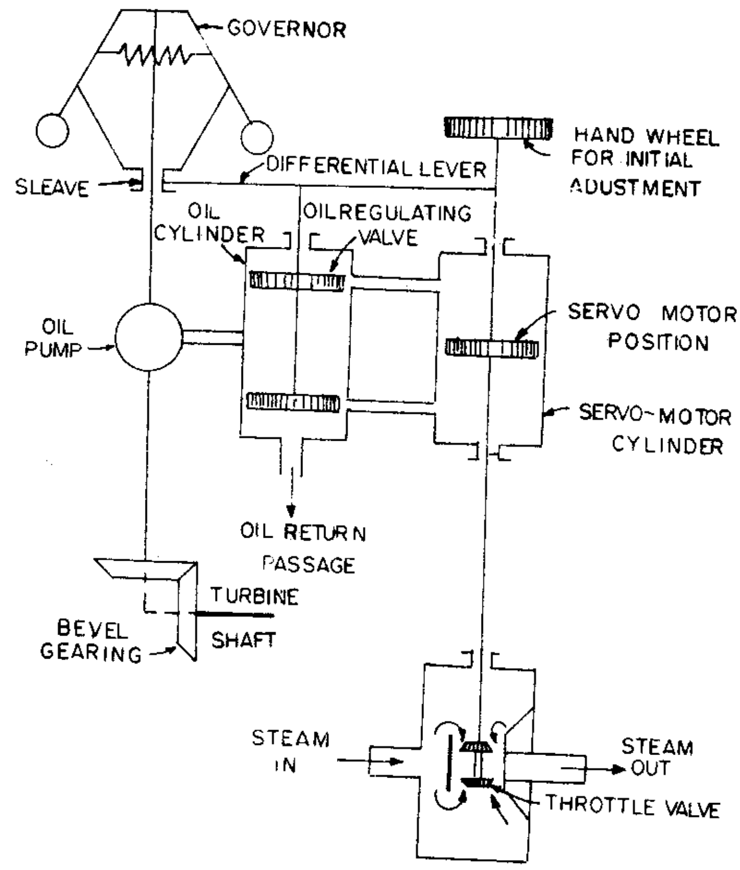


Fig. 16.5. Throttle governing (indirect regulation of throttle valve by servomotor).

The throttle governing is simple in operation but thermodynamically inefficient as the available heat drop is reduced in the irreversible throttling process.

2. Nozzle Control Governing. In this method of control, the steam supplied to the different nozzle groups is controlled by uncovering as many steam passages as are necessary to meet the load by poppet values.

An arrangement often used for large steam power plants is shown in Fig. 16.6. The number of nozzles supplying the steam to the turbine are divided into groups as N_1, N_2 and N_3 and the supply to these nozzles is controlled by the valves V_1, V_2 and V_3.

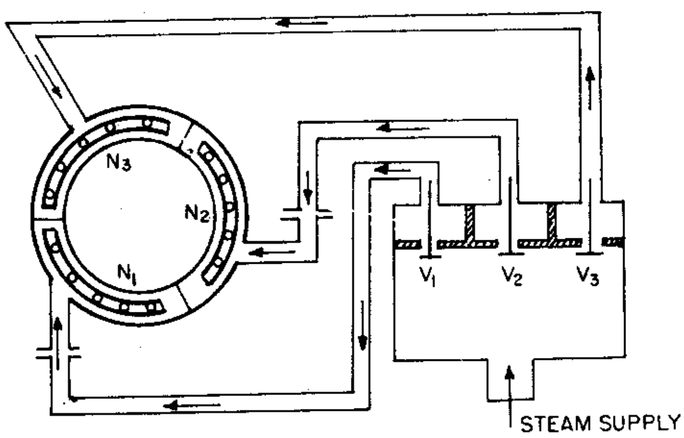


Fig. 16.6. Nozzle control governing.

3. By-pass Governor. More than one stage is used for high pressure impulse turbine to reduce the diameter of the wheel. The nozzle control governing cannot be used for multistage impulse turbine due to small heat drop in first stage. It is also desirable in multistage impulse turbine to have full admission into high pressure stages to reduce the partial admission losses. In such cases by-pass governing is generally employed.

The arrangement of by-pass governing is shown in Fig. 16.7. In this arrangement, for high loads, (higher than 80% full load as 80% full load is most economic load), a by-pass line is provided for the steam from the first stage nozzle box into the latter stage as shown in figure. The by-pass steam is automatically controlled by the lift of the valve. The by-pass valve remains under the control of speed governor for all loads within its range.

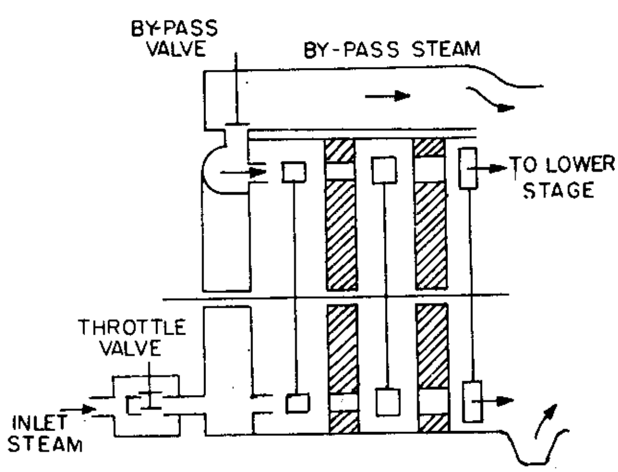


Fig. 16.7. By-pass governing.

Generally in practice a combination of 1 and 2 or 1 and 3 is always preferred.

16.6. LUBRICATION SYSTEM FOR STEAM TURBINES

Lubrication is needed to minimise turnin friction in main bearings, thrust bearings and reduction gears as well as the lubrication also helps to cool the journals and bearing surfaces

Proper lubrication is of utmost importance in the operation of turbine. High journal speed, heat conducted from the steam to the bearings and the possibility of water leaking into the oil are some of the problems that make lubrication difficult. The oil for lubrication should be continuous, under pressure, cool and free from injurious foreign matter.

A lubrication system used for a high capacity turbine (200 MW) is shown in Fig. 16.8. The sytem consists of main centrifugal type of pump as shown in figure which is directly fitted on H.P. turbine shaft and two more driven centrifugal type auxiliary oil pumps (P_1 and P_2).

The oil to the main centrifugal pump is supplied through the jet pumps (J_1) which maintain positive pressure at the suction of main pump.

The auxiliary pump P_1 is full duty rated, a.c.motor driven which is used only during starting, shutting down and in emergencies. It automatically comes in action when the oil pressure in the main relay system falls below a predetermined value.

The second auxiliary pump P_2 is driven by D C. motor. It only comes in action automatically when the pressure of the oil supplied to the bearings falls below a predetermined value.

Automatic alarm signals are fitted in the system to indicate the failure automatic pumps.

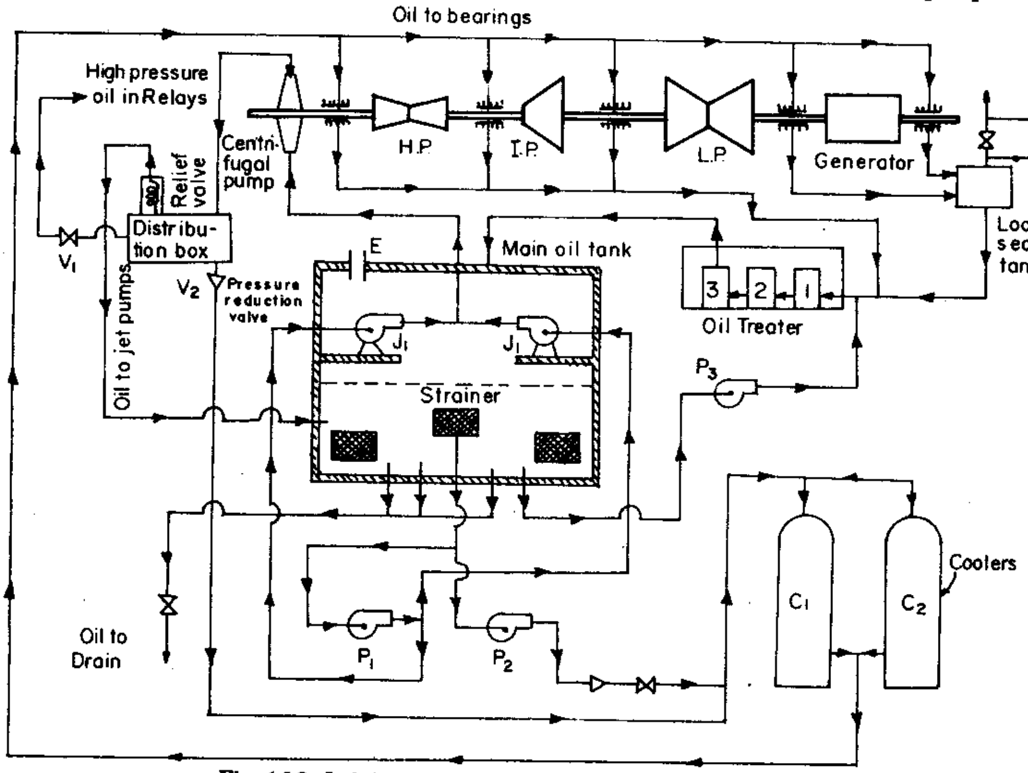


Fig. 16.8. Lubrication system for 200 MW capacity turbine.

The oil is taken from the tank by full duty auxiliary oil pump (P_1) and delivered to the distribution box as shown in figure through non-return valve (not shown). The distributor box is fitted with relief valve through whch surplus oil is returned to the main tank. High pressure oil from the distributor box is supplied to the relays through the valve V_1 as shown in figure. Oil for lubrication purposes passes through the pressure reducing valve V_2 as shown in figure. Oil drained from the bearings is returned back to the main tank through oil treater. It is first passed in the heater (1) to increase the temperature for increasing the fluidity. Then, it is passed through the oil centrifuge, (2) where the dust and solid materials are removed by centrifugal action. Lastly it is passed through the oil purifier, (3) where the dissolved gases and chemical impurities are removed.

All the alternators in the modern power plants are hydrogen cooled. Therefore, there is every possibility of accumulation of hydrogen in the control system. To avoid this, a vapour extractor (E) is fitted on the main oil tank and also on the loop seal tank for the oil return from the alternator shaft. The oil vapour and H_2 collected is discharged to the atmosphere through a pipe by the extractor.

To maintain the purity of the oil, part of oil from the main is continuously recirculated passing through the oil treater with the help of a pump P_3 as shown in figure. Oil drains are also provided at the bottom of main tank. Part of the oil from the tank is drained out after some predetermined days which also carries the solid sludge collected at the bottom of the tank.

Oil coolers are also provided to maintain the quality of the oil (viscosity) before supplying to the bearings. More details about few important components are given below :

1. Oil Pumps. In addition to bearing lubrication, the lubricating system also supplies oil at 8 to 10 bar for hydraulic operation of governing and emergency stop valves. The main pump may be separate motor driven or gear driven from the main shaft or impeller may be directly mounted on the turbine shaft as shown in figure.

One or more separately driven pumps, one of which is likely to be steam-driven, are usally installed for use during periods or start up of shutdown. The auxiliary pumps are commonly arranged to start automotically to keep oil pressure to normal. The pumps must have considerably more capacity than required under normal operation in order to operate control valves quickly without causing oil pressure to drop enough to trip the stop valve.

A small motor driven pump (not shown) is usually provided for units equipped with turning gear to supply oil at low pressure to the turbine bearings only during the period of turning gear operation. This pump also shuts down if other pump fails.

2. Oil Filters and Purifiers. In majority of the lubrication systems, a continuous oil filtering is provided which helps to remove water and dirt continuously. The intermittent use of treatment system is necessary if the foreign material entering into the system is negligible.

The oil filters which are generally used are of centrifugal types and cloth filter type. The centrifugal filters remove dirt, water from the oil using the effect of centrifugal force to remove the heavier particles from the oil. They give good oil cleanliness but require regular cleaning maintenance. The only drawback of this filter is there is some tendency for increased oxidation of oils because of aeration in the filtering process.

The cloth type filters are also commonly used either in combination with centrifugal type or separately. The oil flows through this filter by gravity. The cloth-bag filters have to be cleaned and washed periodically but little attention is required compared with centrifugal type. A spare set of filters is used in the system so that filter operation need not be interrupted.

Large settling tanks with separating trays are used to permit the water in the oil to separate out by gravity.

The oil purifier includes heating apparatus, deaerating tank and chemical treatment if required. It is not possible to give the details of each component in this text.

An oil system of 5000 litres capacity with a purifier's capacity of 500 litres can purify the complete oil within 10 hours.

3. Oil Coolers. The oil coolers are included in the system to maintain required temperature of lubricating oil before supplying to the bearing and control system. The cooling of oil is necessary as the oil absorbs heat passing through the bearings as well as it is purposely heated before passing through the centrifuge for effective removal of solid material. The cooling surface area required depends upon the quantity of water available as well as its temperature.

Vertical coolers are generally used because it helps to collect the sludge to the bottom. The oil should enter the cooler from top (for better sludge collection) and water from bottom because this counter flow provides effective cooling. The oil pressure in the cooler should be higher than water pressure to prevent leakage of water into the oil system. It has been found by test that an increase in oil velocity is more effective to increase an overall heat transfer coefficient than corresponding increase in water velocity. High velocity of oil involves heavy pressure drop particularly during starting when the viscosity of the oil may be six times higher than the viscosity under normal working conditions. Therefore, a special loaded by-pass value across the cooler is provided (not shown in Fig.) to avoid overloading the oil pump. The overall heat transfer coefficient under normal working conditions when oil velocity is 30 m/min and water velocity is 20 m/min. is 200 kW/m^2-°C.

The design and layout of the cooler should permit easy access to the tubes for cleaning. The cleaning of the oil side (outer surface of tubes) is done with degreasing agent such as trichlorethylene or by immersing the tube nest in the solution of NaOH or Na_2CO_3. After cleaning with chemical solution is mentioned above, it is necessary to remove all traces of cleaning solution. This is generally done by means of steam jet with repeated washing.

4. Vapour Extractor. The large turbines which drive hydrogen cooled generators usually have a vapour extractor to remove hydrogen and water vapour which may accumulate in the oil-system. A positive displacement pump is used to exhaust the vapours from the oil tank. The bent form vapour extractor should run to some point where free air movement is sufficient to dissipate any possible hydrogen-gas concentration and where oil vapour condensation will not cause a nuisance.

5. Lubricating Oil. Steam turbine lubricating oil is highly refined mineral oil with good antioxidation and demulsibility properties. Less frequent oil cleaning (4-6 months) is required with inhibited oils than with conventional (1 month). However, even the best oil cannot be used continuously without purification. Either the entire charge is drained out when the unit is shut-down or a small percentage of oil is continuously passed through the purifier as shown in figure.

In starting a new turbine, a great care should be exercised to be sure that all dirt, pipe scale, lint or other foreign material is removed from the oil reservoir and oil piping. The oil pipes should be brushed, cleaned and flushed before erection. When the turbine oil system is erected, the oil system is filled with oil and circulation of oil through the system is started. Temporary bypass is often provided so that oil does not flow through the bearing initially. This initial circulation helps for clearing the pipe system of the lubricating oil. Sometimes, special flushing fluids are passed through the system before filling with new lubricating oil.

16.7. TURBINE TROUBLES

The following troubles may occur during the running of turbines which may cause the damage to the turbines :

(1) Loss of blade shrouding.

(2) Damage of the seal.

(3) Failure of a bearing or whipping of shaft because of improper lubricating-oil pressure ; temperature or viscosity.

Sudden increase in the vibration of turbine is most usual indication of any trouble caused during running of the turbine.

Under these circumstances, it is always advisable to shut the unit (if automatic shutting device is not incorporated in the machine) as soon as possible and find out the cause of trouble.

16.8. BLADE MATERIALS FOR TURBINES

The creap phenomenon is the main criterion in selection of blade material specially for high temperature region. 1% Cr—Mo—V alloy and stainless steels having 12% Cr are widely used. Austenitic alloys are preferred for still higher temperature.

Blades of L.P. stage, though, at the low temperature end have to withstand the effect of corrosion and erosion due to water droplets (0.25 mm), about 10-12% stainless iron is commonly used. New materials such as titanium, plastics reinforced with carbon having a lower specific weight and higher strengths are also considered as they have high tensile strength (70 kg_f/mm^2).

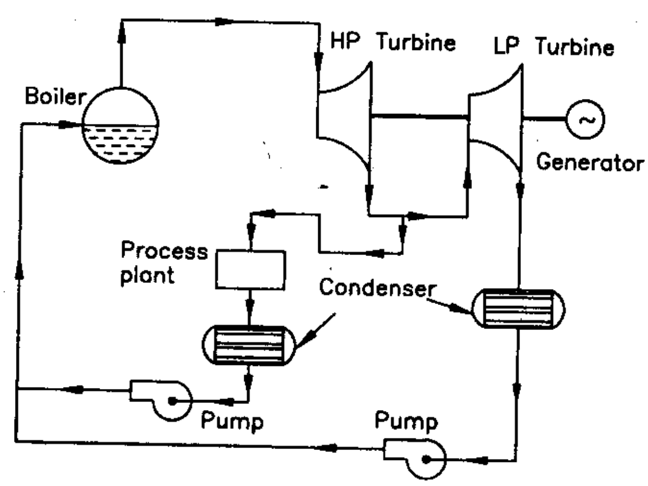


Fig. 16.9. Extraction turbine.

16.9. INDUSTRIAL STEAM TURBINES

Industrial steam turbines supply power to the industries as well as low pressure steam required for processing. Steam is required in paper industry, chemical industry, textile industry and many others for drying, heating etc. According to the type of steam supplied, the industrial steam turbines are classified as follows :

1. Extraction Turbines. In this turbine, a high pressure steam from boiler enters H.P. turbine and expands during work. Part of the steam coming out from H.P. turbine is drawn for use in the industrial process as shown in Fig. 16.9. Remaining steam is further expanded in the L.P. turbine. The exhaust steam from L.P. turbine and the industrial process plant are condensed in different condensers as the condensing pressures are different. The condensate is supplied to the boiler with the help of feed pumps.

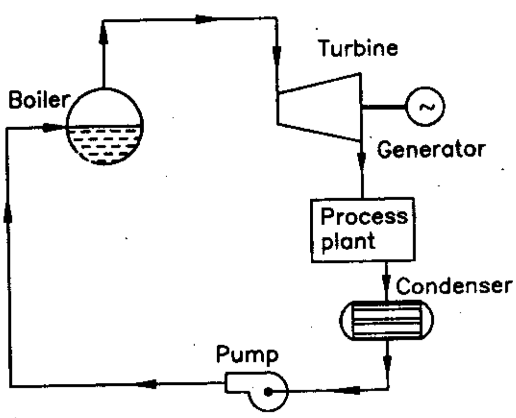


Fig. 16.10. Back pressure turbine.

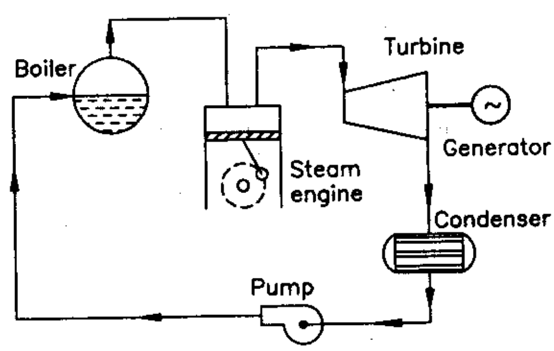


Fig. 16.11. Exhaust turbine.

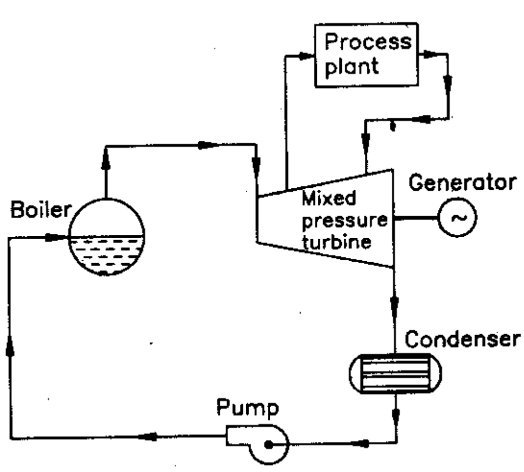


Fig. 16.12. Mixed process turbine.

2. Back Pressure Turbine. An arrangement is shown in Fig. 16.10. The steam after expansion in the turbine is used in processing plant and then condensed in a condenser and fed back to boiler with the help of the pump. The pressure of steam at the exit of the turbine is always above atmospheric pressure, therefore it is known as back pressure turbine.

3. Exhaust Turbine. Sometimes, the exhaust steam coming out of steam engine is used to generate power passing the steam through turbine as shown in Fig. 16.11. The exhaust pressure of the engine is atmospheric whereas the turbine exhausts into vacuum. If the steam from the engine is not utilised in this way, the energy in the steam would be wasted.

4. Mixed Pressure Turbine. In some industries like rolling mills, the steam is required at considerably higher pressure and it is also exhausted at a pressure considerably higher than atmosphere. For such requirements, the steam is extracted at higher pressure from the turbine and again supplied at a lower pressure to the turbine as shown in Fig. 16.12. The steam coming out of turbine finally is condensed and fed back to the boiler.

EXERCISES

16.1. How the turbines are classified ? Explain clearly the difference between impulse and reaction turbine.

16.2. Explain the difference between velocity compounding and pressure compounding of steam turbines. Which is more preferred in practice and why ?

16.3. List out the main components of steam turbine and explain the function of each.

16.4. Discuss the following : low pressure, mixed pressure and bleeder turbine.

16.5. List out the causes of losses in steam turbines and mention the preventive measures to be taken to reduce them.

16.6. What procedure is adopted during starting and stopping the steam turbine of high capacity power plant ? List out the sequence.

16.7. What are the different methods of governing the steam turbine ? Draw a neat diagram of throttle governor and explain its working.

16.8. Draw a neat diagram of lubrication system used for high capacity power plant turbine and explain the working.

16.9. Why protective devices are required for steam turbine ? Explain the working of different types of protective devices with neat sketches.

16.10. What are the common causes of turbine vibrations ?

□□□

17
Condensers

17.1. INTRODUCTION

The generation of electrical energy in steam power plants is relatively inefficient process. In modern fossil fuel plant, approximately 5000 kJ are rejected to the circulating water for each kW-hr of energy produced. With the nuclear power plant, it is 50% greater than that for a fossil fueled plant. This means that more than 10% of heat from the fuel burned in modern fossil fueled plant and that about 2/3rd of the heat generated in nuclear plant is rejected to the cooling water system. Therefore, the quantity of water required for a thermal power plant of 400 MW capacity requires nearly 50 to 100 thousand tons of water per hour. This requirement for cooling water makes water availability a paramount factor in choosing the plant site. Therefore, the plant must be sited near the sources for large quantities of water like ocean, bays, rivers and lakes. The planner would have much more flexibility in power plant siting if the availability of cooling water were not a major consideration.

The use of condenser in the power plant improves the efficiency of the power plant by decreasing the exhaust pressure of the steam below atmosphere. Another advantage of the condenser is that the steam condensed may be recovered to provide a source of good pure feedwater to the boiler and reduces the water softening plant capacity to a considerable extent.

The maximum possible thermal efficiency of a power system is given by $(T_1 - T_2)/T_1$ where T_1 and T_2 are the supply and exhaust temperatures. This expression of efficiency shows that the efficiency increases with an increase in temperature T_1 and with the decrease in temperature T_2. The maximum value of temperature T_1 of the steam supplied to a steam prime-mover is limited by the material consideration. The temperature T_2 (temperature at which heat is rejected) can be reduced if the exhaust of the steam prime mover takes place below the atmospheric pressure. This is because, there is definite relation between the steam temperature and pressure. Low exhaust pressure means low exhaust temperature. The steam cannot be exhausted to the atmosphere if it is expanded in the engine or turbine below atmospheric pressure. Under this condition, the steam is made to exhaust in a vessel known as condenser where the pressure inside is maintained below the atmospheric pressure by condensing the steam with the circulation of the cold water. A closed vessel in which steam is condensed by abstracting the heat from steam and the pressure is maintained below atmospheric pressure is known as condenser. The steam condenser is one of the most essential components of all modern steam power plants.

17.2. ELEMENTS OF STEAM CONDENSING PLANTS

The steam power plants using condenser are shown in Fig. 17.1 (*a*) and (*b*) Fig. 17.1 (*a*) shows that the cooling water used in condenser is not re-circulated again and again but discharged to the downstream side of the river. Whereas Fig. 17.1 (*b*) shows that the cooling water is re-circulated again and again by passing through the cooling tower.

The different components of steam condensing plant are listed below :

1. Condenser. 2. Supply of cooling water. 3. Condenser cooling water pump. 4. Condensate extraction pump. 5. Hot-well. 6. Boiler feed pump. 7. Air extraction pump. 8. Cooling tower. 9. Make up water pump.

17.3. ADVANTAGES OF CONDENSER

The advantages obtained by incorporating a condenser in the steam power plant are listed below :

1. The condensed steam from the condenser is used as feed water for boiler. Using the condensate

as feed for boiler reduces the cost of power generation as the condensate is supplied at higher temperature to the boiler and it reduces the capacity of the feed water cleaning system.

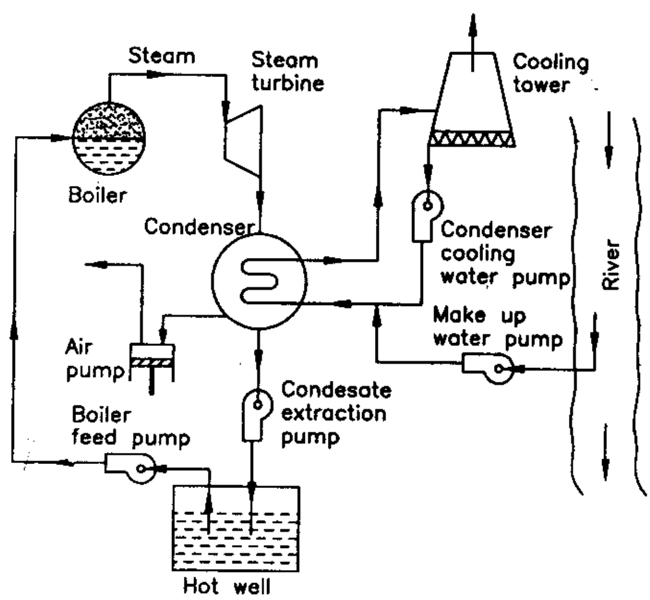
2. The efficiency of the plant increases as the enthalpy drop increases by increasing the vacuum in the condenser.

The specific steam consumption of the plant also decreases as the available enthalpy drop or work developed per kg of steam increases with the decrease in back pressure by using condenser.

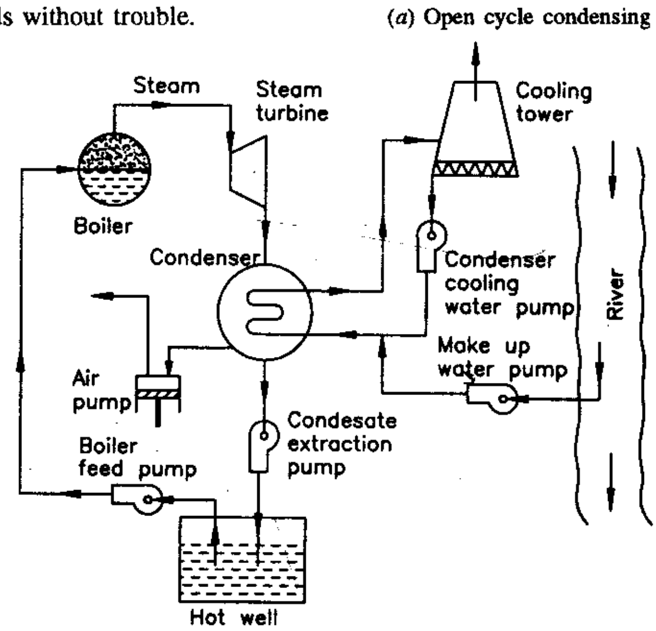
3. The deposition of salt in the boiler is prevented with the use of condensate instead of using the feed water from outer source which contains salt. The deposition of salt in boiler shell also reduces the boiler efficiency. This is particularly important in marine steam power plant.

The use of condenser in steam power plant reduces the overall cost of generation by increasing the thermal efficiency of the power plant.

The efficient condenser plant must be capable of producing and maintaining a high vacuum with the quantity of cooling water available and should be designed to operate for the prolonged periods without trouble.



(*a*) Open cycle condensing system.



(*b*) Closed cycle condensing system.

Fig. 17.1.

The desirable features of good condensing plant are :

(1) Minimum quantity of circulating water.

(2) Minimum cooling surface area per kW capacity.

(3) Minimum auxiliary power.

(4) Maximum steam condensed per m^2 of surface area.

The effect of low vacuum is very pronounced. The efficiency of the power plant depends to a greater extent on the pressure at the exhaust than the high pressure condition of the steam at inlet. The rate of increase of steam consumption per 1 cm fall of vacuum from design turbine condition is between 1.5 to 2% for normal units operating at an absolute pressure of about 72 cm of Hg. This one fact is sufficient to explain the importance of condensing plant in thermal power station.

The quantity of water available and its temperature decides the vacuum which can be maintained in the condenser. The effect of water temperature and water quantity on the vacuum in the condenser is shown in Fig. 17.2 (*a*) and Fig. 17.2 (*b*).

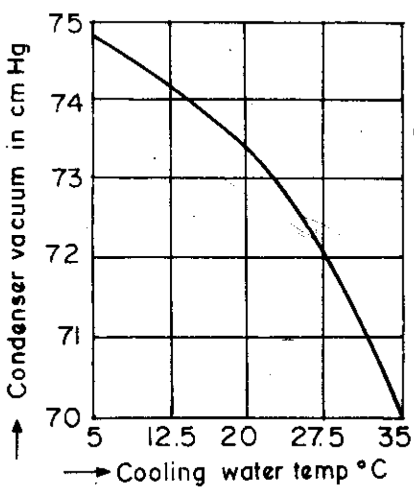


Fig. 17.2. (*a*) Effect on cooling water temp. on condenser vacuum.

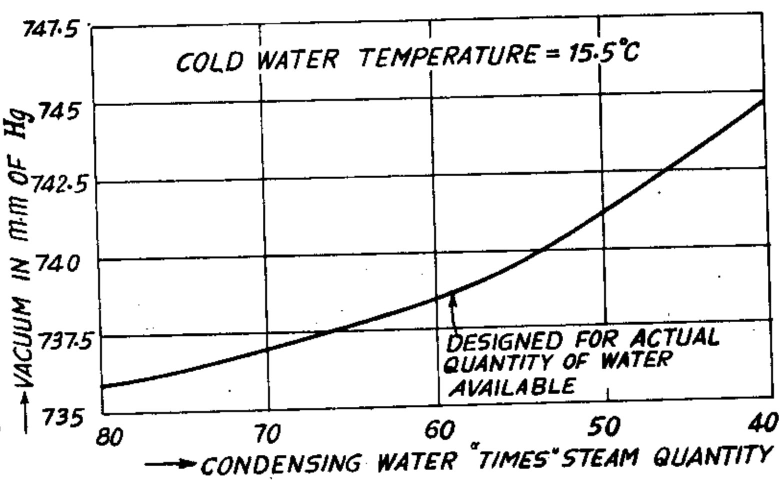


Fig. 17.2 (*b*). Effect of cooling water quantity on condenser vacuum.

17.4. TYPES OF STEAM CONDENSERS

The condensers are mainly classified as Mixing Type or Jet condensers and Non-mixing Type or Surface Condensers.

In mixing type condensers, the exhaust steam from primemover and cooling water come in direct contact with each other and steam condenses in water directly. The temperature of the condensate (condensed steam + cooling water) is same as that of cooling water leaving the condenser. The condensate coming out from the mixing type condenser cannot be used as feed to the boiler as it is not free from salt and pollutants. These types of condensers are generally preferred where the good quality water as feed to boiler is easily available in ample quantity. Mixing condensers are seldom used in modern power plants.

In non-mixing type of condenser, steam and cooling water do not come in direct contact with each other. The cooling water passes through the number of tubes attached to condenser shell and steam surrounds the tubes. These types of condensers are universally used in all high capacity modern steam power plants as the condensate coming out from the condenser is used as feed for the boiler.

A. Mixing or Jet Type Condensers. The jet condensers are mainly divided as parallel flow and counterflow jet condensers.

In parallel flow condensers, the steam and cooling water flow in the same direction whereas they flow in opposite directions in counterflow condensers.

Mixing type condensers are mainly classified into three categories depending upon the arrangement used for the removal of condensate as low level, high level and ejector condenser. These are rarely used in modern high capacity power plants, therefore further description is not given in this text.

(B) Non-mixing Type or Surface Condensers. In this type of condenser, the cooling water and exhaust steam do not come in direct contact with each other as in case of jet condensers. This is generally used where large quantities of inferior water are available and better quality of feed water to the boiler must be used most economically.

The arrangement of the surface condenser is shown in Fig. 17.3 (*a*). It consists of a cast iron air-tight cylindrical shell closed at each end as shown in figure. A number of water tubes are fixed in the tube plates which are located between each cover head and shell.

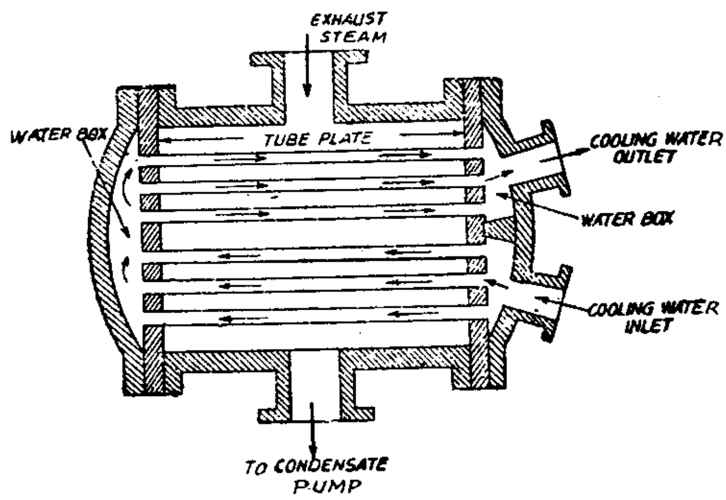


Fig. 17.3 (*a*). Surface condenser.

The exhaust steam from the prime mover enters at the top of the condenser and surrounds the condenser tubes through which cooling water is circulated under force. The steam gets condensed as it comes in contact with cold surface of the tubes. The cooling water flows in one direction through the first set of the tubes situated in the lower half of condenser and returns in the opposite direction through the second set of the tubes situated in the upper half of the condenser as shown in figure. The cooling water coming out from the condenser is discharged into the river or pond. The condensed steam is taken out from the condenser by a separate extraction pump and air is removed by an air pump.

A section of tubes near the air pump suction is screened off by providing a baffle as shown in Fig. 17.3 (*b*) and 17.3 (*c*). The number of tubes used in this section per unit area are more compared with other part of the condenser. The velocity of water through these tubes is also maintained higher. This is done

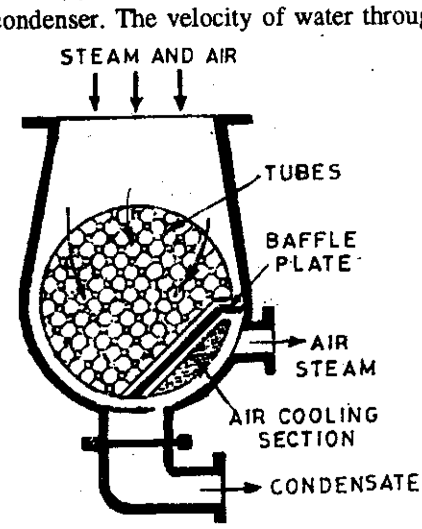


Fig. 17.3 (*b*). Down flow condenser.

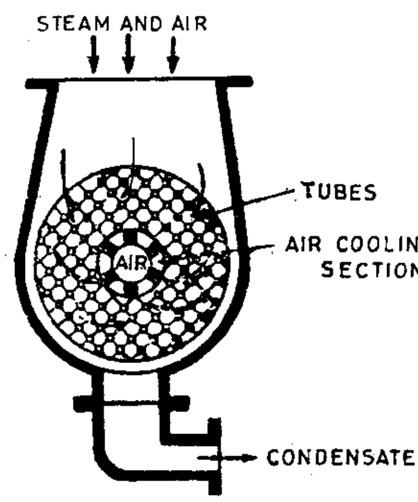


Fig. 17.3 (*c*). Central flow condenser.

to reduce the amount of steam going along with air. The extensive cooling of air in this section increases the density of air going out and reduces the capacity of air pump (size) required by as much as 50%.

The surface condenser requires three pumps when it works on dry-vacuum system, one for circulating the cooling water, one for extracting the condensate and third is required for removing the air. The surface condenser requires only two pumps when it works on wet vacuum system, one for circulating the cooling water and other for extracting the air and condensate together. These types of condensers are widely used for high capacity power plants.

The surface condensers may be classified according tc

(*a*) Number of water passes : single or multipass.

(*b*) Direction of condensate flow and tube arrangement : down flow, and central flow.

A two pass down-flow surface condenser is already shown in Fig. 17.3 (*a*). A sectional view of down-flow condenser is shown in Fig. 17.3 (*b*). The steam enters from the top and flows mainly downward over the tubes. The air is extracted at lower temperature than the condensate by providing a separate cooling section known as air cooler as explained earlier.

A sectional view of central flow condenser is shown in Fig. 17.3 (*c*). In this arrangement, the air cooling section is provided at the centre of the tube nest and air is extracted from this section. This arrangement causes the steam to flow radially towards the centre and passes over the entire periphery of tubes. The formed condensate is extracted from bottom as shown in Fig. 17.3 (*c*). This arrangement is an improvement over downward flow type as it has an access to the whole periphery of the tubes.

In all surface condensers, the cooling water is passed through the tubes and steam surrounds the tubes. The volume occupied by the tubes in the condenser shell is hardly 10% of the condenser shell volume. The specific volume of the steam at the condenser pressure is very large and if the steam is to be passed through the condenser tubes, then the number of tubes (volume occupied by the tubes) required is extremely large and increases the capital cost of condenser.

A good surface condenser should have a low pressure drop, maximum effective surface arrangement and should be economical in first cost. The total surface area required is reduced by utilizing the following principles.

(*a*) Open spacing is provided at the steam inlet to obtain a blanket of steam around the most effective tubes.

(*b*) Steam penetration lanes are provided to permit the steam to drive deep into the centre of the condenser.

(*c*) Higher loading per m^2 of surface area is provided.

(*d*) Deaerating and reheating types of hot-wells are provided.

In modern condensers, a steam bypass lane is provided along the side of the shell to pass steam to the hot-well for reheating the condensate. This also helps for deaeration of condensate before passing to the boiler.

Requirements of modern surface condensers. The requirements of an ideal surface condenser used for power plants are listed below.

1. The steam should be evenly distributed over the whole cooling surface of the condenser vessel with minimum pressure loss.

2. There should not be undercooling of condensate. To achieve this, the quantity of cooling water circulated should be so regulated that the temperature of cooling water leaving the condenser is equivalent to the saturation temperature of the steam corresponding to the steam pressure in the condenser.

3. The water should be passed through the tubes and steam must surround the tubes from outside for the reason mentioned earlier. This also helps to prevent the deposition of dirt on the outer surface of the

tubes. Any deposition collected on the inside surface of the tubes can be removed with the help of mechanical brushes just by removing the end cover plates.

4. There should not be air-leakage atall in the condenser as it destroys the vacuum in the condenser and reduces the work done per kg of steam. The presence of air also reduces the heat transfer rates in the condenser very rapidly. If at all there is any leakage of air, an arrangement (air-pump) should be made to remove it as rapidly as possible with least expenditure of energy.

5. An arrangement for cooling the air to a maximum possible extent should be provided in the condenser before its extraction. This reduces the loss of potential condensate and also reduces the capacity of air-extraction pump.

6. The rise in temperature of the cooling water passing through the condenser is limited to 10°C. Therefore quantity of water and surface area of the condenser causing the heat flow required are large.

$$Q = m_w(T_{wo} - T_{wi}) = UA_s(LMTD)$$

Where m_w is the mass of cooling water per sec passing through condenser tubes and T_{wi} and T_{wo} are the inlet and outlet temperatures of the cooling water.

U, A_s and $LMTD$ are overall heat transfer coefficient (W/m^2-°C), surface area of tubes and log mean temperature causing the heat flow from steam to cooling water.

The temperature T_{wo} should be below the saturation temperature of steam corresponding to partial pressure of steam in the condenser.

(C) Non-Conventional Direct Contact Condensers. This is a new concept in power industry where condensate is used to condense the exhaust steam in the condenser.

The arrangement of the system is shown in Fig. 17.4

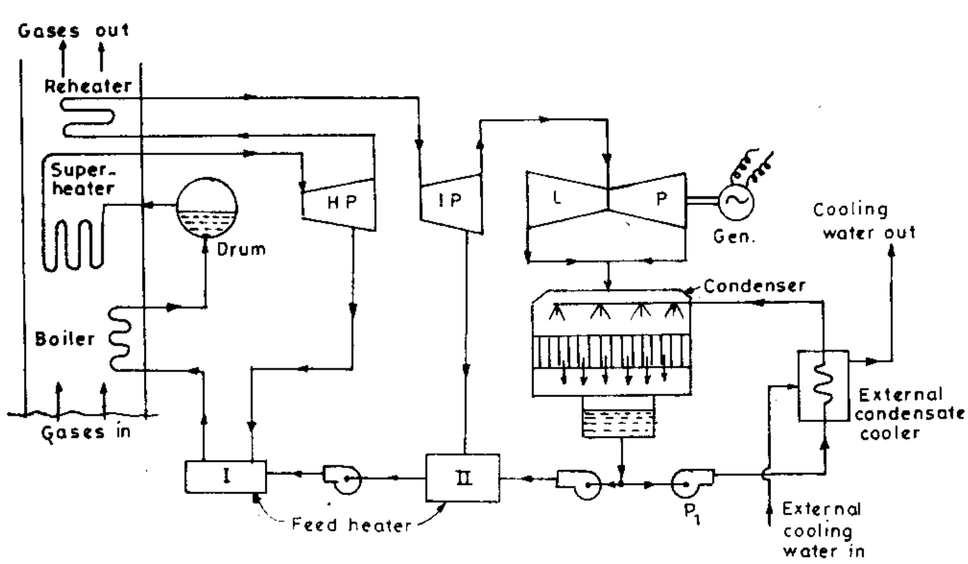


Fig. 17.4.

In this arrangement, part of the condensate coming out of condenser is cooled by external cooler as shown in figure and the same is used again in the condenser in the form of spray and the condenser is of mixing type.

This system increases unit availability by upto 3% compared to conventional condenser. The other benefits include a slightly lower heat rate, ease of maintenance, cleaner heat transfer surface, shorter condenser height, better part load performance and better control of dissolved O_2. The other advantages are listed below :

(1) The potential leakage is less than conventional condenser as number of joints are less. Therefore the outages are reduced.

(2) The pressure of the condensate side is higher than cooling water side (in cooler) which presents the leakage of contaminants into the condenser.

(3) The heat transfer rates are doubled and fouling factors are five times lower than those for conventional condensers.

(4) When cleaning is necessary, individual module of this system can be taken out of service with no effect on performance.

The major drawbacks are (*i*) total heat transfer surface (condenser and cooler together) is 30% more than conventional (*ii*) Need of pump around loop (P_1) consumes extra power and increases system complexity. In addition to this (*iii*) adequate filtration and treatment for biofouling of cooling water is necessary to prevent plugging.

(D) Evaporative Condensers. These condensers are more preferable where acute shortage of cooling water exists. The arrangement of the condenser is shown in Fig. 17.5. Water is sprayed through the nozzles over the pipe carrying exhaust steam and forms a thin film over it. The air is drawn over the surface of the coil with the help of induced fan as shown in figure. The air passing over the coil carries the water from the surface of condenser coil in the form of vapour. The latent heat required for the evaporation of water vapour is taken from the water film formed on the condenser coil and drops the temperature of the water film and this helps for heat transfer from the steam to the water. This mode of heat transfer reduces the cooling water requirement of the condenser to 10% of the requirement of surface condensers. The water particles carried with air due to high velocity of air are removed with the help of eliminator as shown in figure. The make-up water (water vapour and water particles carried with air) is supplied from outside source.

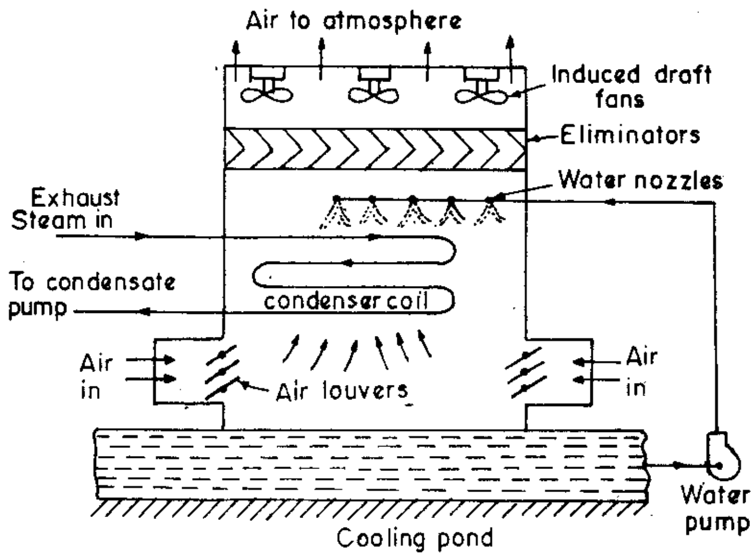


Fig. 17.5. Evaporative Condenser.

The quantity of water sprayed over the condenser coil should be just sufficient to keep the condenser coil thoroughly wetted. The water flow rate higher than this will only increase the power requirement of water pump without increasing the condenser capacity. This type of condenser works better in dry weather (low WBT) compared with wet weather as the water vapour carrying capacity of dry air is higher than wet air at the same temperature.

The arrangement of this type of condenser is simple and cheap in first cost. It does nor require large quantity of water therefore needs a small capacity cooling water pump. The vacuum maintained in this condenser is not as high as in surface condensers therefore the work done per kg of steam is less with this condenser compared with surface condenser. These condensers are generally preferred for small power plants and where there is acute shortage of cooling water.

17.5. AIR LEAKAGE, ITS EFFECTS ON THE PERFORMANCE OF CONDENSER AND METHODS OF ITS REMOVAL

The sources of air leakage in the condenser are listed below :

1. The air leaks through the joints, packings and glands into the condenser where the pressure is below the atmospheric pressure. The amount of air leakage through these sources depends upon the quality of workmanship.

2. The feed water contains air in dissolved condition. The dissolved air gets liberated when the steam is formed and it is carried with the steam into the condenser.

Normally the quantity of air leakage in surface condenser is 0.05% of steam condensed when the steam turbine is used as prime mover.

The effects of air leakage in a condenser are listed below :

1. It increases the pressure in condenser or back pressure of the prime-mover and reduces the work done per kg of steam.

2. The pressure of air lowers the partial pressure of steam and its corresponding temperature.

The latent heat of steam increases at low pressure. Therefore, more quantity of water is required to condense one kg of steam as the quantity of latent heat removed is more. There is greater possibility of under-cooling the condensate with the reduction in partial pressure of steam due to the presence of air. This phenomenon reduces the overall efficiency of the power producing plant.

3. The heat transfer rates are greatly reduced due to the presence of air because air offers high resistance to heat flow. This further necessitates the more quantity of cooling water to maintain the heat transfer rates. Otherwise, it reduces the condensation rate and further increases the back pressure of the prime mover.

It is obvious from the above discussion that the preventive measures should be taken to remove leaking air from the condenser to avoid its bad effects.

The air from the condenser is removed with the help of airpumps. The primary function of the air pump is to maintain the vacuum in the condenser which corresponds to the exhaust steam temperature by removing the air. Another function of the pump is to remove the condensate coming out from the bottom of the condenser.

An air-pump which removes both air and condensate together is called 'Wet Air-Pump' while the air-pump which removes only the moist air is known as 'Dry-Air-Pump'.

The types of air-pump which are commonly used are listed below :

1. Steam Ejectors (Generally dry).
2. Rotary Type (Generally dry).
*3. Reciprocating Type (dry or wet).

17.6. VACUUM EFFICIENCY

The lowest pressure (or highest vacuum) which can exist in a condenser is the saturation pressure of steam corresponding the steam temperature entering into the condenser. But the actual pressure in the condenser is always greater than the ideal pressure by an amount equal to the partial pressure of air present in the condenser.

Assume p_s = Saturation pressure of steam in bar corresponding to the steam temperature entering into the condenser.

p_t = Total pressure of the air and steam in the condenser $(p_a + p_s)$.

p_b = Atmospheric or barometric pressure.

*For further details of these equipments, the readers are requested to see the book on Thermodynamics and Heat Engines by the same authors.

Ideal vacuum possible without air leakage

$$= (p_b - p_s)$$

Actual vacuum existing in condenser due to air leakage

$$= p_b - p_t = p_b - (p_a + p_s)$$

where p_t is the actual pressure in the condenser.

The vacuum efficiency is defined as the ratio of actual vacuum to ideal vacuum.

$$\text{Vacuum efficiency} = \frac{p_b - (p_a + p_s)}{p_b - p_s} = \frac{p_b - p_t}{p_b - p_s} \qquad \text{...(17.1)}$$

If there is no air leakage into the condenser, then $p_a = 0$ and, therefore, the vacuum efficiency becomes 100%.

The leakage of air and insufficient quantity of cooling water decrease the vacuum efficiency of the condenser.

17.7. DALTON'S LAW OF PARTIAL PRESSURE USED FOR CONDENSER ANALYSIS

Dalton's law of partial pressure states that "the total pressure exerted by a mixture of gases or a mixture of gas and vapour is equal to the sum of individual partial pressures of the constituents of the mixture if individual partial pressures of the constituents of the mixture when individual alone occupies total volume occupied by mixture having the same temperature of mixture."

This is explained by taking the following example.

The total pressure in the condenser is the sum of the partial pressures of steam and air.

According to Dalton's law of partial pressure

$$p_t = p_s + p_a \qquad \text{...(17.2)}$$

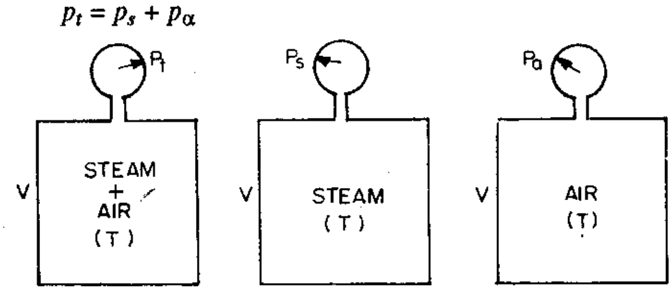


Fig. 17.6.

where V = volume of condenser shell.

T = Temperature of mixture (air + steams) in the condenser.

p_t = Actual total pressure in the condenser.

p_s = Partial pressure of steam in condenser.

p_a = Partial pressure of air in condenser.

m = Total mass of mixture (air + steam) in the condenser shell.

m_s = Mass of steam in condenser shell.

m_a = Mass of air in condenser shell.

v_s = Specific volume of saturated water vapour at temperature T and pressure P_s.

v_a = Specific volume of air at temperature T and Pressure P_a.

$V = m_s . v_s = m_a . v_a$.

$$\therefore \quad \frac{m_a}{m_s} = \frac{v_s}{v_a} \qquad \text{...(17.3)}$$

The mass of air per m^3 of the condenser shell

$$= \frac{m_a}{V} = \frac{1}{v_a} \quad ...(17.4)$$

and the mass of water vapour per m^3 of the condenser shell

$$= \frac{m_s}{V} = \frac{1}{v_s} \quad ...(17.5)$$

The total mass of mixture in the condenser shell can be written as

$$m = m_a + m_s$$

$$= m_a\left(1 + \frac{m_s}{m_a}\right) = m_a\left(1 + \frac{v_a}{v_s}\right) \quad ...(17.6)$$

or

$$m = m_s\left(1 + \frac{m_a}{m_s}\right) = m_s\left(1 + \frac{v_s}{v_a}\right) \quad ...(17.7)$$

17.8. THERMODYNAMIC ANALYSIS OF CONDENSER

Figure 17.7 (*a*) shows the energy diagram for surface condenser. The suffixes *a*, *s* and *c* denote air, steam and condensate.

Applying the energy equation to the system (neglecting kinetic and potential energy changes), the heat transferred to the cooling water Q_w is given by

$$Q_w = (m_{s1} \,.\, H_{s1} + m_{a1}h_{a1}) - (m_{a2} \,.\, H_{s2} + m_{a2}h_{a2}) - m_c h_c.$$

As $\quad m_{a1} = m_{a2}$ and $m_{s1} = m_{s2} + m_c$

$$\therefore \quad Q_w = (m_{s1} \,.\, H_{s1} - m_{s2} . H_{s2} - m_c h_c) + m_{a1}(h_{a1} - h_{a2})$$

$$= (m_{s1} H_{s1} - m_{s2} H_{s2} - m_c h_c) + m_{a1} C_{p_a} (T_{a1} - T_{a2})$$

The first bracket of the right hand side of the above equation represents the net heat lost by the incoming steam and second term represents the heat lost by the incoming air.

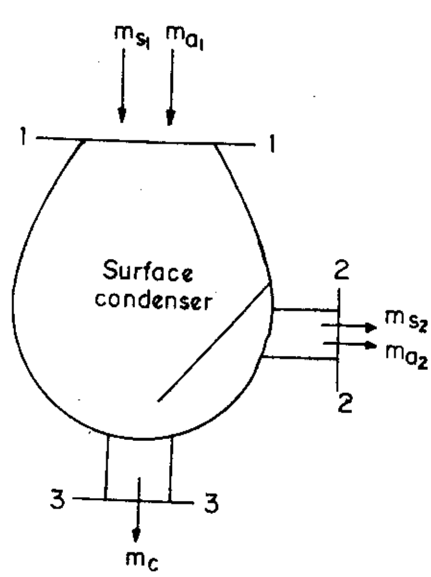


Fig. 17.7 (*a*).

Generally the heat lost by air is negligible compared with heat lost by steam and $m_{s2} << m_{s1}$ and therefore $m_{s1} = m_c$.

$\therefore$ For all practical purposes, we can write

$$Q_w = m_w C_{pw} (T_{wo} - T_{wi}) = (m_{s1} \, H_{s1} - m_c h_c) = m_c (H_{s1} - h_c)$$

as $m_{s1} = m_c$

$$\therefore \quad \frac{m_w}{m_c} = \frac{m_w}{m_s} = \frac{H_{s1} - h_c}{C_{pw}(T_{wo} - T_{wi})} = \frac{h_{w1} + x_2 h_{fg1} - h_c}{C_{pw}(T_{wo} - T_{wi})} \quad ...(17.8)$$

where T_{wi} and T_{wo} are the inlet and outlet temperatures of the cooling water and m_w is the mass of cooling water passing through the condenser.

The equation 17.8 gives the mass of cooling water required per kg of steam condensed.

17.9. DESIGN OF CONDENSERS

In designing the condenser, the designer has to find out quantity of circulating water required per kg of steam condensed and the surface area of the condenser. The design totally depends upon the type of the water available, the temperature of water available and arrangement of the condenser. A best design of a condenser is that which requires minimum quantity of cooling water at available temperature to condense one kg of steam.

Surface Condensers. The water flow for single pass condenser and temperature variations are represented in Fig. 17.7.

Considering there is no heat loss from the system to the surroundings, we can write

Heat lost by steam = Heat gained by water

$$Q = m_s\,(h + xh_{fg} - T_c) = m_w\,C_{pw}\,(T_{wc} - T_{wi})$$

$$\therefore \quad \frac{m_w}{m_s} = \frac{h + xh_{fg} - T_c}{C_{pw}\,(T_{wc} - T_{wi})} \quad ...(17.9)$$

The condenser vacuum to be maintained (or T_s) totally depends upon the temperature of cooling water (T_{wi}) available. The pressure in the condenser should be such that the saturation temperature of steam at condenser pressure should be greater than T_{wi} by 10°C to 15°C. The maximum value of T_{wo} is equal to T_s but in actual practice, it is not possible to allow an increase in water temperature to T_s even it reduces the quantity of water flow because it requires long condenser tubes and increases the cost and therefore it is not economically justified.

The total heat lost by steam is given to the water through the tubes of condensers, therefore it can be written as

$$Q = UA\theta \quad ...(17.10)$$

where *U(overall heat transfer coefficient)

$$= \frac{d_o}{d_i} \cdot \frac{1}{h_i} + \left(\frac{d_o - d_i}{d_o + d_i}\right)\frac{d_o}{K} + \frac{1}{h_o} \quad ...(17.11)$$

where d_i and d_o are inlet and outlet diameters of the condenser tubes and h_i and h_o are inside and outside heat transfer coefficients and K is the thermal conductivity of tube material

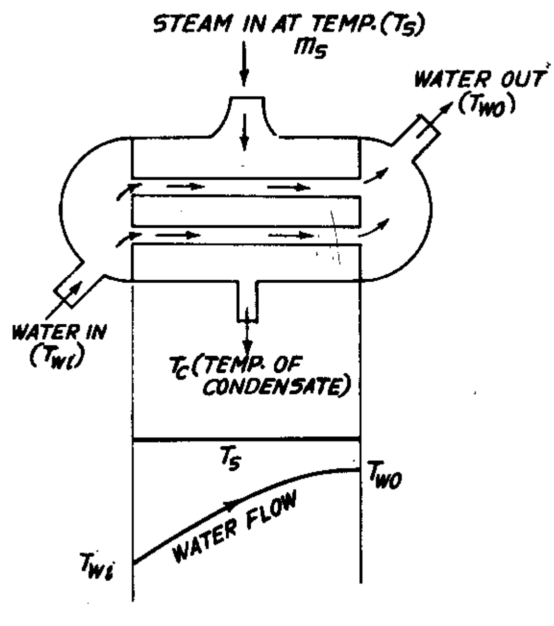


Fig. 17.7 (*b*). Temperature distribution in single pass condenser.

$$*\theta = \frac{(T_s - T_{wi}) - (T_s - T_{wo})}{\log_e\left(\dfrac{T_s - T_{wi}}{T_s - T_{wo}}\right)} \quad \frac{(T_{wo} - T_{wi})}{\log_e\left(\dfrac{T_s - T_{wi}}{T_s - T_{wo}}\right)} \quad ...(17.12)$$

A = Surface area through which heat is transferred from steam to water.

If the quantity of heat to be transferred (Q) is known and U and θ are known (as flow conditions and temperatures encountered are specified) then the surface area required to transfer the heat can be calculated.

The attainable heat transfer rate dictates the size and, therefore, the cost of the condenser. The size of the condenser for the given heat flow can be reduced by increasing the overall heat transfer coefficient. The overall heat transfer coefficient can be inceased by increasing the velocity of water through the tube. But any increase in velocity increases the pressure loss across the condenser and, therefore, a higher pumping cost. There is **optimum velocity which gives most economical heat transfer. The most economical water velocity generally lies between 1.5 m/sec to 2 m/sec.

The condensate temperature in a modern condenser should be within 1°C of the vacuum temperature. In order to achieve this standard of performance, the condensate which has formed on the top of the tubes,

*The students are advised to see the book on 'Heat Transfer' by the same authors for the detail derivation of the above expressions.

**For the proof of a optimum velocity, the readers are requested to see the book on 'Refrigeration and Air-Conditioning' by the same authors.

is deflected by means of baffles, to ensure, that the lower tubes do not have a cooling effect. Another means of achieving this design requirement is to lead a portion of the steam directly to the bottom of the condenser shell so that it warms up the condensate. This is a constructional problem than a design one.

The length of the tube and diameter are known then we can find the number of the tubes required by using the following expression

$$\pi d_o L . n = A_s$$

where n is the number of tubes.

In modern power plants, a single pass condenser is not desirable as length of the tubes required is considerably long which is impracticable for manufacture and construction of condenser becomes difficult. Therefore multipass condensers as shown in Fig. 17.8 are commonly used in practice.

The value of θ is given by the same expression as mentioned earlier. In this arrangement, the same quantity of water flows twice through first set of tubes and second set of tubes. Therefore the mass of water flow per second is given by

$$m_w = \frac{\pi}{4} (d_i)^2 \, v \, . \, \rho \times n \qquad ...(17.14)$$

where v = velocity of water through the tubes (m/sec)

ρ = density of water

n = number of tubes in first set.

The number of passes (p) required is given by

$$p = \frac{A_s}{na} \qquad ...(17.15)$$

where A_s is the total surface area required to transfer the heat and it can be calculated by using the expression.

$$Q = UA_s\theta \qquad ...(17.16)$$

where n = number of tubes per pass

a = surface area of each tube ($\pi d_o L$).

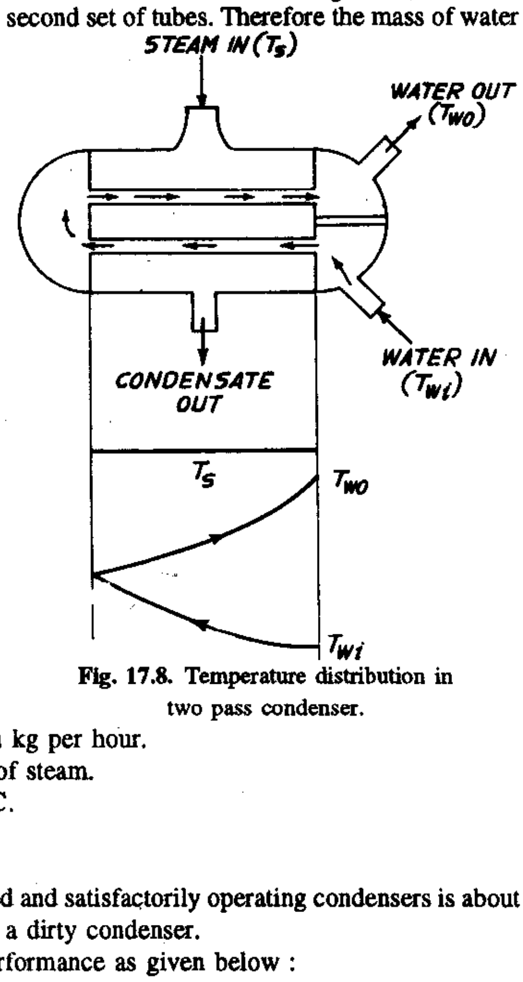


Fig. 17.8. Temperature distribution in two pass condenser.

Performance of Condenser. One authority (Evans) has suggested the following performance factor for condenser.

K (performance factor)

$$= \frac{m_s . H_s}{\theta_m A.V.} \qquad ...(17.17)$$

where m_s = mass of steam condensed in kg per hour.

H_s = heat removed in kJ per kg of steam.

θ_m = mean temp. difference in °C.

A_s = total surface area in m^2.

V = water velocity in m/sec.

The value of this performance factor for well designed and satisfactorily operating condensers is about 6300. The lower value than 6300 will generally indicate a dirty condenser.

Evans has also suggested another coefficient of performance as given below :

$$K = \frac{1800}{(T_s - T_{w_o})(T_s - T_{w_i} + 9)} \qquad ...(17.18)$$

This takes into accont water and steam temperatures which is an advantage. An operating engineer is able to determine when and where the plant is at fault just noting the temperatures mentioned in the above formula.

The condenser efficiency is given by

$$\eta_c = \frac{T_{wo} - T_{wi}}{T_s - T_{wi}}$$

17.10. CORROSION AND SCALE FORMATION IN CONDENSER TUBES AND THEIR PREVENTION

The efficiency of the condenser tubes depends upon maintenance of heat transfer surface cleanliness, adjustment of water flow for best economy and reduction of air-leakage to a minimum. The importance of water flow and prevention of air leakage are already discussed. The success of heat transfer with minimum power consumption for a long time mostly depends upon the cleanliness of the condenser tubes. The corrosion and scale formation are the common phenomenon in condenser tubes during operation due to the action of chemical compounds and deposits collected on the tube surfaces carried with the water. The life of the tubes is also reduced due to erosion which is the effect of abrasive materials (like sand) carried with cooling water.

In an average condenser installation on a river or lake, provision must be made for cleaning the condenser tubes. The fouling of tubes occurs because of algae, organic matter, leaves or other floating debris. Grills and screens remove most of the floating debris, even then small particles will eventually accumulate on the tubes and reduce the heat transfer. It is also desirabe to clean the condenser while it is under load. A single pass condenser during working condition can be cleaned by using back-washing. A valve arrangement is generally provided for back-washing purposes.

With most waters, there is general tendency for algae growth to build up on the tube surfaces. Algae growth is considerably more rapid under warm water conditions therefore summer periods are of the greatest trouble from this source in North American power plants. The algae often serves as a binder for mud or scale and if algae deposits are removed or controlled, other deposits are also minimised as well.

In closed type cooling system, where the cooling water is concentrated by evaporation, the possibility of scale formation is more if the water is not chemically treated.

Two general methods of treament are used for condenser tubes cleaning. First is the sterilization of the heat exchange surface of the condenser. This sterilization can be accomplished by a number of commercially available compounds as copper sulphate, chlorine, chlorinated phenols or mercurials.

The another method of tubes cleaning is 'Acid Cleaning' using an inhibited acid. This involves the handling of a large quantity of acid and the possibility of corroding tubes.

The condenser cleaning is costly and unpleasant when done by any means and it is generally more economical to prevent the deposits than to remove them after they are formed.

Another difficulty commonly experienced is the corrosion of condenser tubes and many times failures during operation. This is more undesirable and dangerous than scale formation, therefore every care should be taken to prevent the tubes from corrosion.

One of the causes of corrosion is the deposits of alkali and iron sulphates which slowly eats the tube metal and failure occurs. The corrosion is also accelerated with an increase in cooling water temperature particularly during summer. This is because the deposition rate increases with an increase in water-temperature level.

The basic reason of corrosion is the formation of sulphur trioxide in the water due to chemical reactions. The formed shlphur trioxide dissolves in water forming H_2SO_4 which is highly corrosive to majority of the metals. The sulphur trioxide also helps for forming the deposits on the tube surfaces.

A liquid organic base additive known as 'KRYDA' is developed by Dr.Bartoletti and Dr.Whicheer which prevents the formation of sulphur trioxide. The important property of this additive is that it deactivates the oxidation catalyst agent and hence the formation of sulphur trioxide will be controlled. With the addition of this additive, deposits and corrosion in high temperature zones are practically eliminated. The additive mentioned above also removes already present deposits in the form of dust.

The corrosion of metal tubes is reduced by proper selection of materials for tubes. If the cooling water is corrosive or has other undesirable properties, the use of resistive materials for tubes and a protective coating for piping and condenser water boxes is usually the most economical solution. Epoxy coatings applied to the surfaces of the water box, tube sheet and inlets of the tubes has proved to be a relatively inexpensible method to extend surface condenser life by minimizing corrosion/erosion action on metallic surfaces. These coatings, used in conjunction with a degree of cathodic protection to accommodate minute pinholes in the epoxy, are in good condition even after six years of condenser operation and are providing the required corrosion/erosion protection to the condenser metal surfaces. Intermittent chlorination of water may by used to prevent or retard the growth of algae and sea water crustaceans. The different types of materials used for condenser tubes to reduce the corrosion effect are discussed below.

17.11. MATERIALS FOR STEAM CONDENSERS

The applications of stainless steel tubing for surface condensers are approximately 30 years old. The major growth of this application has occurred in past two decades only. Originally these materials were only considered for highly corrosion environments or areas exposed to severe erosion. The cost of stainless steel tubes and available heat transfer data, a decade ago, restricted their use to the really difficult problems areas. Since that time, a number of important advances have been achieved which have permitted a more use of these materials for condenser applications.

The determinations of the overall heat transfer properties of stainless steel condenser tubes in the early 1960 led to more extensive use of these materials. The popular types are 304 (72% iron, 19% chromium, 9% nickel) and 316. 304 is used in cooling water environment with low chloride concentrations and 316 is used for sea water environments.

The inherent ability of the stainless steel to conduct heat from one surface to another is not very favourable in comparison with many other materials. If the thermal conductivity of the metal were the only criteria, there would be only few stainless steel applications. When one considers the sum of the resistances through which heat must pass from steam side to cooling water, the performance of the stainless steel becomes highly competitive with other materials. According to the Inco power conference held in 1964, the resistance of the stainless steel tube wall accounted only 2% of the total resistance as shown in Fig. 17.9.

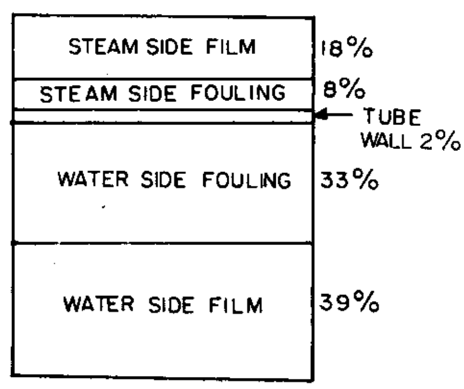


Fig. 17.9. Total thermal resistance of a condenser tube.

The heat transfer characteristics of stainless steel and Admiral brass are shown in Fig. 17.10.

In case of stainless steel tubes, the fouling is due to the formation of deposits from the cooling water only but the fouling of the brass is caused by deposits and corrosion of the inside tube surface also.

Recently aluminium bronzes for all of the cooling loop components from screen to condensers are developed by Ampco Co. They have proved their ability to resist erosion, corrosion, pitting and cavitation and stress corrosion cracking.

The stainless steels are not subjected to a general corrosion attack with the formation of a corrosion product and as a result, a good base on which deposits can form is not readily available. For this reason, the stainless steels generally remain clean in service and excellent performances have been reported.

The corrosion product which forms on brass tubes is frequently predictable based on water chemistry and known corrosion mechanism. This is not the case with deposits which form on stainless steel tubes. Many factors have been found to effect the type of desposits, among these are dissolved solids, pH value, chlorination. bacteria, and water velocity.

The overall corrosion resistance of stainless steel, 304 type is excellent for condenser tube service both the interior and exterior surface resist the formations of corrosion product which has an important influence on the heat transfer characteristics of tubes. It offers excellent erosion and corrosion resistance in fresh water, immunity to NH_3 and sulfide attack and the elimination of potentially troublesome copper ions in feed water. The stainless steels are not completely free from corrosion attack in condenser applications. Pitting

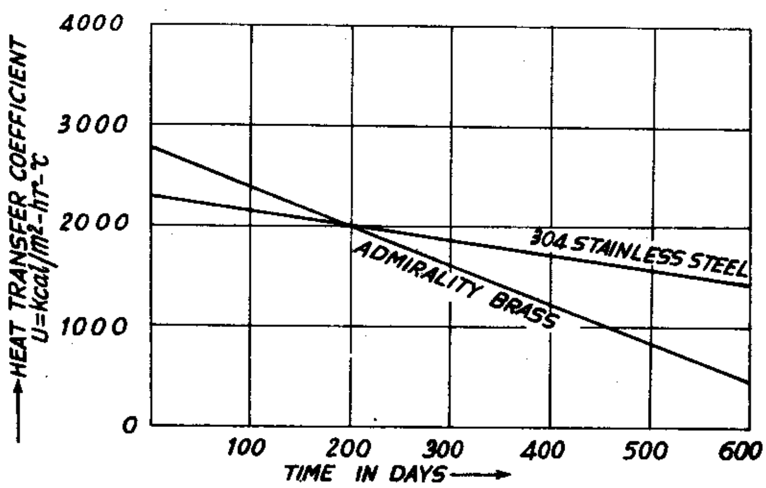


Fig. 17.10.

corrosion in stainless steel is usually attributed to chlorides. Tubes exposed to high chloride water are more sensitive to rapid and severe pitting when fouling occurs. The cost of the material is one of the important factors for its selection. The economical advantage of stainless steel condenser tubing cannot be attributed to the cost of material, and other factors like corrosion resistance, heat transfer and cleanliness of operation must also be considered. Stainless steel offers a distinct advantage over other materials.

The another promising material particularly for sea water is 90/10 copper/nickel alloy. It exhibits excellent corrosion resistance in fresh, salt and blackish water. It performs well at high water velocities (3 m/sec). It shows good immunity to stress corrosion cracking in both chloride and ammonia environments. It has high resistance to general corrosion and erosion and resists pitting attack well in all types of polluted water. The sulphides produced from naturally occurring sulphates by bacteria under an aerobic condition are specially aggressive to copper-bearing alloys.

Titanium gets the highest marks in virtually all areas of concern to condenser designers. It exhibits excellent resistance to corrosion, erosion and pitting as well as to deposits, chloride and ammonia induced attack and steam impingement. The major problem is its relatively high cost.

The thermal performance of condenser tubes is seriously affected by biofouling, particularly with sea water. Slime and algae accumulation on the tube surface, tube blockage by barnacles, mussels and sea weeds impair the thermal performance of the tubes. Warm water recirculation and chlorine dosing are the common methods used for preventing the biofouling.

SOLVED PROBLEMS

The following formulae are used for solving the problems :

1. Pressure equivalent of 1 cm of Hg = 0.01359 bar.
2. $\rho_a = \frac{1}{v_a}$ and $\rho_s = \frac{1}{v_s}$.
3. Vacuum efficiency $= \frac{p_b - (p_a + p_s)}{p_b - p_s} = \frac{p_b - p_t}{p_b - p_s}$.
4. Condenser efficiency $= \frac{T_{w_o} - T_{w_i}}{T_s - T_{w_i}}$

5. $\dfrac{m_a}{m_s} = \dfrac{v_s}{v_a}$.

6. $m = m_a + m_s = m_a\left(1 + \dfrac{v_a}{v_s}\right) = m_s\left(1 + \dfrac{v_s}{v_a}\right)$.

7. $\dfrac{m_w}{m_s} = \dfrac{h_{w1} + x_1 h_{fg1} - h_c}{C_{pw}\,(T_{w_o} - T_{w_i})}$ for surface condenser.

8. The volume delivered by vacuum pump

$$V = \frac{\pi}{4} D^2\, LN\,.\,\eta_v \text{ for single acting pump.}$$

9. Cooling Tower Approach = T_{wb_t} (incoming air) – T_{w_o}.

10. Cooling Tower range = Hot water temperature entering the cooling tower-cold water temperature leaving the cooling tower
= $(T_{wo} - T_{wi}$

11. Heat Transfer in cooling tower is given by an equation

$$m_{w1}\, C_{pw}\,(T_1 - T_2) = \frac{V}{v_{s1}}\,[(H_{a2} - H_{a1}) - (\omega_2 - \omega_1)\, C_{pw}\, T_2] \text{ where } T_1 > T_2$$

and where suffixes 1 and 2 represent the inlet and outlet conditions of water and air and v_{s1} is the specific volume of entering air.

12. $m_{w1} = \dfrac{m_s\,(\Delta H)}{(h_{w2} - h_{w1})}$.

13. Design of surface condenser $Q = UA\theta$.

where $$U = \frac{d_o}{d_i}\frac{1}{h_i} + \left(\frac{d_o - d_i}{d_o + d_i}\right)\frac{d_o}{K} + \frac{1}{h_o}$$

and $$\theta = \frac{\theta_i - \theta_o}{\log_e\left(\dfrac{\theta_i}{\theta_o}\right)}$$

and $A = \pi d_o L\,.\,n$.

For multipass-condenser

$$m_w = \frac{\pi}{4}\,(d_i)^2\,.\,v\,.\,\rho \times n$$

where n is the number of tubes in single pass.

$$p \text{ (Number of passes)} = \frac{A}{n\,.\,\pi d_o L}.$$

Problem 17.1. *The steam at 100 bar and 400°C is supplied to a steam-turbine. The isentropic efficiency of the turbine is 80%. The condenser pressure is 0.1 bar. If the plant capacity is 120 MW and the specific steam consumption is 4 kg/kWh ;*

(*a*) *determine the cooling water required per hour in the condenser assuming that there is no under-cooling in the condenser.*

(*b*) *and assuming that there is 5°C under-cooling in the condenser.*

Rise in temperature in cooling water is limited to 10°C in both cases.

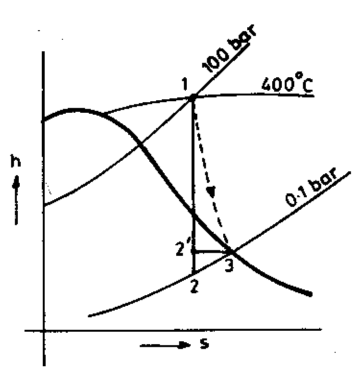


Fig. Prob. 17.1.

Sol. The point '1' is marked on $h - s$ chart and then point

'2' is located by drawing vertical line from '1' till it cuts to 0.1 bar pressure line.

The point 2′ is marked as 1 – 2′ = 80% of 1 – 2 then point 3 is marked drawing horizontal line through point 2′ till it cuts 0.1 bar pressure line.

The steam to be condensed in the condenser

$$= 4 \times (120 \times 10^3) \text{ kg/hr}$$

For condenser $= 4.8 \times 10^5$ kg/hr

heat gained by water = Heat lost by steam.

$$\therefore \quad m_w C_{pw} (T_{wo} - T_{wi}) = m_s (h_3 - h_{f3})$$

$$m_w \times 4.2 \times 10 = 4.8 \times 10^5 (1970 - 191.8) = 8.56 \times 10^8$$

where h_3 is taken from $h - s$ chart and h_{f3} is taken from steam table at 0.1 bar.

$$\therefore \quad m_w = \frac{8.56 \times 10^8}{4.2 \times 10} \text{ kg/hr} = \frac{8.56 \times 10^8}{4.2 \times 10 \times (1000 \times 60)} \text{ tons/min}$$

$$= \mathbf{396.7 \text{ tons/min}}$$

Problem 17.2. *The steam at 100 bar and 500°C is supplied to a steam turbine of 30 MW capacity. The vacuum maintained in the condenser is 71 cm of Hg when the barometer reads 76 cm of Hg. The work developed is equivalent to isentropic heat drop during expansion in the turbine. The steam coming out of the turbine is condensed in a surface condenser using river water for cooling. The rise in temperature of the cooling water is limited to 10°C. The temperature of the water at the inlet of the condenser is 20°C.*

Assuming the following :

Density of water = 1080 kg/m^3

Specific heat of water = 4.6 kJ/kg°C.

The condensate comes out of condenser as saturated liquid. Find (a) the mass of steam supplied to the turbine per hour ; (b) Capacity of the pump to circulate water ; and (c) the heat transfer area of the condenser if the overall heat transfer coefficient is 400 W/m^2*-°C.*

Neglect the heat losses in the system.

Sol. The pressure in the condenser

$= (76 - 71) \times 0.01359 = 0.068$ bar

Locate the point 1 and 2 as shown in Fig. Prob. 17.2 on H-s chart as 1—2 is isentropic expansion.

From chart $h_1 = 3389$ kJ/kg, and $h_2 = 2054$ kJ/kg.

If m_s is the mass of steam supplied to the turbine per hour, then

$$m_s (h_2 - h_1) = 30 \times 1000$$

$$\therefore \quad m_s = \frac{30 \times 1000}{(3389 - 2054)} = 22.5 \text{ kg/sec}$$

The dryness factor of steam (x_2) coming out of turbine and entering into condenser = 0.782 (from $h - s$ chart).

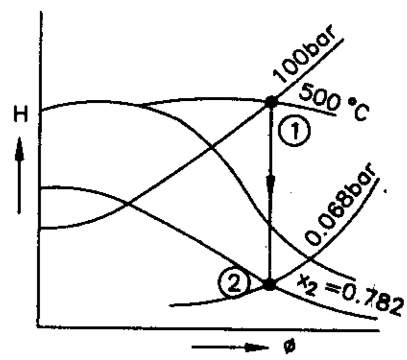


Fig. Prob. 17.2.

Heat lost by steam in condenser = Heat grained by water

$$\therefore \quad (h_2 - h_{f2}) m_s = m_w \cdot C_{pw} (T_{w_o} - T_{w_i})$$

$$\therefore \quad m_w = \frac{(2054 - 159.6) \times 22.5}{4.6 \times 10} = \mathbf{926.6 \text{ kg/sec.}}$$

$\therefore$ Capacity of the pump $= \mathbf{926.6 \text{ kg/sec}}$

$$Q = (h_2 - h_{f2})\, m_s = UA\theta.$$

where
$$\theta = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)} = \frac{(38.0 - 20) - (38.0 - 30)}{\log_e\left(\frac{18.0}{8.0}\right)} = \mathbf{12.35°C}$$

$\therefore$
$$A = \frac{(h_2 - h_{f2})\, m_s}{U \cdot \theta} = \frac{(2054 - 159.6) \times 926.6}{400 \times 12.35} = \mathbf{353.3\ m^2}.$$

Problem 17.3. *A steam turbine develops 3000 kW when the steam is supplied at 10 bar and 250°C. The vacuum in the condenser is maintained at 65 cm of Hg. The barometer reads 75.2 cm of Hg. The rise in temperature of cooling water is limited to 15°C. The temperature of the condensate coming out of condenser is 35°C.*

Neglecting all losses in the system and assuming the expansion through the turbine is isentropic, find

(*a*) *Specific steam consumption of the plant,*

(*b*) *Thermal efficiency of the plant, and*

(*c*) *Quantity of cooling water circulated through the condenser per hour.*

Assume there is no leakage in the condenser.

Sol. (*a*) The total pressure in the condenser

$$p_t = (75.2 - 65) \times 0.01359 = \mathbf{0.1359\ bar}$$

$$p_t = p_s = \mathbf{0.1359\ bar} \text{ as } p_a = 0.$$

The condition of steam can be calculated by using the same procedure as discussed in last problem

$\therefore$ $$x_2 = \mathbf{0.846}$$

The work done per kg of steam in the turbine assuming the expansion is isentropic

$$\Delta h = h_1 - h_2 = 2984 - 2234 = \mathbf{750\ kJ/kg}$$

The steam passing through the turbine is given by

$$m_s \times (\Delta h) = 3000$$

$\therefore$ $$m_s = \frac{3000}{750} = \mathbf{4\ kg/sec} = 14400 \text{ kg/hr}$$

$\therefore$ $$\text{Specific steam consumption} = \frac{14400}{3000} = \mathbf{4.8\ kg/kW\text{-}hr}$$

(*b*) Thermal efficiency of the plant

$$= \frac{\text{Isentropic work}}{h_1 - h_{f2}} = \frac{750}{2984 - 147} \times 100 = \mathbf{26.4\%}$$

(*c*) Cooling water supplied in tons per hour

$$m_w = \frac{m_s(h_2 - h_{f2})}{(T_{we} - T_{wi})\, C_{pw}} = \frac{14400(2234 - 147)}{4.2 \times 15 \times 1000}$$
$$= \mathbf{447\ tons/hr.}$$

Problem 17.4. *The following data collected from a steam turbine of 30 MW capacity thermal power plant.*

Steam condensed = 50,000 kg/hr.

Temperature of steam in condenser = 40°C.

Dryness of steam entering into condenser = 0.85.

The air leakage in the condenser = 150 kg/hr.

Temperature of the condensate = 35°C.

Temperature at the suction of the air pump = 32°C.

Barometer reads 76 cm of Hg = 1.013 bar

Find

(*a*) *Vacuum gauge reading in the condenser.*

(*b*) *Capacity of dry-air pump.*

(*c*) *Loss of steam in kg per hour.*

(*d*) *Quantity of cooling water passed through the condenser per hour if the rise in cooling water temperature is limited to 10°C.*

Sol. (*a*) The total pressure in the condenser is given by

$$p_t = p_s + p_a$$

where p_s (pressure at 40°C saturation temp.)

= 0.0752 bar (from steam tables).

v_s (specific volume of steam at 40°C saturation temp.)

= 19.5 m³/kg (from steam tables)

Volume of 50,000 kg of steam is given by

$$V = 50{,}000 \times xv_s = 50{,}000 \times 0.85 \times 19.5 = 828750 \text{ m}^3$$

The same volume is also occupied by 150 kg of air, therefore, the pressure of air in the condenser is given by

$$p_a = \frac{m_a RT}{V}$$

$$= \frac{150 \times 287 \times (40 + 273)}{828750 \times 10^5} = \mathbf{0.000163 \text{ bar}}$$

$$\therefore \quad p_t = 0.0752 + 0.000163 = \mathbf{0.07536 \text{ bar}}$$

∴ Vacuum in condenser in cm of Hg.

$$= \frac{(1.013 - 0.07536)}{0.01359} = \mathbf{68.95 \text{ cm of Hg.}}$$

(*b*) Partial pressure of steam at air-pump suction is the saturation pressure of steam corresponding to 32°C

= **0.0485 bar** (from steam tables)

$$\therefore \quad P_{a1} = p_t - P_{s1} = 0.07536 - 0.04850 = \mathbf{0.02686 \text{ bar}}$$

Volume of air at 32°C and 0.02686 bar is given by

$$V_1 = \frac{m_a RT}{p_{a1}} = \frac{150 \times 287 \times (32 + 273)}{0.02686 \times 10^5} = \mathbf{4848.4 \text{ m}^3/\text{hr}}$$

∴ Air pump capacity = **4848.4 m³/hr.**

(*c*) v_{s1} (specific volume of steam at 32°C saturation temperature)

= 29.6 m³/kg (from steam tables).

∴ Loss of steam in kg per hour

$$= \frac{V_1}{v_{s1}} = \frac{4848.4}{29.6} = \mathbf{163.8 \text{ kg/hr.}}$$

(*d*)
$$m_w = \frac{m_s(h_1 - h_{f2})}{C_{pw}(T_{w_o} - T_{w_i})}$$

where h_1 (heat of steam at 40°C saturation temp. and 0.85 dry)

$$= h_{f1} + x_1 h_{fg1} = 168 + 0.85 \times 2414 = \mathbf{2219.7 \text{ kJ/kg.}}$$

$$\therefore \quad m_w = \frac{50{,}000(2219.7 - 147)}{4.2 \times 10 \times 1000} \text{ tons/hr.} = \frac{50{,}000 \times 493}{4.2 \times 1000 \times 10} = \mathbf{587 \text{ tons/hr.}}$$

Problem 17.5. *In a small capacity of thermal power plant, the condenser is provided with a separate air-cooling section. The temperature of steam entering the condenser is 56°C and temperature at the air-pump suction is 46°C. The barometer reads 76 cm of Hg.*

Find (a) The vacuum in the condenser.

(b) If the discharge of dry air-pump is 90 m³/min, find the air-leakage in condenser in kg/hr.

(c) Loss of condensate due to the water vapour carried with air through air-pump.

Sol. (*a*) p_s (pressure of steam at saturation temperature of 56°C).

$$= 0.1684 \text{ bar (from steam tables)}$$

$$= \frac{0.1684}{0.01359} = \mathbf{12.4 \text{ cm of Hg.}}$$

p_a (partial pressure air at the inlet of condenser) = 0.

$$\therefore \quad p_t = p_s = \mathbf{12.4 \text{ cm of Hg.}}$$

∴ Vacuum in the condenser

$$= 76 - 12.4 = \mathbf{63.5 \text{ cm of hg.}}$$

(*b*) p_s (pressure of steam at saturation temperature of 46°C)

$$= \mathbf{0.1028 \text{ bar}} \text{ (from steam tables)}$$

As the total pressure in the condenser remains same

∴ p_a (air pressure near air suction pump)

$$= p_t - p_s = 0.1684 - 0.1028 = \mathbf{0.0656 \text{ bar}}$$

The air leakage in the condenser per hour is given by

$$m_a = \frac{p_a V}{RT} = \frac{0.0656 \times 10^5 \times (90 \times 60)}{287 \times (46 + 273)} = \mathbf{386.9 \text{ kg/hr.}}$$

(*c*) Loss of condensate per hour

$$= \frac{V}{v_s} = \frac{90 \times 60}{14.56} = \mathbf{371 \text{ kg/hr.}}$$

where v_s is the specific volume of steam at 46°C saturation temperature (from steam table).

Problem 17.6. *The air leakage into a surface condenser operating with a steam turbine is estimated 84 kg/hr. The vacuum near the inlet of air-pump is 70 cm of Hg when barometer reads 76 cm of Hg. The temperature at the inlet of vacuum pump is 20°C. Find :*

(a) Minimum capacity of the air pump in cu-metres per hour.

(b) Find the dimensions of the reciprocating air-pump to remove the air if it runs at 200 r.p.m.

Take L : D = 3 : 2 and volumetric efficiency = 80%.

(c) Also find the mass of vapour extracted per minute with air.

Sol. (*a*) The total pressure at the point (*A*) is given by

$$p_t = \frac{(76 - 70)}{76} \times 1.013 = \mathbf{0.08 \text{ bar.}}$$

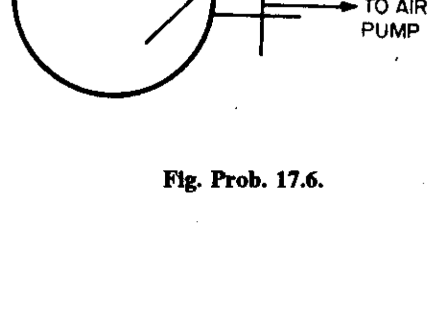


Fig. Prob. 17.6.

The partial pressure of water vapour at point (*A*) is the pressure corresponding a saturation temperature of 20°C.

$$p_s = \mathbf{0.0238} \text{ bar (from steam table)}$$

∴ Partial pressure of air at point (*A*) is given by

$$p_a = p_t - p_s = 0.08 - 0.0238 = \mathbf{0.0562 \text{ bar.}}$$

Total volume of 84 kg of air at 20°C and 0.0562 bar is given by

$$V = \frac{m_a RT}{p} = \frac{84 \times 287 \times (20 + 273)}{0.0562 \times 10^5} = \mathbf{1256.8 \text{ m}^3\text{/hr.}}$$

∴ Capacity of air pump = **1256.8 m³/hr.**

(*b*) $$V = \frac{\pi}{4} D^2 L \times N \times \eta_v$$

where *D* and *L* are diameter and stroke of air pump.

N is the r.p.m. of the pump and η_v is the volumetric efficiency

$$\therefore \quad \frac{1256.8}{60} = \frac{\pi}{4}\left(\frac{D}{100}\right)^2 \times \frac{1.5\,D}{100} \times 200 \times 0.8$$

where *D* is in cm

$$\therefore \quad D^3 = \frac{1256.8 \times 4 \times 10^6}{60 \times 1.5\pi \times 200 \times 0.8} = 88945$$

$$\therefore \quad D = \mathbf{43 \text{ cm}}$$

$$\therefore \quad L = 1.5\,D = 1.5 \times 43 = \mathbf{64.5 \text{ cm.}}$$

(*c*) Mass of water vapour going with air in the air pump per hour

$$= \frac{V}{v_s}$$

where v_s is the specific volume of saturated steam at 20°C (taken from steam tables).

$$\therefore \quad m_s = \frac{1256.8}{57.83} = \mathbf{21.73 \text{ kg/hr.}}$$

Problem 17.7. *The steam condensed in a surface condenser is 12500 kg per hour and the amount of air leakage is 5 kg/hr. The vacuum near the suction pump is 70 cm of Hg and temperature is 34°C. The air and condensate is removed by a wet air-pump. Find the capacity of wet air pump if the volumetric efficiency of the pump is 80%.*

Take barometric pressure = 76 cm of Hg.

(*b*) *If the air-pump runs at 100 r.p.m., find the dimensions of pump required.*

Take L = 1.5 D and assume pump is single acting.

Sol. p_s (pressure of steam corresponding to 34°C saturation temperature)

= 0.0542 bar (from steam tables).

p_t (pressure in condenser) = (76 – 70) × 0.01359 = **0.0815** bar.

p_a(partial pressure of air) = $p_t - p_s$ = 0.0815 – 0.0542 = **0.0273** bar

Volume of air in the condenser per hour is given by

$$V = \frac{m_a RT}{p_a} = \frac{5 \times 287 \times (34 + 273)}{0.0273 \times 10^5} = 161.4 \text{ m}^3\text{/hr} = \mathbf{2.69 \text{ m}^3\text{/min.}}$$

Volume of condensate formed per minute

$$= \frac{12500}{60 \times 1000} = \mathbf{0.2083 \text{ m}^3\text{/min.}}$$

Total volume of air and condensate removed per minute by the pump

$$= 2.69 + 0.2083 = \mathbf{2.898\ m^3/min.}$$

$$V = \frac{\pi}{4} D^2 \times L \times \eta_v \times N$$

$$\therefore \quad \frac{\pi}{4}\left(\frac{D}{100}\right)^2 \times \frac{1.5D}{100} \times 0.8 \times 100 = 2.898 \text{ where } D \text{ is cm}$$

$$\therefore \quad D^3 = 30.74 \times 10^3$$

$$\therefore \quad D = \mathbf{31.3\ cm} \text{ and } L = 1.5\ D = 1.5 \times 31.3 = \mathbf{47\ cm.}$$

Problem 17.8. *The temperature of steam entering the surface condenser is 38°C. The temperature of air entering the air pump is 34°C and temperature of condensate is 36°C. The air and condensate are removed by separate pumps. Find the capacity of the air pump required if the air leakage is 3 kg per hour. The air before entering into the air pump is cooled by a screened-off section of condenser.*

(*b*) *Find the percentage change in air pump capacity and loss of water vapour carried with air if the air and condensate are removed by a single pump.*

Assume there is no air leakage at the entry of the condenser and vacuum throughout the condenser is same. The condensate removed per hour is 8000 kg/hr.

Sol. (*a*) p_s (pressure of steam corresponding to 38°C temperature of saturated steam).

$$= \mathbf{0.0676\ bar} \text{ (from steam table.)}$$

$$p_a = \mathbf{0.0} \text{ at the entry of condenser as given in problem}$$

$$\therefore \quad p_t = \mathbf{0.0676\ bar.}$$

This pressure (0.0676 bar) is constant throughout the condenser.

p_s (pressure of steam corresponding to 34°C (temp. of saturated steam).

$$= \mathbf{0.0542\ bar} \text{ (from steam table)}$$

$\therefore$ p_a (partial pressure of air at the entry of air pump)

$$= p_t - p_s = 0.0676 - 0.0542 = \mathbf{0.0134\ bar.}$$

The volume of 3 kg of air at 34°C and 0.0134 bar pressure is given by

$$V_1 = \frac{m_a RT}{p_a} = \frac{3 \times 287 \times (34 + 273)}{0.0134 \times 10^5} = \mathbf{197.3\ m^3/hr.}$$

(*b*) If the air and condensate are extracted by a common pump then the temperature of air going out will be 36°C.

p_s (pressure of steam corresponding to 36°C temperature of saturated steam)

$$= \mathbf{0.0606\ bar} \text{ (from steam table)}$$

$$p_a = p_t - p_s = 0.0676 - 0.0606 = \mathbf{0.007\ bar.}$$

The volume of 3 kg of air at 36°C and 0.007 bar pressure is given by

$$V_2 = \frac{m_a RT}{p_a} = \frac{3 \times 287(36 + 273)}{0.007 \times 10^5} = \mathbf{380\ m^3/hr.}$$

Volume of 8000 kg of condensate at 36°C

$$= 8000 \times 0.001006 = \mathbf{8\ m^3/hr.}$$

Total volume removed by wet air pump

$$= 380 + 8 = \mathbf{388\ m^3/hr}$$

$\therefore$ Percentage increase in air-pump capacity

$$= \frac{388 - 201}{201} \times 100 = \mathbf{97\%.}$$

Mass of water vapour carried with air when dry air-pump is used to remove the air.

$$= \frac{V_1}{v_{s_1}} \text{ where } v_{s_1} \text{ is specified volume of steam at 34°C saturated steam temperature.}$$

$$= \frac{201}{26.5} = \mathbf{7.55\ kg/hr.}$$

Mass of water vapour carried with air when wet air pump is used to remove the air

$$= \frac{V_2}{v_{s_2}} \text{ where } v_{s_2} \text{ is the specific volume of steam at 36°C saturated steam temperature.}$$

$$= \frac{388}{24} = 16.20 \text{ kg/hr}$$

∴ Percentage increase in loss of water vapour.

$$\therefore \quad \frac{16.20 - 7.55}{7.55} \times 100 = \mathbf{114.5\%}.$$

Problem 17.9. *A steam turbine of 12,500 kW capacity requires 5 kg of steam per hour per kW. The quantity of air leakage into the condenser is 1 kg per 1000 kg of steam used by turbine. The vacuum in the condenser is 70 cm of Hg when the barometer reads 76 cm of Hg. The temperature at the suction of the air-pump is 30°C. The surface condenser used is fitted with a separate condensate pump and air-pump. Find,*

(*a*) *The capacity of the air-pump required per minute.*

(*b*) *The mass of water vapour carried by air in kg per hour.*

(*c*) *The quantity of cooling water required per minute in tons if the rise in temperature of the water is limited to 8°C. The quality of steam entering the condenser is 0.9 dry and there is no under-cooling in the condenser.*

Sol. (*a*) The pressure in the condenser is given by

$$p_t = p_a + p_s = \left(\frac{76 - 70}{76}\right) \times 1.013 = 0.08 \text{ bar}$$

p_s (partial pressure of steam at 30°C)

$$= 0.04325 \text{ bar (from steam tables)}$$

∴ p_a (partial pressure of air)

$$= 0.08 - 0.04325 = \mathbf{0.03675\ bar}$$

Air leakage into the condenser per minute

$$= 20{,}000 \times 5 \times \frac{1}{1000} \times \frac{1}{60} = \mathbf{1.67\ kg/min.}$$

Volume of 1.67 kg of air at a temperature of 30°C and pressure of 0.03675 bar.

$$V = \frac{m_a RT}{p_a} = \frac{1.67 \times 287 \times (30 + 273)}{0.03675 \times 10^5} = \mathbf{39.5\ m^3}.$$

∴ Capacity of air pump = 39.5 m^3/min.

(*b*) The mass of water vapour carried with air in kg per hour is given by

$$m_s = \frac{V}{v_s}$$

where v_s (specific volume of saturated steam at 30°C)

$$= 32.8 \text{ m}^3\text{/kg (from steam table)}$$

$\therefore \quad m_s = \dfrac{39.5 \times 60}{32.8}$ kg/hr = **72.25 kg/hr.**

(c) The quantity of cooling water required per minute in kg is given by

$$m_w = \frac{m_s\,[h_{f1} + x_1 h_{fg1} - h_c]}{C_{pw}(T_{wo} - T_{wi})}$$

If thère is no under-cooling, then,

$$h_c = h_{f1}$$

$$\therefore \quad m_w = \frac{m_s \times x_1 h_{fg1}}{C_{pw}(T_{w_o} - T_{w_i})}$$

$$m_s = \frac{20{,}000 \times 5}{60} \text{ kg/min. (given)}$$

$$x_1 = \mathbf{0.9} \text{ given}$$

$$h_{fg1} = 2438 \text{ kJ/kg at 30°C Saturated temperature (from steam tables)}$$

and $\quad T_{w_o} - T_{w_i} = 8°C$ given.

$$\therefore \quad m_w = \frac{20{,}000 \times 5}{60} \times \frac{0.9 \times 2438}{4.2 \times 8} \text{ kg/min.}$$

$$= \frac{20{,}000 \times 5}{60} \times \frac{0.9 \times 2438}{4.2 \times 8} \times \frac{1}{1000} \text{ tons/min.} = \mathbf{108.8 \text{ tons/min.}}$$

Problem 17.10. *The condition of steam entering into a jet condenser is 0.9 dry. It is condensed by using the water at 15°C. The mass of air in the condenser is 30% of the mass of air and water vapour in the condenser. Assuming that the flow of water is adjusted in such a way that only latent heat of steam is absorbed by the cooling water, find :*

(a) The temperature of mixture of condensate and water leaving the condenser.

(b) The mass of water required per kg of steam condensed.

Take R (for air) = 287 Nm/kg-K.

R (for steam) = 462.8 Nm/kg-K.

Condenser vacuum = 61.3 cm of Hg.

Barometer Reading = 76 cm of Hg.

Sol. According to Delton's law of partial pressure, the pressure exerted by each constituent is directly proportional to the percentage mass of the consequent in the mixture when the constituent temperature is same as mixture temperature.

$$\therefore \quad p_a = 0.3\, p_t \text{ as given in problem}$$

$$p_t = (76 - 61.3) \times 0.01359 = \mathbf{0.2 \text{ bar}}$$

$$\therefore \quad p_a = 0.3 \times 0.2 = \mathbf{0.06 \text{ bar}}$$

$$\therefore \quad p_s = p_t - p_a = 0.20 - 0.06 = \mathbf{0.14 \text{ bar}}$$

T_s (saturation temperature of steam at 0.14 bar pressure)

$$\approx \mathbf{52°C} \text{ (from steam tables).}$$

As only latent heat of steam is to be removed and there is no under-cooling of the condensate, the outlet temperature of the cooling water must be also equal to 52°C.

$\therefore$ Mixture temperature coming out of condenser = 52°C.

For ideal jet condenser, when there is no under-cooling of condensate, the quantity of cooling water required is given by

$$\frac{m_w}{m_s} = \frac{h_{f1} + x_1 h_{fg1} - C_{pw} \times T_{w_o}}{C_{pw}(T_{w_o} - T_{w_i})}$$

$$T_{w_o} = T_s = 52°C$$

as required condition for no under-cooling of condensate.

The value of h_{f1} and h_{fg1} should be taken from steam table at steam pressure of 0.14 bar.

$$\therefore \quad \frac{m_w}{m_s} = \frac{218.4 + 0.9 \times 2381.4 - 4.2 \times 52}{4.2[52 - 15]} = \frac{0.9 \times 2381.4}{37} = \mathbf{13.8}$$

∴ Minimum quantity of cooling water required per kg of steam condensed neglecting heat given to air is **13.8 kg.**

Problem 17.11. *Find the surface required in a surface condenser dealing with 30 × 10³ kg of steam per hour 90% dry at 0.04 bar absolute pressure. The cooling water enters the condenser at 15°C and leaves at 25°C. Assume the overall heat transfer coefficient 3000 W/m²-°C.*

If this condenser has two water passes composed of tubes of 2 cm outside diameter and 1.2 mm thick with water speed of 1.5 m/sec determine the number of tubes in each pass and the length of each tube.

Sol. The saturation temperature of steam at 0.04 bar is 28.6°C. (from steam tables).

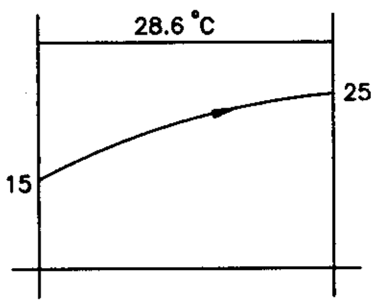


Fig. Prob. 17.11 (*a*).

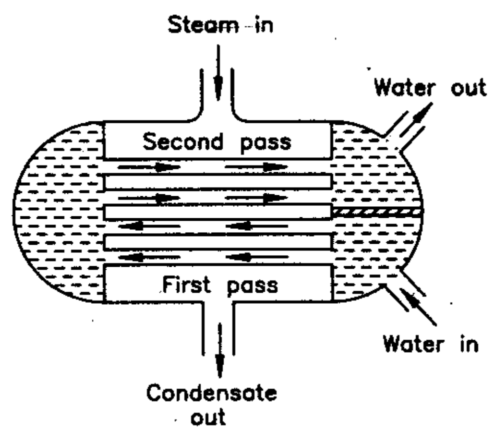


Fig. Prob. 17.11.

Total heat lost by steam per hour = Total heat gained by water

$$\therefore \quad 30 \times 10^3 \times 2440 \times 0.9 = m_w \times 4.2\,(25 - 15)$$

$$\therefore \quad \mathbf{m_w = 1.57 \times 10^6 \ kg/hour.}$$

The *LMTD* can be calculated by using the following expression.

$$LMTD = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)} = \frac{(28.6 - 15) - (28.6 - 25)}{\log_e\left(\frac{28.6 - 15}{28.6 - 25}\right)} = \mathbf{7.52°C.}$$

The surface area required can be calculated by using the following equation

$$Q = AU\,(LMTD)$$

$$\therefore \quad 1.57 \times 10^6 \times \frac{2440 \times 0.9}{3600} = A \times 3000 \times 7.52$$

$$\therefore \quad \mathbf{A = 424.5 \ m^2}$$

The arrangement of two passes is shown in Fig. Prob. 17.11.

Assuming the number of tubes in one pass is '*n*' we can write down the following equation

$$\frac{\pi}{4}(d_i)^2 \times v \times \rho \times n \times 3600 = 1.57 \times 10^6$$

$$d_i = 20 - 2.4 = 17.6 \text{ mm}$$

$$\rho \text{ (density of water) at } \left(\frac{15 + 25}{2}\right) = 20°\text{C} = 998.2 \text{ kg/m}^3$$

$$\therefore \quad \frac{\pi}{4}\left(\frac{1.76}{100}\right)^2 \times 1.5 \times 998.2 \times n \times 3600 = 1.57 \times 10^6$$

$$\therefore \quad n = \mathbf{1198}$$

Total number of tubes in both passes = $2n$ = 2396

The length of each tube can be calculated as follows

$$\pi d_o L \times 2n = 424.5$$

$$\pi\left(\frac{2}{100}\right) \times L \times 2396 = 424.5$$

$$\therefore \quad L = \mathbf{2.83 \text{ metres.}}$$

Problem 17.12. *A single pass surface condenser of a small power plant condenses 10 tons of saturated steam at 0.2 bar per hour. The rise in the water temperature is limited to 20°C and temperature difference between the steam and water at outlet is 10°C. Taking overall heat transfer coefficient = 4 kW/m²-°C, find out the area of the condenser required.*

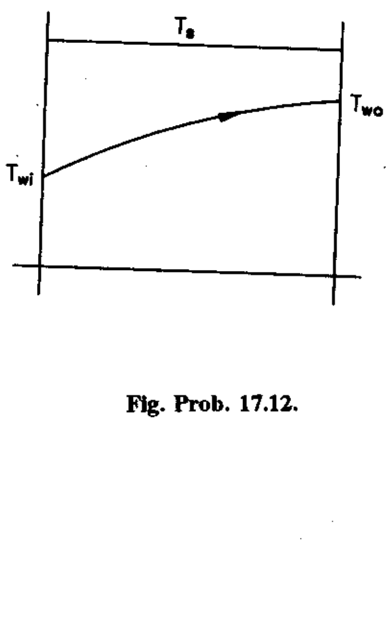


Fig. Prob. 17.12.

Sol. The heat lost by the steam in the condenser (Q) is given by

$$Q = m_s \,.\, h_{fg}$$

where h_{fg} is latent heat of steam at 0.2 bar pressure

$$\therefore \quad Q = \frac{10 \times 1000}{3600} \times 2358.3 = \mathbf{6550.8 \text{ kW}}$$

The Q is transferred through the tubes to the water and it is also given by

$$Q = UA\ (LMTD) \quad \text{...(a)}$$

where
$$LMTD = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)}$$

The given data is

$$T_s - T_{wo} = 10°\text{C and } T_{wo} - T_{wi} = 20°\text{C}$$

Using the above two equations, we get

$$\theta_i = T_s - T_{wi} = 30 \text{ and } \theta_o = T_s - T_{wo} = 10$$

$$\therefore \quad LMTD = \frac{30 - 10}{\log_e (3)} = \mathbf{18.2°C}$$

Now subsituting all the values in equation (a)

$$6550.8 = 4 \times A \times 18.2$$

$$\therefore \quad A = \frac{6550.8}{4 \times 18.2} = \mathbf{90 \text{ m}^2}$$

Problem 17.13. *A single pass surface condenser is designed to condense 5000 kg dry saturated steam at 50°C per hour. The ID = 15 mm and OD = 18 mm of the tubes used in condenser. The condensed water is available at 20°C and its rise in temperature is limited to 10°C. The velocity of the water is limited to 2 m/sec.*

Taking the following data, calculate the number of tubes required if its length is limited to 3 m
h_o (steam side) = 5000 J/m^2-s-°C
h_i (water side) = 3200 J/m^2-s-°C
f_i (water side fouling water) = 0.0002 m^2-°C/W
K (conductivity of tube material) = 80 W/m-°C. Assume there is no under-cooling of the condensate.

Sol. Q (heat lost by steam)

$$= m_s\, h_{fg} \text{ (at 50°C saturated temperature)}$$

$$= \frac{5000}{3600} \times 2383 = 3310 \text{ kJ/sec} = \mathbf{3310\ kW}$$

The Q is also given by

$$Q = U_o A\ (LMTD) \qquad \ldots(a)$$

$$LMTD = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)} = \frac{30 - 20}{\log_e\left(\frac{30}{20}\right)} = \mathbf{24.7°C}$$

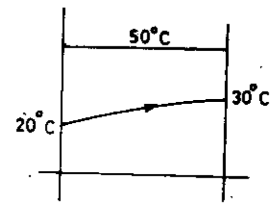


Fig. Prob. 17.13

$$\frac{1}{U_o} = \left(\frac{1}{h_i} + f_i\right)\frac{d_o}{d_i} + \frac{1}{h_o} + \frac{d_o - d_i}{d_o + d_i}\left(\frac{d_o}{k}\right)$$

where f_i is inside fouling factor

$$\frac{1}{U_o} = \left(\frac{1}{3200} + 0.0002\right) \cdot \frac{18}{15} + \frac{1}{5000} + \frac{3}{33} \times \frac{18}{1000 \times 80}$$

$$= 6.15 \times 10^{-4} + 2 \times 10^{-4} + 0.2 \times 10^{-4} = 8.35 \times 10^{-4}$$

$$\therefore \quad U_o = \mathbf{1197.6\ W/m^2\text{-}°C}$$

Now substituting the values in equation (a)

$$3310 \times 10^3 = 1197.6 \times A \times 24.7$$

$$\therefore \quad A = \frac{3310 \times 10^3}{1197.6 \times 24.7} = \mathbf{111.9\ m^2}$$

If there are n number of tubes, then

$$\pi d_o L \times n = A = 111.9$$

$$\therefore \quad \pi \times \frac{18}{1000} \times 3 \times n = 111.9$$

$$\therefore \quad n = \frac{111.9 \times 1000}{\pi \times 18 \times 3} \approx \mathbf{660}$$

Problem 17.14. *A surface steam condenser is to be designed for 120 MW plant when the steam is supplied at 150 bar and 600°C and condenser pressure is 0.08 bar. The water side and steam side heat transfer coefficients are 1000 W/m^2-°C and 5000 W/m^2-°C respectively. The inlet and outlet temperatures of water are 25°C and 35°C. The condensate is saturated water. Determine the number of tubes required if ID = 2.5 cm and OD = 2.9 cm and length of the tube is limited to 5 m. Arrangement of the condenser is single-pass shell type. Assume there are no losses and expansion in the turbine is isentropic.*

Sol. The isentropic expansion of steam through turbine is shown in Fig. Prob. 17.14 (b) on $h - s$ chart and arrangement of the condenser is shown in Fig. Prob. 17.14 (a).

The condensing temperature = Saturation temperature of steam at 0.08 bar
= **41.5°C** (from steam table)

The points 1 and 2 are marked as shown and enthalpies are noted

$$h_1 = 3580 \text{ kJ/kg}, \ h_2 = 2080 \text{ kJ/kg}$$
$$h_{f2} = 174 \text{ kJ/kg (from steam table)}$$

If m_s is the mass of steam flowing through the turbine per second, then we can write

$$m_s\,(h_2 - h_1) = 120 \times 1000$$

$$\therefore \quad m_s = \frac{120 \times 1000}{3580 - 2080} = \mathbf{80\ kg/sec}$$

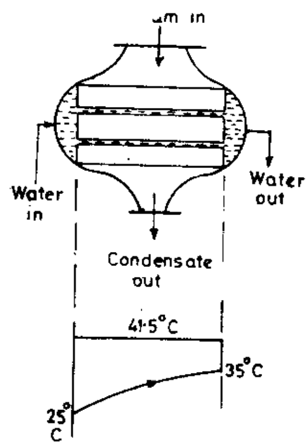


Fig. Prob. 17.14 (*a*)

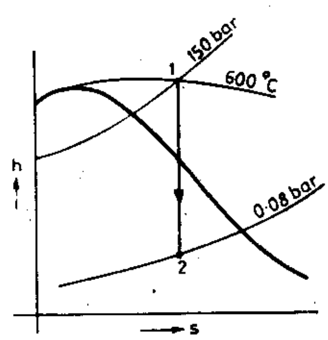


Fig. Prob. 17.14 (*b*)

The heat lost in the condenser by the steam is given by

$$Q = m_s\,(h_2 - h_{f2})$$
$$= 80\,(2080 - 174) = \mathbf{80 \times 1906\ kJ/sec}$$

The heat flow is also given by

$$Q = U_o A\,(LMTD) \qquad \ldots(a)$$

where U_o is given by

$$\frac{1}{U_o} = \frac{1}{h_i} \cdot \frac{d_o}{d_i} + \frac{1}{h_o}$$

where U_o is overall heat transfer coefficient referred to outer surface of the tubes

$$= \frac{1}{1000} \cdot \frac{2.9}{2.5} + \frac{1}{5000} = .00116 + 0.0002 = 0.00136$$

$$\therefore \quad U_o = \mathbf{735.3\ W/m^2\text{-}^\circ C}$$

$$LMTD = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)} = \frac{(41.5 - 25) - (41.5 - 35)}{\log_e\left(\frac{41.5 - 25}{41.5 - 30}\right)} = \mathbf{13^\circ C}$$

Now substituting all the values in equation (*a*)

$$80 \times 1906 = 735.3 \times A_s \times 13$$

$$\therefore \quad A_s = \frac{80 \times 1906}{735.3 \times 13} = \mathbf{15.95\ m^2}$$

The A_s is given by

$$A_s = (\pi d_o L) \times n \text{ where } n \text{ is number of tubes}$$

$$\therefore \quad n = \frac{15.95}{\left(\pi \times \frac{2.9}{100} \times 5\right)} = \mathbf{35\ tubes}$$

Problem 17.15. *A surface type steam condenser is designed to condense 300 tons of steam per hour at 0.04 bar pressure and 0.9 dry. The cooling water inlet and outlet temperatures are 15°C and 25°C respectively. The overall heat transfer coefficient in the condenser = 3 kJ/m²-°C.*

The condenser has two water passes and tubes used are d_i = 17.6 mm and d_o = 20 mm. Determine the number of tubes used in one pass and length of each tube if the water speed in the condenser tube is limited to 2.5 m/sec.

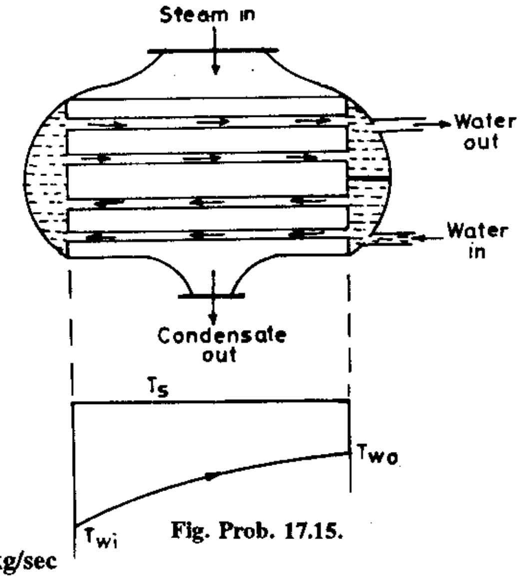


Fig. Prob. 17.15.

Sol. The condenser arrangement and temperature distribution is shown in Fig. Prob. 17.15.

The given data is

$$T_{wi} = 15°C,\ T_{wo} = 25°C$$

T_s = 28.6°C (from steam table at 0.04 bar)

h_{fg} (at 0.04 bar)

= 2433 kJ/kg (from steam table)

Q = Heat lost by steam = Heat gained by water

$$\therefore \quad m_s h_{fg} x = m_w C_{pw} (T_{wo} - T_{wi})$$

$$\left(\frac{300 \times 1000}{3600}\right) \times 2433 \times 0.9 = m_w \times 4.2(25 - 15)$$

$$\therefore \quad m_w = \frac{300 \times 1000}{3600} \times 2433 \times 0.9 \times \frac{1}{4.2 \times 10}$$

$$= \mathbf{4.35 \times 10^3\ kg/sec}$$

The Q is also given by

$$Q = UA\ (LMTD) \qquad ...(a)$$

where *LMTD* is given by

$$LMTD = \frac{\theta_i - \theta_o}{\log_e\left(\frac{\theta_i}{\theta_o}\right)} = \frac{(28.6 - 15) - (28.6 - 25)}{\log_e\left(\frac{28.6 - 15}{28.6 - 25}\right)} = \mathbf{7.5°C}$$

Substituting the values in equation (*a*)

$$Q = m_s\ (h_{fg}.x) = \text{UA}\ (LMTD)$$

$$\therefore \quad \frac{300 \times 1000}{3600} \times 2433.9 = 3 \times A\ (7.5)$$

$$\therefore \quad \text{A} = \mathbf{9015\ m^2}$$

Assuming these are '*n*' tubes in each pass, we can write

$$\frac{\pi}{4}(d_i)^2\ v\rho \times n = m_w = 4.35 \times 10^3$$

$$\frac{\pi}{4}\left(\frac{17.6}{1000}\right)^2 \times 2.5 \times 1000 \times n = 4.35 \times 10^3$$

$$\therefore \quad n = \frac{4.35 \times 10^3 \times 4 \times 10^6}{\pi \times (17.6)^2 \times 2.5 \times 1000} \approx \mathbf{7147}$$

Total number of tubes in both passes

$$= 7147 \times 2 = \mathbf{14294}$$

If L is the length of each tube, then we can write

$$\pi d_o L \times (2n) = A$$

$$\therefore \quad \pi \times \frac{2}{100} \times \text{L} \times 14294 = 9015$$

$$\therefore \quad L = \mathbf{10.04}\ \text{metres}$$

Problem 17.16. *A surface condenser with separate air pump for removing the air only and providing separate air-cooling section is designed to condense 20 tons of steam per hour. The air leakage per hour is 6 kg. The temperature of the condensate is 36°C and temperature near the suction of air pump is 28°C. The steam enters into the condenser at 39°C and dry-saturated condition. Find*

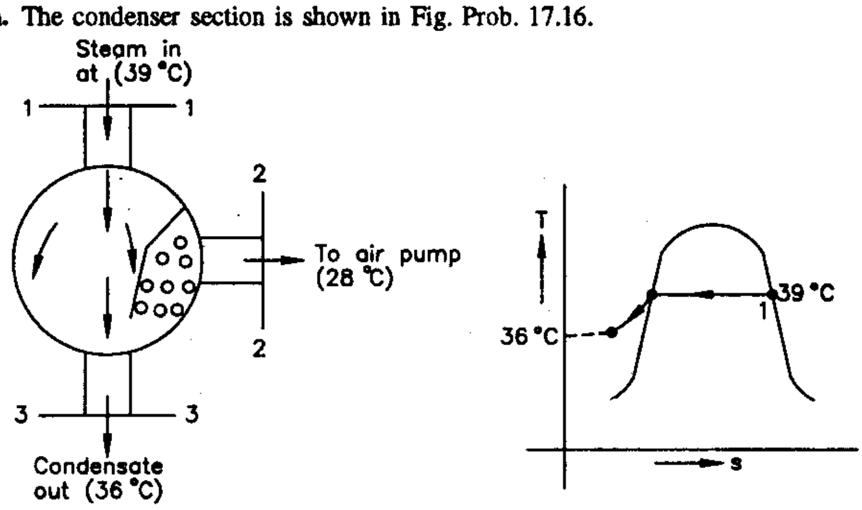
(*i*) *Percentage reduction in air-pump capacity due to separate air-cooling section.*

(*ii*) *Minimum quantity of cooling water if the rise in temperature is limited to 15°C.*

(*iii*) *Saving in the condensate and heat supply in the boiler per hour due to incorporation of air cooling section.*

The loss of condensation is made up with water at 15°C. (*B.U., Dec. 2001*)

Solution. The condenser section is shown in Fig. Prob. 17.16.



(*a*) (*b*)

Fig. Prob. 17.16.

Important Note. The total pressure in the condenser at any section remains same. The total pressure at the entry of the condenser is the sum of pressure corresponding to saturation temperature of 39°C and the pressure exerted by the air.

Considering Section 1 – 1

$$T_1 = 39°\text{C},\ p_{s1} = 0.06991 \text{ bar and } v_{s1} = 20.56 \text{ m}^3/\text{kg (from steam tables)}$$

$$h_{s1} = 2572.6 \text{ kJ/kg}$$

V_{s1} (volume of steam entering the condenser)

$$= m_s v_{s1} = 20 \times 10^3 \times 20.56 = \mathbf{411.14 \times 10^3\ m^3/hr}$$

By Dalton's law

$$V_{a1} = V_{s1} = 411.14 \times 10^3 \text{ m}^3/\text{hr}$$

Applying the equation of state to the air at 1 – 1

$$p_{a1} = \frac{m_{a1} R_a T_1}{V_{a1}} = \frac{6 \times 287(39 + 273)}{411.14 \times 10^3}$$

$$= 1.32 \text{ N/m}^2 = 1.31 \times 10^{-5} \text{ bar}$$

$\therefore$ $$p_t = p_{s1} + p_{a1} = 0.06991 + 1.31 \times 10^{-5}$$

$$= \mathbf{0.069923\ bar}$$

Considering Section 2 – 2

$$T_2 = 28°C$$

$$p_{s2} = 0.0378 \text{ bar and } v_{s2} = 36.728 \text{ m}^3\text{/kg (from steam tables)}$$

and $h_{s2} = 2552.7$ kJ/kg

$$p_{a2} = p_t - p_{s2} = 0.069923 - 0.0378 = \mathbf{0.032123 \text{ bar}}$$

Applying the equation of state to air at section 2 – 2

$$p_{a2}V_{a2} = m_a R_a T_2$$

$$0.032123 \times 10^5 \times V_{a2} = 6 \times 287(28 + 273)$$

$$V_{a2} = \mathbf{153.575 \text{ m}^3\text{/hr}}$$

As per Dalton's law

$$V_{s2} = V_{a2} = \mathbf{153.573 \text{ m}^3\text{/hr}} \quad ...(1)$$

Let m_{s2} is the mass of steam carried away with air at section 2 – 2.

$$\therefore \quad V_{s2} = m_{s2} \cdot v_{s2}$$

$$\therefore \quad m_{s2} = \frac{153.575}{36.728} = \mathbf{4.35 \text{ kg/hr}} \quad ...(a)$$

Consider section 3 – 3 and assume air-pump is not used, then

$$T_3 = 36°C, \; p_{s3} = 0.0594 \text{ bar}, \; v_{s3} = 23.967 \text{ m}^3\text{/kg (from steam table)}$$

Then $p_{a3} = p_l - p_{s3} = 0.069923 - 0.0594 = \mathbf{0.010523 \text{ bar}}$

Applying equation of state to air at 3 – 3

$$p_{a3}V_{a3} = m_a R_a T_3$$

$$0.010523 \times 10^5 \times V_{a3} = 6 \times 287(36 + 273)$$

$$\therefore \quad V_{a3} = \mathbf{503.65 \text{ m}^3\text{/hr}}$$

$$V_{s3} = V_{a3} = 503.65 \text{ m}^3\text{/hr} \quad ...(2)$$

But $V_{s3} = m_{s3} \cdot v_{s3}$

$$503.653 = m_{s3} \times 23.967$$

$$\therefore \quad m_{s3} = \mathbf{21.09 \text{ kg/hr}} \quad (b)$$

From the equations 1 and 2, the percentage reduction in air pump capacity

$$= \frac{V_{a3} - V_{a2}}{V_{a3}} \times 100 = \frac{503.653 - 153.575}{503.653} \times 100 = \mathbf{69.5\%}$$

Determination of Cooling Water Requirement

Applying energy balance to condenser.

Heat lost by the steam and air = Heat gained by water + Heat in condensate

$$(m_{s1}h_{s1} - m_{s2}h_{s2}) + (m_{a1}h_{a1} - m_{a2}h_{a2}) = m_w (h_{w2} - h_{w1}) + m_c h_c$$

$$m_w C_p (\Delta T_w) = [20 \times 10^3 \times 2572.6 - 4.35 \times 2552.7] + [6 \times 1.005(39.0) - 6 \times 1.005 \times 28] - 20 \times 10^3 \times 4.18(36.0)$$

where $m_c = m_{s1}$ (assumed)

$$m_w \times 4.18 \times 15 = 2 \times 10^3(2572.6 - 0.555) + 6 \times 1.005(39 - 25) - 20 \times 10^3 \times 150.5$$

$$= 2 \times 10^3(2572.6 - 0.555 - 150.5) + 6 \times 1.005(14)$$

$$\therefore \quad m_w = 772.35 \times 10^3 \text{ kg/hr} = \mathbf{772.35 \text{ tons/hr} = 214.5 \text{ kg/sec}}$$

$\therefore$ Saving in the condensate

$$= m_{s3} - m_{s2} = 21.09 - 4.35 = \mathbf{16.74 \text{ kg/hr}}$$

Saving in heat supplied

$$Q = 16.74 \times 4.18(36 - 15) = \mathbf{1469.44 \text{ kJ/hr}}$$

EXERCISES

17.1. What are the principal reasons for operating steam turbines with condenser ?

17.2. Classify the condensers and explain the essential difference in each type.

17.3. Describe with neat sketches the operation of the following condensers.
(*a*) Jet condenser (*b*) Surface condenser, and (*c*) Evaporative condenser.

17.4. Discuss the merits and demerits of surface condensers over jet condensers.

17.5. Define the term 'vacuum efficiency' applied to a condenser. Discuss the factors which affect the vacuum efficiency of a condenser.

17.6. The vacuum maintained in a condenser used with steam engine is always less than the vacuum maintained in a condenser used with steam turbine. Explain the reasons.

17.7. The vacuum which can be maintained in surface condenser is as high as 73.5 cm of Hg where the maximum vacuum which can be maintained in jet condenser is limited to 68 cm of Hg. Explain the causes.

17.8. What are the sources of air leakage into a condenser ? Briefly state the effects of air leakage on the performance of a condenser.

17.9. State the usual sources of inefficiency in condensers in which a high vacuum is required at the exhaust end of a steam turbine.

17.10. The vacuum efficiency of a surface condenser is always higher than the vacuum efficiency of jet condenser. State the causes.

17.11. Describe the various methods used to obtain the maximum possible vacuum in condensers used in modern steam power plants.

17.12. Define Dalton's law of partial pressure and explain how it applies to a condenser of a steam power plant.

17.13. Describe with neat sketches the different forms of a surface condenser used in steam power plants.
List out the factors which are responsible for loss of efficiency in a surface condenser.

17.14. Describe the different methods used for maintaining the vacuum in the condensers. Discuss their relative merits and demerits.

17.15. Explain clearly the functions of the following components used in modern steam power plants.
(*a*) Air ejecter, (*b*) Inter-cooler, (*c*) After cooler.

17.16. Discuss briefly the causes of failure or insufficient vacuum in a surface condenser equipped with steam ejector for steam turbine power plant.

17.17. Explain the benefits of fitting an air-cooling section to the steam-condenser.

17.18. Describe either (*a*) an arrangement suitable for reducing the vapour loss at the air extraction of a condenser or (*b*) A sideaerator. Explaining how it is incorporated into a circuit.

17.19. Find out the theoretical formula for finding out the minimum quantity of cooling water required to be circulated through the condenser per minute.

17.20. What is the function of a cooling tower in a modern steam power plant ? Describe with a neat sketch the working of a mechanical type cooling tower.

17.21. The heating of water used for steam condensation in jet condenser up to the saturation temperature of steam in condenser is preferred where it is not preferred in surface condenser even if it reduces the quantity of cooling water required. State the reasons.

17.22. The rise in temperature of the cooling water used in surface condensers is limited to 10°C. State the reasons.

17.23. Why single pass condensers are preferred with once through cooling system ? What are the design factors in case of single pass condenser which differ from multi-pass condenser ?

17.24. What are the causes of corrosion and scale formation is condenser tubes ? Discuss the different methods used for their prevention.

17.25. List out the different materials used for condenser tubes and discuss the merits and demerits of each from the heat transfer point of view and resistance to corrosion.

17.26. How the corrosion and scale formation are reduced in once through cooling system ?

17.27. The following readings were taken during a test on a condenser.

Vacuum in the condenser = 700 mm of Hg.
Barometer reading = 762 mm of Hg.
Temp. of steam entering into the condenser = 35°C.
Cooling water circulated = 46270 kg/hr.
Inlet and outlet temps. of cooling water = 16.7°C and 31°C.
Condensate collected = 1200 kg/hr.
Find (*a*) mass of air present per cu-m of condenser volume.
(*b*) the condition of steam entering into the condenser.
(*c*) the vacuum efficiency of the condenser.

17.28. Steam enters into a condenser where the vacuum is maintained at 664.5 mm of Hg when the barometer reads 760 mm of Hg. The condition of steam entering into the condenser is 0.9 dry and the temperature of condensate coming out of condenser is 35°C. The rise in temperature of circulating water is 10°C. Find the quantity of circulating water per kg of steam condensed and condenser efficiency. Neglect the air pressure at the entrance of condenser.

17.29. A surface condenser handles 13625 kg of steam per hour. The quantity of steam entering the condenser is 0.88 dry. The vacuum in the condenser is 694 mm of Hg while barometer reads 760 mm of Hg. The air leakage is 7.25 kg/hr. The temperature at the suction of air-pump is 36°C. Find (*a*) dimensions of single acting dry air-pump to remove the air from the condenser when it is running at 60 r.p.m. Take the volumetric efficiency of the pump as 85% (*b*) the surface area of the condenser required if the rate of heat flow from steam to water is 3900 kJ/m^2-sec. Take $L = 1.25\ D$.

17.30. A surface condenser is used to condense 12500 kg of steam per hour. The air leakage estimated is 1 kg per 2500 kg of steam. The temperature near the suction of air-pump is 34.25°C. The vacuum in the condenser is 705 mm of Hg. when barometer reads 760 mm of Hg. If the air and condensate are removed by the same air-pump find the dimensions of air pump if its volumetric efficiency is 80%. The pump is single acting and runs at 55 r.p.m. Take $L = 1.25\ D$. (*b*) Also find the quantity of cooling water required per hour if the steam entering the condenser is dry and saturated and rise in temperature of water is limited to 10°C. (*c*) the loss of condensate due to loss of water vapour carried with air.

17.31. A surface condenser with separate air pump for removing the air only and providing separate air-cooling section is designed to condense 20 tons of steam per hour. The air leakage per hour is 6 kg. The temperature of the condensate is 36°C and temperature near the suction of air pump is 28°C. The steam enters into the condenser at 39°C and temperature near suction of air pump is 28°C. The steam enters into the condenser at 39°C and dry-saturated condition. Find (*a*) percentage reduction in air-pump capacity due to separate air-cooling section (*b*) minimum quantity of cooling water if the rise in temperature is limited to 15°C. (*c*) saving in the condensate and heat supply in the boiler per hour due to incorporation of air-cooling section. The loss of condensate is made up with water at 15°C.

17.32. A jet condenser is designed to condense 2.4 tons of steam per hour. The vacuum in the condenser is 684 mm of Hg while the barometer reads 754 mm of Hg. The mass of air entering into the condenser is 16.8 kg per hour. The mass of cooling water is 25 kg per kg of steam condensed. The temperature of cooling water is 24°C before entering into the condenser. The cooling water contains air in dissolved condition. The air volume in water is 6% of the volume of total entering water into the condenser at atmospheric conditions. Find the capacity of pump required to remove air, condensate and cooling water assuming its volumetric efficiency is 80%.

17.33. A jet type condenser is used to condense 4000 kg of steam per hour. The vacuum in the condenser is 708.5 mm of Hg while the barometer reads 760 mm of Hg. The quantity of cooling water injected is 40 kg per kg of steam. The volume of dissolved air in the water is 5% of the volume of water at 1 bar and 15°C. The air coming with steam and leakage is 3 kg/hr. The temperature at the suction of dry air pump is 30°C. Assuming the volumetsic efficiency of dry air pump 80%, find the capacity of the pump per minute.

17.34. A barometric jet condenser is used to condense 1200 kg of wet steam per hour. The quantity of cooling water injected at 15°C is 15 kg per kg of steam entering into condenser. The temperature of cooling water and condensate as they leave the condenser is 50°C. The vacuum in the condenser is 66 cm of Hg while the barometer reads 76 cm of Hg. Find (*a*) the dryness fraction of steam. (*b*) A separate air pump of 3 m^3 per minute capacity is used to remove the air from condenser. The temperature at the suction of air pump is 40°C. Assuming the vacuum is uniform throughout the shell of condenser, find the mass of air entering into the condenser per hour. (*c*) Also find the effective length of the tail pipe required for the condenser.

17.35. A surface condenser is designed to handle 90,000 kg of steam per hour when the steam enters the condenser at 0.08 bar and 0.9 dry. The inlet and outlet temps of water are 25°C and 31°C. The tubes sizes used are 25 mm *OD* and 22 mm *ID* and length of each tube is 5 m. Assuming the velocity of water through the tubes is 1.5 m/sec, overall heat transfer coefficient is 2800 W/m^2-°C and there is no under-cooling. Find the total number of tubes used and number of passes.

17.36. A condenser is to be designed to condense a certain mass of steam per hour at a temperature of 50°C. The necessary heat transfer rate to condense the vapour is 2×10^6 kJ/hr. Water flowing through tubes of 22 mm inside diameter and 26 mm of outside diameter 10 metre long are to be used with the cooling water at a rate such that the Reynolds number is 10,000. The water enters at 15°C and leaves at 30°C. Assume the heat transfer coefficient of steam side 7500 W/m^2-°C and neglect the resistance of tubes. Find the water required per hour. Also find the number of tubes required per pass and number of passes. Take the other required properties from the steam tables.

17.37. Steam is to be condensed at 0.03 bar and 0.94 dry in a shell and tube condenser consisting of 20 m length of copper tube (*OD* = 25 mm and *ID* = 22 mm). Water is available at 15°C for condensing the steam.

Assuming the following data :

h (water side) = 4000 W/m-°C.

h (steam side) = 10,000 W/m^2-°C.

Resistance of scale (steam side) = 0.0002 m^2-°C/W.

Resistance of scale (water side) = 0.0006 m^2-°C/W.

Total number of tubes used = 200.

Assume all ideal conditions.

□□□

18

Cooling Ponds and Cooling Towers

18.1. NECESSITY OF COOLING THE CONDENSER WATER

The cooling water system is one of the most important systems of power plant and its availability predominantly decides the plant site. The high cost of water makes it necessary to use cooling towers for water cooled condensers.

The cooling water requirement in an open system is about 50 times the flow of steam to the condenser. Even with closed cooling system using cooling towers, the requirement for cooling water is also considerably large as 5 to 8 kg/kW-hr. This means a 1000 MW station will require about 100 thousand tons of circulating water per day even with the use of cooling towers. This huge requirement of cooling water is equivalent to a requirement of big city like Bombay.

As the cooling water takes the latent heat of steam in the condenser, the temperature of the water increases. The hot water coming out of the condenser cannot be used again in a closed system without pre-cooling. This is because, the hot water coming out if used again will not be able to absorb the heat as T_{wo} reaches near to T_s saturation temperature of steam at condenser pressure and the condenser vacuum cannot be maintained. Therefore it is absolutely necessary to precool the water coming out of condenser before using again.

18.2. CONDENSER WATER COOLING SYSTEMS

The cooling systems which are commonly used in practice according to the availability of the water are listed below :

Cooling System
- Open or once through system
- Closed system
  - Water cooling
    - Cooling ponds
    - Cooling towers
  - Air-cooling
    - Direct dry type cooling tower system
    - Indirect air cooling system
- Mixed system

1. Open or Once Through or River Water System. In this system, the water is drawn directly from the upstream side of the river, pumped through the condensers and then discharged to the downward side of the river at temperature 5 to 10°C in excess of the inlet temperature. The temperature of the discharged water should be kept within safe limits to prevent harm to fishes. The limit of discharged water temperature is specified by the Fisheries Board. On one river in England, the discharged water temperature was limited to 24°C for fishing reasons and this necessitated the use of single pass-condensers.

The arrangement of the open system is shown in Fig. 18.1. The position of inlet and outlet should be chosen in such a way that there should not be recirculation of hot water which impaires the efficiency of the condensing plant. Such re-circulation is possible in small rivers and canals therefore the distance between the inlet and discharged points should be as large as one kilometre or more. This type of cooling system can be used only where required quantity of water is available throughout the year.

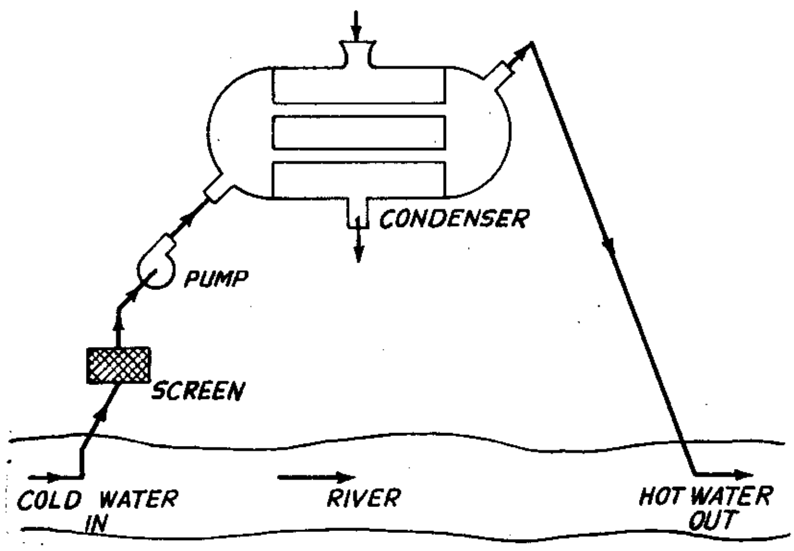


Fig. 18.1. Once through cooling system.

2. Once through system when sea or tidal river water is used for cooling. The plants on tidal waters (using water from the tide or sea) have a special problems in construction of cooling water system and in avoiding re-circulation of the cooling water from the outlet back to the intake. In many plants the intakes and outlets are separated as much as 3 kilometres.

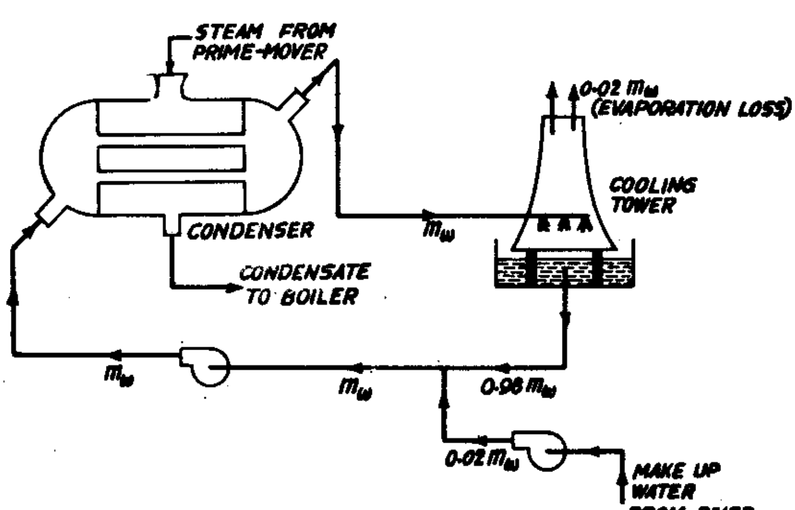


Fig. 18.2. Closed cooling system.

In tidal waters, the water flows in one direction for some time and in another direction for some time, therefore it requires a special arrangements for the change in flow direction. The stations taking water from river or tidal source usually have the ends of both section and discharge pipes submerged below the lowest recorded tidal level. The water circulation system will operate at all times and at all states of tide if the distance between the lowest level and highest point of the circulating water system does not exceed 10 metres.

Closed System. When enough quantity of water is not available for cooling requirement from river, then closed type system is universally adopted.

In closed system, the hot water coming out of condenser is cooled either by spraying in the pond or passing through the cooling tower instead of discharging to the downward side of the river. The quantity of water required is collected from the river during flood period or when sufficient water is available with required purity and same water is used again and again for condenser by passing through the cooling towers. Such arrangement of cooling system is shown in Fig. 18.2.

With this system of cooling also, an external source of water is needed to replace tower evaporatign and carry-over losses. The quantity of water lost by evaporation and carry-over varies between 2 to 5% of that circulated depending upon the design of tower.

Even 5% of make up water is appreciable quantity (five thousand tons per hour for 1000 MW capacity plant) therefore the power plants using closed cooling system obtain their make-up water from rivers and a similar intake arrangement is adopted on a smaller scale as used for open system.

Mixed System. Mixed type cooling system uses river water as well as cooling towers simultaneously. This system embodying the river and cooling towers overcomes the difficulty of re-circulation and meet the requirements of a fishery board on a fairly small river. The arrangement of this system is shown in Fig. 18.3. The part of the water from the condensers is discharged directly into the downstream of the river, and part of the water is pumped to the cooling tower, where it is further cooled and then discharged to the downward side of the river. In this way the water discharged to the river is maintained at a suitable temperature and re-circulation troubles are eliminated. The advantages of this system are listed below :

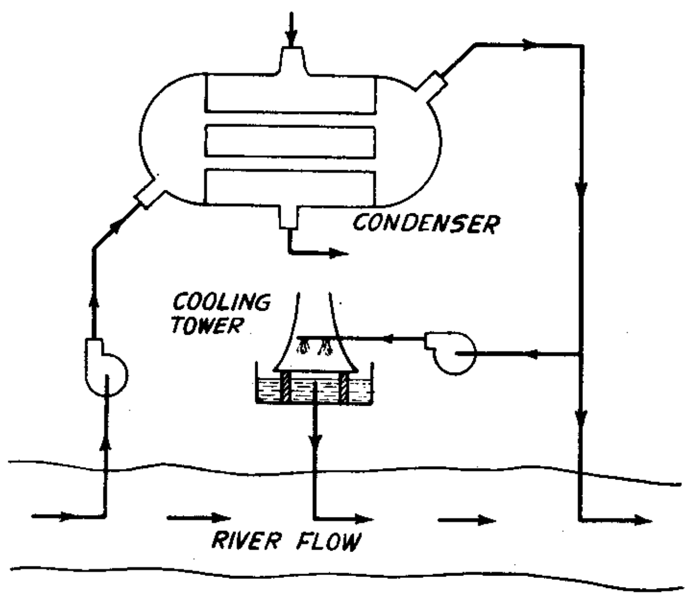


Fig. 18.3. Mixed type or combined river and cooling tower system.

1. The size of cooling towers can be reduced where the side area is limited.
2. The quantity of cooling water required is reduced as re-circulation is eliminated.
3. The turbine plant efficiency is increased.
4. A saving in discharge pipework and pump power is possible as the towers are situated at considerable small distance from the station.

This system should be adopted only when there is possiblity of recirculation and it is necessary to meet the requirements of a fishery board. Another type of system where river water and cooling tower both are used simultaneously to take the condenser load is shown in Fig. 18.4.

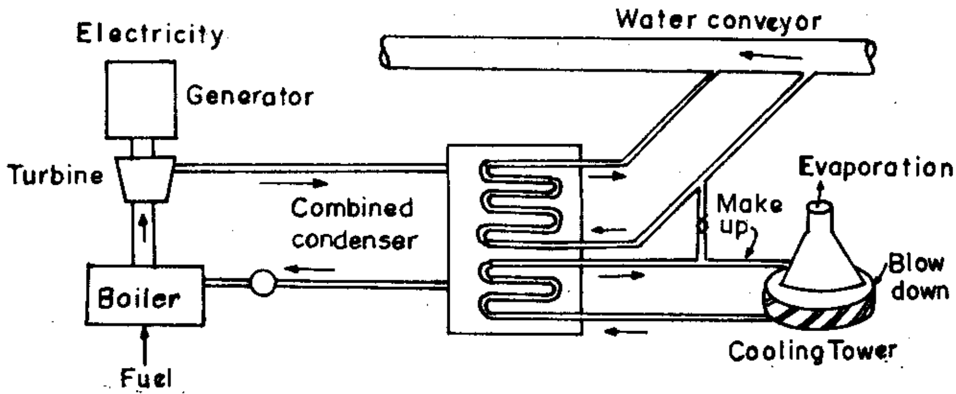


Fig. 18.4. Scheme Diagram of Proposed Combined Condenser Method (OTC-Tower)

Steam condensing plant when adequate cooling water is not available for whole the year. This system is used as open or closed system. Sometimes, the adequate water in the river or canal is not available throughout the year for once through cooling system. Many times, the river can supply adequate water 80 to 90% period of the year but it runs dry during the remaining period of the year. Therefore, under such circumstance, the once through cooling system can be used only for 80 to 90% of the year and for the remaining period closed system must be used. In this type of cooling system, the cooling towers are used only during the period of inadequate water supply.

The example of such case is the Pinki Power Project at Kanpur. For 11 months, normal once through system is used for condenser cooling and for remaining one month, cooling towers are used when the Ganga's canal runs dry. The water from Ganga's canal is pumped with the help of three pumps, each having a capacity of 8000 cu-m/hr. The water after passing through the condenser for cooling is discharged to the downstream of the river as shown in Fig. 18.5.

During the time when the canal is closed, the closed circulation system is used. The water coming out of condenser is passed through the cooling towers having six sets as shown in figure. Make-up water during the canal closure is obtained through the make-up water pump. The seepage water in the canal will be sufficient to provide this make up.

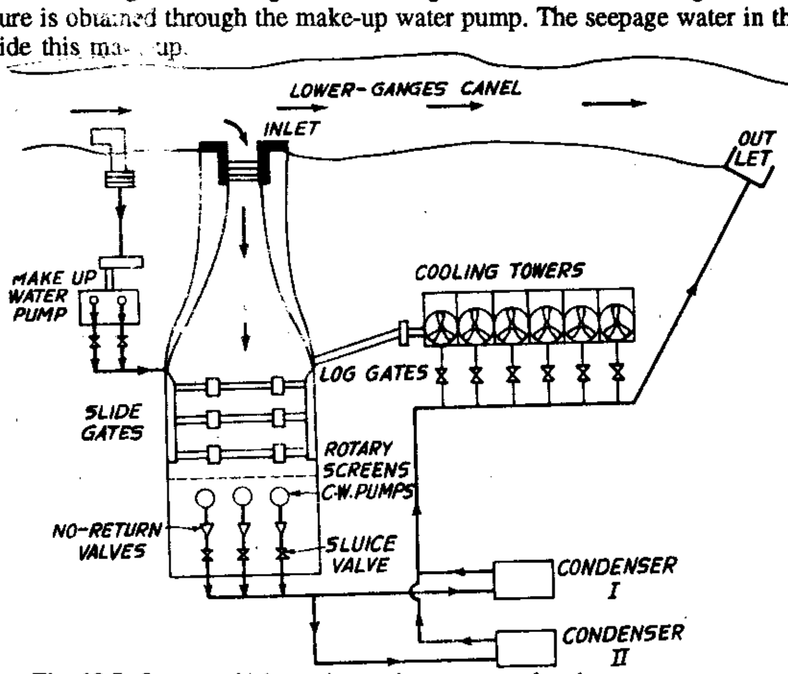


Fig. 18.5. System which can be used as open or closed system.

18.3. WATER COOLING METHODS AND MECHANISM OF COOLING

The cooling of water coming out from the condenser is done by either cooling pond or cooling tower. The hot water gets cooled when exposed to atmospheric air, that is because, the atmospheric air has definite capacity to absorb water vapour at given temperature and the heat required for water vapourization is taken from the remaining water and water gets cooled. The detailed heat transfer mechanism between the hot water and atmospheric air is discussed below.

Mechanism of cooling. Whenever the moisture is exposed to air either in form of droplets or sheet, part of it is evaporated. As the liquid changes to vapour, the heat required for evaporation is taken from the remaining water and water gets cooled.

When the hot water comes in contact with air, the heat from the water to air is transferred as sensible heat as water temperature is higher than air temperature and evaporation heat as WBT of air is lower than DBT (dry bulb temperature) of air. The difference between the DBT and WBT (wet bulb temperature) indicates the capacity of air absorbing the water vapour. The rate of heat transfer between the hot water and air depends upon the initial temperature of hot water, temperature of atmospheric air, relative humidity of air, the movement of air and solar radiation. Higher the DBT, lower the WBT, and higher air movement gives better cooling of water.

The conditions of atmospheric air and the air after cooling the water coming in contact are represented on psychrometric chart as shown in Fig. 18.6.

The net heat absorbed per kg of air from the hot water is given by $(H_2 - H_1)$ as shown in figure. On the average about 80% of total heat lost by water is removed by evaporation and 20% by sensible heat transfer.

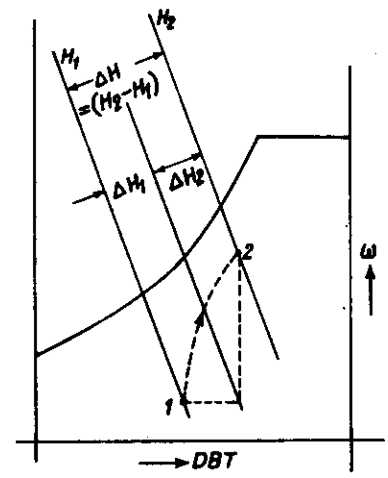


Fig. 18.6. Representation of evaporative cooling on psychrometric chart.

Total heat transferred = Heat of evaporation + sensible heat

$$\Delta H = \Delta H_2 + \Delta H_1$$

A cooling tower is an evaporative cooler. In a tower, a large surface area of water is exposed to the air that is passing through the tower so that some of the water will evaporate into the air.

A difference in vapour pressure between the saturated water and unsaturated air is what triggers this evaporation. The heat energy necessary for evaporation is taken from the water itself and lowers the temperature of the remaining water.

In all thermal power plants using steam as working fluid, either cooling ponds or cooling towers are used for condenser water cooling. The constructional details and operating features are discussed below.

The warmed water coming out of condenser rises to the surface of the body of water to which it discharges, because of its lower density. It spreads in a thin film on the surface which helps for evaporation and cooling of water. A few plants depend entirely on such surface coolingwithout using sprays or cooling towers. This method of cooling proves feasible where enough water surface is naturally available like lakes, ponds and sea water surface.

18.4. INTRODUCTION TO COOLING PONDS

It is sometines advisable not to locate a power plant where condensing water facilities in the usual form of river are not available. In such cases, the choice goes to spray pond or cooling towers.

The spray cooling pond is one of the simplest methods of cooling the condenser water although it is not efficient. Thousands of spray ponds were put into operation until cooling towers came into picture in 1920s.

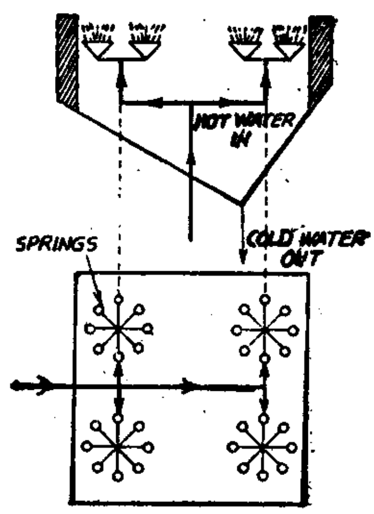


Fig. 18.7. Cooling Pond.

The construction of spray pond used for this purpose is shown in Fig. 18.7. The hot water coming out of condenser is sprayed through the nozzles to expose maximum surface area of water to air for effective cooling. The spray ponds are always surrounded with wooden walls to prevent the wind from carrying the water particles.

The following principles must be adopted for the design of cooling ponds.

1. The long dimension should be placed broad side to the prevailing wind.
2. Spray nozzles should be placed 1 to 2 metres above the water surface to obtain maximum cooling.
3. The nozzle arrangement should be such that there is no interference between the different sprays produced.
4. The distributing pipes may be spaced 6 to 7 m apart.
5. A sufficient space (15 to 20 metres) should be allowed between the last nozzle and the edge of the pond to prevent excessive spray loss.
6. The pressure of 1.5 bar should be used at nozzles for better atomization of water.
7. A surface area of 2.5 to 3 m^2 per litre flow of water per second should be provided.
8. Louvered fence should be provided to reduce the carry-over losses.

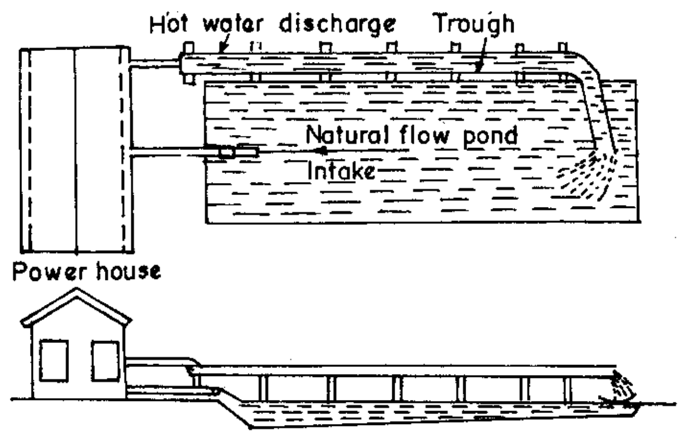


Fig. 18.8. Non-directed flow natural cooling pond suitable for long and narrow lots.

The spray cooling ponds are still preferred where sufficient area at power plant site is available because the area required for spray cooling pond is 25 to 50 times the area of cooling tower. The another advantage of this cooling system is, the pumping costs are less than cooling towers as pumping head required is considerably less.

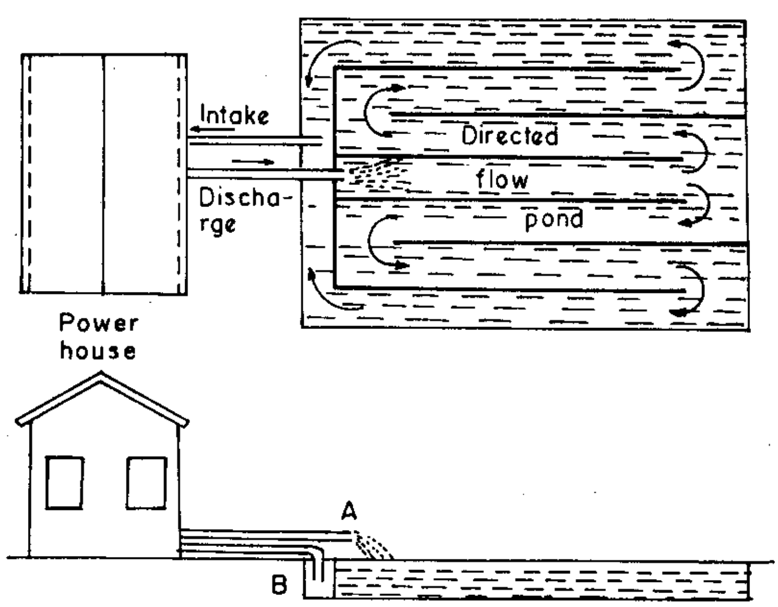


Fig. 18.9. Directed flow natural cooling pond.

Types of Cooling Ponds. The common types of cooling ponds used in practice are listed below :

(1) Single Deck and Double Deck Systems. In a single deck system, spray nozzles are arranged at the same elevation as shown in Fig. 18.8. Its effective cooling effect is less than double deck system.

In this system, spray nozzles are arranged at different elevations as shown in Fig. 18.9. Its cooling effect is more than single deck system as water comes in contact with air at lower temperature.

(2) Natural and Directed Flow System. In natural flow system, water coming out from the condenser is just allowed to flow into the pond as shown in Fig. 18.10. This system is rarely used now-a-days.

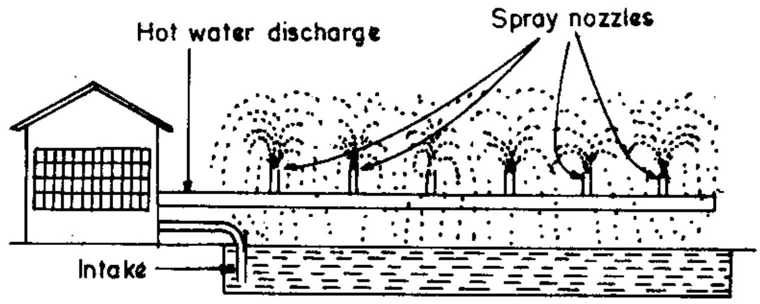


Fig. 18.10. Single deck spray pond.

In directed flow system, the hot water coming out of condenser enters the middle channel as shown in Fig. 18.11. and on reaching the far end divides into two currents, being directed by the baffle walls so as to traverse the pond several times before uniting at the intake point. The water gets more time and passes over a more surface, so the cooling achieved is very effective.

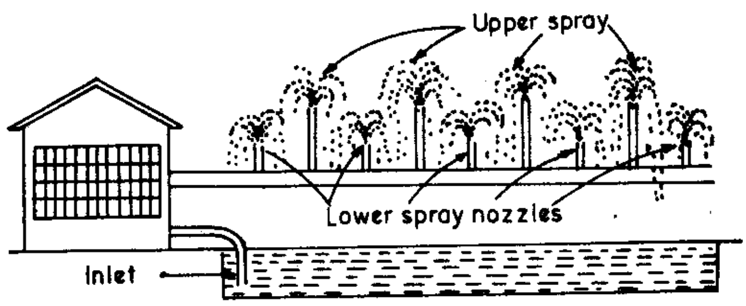


Fig. 18.11. Double deck spray pond.

(3) Open and Louvre Fence. In case of open pond, the drift losses will be more if the wind velocity is high. This can be avoided by providing Louvre fence as shown in Fig. 18.12.

A few disadvantages of cooling ponds are listed below.

1. The area required for cooling is considerably large.
2. Spray losses due to evaporation and windage run high.
3. There is no control over the temperature of cooled water. The cooling effect is reduced with the decrease in wind velocity and if the load on the plant increases, the pond does not respond to the requirement. When the maximum cooling is required during increased load, it provides minimum cooling in absence of wind flow.
4. The cooling efficiency is low compared with cooling tower.

Spray cooling pond system is rarely used nowadays in modern thermal power plants of very high capacity as it requires large surface area which is rarely available.

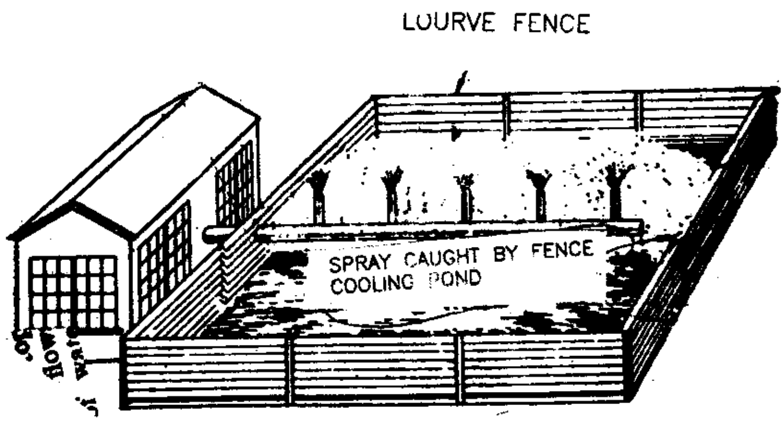


Fig. 18.12.

Cooling Towers. The demand for electric power has increased to such an extent that it is now no longer justifiable to site the large power station near river sides. The present trend is to locate the station near the load centre with the use of large and highly efficient cooling towers.

The cooling towers are desired when positive control on the temperature of water is required, the space occupied by the cooling system is considerable factor and the plant is situated near load centre and far away from the adequate natural resources of cooling water.

The principle of cooling the water in cooling tower is similar to the evaporative condenser or spray pond. The rate of evaporation of water in cooling tower and subsequent reduction in water temperature depends upon the following factors :

(1) Amount of water surface area exposed.
(2) The time of exposure.
(3) The relative velocity of air passing over the water droplets formed in cooling tower.
(4) The R.H. of air and difference between the inlet air WBT and water inlet temperature.
(5) The direction of air flow relative to water.

Higher the surface area, more time of exposure, lower relative humidity, higher difference between WBT of air and water inlet temperature, and cross-flow give effective cooling and reduce the tower size.

The temperature difference between WBT of incoming air and outgoing temperature of water is known as 'Cooling Tower Approach'. Lower cooling tower approach is always desirable as it is an indication of effective cooling. Higher the quantity of water circulated, lower will be the approach of the tower. The quantity of water circulated economically is also limited by the power requirements of the pump.

A cooling tower is a semi-enclosed device for evaporative cooling of water by contact with air. It is a wooden, steel or concrete structure and corrugated surfaces or troughs or baffles or perforated trays are provided inside the tower for uniform distribution and better automization of water in the tower. The hot water coming out from the condenser is fed to the tower on the top and allowed to tickle in form of thin sheets or drops. The air flows from bottom of the tower or perpendicular to the direction of water flow and then exhausts to the atmosphere after effective cooling. To prevent the escape of water particles with air, draft eliminators are provided at the top of the tower.

The cooling towers are mainly divided into two groups as Natural draft or Atmospheric cooling towers and Mechanical draft towers as per the air flow through the towers.

18.5. ATMOSPHERIC OR NATURAL DRAFT COOLING TOWERS

In natural draft cooling towers, the air flows naturally without fan through tower and provides the required cooling.

The natural draft towers are further divided into three types.

1. Natural Draft Spray Filled Tower. The arrangement of this type of tower is shown in Fig. 18.13. The air enters through the lovered sides and flows across the unit in a transverse direction. The air circulation through the tower depends on wind velocity. The capacity of this tower varies from 50 to 100 litres per minute per m^2 of base area depending upon the air velocity.

These towers are used only for diesel plants and where prevailing winds are not cut-off by obstructions. This is not used for high capacity thermal plants as cooling range is limited, wind losses are high and there is no control over the outlet temperature of water.

2. Natural Draft Packed Type Tower. The construction of this tower is similar to spray filled tower except that water distributing toughs of fills are used which helps to break the water into small droplets. In this tower also, the flow of air is cross-wise to the flow of water.

These towers are also rarely used for thermal power plants as original cost (due to height requirement) and pumping head required are high. Tower's extreme length and height and narrow width require anchoring to withstand high winds.

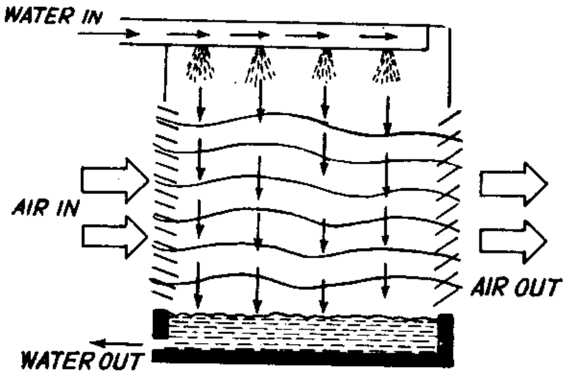


Fig. 18.13. Atmospheric spray-filled cooling tower.

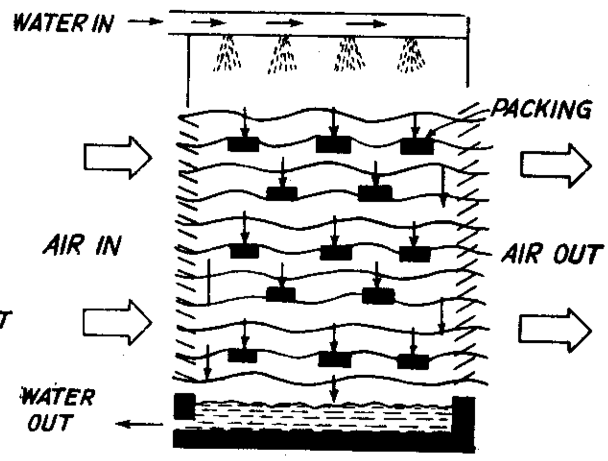


Fig. 18.14. Packed atmospheric cooling tower.

3. Hyperbolic Cooling Tower. The first hyperbolic natural draft reinforced concrete tower was designed by Prof. Van Itesson of Dutch state and installed at Emma Collisey in 1916.

The hyperbolic cooling towers are widely used in Europe. First unit of this type was installed in U.S.A. at Big-Sandy station of Kentucky Power Co. It is capable of handling 120×10^3 gpm and cools the water from 43°C to 30°C. It has minimum diameter of 39.5 m and maximum diameter of 74.5 m and is 400 metres high. It serves to the station of 265 MW capacity. Another notable cooling tower of this type is Gundremmingen, 160 m high, 135.2 m bottom and 83.2 m top diameter established in 1978 at Germany.

The arrangement of hyperbolic cooling tower is shown in Fig. 18.15. It is steel reinforced concrete structure mostly slack (empty space) and the bottom 10 m above the air-intake contains packing over which warm water flows. The shape of the stack is circular in plan and hyperbolic in profile. The operation of this tower is much like that of other natural draft spray cooling towers with hot water cascading over timber splash type filling through which cooler air moves.

Any cooling tower breaks the warm water into a spray or a film of water and expose it to a flow of air. Some of the heat in the water is directly transferred to the air stream and some of the water evaporates, providing significant cooling effect. Then the moist warm air goes to the atmosphere and the cooled water is collected and returned to the power plant.

As in natural cooling tower, there is no fan to create the flow of air in the hyperbolic cooling tower.

But the flow of air through it is established by chimney action. The flow is created by the density difference between atmospheric air and the air inside the tower which has been warmed by the hot circulating air. Higher the RH of the air, it contains more water vapour (H_2O) which is lighter than air and the net density of high humidity air becomes lower than the surrounding air. This difference in density provides necessary pressure head for creating the flow. The difference in density is considerably small so the height of the cooling tower required to create positive flow of air is considerably large compared with mechanical towers.

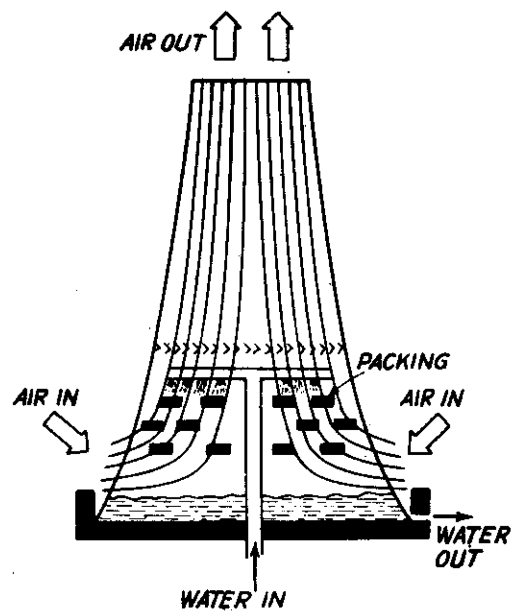


Fig. 18.15. Hyperbolic cooling tower.

The advantages of this tower over mechanical towers are listed below :

1. The hyperbolic towers have a cooling capacity comparable with that of multicell installation of induced draft cooling towers and they also require considerable less ground area.

2. Since no fans are needed, power cost and auxiliary equipments are eliminated and therefore operating and maintenance costs are consequently reduced. It gives more or less trouble-free operation.

3. Hyperbolic tower's chimney shape creates its own draft assuring efficient operation even when there is no wind.

4. Ground fogging and warm air-recirculation which are often the facing problems with mechanical draft installations are also avoided in hyperbolic towers.

5. The towers may be as high as 125 m and 100 m in diameter at the base with the capability of withstanding winds of well over 100 mph which is hurricane speed. These structures are more or less self-supported structures.

6. The enlarged top of the tower allows water to fall out of suspension.

Hyperbolic natural-draft towers work best when the difference between cold water and WBT of air ($T_{wbt} - T_{wi}$) is equal to or is greater than the difference between hot water and cold water temperature, *i.e.*, when approach is equal to or greater than the range. Therefore, they are preferred when operating conditions couple low WBT and high relative humidity.

The major drawbacks of this tower are listed below :

1. Its initial cost is considerably high.

2. Its performance varies with the seasonal changes in DBT and RH of air.

While initial cost may be higher, the saving in fan power, longer life and less maintenance always favour for this type of tower. It is also more favourable over mechanical draft as central station size increases.

The hyperblic towers are almost often selected over the mechanical draft towers under the following operating conditions.

1. A combination of low WBT and high inlet and outlet water temperature exists, *i.e.*, broad cooling range and long cooling approach.

2. Heat load is heavy during winter.

A natural draft hyperbolic tower is used mainly for large cooling duties, for example, in power stations where close approach to WBT and accurate water temperature control are not very essential though its initial cost is high, it has an advantage of low maintenance and operation costs because of the absence of fans and other power consuming mechanical devices.

The construction technology is so developed that a natural draft cooling tower of 135 m height and 100 m diameter with hardly 25 cm thick wall is designed and constructed by ECC Construction Group in India. If the dimension of egg were proportionately increased, the cooling tower shell thickness would be one-third that of an egg. ECC has advanced construction technology and comprehensive engineering services to undertake such challenging projects.

18.6. MECHANICAL DRAFT COOLING TOWERS

The natural draft towers are totally replaced by mechanical draft towers as they provide closer approach to WBT, give higher efficiency, reduce spray and windage losses and require reduced ground area.

Mechanical draft towers require less space and less piping than natural draft towers. Lower water temperatures supplied by mechanical draft towers usually boost overall plant economy enough to cover the added operating charges and higher initial cost of installation.

The mechanical draft towers use fans to move the air through the tower instead of depending on natural draft or wind velocity. This speeds cooling and increases the efficiency of tower by increasing the air-velocity over wet surfaces and through the tower. The correct amount of air at required velocity can be readily selected to meet job cooling requirements. With the use of mechanical tower, much smaller equipment can be used to handle air-flow under fairly well controlled operational conditions.

The mechanical towers are subdivided as forced draft, and induced draft. The induced draft is further divided as counterflow and cross-flow types. The details of each are disussed below.

(1) Forced Draft Tower. The arrangement of the forced draft tower is shown in Fig. 18.16. The fan is located at the base of the tower and air is blown by the fan up through the descending water. The entrained water is removed by draft eliminators on top.

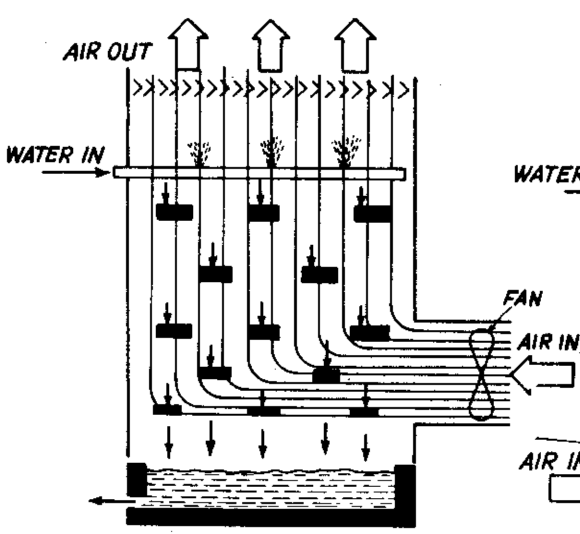


Fig. 18.16. Forced Draft cooling tower.

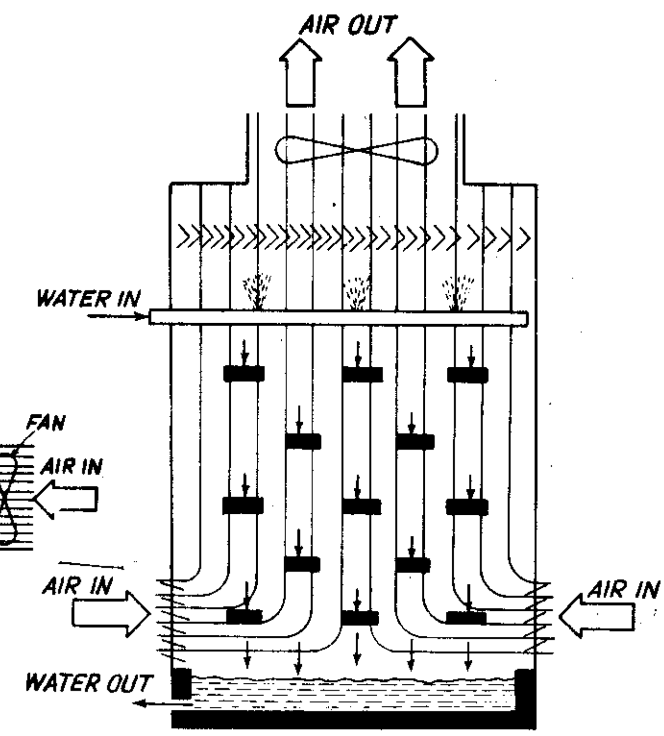


Fig. 18.17. Induced Draft cooling tower.

(2) Induced Draft Counter-flow Tower. The arrangement of the induced draft is shown in Fig. 18.17. The difference lies only in supply of air. In this case the fan is located at the top of the tower and air enters through the louvers located on the tower's side as shown in figure and is drawn up and discharged through the fan casing to the atmosphere.

Such type of cooling towers made with combination of fibreglass, PVC and stainless steel developed by paharpur Cooling Towers Ltd., Calcutta are available in 38 models covering a capacity range of 150 TR in a single unit as shown in Fig. 18.17 (*a*).

E-mail : paharpur.ccu@gncal.globalnet.ems.vsnl.net.in

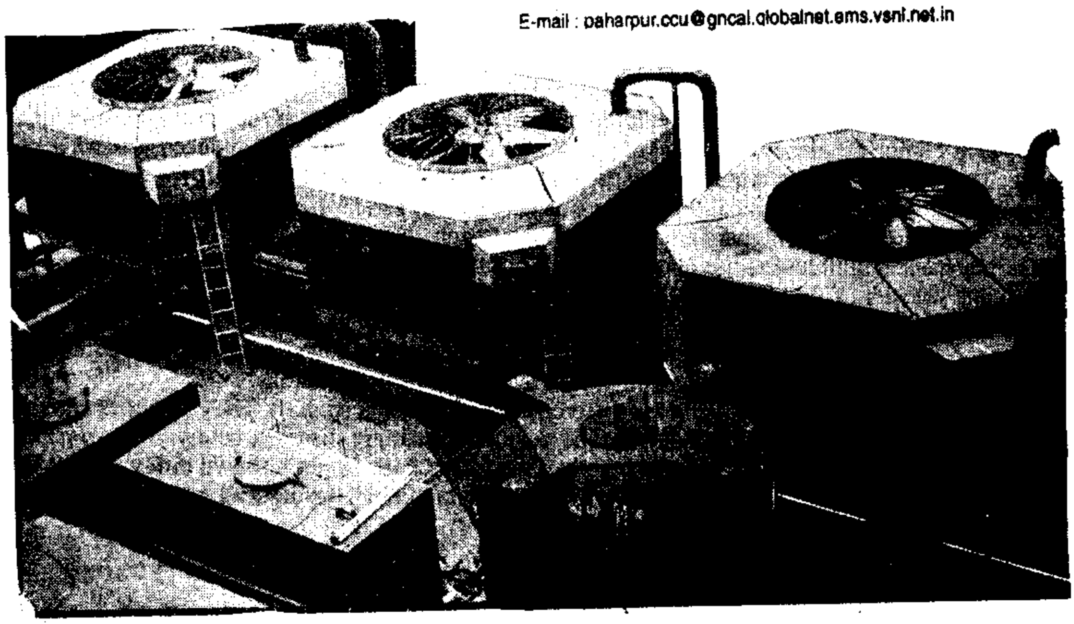


Fig. 18.17. (*a*)

(3) Induced Draft Cross-flow Tower. The arrangement of this tower is shown in Fig. 18.18. This tower provides horizontal air flow as water falls (cross-flow of air to water) down the tower in the form of small drops over filling. The fan centred at top of unit draws air through two cells that are paired to a suction chamber partitioned beneath the fan. The drift eliminators turn air toward out-let fan as air leaves the water sprays. The outstanding feature of this tower is lower air static pressure loss as there is less resistance to air flow.

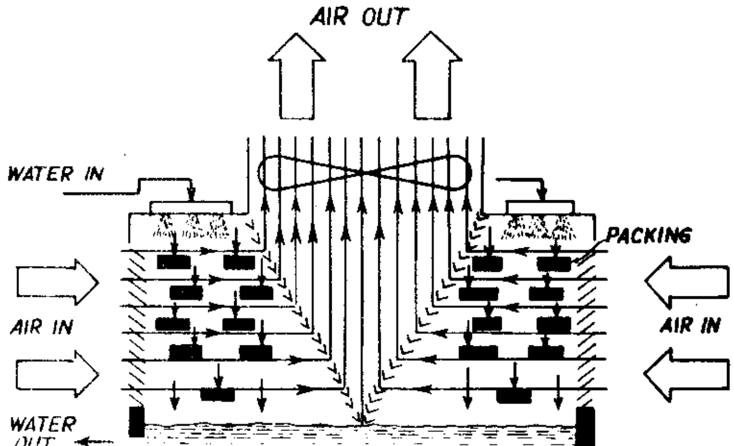


Fig. 18.18.

From a thermodynamic standpoint, the counterflow arrangement is more efficient than cross-flow because its enthalpy potential difference is higher. In cross-flow arrangement, the water flows by gravity through a distribution pan and descends through a large fill area. Therefore, the draft losses are less compared with counter-flow arrangement and resulting in lower power requirements.

The cooling of water in all types of towers is due to the heat lost by the water to the air by convection, conduction and evaporation as it comes in contact with the air. First there is direct exchange of heat between the ascending air and falling drops of water through convection and conduction. Then there is cooling produced by the evaporation over the whole surface of water drops. Such rapid evaporation over the water films in the tower induces cooling. Further, the steam so produced being lighter than air, intensifies the draft. There is some cooling by evaporation on the face of all wet parts of the laths and of the internal fills.

The evaporation and effective cooling of air is greater when the air coming from outside is warmer but drier than when it is cold but already saturated.

The effective cooling of water depends upon the DBT and WBT of atmospheric air, the water inlet temperature, size and height of tower, water distribution system ; velocity of air and its quantity and the fill arrangement. As the controlling factors are so many, it is always difficult to design a cooling tower for given conditions.

Round Cooling Towers. A new profile, round mechanical cooling tower designed by Marley's Co. offers a unique design alternative for multiple-cooling tower installations on limited site areas. The round configuration permits greater siting flexibility using less plan area than comparable rectangular cooling towers as shown in Fig. 18.19. Saving in the space reduces the cost of circulating water supply and return piping which can be significant. Also, round shape makes siting essentially independent of wind direction. The round towers are more preferred when higher plume dischange is a primary condition. The clustered fan arrangement produces a single concentrated, buoyant plume which rises to a higher level before dispersing. It gives proven thermal performance and minimum long term maintenance costs.

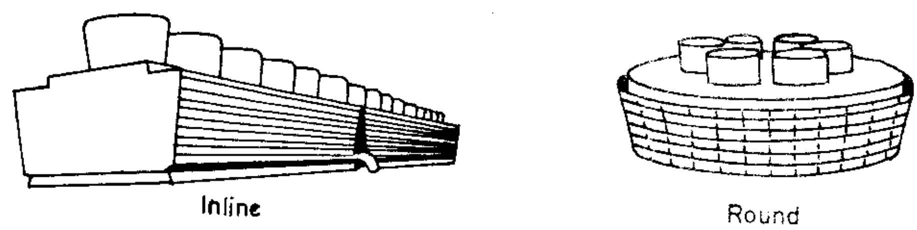


Fig. 18.19.

Another type of forced draft counterflow round tower presently in consideration is shown in Fig. 18.20. With this design, we have land use economy and short pipe runs as basic advantages. In addition to this, by moving the fans from the top to the periphery, we remove them from 100% humidity flume where the possibility of blade rusting is very high. They are also easily accessible for maintenance and service. This tower can give better plume rise from the single discharge compared with induced draft round tower.

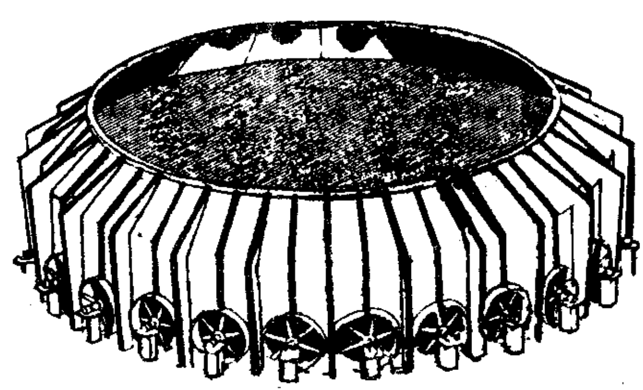


Fig. 18.20. The Forced Draft counterflow round cooling tower.

Comparison of Forced and Induced Draft Towers

Forced Draft Towers. The advantages of this type of towers are listed below :

1. This is more efficient than induced draft as some of the air velocity is converted into static pressure in the tower and recovered in the form of useful work.
2. The vibration and noise are minimum as mechanical equipments are set on a solid foundation.

3. As it handles dry air, problems of fan blade erosion are avoided.

4. It is more safe as it is located on the ground level.

The major disadvantages of this tower are listed below.

1. There is possibility of recirculation of hot, humid exhaust coming out from the top of the tower through the low pressure air intake region. The recirculation can cut the efficiency as much as 20%. To avoid this, a cross-flow tower as shown in Fig. 18.21 can be used.

2. During cold weather, ice is formed on nearby equipments and buildings or in the fan housing itself. The frost in the fan outlet can brake the fan blades.

3. The fan size is limited to 4 metres.

4. The power requirement of F.D. fan system is approximately double that of I.D. system for the same capacity.

Induced Draft Tower. The advantages of this tower over forced draft are listed below.

1. The main advantage is that coldest water comes in contact with the driest air and warmest water comes in contact with the most humid air.

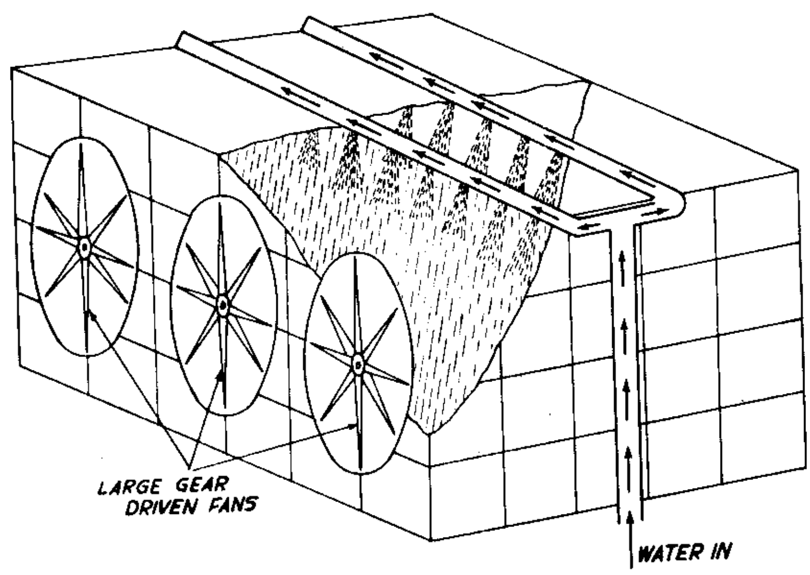


Fig. 18.21. Cross-flow forced type of cooling tower.

2. The recirculation is seldom a problem with this tower as outlet fan discharges the heated and humid air directly away from the air-intakes below the tower.

3. The size of 20 m in diameter can be used.

4. Claims are made that the I.D. fan tower has the advantages of lower first cost, requires less space, is capable of cooling through a wide range.

5. The first cost is lower due to the reduction in pump capacity required and smaller length of water pipes. The power consumption per kg of water cooled is less compared with F.D. fan system.

The disadvantages of this water cooling system are listed below.

1. The static pressure loss is higher as restricted area at base tends to choke off the flow of high velocity air. This requires higher power motor to drive the fan compared with forced draft handling equivalent amount of air.

2. The air velocities through the packings are unevenly distributed and it has very little movement near the walls and centre of the tower.

The forced draft cooling tower is competitive to induced draft type in terms of initial cost for cooling capacity up to 15000 litres of water per minute. For larger quantity, the economics shifts in favour of the induced draft tower.

The water loading of double-flow induced draft tower is considerably higher (500 litres/min/m^2) compared with forced draft tower (300 litres/min/m^2). The only disadvantage is that it promoters the growth of algae as the entire water feed is exposed to air.

The trend of increase in number of natural cooling towers is rapid as well as spread over a wide area of the USA which can be seen from Fig. 18.22.

Comparison between natural and mechanical draft tower. Due to current economic and regulatory forces and continued trend towards higher MW power plants, the natural draft cooling tower has emerged as a major factor in power design.

The number of natural draft towers in USA was hardly 13 in 1970 which went to 129 by the end of 1976. This itself shows that the power industry is moving for natural draft towers instead of for forced draft. The number of natural draft towers in USA is hardly 20% of the total number of cooling towers but the MW capacity they represent is far higher as natural draft towers are specified for the larger plants. The natural draft cooling towers are specified for almost 50% of new generating capacity requiring cooling towers.

(*a*) Natural draft towers installed in 1983 in USA. (*b*) Natural draft towers completed in USA by 1975.

Fig. 18.22.

Both the types of cooling towers are used when the use of once through cooling system is possible either because of thermal effluent problems or lack of sufficient water. Essentially, both types of cooling towers serve the same purpose, to cool the water that flows through the condenser of a power plant operating in a closed cycle mode.

Both types of towers move large quantities of air over the tower packing so that the water breaks into tiny droplets and provides maximum surface for water-air contact for better cooling. Usually, the pumping height or head is in the range of 15 m for both types.

But the adoption of one over the other depends upon many factors as listed below.

The major differences lie in size, shape, installed cost, operating cost, maintenance cost availability and suitability for particular types of plants and plant sites.

1. Size and shape. The mechanical draft tower is 15 m high and runs thousands of metres in length. It looks like a long rectangular box. By contrast, natural draft tower may be 150-200 m high and 100-150 m in diameter and has familiar hyperbolic shape.

The natural draft tower requires less land per kW capacity and it need not be located away from the main power house like meachanical draft towers because of its plume. Natural draft towers are generally prefered where space is limited or expensive.

The difference in the heght of the two towers has become very important factor, particularly with regard to plume dispersion. The problem of local fogging, recirculation and icing is eliminated with natural draft tower as it can release its cloud of water vapour at a considerable height of 200 m. Its adoption for a plant located in a valley has become universal as the 200 m height of the tower allows the plume to be released above the hills and plume is widely dispersed rather than setting back into the valley and causing chronic fogging.

2. Installed cost. The natural draft tower costs nearly twice of mechanical draft tower. The gap is reduced with increasing plant size but the difference still remains substantial.

3. Operating cost. The water pumping cost remains same for both the towers as the pumping head is same for both. But mechanical draft tower requires additional energy to run the fans, therefore the energy costs for mechanical tower are considerable. The energy required to run the fans amounts to 0.5% of the station operating capacity (400 MW plants require 2 MW to run the fans). For large plant size, this energy saving becomes significant if natural draft towers are used.

4. Maintenance cost. Natural draft towers require minimum maintenance cost because the construction is more permanent and less mechanical equipment is involved. Natural draft tower does not require fans, motors and gears therefore the possibility of failure is very remote with natural draft towers. They provide better availability. They are also highly resistant to ice damage and less susceptible to operator error.

5. Site suitability. Natural draft towers are generally preferred for valley locations and sites of the plants near highways and navigable waterways where fogging cannot be tolerated for safety reasons.

The main force behind the adoption of natural draft towers for the coming-up power plants throughout the world is purely economic. The high cost of fan energy penalizes the mechanical draft tower and as long as energy costs remain high, favour goes to natural draft tower.

The mechanical draft towers are preferred for small capacity plants whereas natural draft towers are preferred for large capacity plants. The actual capacity, for which the natural draft tower is preferred, generally depends on design points and optimization analysis for a particular plant. But it is safe to say that the current rate of inflation and energy costs make the natural draft tower more economic even for small plants. The cross-over point may be 400 MW and many descend if the inflation rate and cost of energy continues to increase.

Other factors which are enlarging the domain for the natural draft tower are availability of land, ordinances against low level discharge and weather conditions.

Comparison between counter-flow and cross-flow. Cooling towers are also classified by the motion of air in relation to the hot water. Towers where air and hot water mix at 180° with air moving vertically through the packing are called counter-flow towers. In cross-flow towers, air and water mix at 90°, with air moving horizontally through the packing. Fig. 18.23 and Fig. 18.24 show cross-flow and counter-flow natural draft towers and Fig. 18.25 and Fig. 18.26 show cross-flow and counter-flow mechanical draft towers.

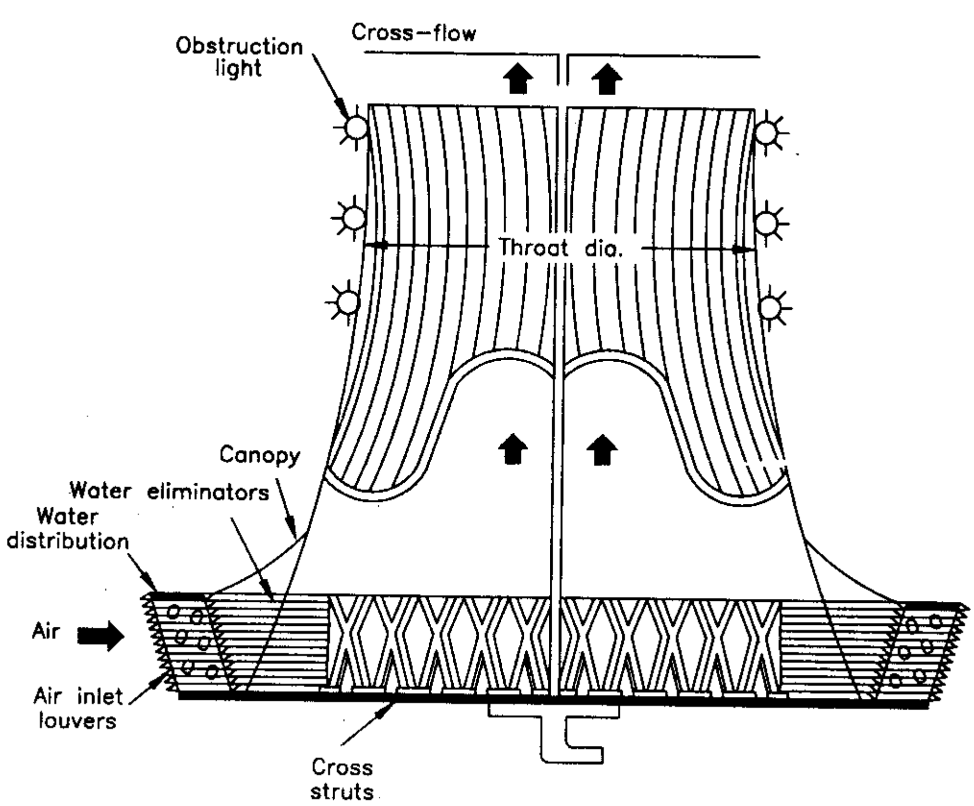


Fig. 18.23. Counterflow natural tower.

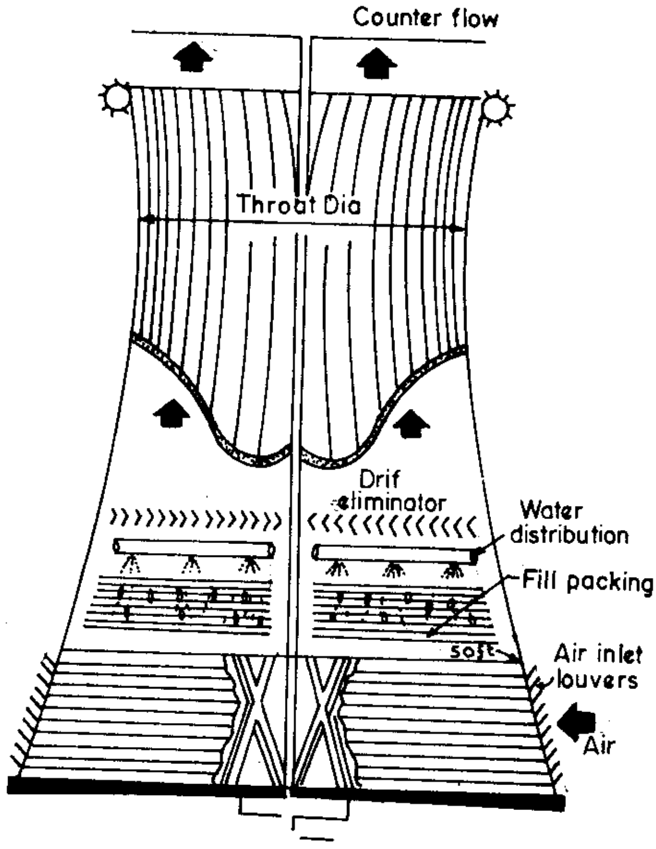


Fig. 18.24. Cross-flow natural towers.

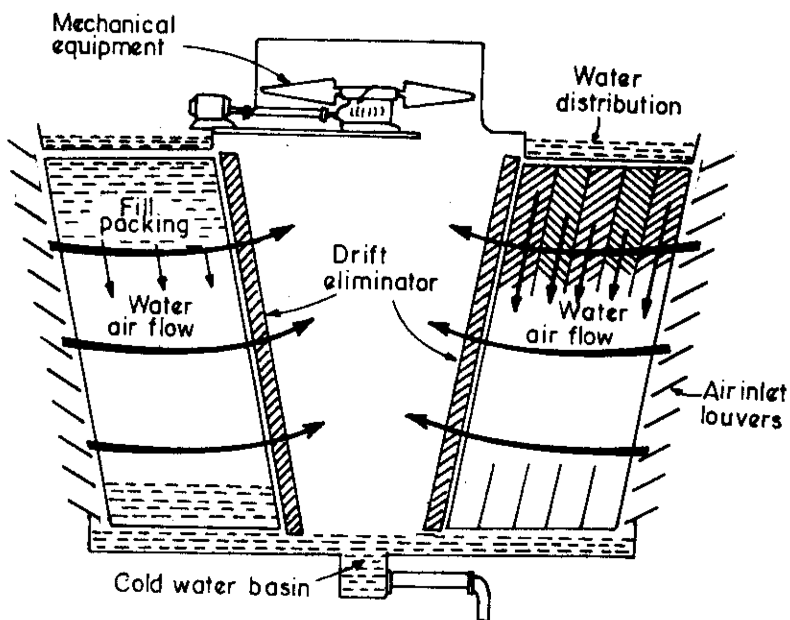


Fig. 18.25. Cross-flow mechanical tower.

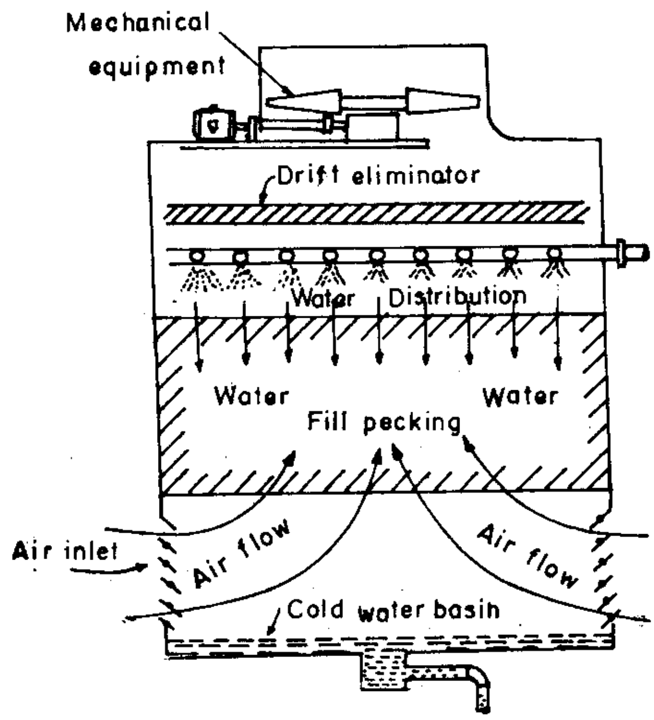


Fig. 18.26. Counter-flow mechanical tower.

Both cross-flow and counter-flow designs have relative strengths and weaknesses. Both designs are comparatively adopted in USA but the present trend is for counter-flow as it has more advantages over cross-flow particularly with new type of fills developed.

*In the past, cross-flow were used because of lower material costs which were compensated for higher operating costs. But the cost of material has not risen as fast as fuel costs and generating equipment costs, so the utimate result is to shift in favour of counterflow. This has been a slow but steady and irreversible process.

*The counter-flow is inherently more effective as all the air eventually approaches saturation at the hottest water temperature, whereas in cross-flow, much of the air is under-utilised because it approaches saturation in the lower portion of the tower at temperatures, well below the inlet water temperature. On the other hand, the cross-flow offers the basic mechanical advantage of an open-sided structure well suited to higher rates than those found in counter-flow towers of comparable cooling duty.

*Cross-flow were more economical than counter-flow when *splash type fill* was used. One main reason for this was that wide counter-flow cooling towers require high air inlet opening at the bottom which are less effective in the cooling process than the parts of the tower above the air inlet openings. After 1970 onwards, film type fill was introduced in the field which was better suited for counter-flow because film cooling provided more effective wetted surface than splash type. In counter-flow, fill is oriented in a horizontal plane where the length and width of the tower are much greater than that of fill thickness. The fill thickness is minimised to reduce air side static pressure head loss, as well as water side pump head. The water loading in cross-flow is higher than counter-flows as the fill is oriented in an upright configration over the entire height of the sidewalls below the water distribution pan where fill thickness is minimised to reduce the air side pressure loss. This concentrates the water flow downward through the narrow fill section.

*Cross-flow towers are less resistant to ice build-up than counter-flow types. In larger capacity stations, great emphasis is placed on reliability so this characteristic is often taken as a significant evalution factor. The present fill materials are far more resistant to ice damage than wood, they are not wholly immune and a failure can be quite dramatic.

*The critical design basis for cooling tower is the conditions encountered during summer high ambient WBT. It is a characteristic of a cooling tower that as the air passing through the tower approaches a saturated discharge condition, the energy driving force between air saturated at water interface and main body air stream is very much influenced by change in air temperature. Thus a tower of low fill capability and high air rate will be less sensitive to reduction in inlet WBT than a tower of higher fill capability and lower air rate. A cross flow tower always operates at higher air rate than does a counter-flow tower for a comparable duty. This is the main reason that a higher air flow-rate cross-flow tower loses less in cold water approach to WBT than does a counter-flow of comparable duty when the WBT is lowerd, as in going from a summer condition to winter condition.

18.7. AIR-COOLED OR DRY-TYPE COOLING SYSTEM

The disposal of waste heat from thermal electric generating plants has always remained an important consideration in plant design and site selection, but in recent years, waste heat disposal has become a major concern with the public comfort. Awareness of potential thermal pollution problems has focussed attention on the task of properly handling this waste heat.

The problem of thermal pollution and importance of its control can be understood with the following example. The heat rejection in the year 2000 in U.S.A. from power plants would take care of the energy requirements for some 20 cities, the size of New York if it is utilised. This problem partly can be solved with the use of evaporative cooling systems (evaporative condenser and cooling towers). But with the increasing demand of water, the amount of water required for evaporative type cooling system may no longer be accepted. It is estimated that the use of evaporative cooling towers to dissipate all of the heat that would be rejected from the power plant in U.S.A. in the year 2000 would consume a water supply approximaterly equivalent to the total annual flow in the Missouri river at Omaha. The water evaporation rate in cooling towers for a fossil fueled plant is approximately 0.6 gal per kW-hr generated which means that the water evaporation in cooling towers for a 800 MW fossil fueled plant operating at 75% annual plant factor would be approximaterly 9700 acre-ft per year.

The planner would have much more flexibility in power plant siting if the availibility of cooling water were not a major consideration.

The dry cooling system (air-cooling), offers a solution to the problem of thermal pollution of water. The dry cooling system rejects the heat directly to the atmosphere which is the largest heat sink available, eliminating the consumption of large volumes of water by evaporation or the thermal pollution of an even larger amount if once through process is used.

Two different dry-type cooling systems which are commonly used are discussed as given below.

18.8. INDIRECT DRY TYPES OR HELLER COOLING SYSTEM

The priniciple components of this system are shown in Fig. 18.27. The condensate collected in the condenser is party fed to the boiler and partly passed through the cooling coils. The hot condensate passed through the cooling coils is cooled by passing the air with the help of fan as shown in figure. The cooled water coming out of cooling coil is spread through the nozzle into the condenser. The steam coming out of turbine is condensed coming in direct contact with water spread through the nozzle.

The water turbine included in the circut as shown in figure is to recover some of the pressure and elevation head between the cooling coils and condenser.

As the circulating water does not come into direct contact with the cooling air, therefore, there is no evaporative loss of water as in the wet type cooling tower.

This system of cooling is always referred to as Heller system as this concept of indirect system of

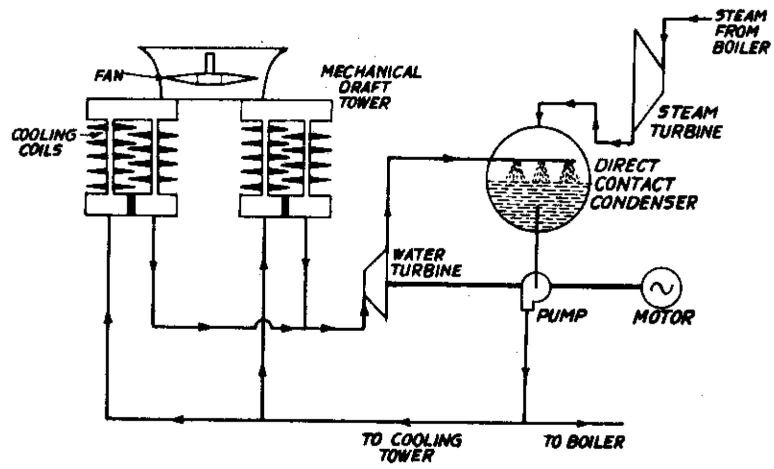


Fig. 18.27. Indirect dry-type cooling tower condensing system with mechanical draft tower.

condensation by air was first pressented by Lazlo Heller, Prof. of the Technical University of Budapest (Hungary) in 1956 at Vienna world power conference.

Direct Dry Type Cooling System. The condensation of steam coming out of turbine is effected with the use of atmospheric air instead of water as in case of once through cooling system.

The principle components of this system are shown in Fig. 18.28. The turbine exhaust is passed through the air-cooled coil as shown in figure. The air is passed over the finned coil surfaces with the help of fan and condenses the steam. The condensed steam is collected in the condensing headers and then it is passed to the boiler with the help of feed pump.

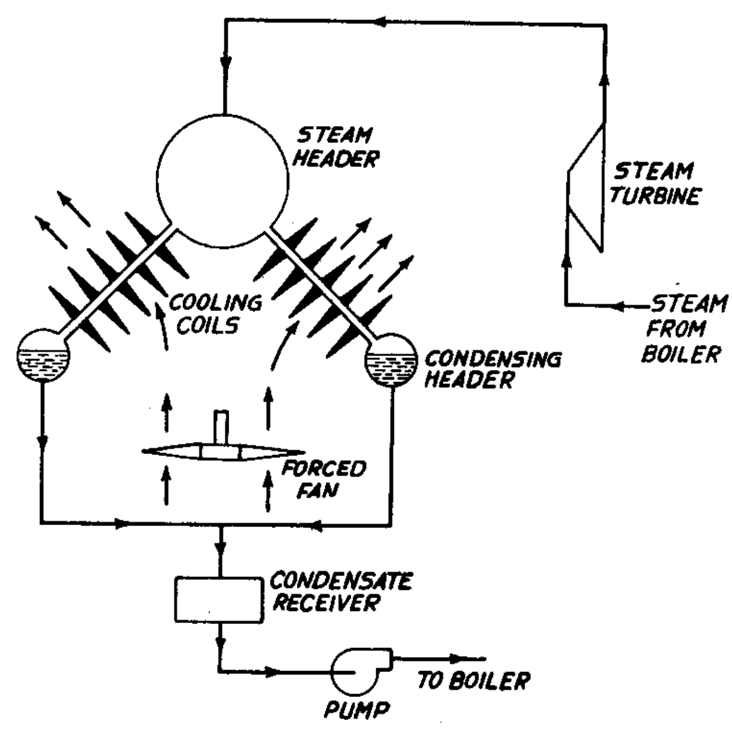


Fig. 18.28. Direct dry-type cooling system.

The Wyodak steam electric power plant near Gillette, Wyo, is one of the wonders in the power industry which uses air-cooled condensers for its 330 MW capacity plant. The plant was commissioned in June 1978. The plant would require about 4000 gpm of make-up water if it should have used an evaporative cooling towers. The air-cooled condensers reduced this requirement to about 300 gpm reducing the plant water need by 92.5%.

The air-cooled condensers cover an area equal to a city block and towers five-stories above the ground. It operates like a car radiator. Steam from the turbine exhaust is distributed to finned carbon steel tubes which are grouped in rectangular bundles and are installed in frame modules above the air circulation fans. A fan deck is 112 m long × 58 m wide and is situated 20 m above ground level. This height was required to admit 45 million cfm of air needed to condense 0.95 million kg/hr of steam. Twelve rows of fans, 69 in number, each 7 m in diameter, force air over 1-million m^2 of finned tube surface, condensing the steam into water.

The water was very much scarce at the site, a water treatment plant is installed to handle the effluent from the city of Gillette's. An average of 1100 gmp of effluent, pumped 4-miles from Gillette, is treated to provide water for boiler make-up auxiliary cooling and fire protection. The treatment consists of chlorinating, softening, filtering, solid removal by demineralization.

The principal difference between the direct and indirect dry type cooling systems, the direct system has to handle very large volume of steam compared with the smaller volume of circulating water in the indirect system.

The use of direct type cooling system is limited to the unit turbine size of 200 MW capacity ; whereas the indirect type can be used to all unit sizes.

Economics of Dry Cooling System. The purpose of dry cooling system is not to replace the conventional one or wet cooling tower system but to provide a means of cooling where adequate make-up circulating water is not available at all or available at a very high cost.

The capital cost of dry-cooling system is considerably higher than conventional evaporative cooling tower system for the same duty, therefore the adaptability of dry cooling system to a particular power plant requires a considerable economic study. The factors to be taken into account for economic analysis are the cost of fuel, the cost of make-up water, alternative available sites, water pumping charges and the cost of transmitting the power to the load centre.

Owing to the freedom in the site selection permitted by the use of dry cooling towers, their use can bring a considerable overall economic advantage in spite of the first cost.

The greater flexibility in power plant siting which is offered by dry cooling system may make possible power transmission saving and fuel transport cost which would offset all of the cost difference between wet and dry systems. Its use may permit an additional generating unit to be built at an existing station though there is no adequate water for an additional wet cooling tower. This would permit the utility to realise the economics of an additional unit at an existing facility.

The cause for adopting the dry cooling system for the new power plant is not one merely of shortage of cooling water but it is necessary to consider whether it is cheapter to build near the fuel source or load centre rather than the availability of cooling water to be the determining factor in the site selection.

Some electric generating power plants with dry type cooling towers in operation are listed below.

Location	*Rating in MW*	*Year of Commissioning*
Rome (Italy)	2-30	1957
Cologne (Germany)	28	1958
Rugbey (England)	120	1962
Quetta (Pakistan)	7-5	1964
Ludwigshafen (Germany)	38	1966
Ibbenburen (Germany)	150	1967
Wyodak (USA)	22	1969
Gyongyos (Hungary)	2-100	1969
Utrillas (Spain)	160	1970
Razdan (Russia)	3-220	1972

18.9. DRY-WET COMBINED COOLING TOWERS

Several different types of closed cycle cooling systems have been proposed to meet the increasing needs of the power plants within the constraints imposed by water resources limitations and environmental restrictions. Whatever be the situation, the need for enlarged make up water supplies for electric power generation is a growing problem to the power industry and, therefore, the answer must be to design cooling systems that make the most efficient use of the available cooling water. One of the most promising of these is the dry-wet combination cooling system.

The combination of dry and wet cooling water is a new approach to reduce the water requirement in the cooling circuit and simultaneously reduce the working cost of the cooling system (Rs/kWh). A 1000 MW nuclear power plant operating at rated load using all-wet system requires about 20 million gallons of make-up water each day (0.83 gallons/kWh). Therefore the capital cost of water saving with wet-dry system compared with all-wet or all-dry system is frequently an important consideration.

A dry cooling system (only air-cooled) is divided as 'Indirect' and 'Direct' cooling system as mentioned in previous article. In 'direct' system, the exhaust steam from the turbine is condensed directly in the air-cooled condenser. In 'indirect' system, the exhaust steam from the turbine is first condensed with the use of cooling water and heated cooling water is cooled with the help of air (without coming in contact as in heat exchangers) and then it is recirculated back to condenser. Direct and indirect air-cooled cooling systems do not work efficiently during hot summer and are incapable to the fluctuating loads. Conventional wet type system does not work efficiently during cold weather. Power plant cooling capacity employing evaporative type wet towers can be supplemented with dry towers (air-cooling). A new condenser water cooling system is proposed to use both conventional wet towers and conventional dry-towers to meet every conceivable make-up water situation from 100% wet to 100% dry. The wet-tower is designed to do its greatest work during the summer while the dry tower does its greatest work during cold weather. The capability of these two cooling systems is combined to condense the turbine exhaust steam in duel-service condenser. The net result is a new cooling type system that can use conventional low exhaust pressure turbines and make reduced make-up water situation that a utility faces. It will use every available gallon of water for evaporative cooling at the higher ambient temperature conditions and discharge the remaining waste heat directly into the atmosphere as sensible heat. In this way, dry-wet cooling system combines low water use features of dry systems and high cooling efficiency feature of wet system in any operationally flexible combination. Any type of make-up water supply to the wet tower can be used as potable, blackish or sea water.

There are many different configurations into which dry-tower and wet-tower can be arranged to form a combined dry-wet cooling tower system. Dry-wet cooling tower systems are mainly classified as parallel path and serious path towers according to their air side design.

Arrangement of Dry-Wet Towers. In parallel path towers, the air flows in parallel streams through the dry and wet sections. This results in a system with a relatively low air pressure drop. Both sections are benefited thermally since each receives cool ambient air. The air steams coming out from wet and dry sections may remain separate as shown in Fig. 18.29 (*b*) or mix as shown in Fig. 18.29 (*a*).

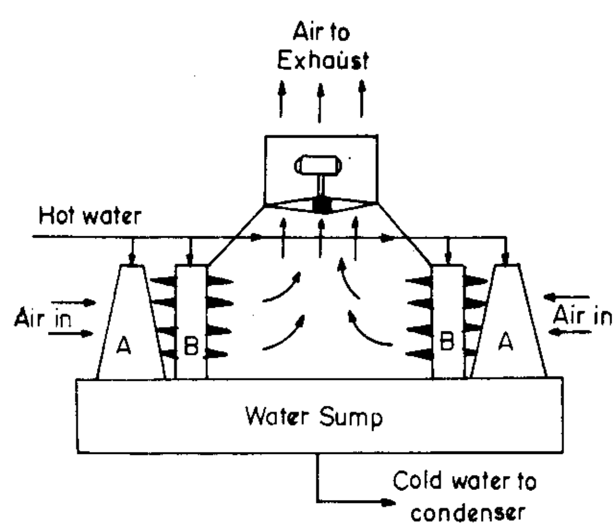


Fig. 18.29. (*a*) Conventional Wet Type.

Combination of dry-wet towers will optimize with larger dry sections in colder climates while higher wet sections will be required in warmer climates. The dry-only and dry-wet modes of operation are more often attractive in colder climates than in warmer and consequently the water evaporation and fogging magnitudes are lower. Furthermore, for a dry-wet tower of a given size, the energy and fuel consumption are reduced in colder climates as compared with warmer climates.

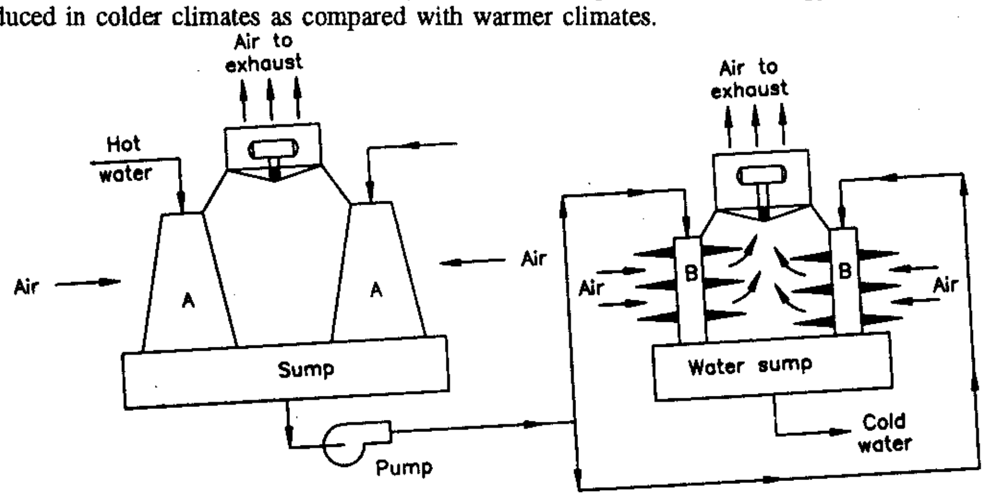


Fig. 18.29. (*b*) Conventional Dry Tower.

The water evaporation in wet tower and fogging potential are closely related. It is possible to constrain the operation of the dry-wet tower to maintain minimum water loss and prescribed level of fogging.

Increassing the dry surface area achieves more economical cooling since the fogging magnitude limit is not violated. Thus the total cost of the system drops as the dry surface area is made larger until increasing capital costs offset the decreasing operating costs. Generally, imposition of severe fogging constraints make combination tower with larger dry-sections and higher capital costs economically superior to wet towers.

The selection of basic wet section size depends upon the prevailing meteorological conditions, the optimum size of the dry section to be added to it depends primarily upon the water cost. The increased capital cost of the combination towers is then compensated for by the lower operating costs resulting from water and fuel conservations.

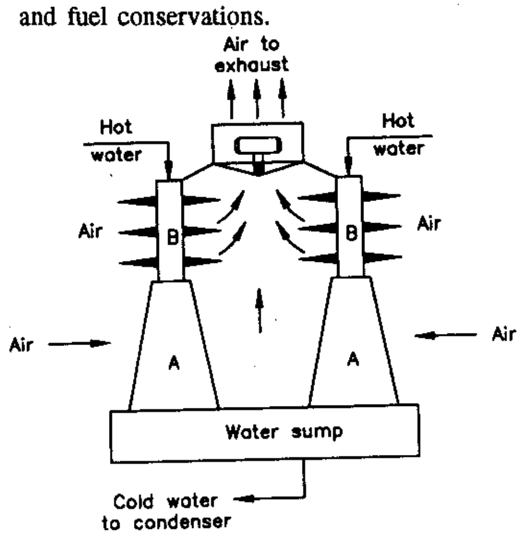


Fig. 18.30. (*a*) Arrangement of wet-dry tower with parallel flow of water and series flow of air.

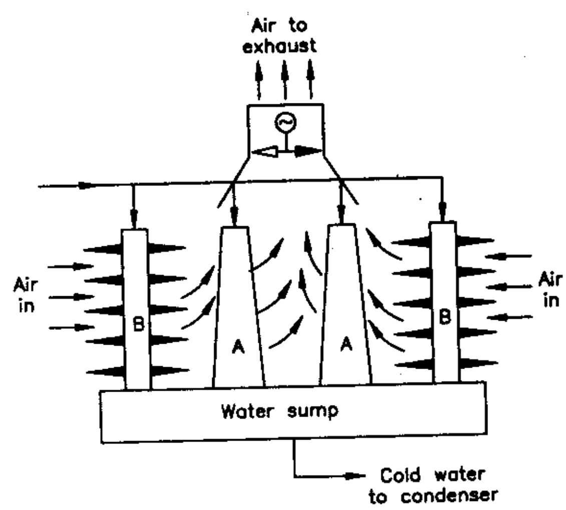


Fig. 18.30. (*b*) Arrangement of dry-wet tower with parallel flow of water and series flow of air.

The air flow rate through dry-tower sections is the most important single factor influencing the economic and thermodynamic performance of various dry-wet cooling tower arrangements. This factor explains the difference in performance between the configurations shown in Fig. 18.30 (*a*) and Fig 18.30 (*b*) as well as the difference between parallal path and serious path configurations. In general, the configuration shown in Fig 18.30 (*b*) is economically superior to that with combined sections except when water conservation or fog abatement is of overriding importance. Parallel path towers are generally far more economical than series path towers.

The dry-wet parallel flow towers currently manufactured appear to offer significant water saving but at a higher capital cost than mechanical draft wet towers with similar capacity. If the dry sections are not sized larger enough, the water saving and fog abatement are achieved at the expense of higher cold water temperatures coming out of tower and therefore poor power plant efficiencies.

Other two arrangements of series and parallel are discussed below. A series arrangement of dry-wet cooling system is shown in Fig. 18.31. The water coming out of condenser is first passed through dry-cooling tower, where it is cooled by passing through a finned heat exchanger with the help of air. Then the water coming out of dry tower is passed through wet cooling tower as shown in figure where it is cooled in the direct contact of air by evaporative cooling. The water collected in the basin is supplied to the condenser with the help of pump.

Another parallel arrangement of dry-wet system is shown in Fig. 18.32. The load of condenser (condensing of steam) is partly taken by the dry tower and partly by wet tower. In the right-hand circuit, the water used for condensing steam is cooled in an indirect air-cooler (dry-system) and cooled water is again circulated through condenser with the help of pump. In the left-hand circuit, the water used for condensing the steam is cooled in a direct current of air (wet-system) and cooled water collected in the basin is again circulated through the condenser with the help of pump.

In both systems, the water evaporated by the wet-section of the system is compensated from the external source.

The economy of the combined system lies in the capital and working cost. In addition to this, the combined system conserves the water required for the power plant. This system allows the flexibility for locating the power plant at any required place where there is a shortage of water.

The estimated costs of all-wet, all-dry and combined wet and dry systems are tabulated below for the condenser load of 2×10^9 kJ/hr when ambient conditions are 35°C DBT and 24°C WBT where tower approach is 9.5°C.

Middle Town, USA (Location of the Plant)

Costs	*100% Wet*	*100% Dry*	*Dry-Wet*	
			50% wet & 50% dry	*70% wet & 30% dry*
Direct cost \$ 10^6	21.356	87.398	53.39	41.64

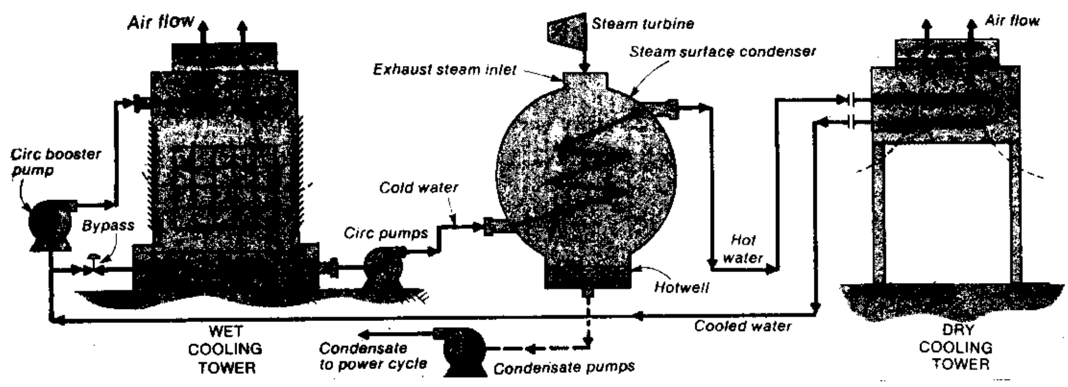


Fig. 18.31. Series water flow, with all water passing first through the dry tower, then the wet tower, is one possible arrangement.

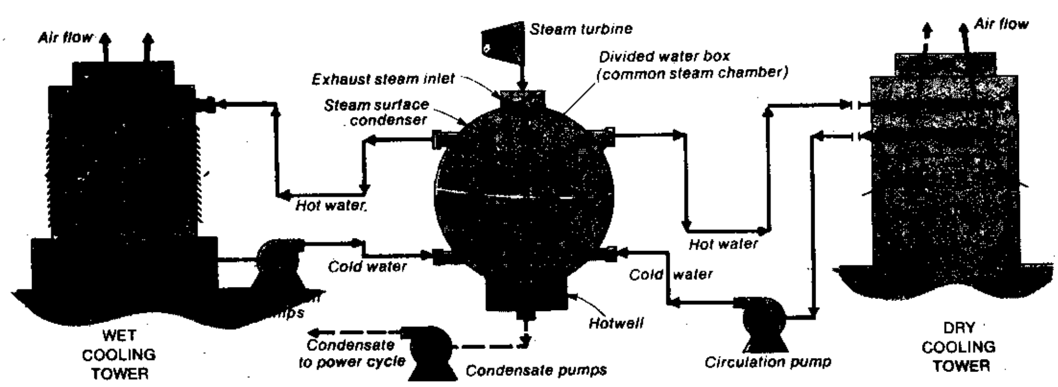


Fig. 18.32. Parallel water flow, an alternative method, separates the flows to each tower, requires a divided-waterbox condenser.

Location of Cooling Tower. The following points should be carefully considered when locating the cooling tower.

(1) The tower site should be such that it allows unrestricted air flow to the tower. As much open space as far as possible should be allowed between the cooling tower louvers and nearby structures.

(2) It is always advisable to place a cooling tower away from office building, laboratory and control room in the plant because of the possibility of the moist air and noise of the tower causing inconvenience to the working people.

(3) Prevailing wind direction should be studied so as to minimize their circulation effect of hot air from the tower which should be carried away from the tower and should not re-enter through the louvers.

(4) The tower should be so located that the piping runs to and from it should be minimum. The tower should be as near as possible to the source of make-up water.

(5) The placement of the tower should also be studied with reference to nearby chimney and effluent from processes, particularly if these are hot or oily.

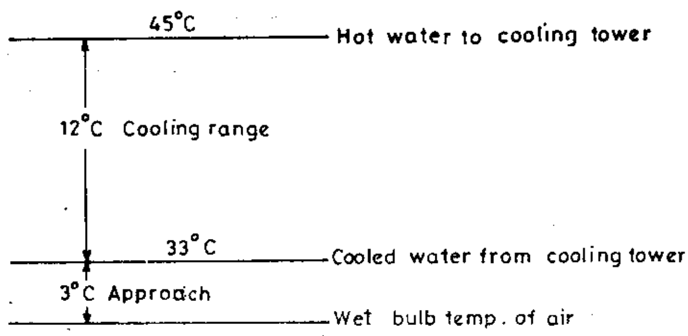


Fig. 18.32 *(a)*.

Advantages and Disadvantages of Dry-Wet Cooling Systems

Advantages. 1. Dry-wet towers offer the possibility of siting the power plants at some locations where adequate water supply for only wet towers may not be available. This flexibility in plant location allows plant to be built nearer to the fuel sources or areas of high power demand thereby allowing savings in fuel transportation or power transmission costs.

2. Dry-wet cooling system takes the advantage of low water use features of dry systems and high cooling efficiency feature of wet system.

3. The flexibility in operating the system in various modes such as dry only, wet-dry or dry-wet would enable system operators to take an advantage of differing meteorological conditions at the site.

4. The dry-wet cooling system is envisaged as a system that would incorporate the high heat rate reflection potential and consequently low turbine back pressure of wet system and yet would not result in the high water losses and other problems of wet system as fogging.

5. Some electrical utilities would like to extend existing facilities but do not have the extra additional cooling water supplies available at the site. The configurations shown in Fig 18.30. *(a)* or Fig. 18.30 *(b)* can be built just by adding the dry-section in the existing wet cooling systems without many changes and the capacity of the existing system can be economically increased. There is a fan for each dry and each wet section as shown in Fig. 18.30 *(a)* and, therefore, by shutting of the fans in either the dry section or the wet section or by re-routing water flows, three modes of operation (dry only, wet only, dry-wet combined) can be realised.

6. Just by changing the positions of the towers in each section for the configurations shown in Figs. 18.30 *(a)* and 18.30 *(b)* determine the operation of the system in all these modes and thereby increase the operating flexibility. Therefore, it is more easy to shift the system from one combination (say wet) to other (say dry) just by operating the louvers in different sections.

Disadvantages. Dry-wet towers have some of the drawbacks associated with both dry as well as wet towers.

1. The capital costs of dry-wet towers are higher than similar duty wet towers.

2. There are start-up and shut-down problems as freezing is commonly associated with dry towers and therefore special arrangements are required to avoid this.

3. There are still some water consumption, fogging and related problems similar to wet towers.

4. To evaluate the proposed combination of dry-wet cooling system with regard to the environmental consequences, it is necessary to calculate water consumption and fogging magnitude at all meteorological conditions that can occur at proposed site. Both capital and operating costs are usually estimated by assuming a set of 'fixed' design conditions, expected to prevail for small fraction of the working period. Such *fixed conditions* designs would not realistically incorporate the advantages of inherent flexibility of dry-wet tower and would not adequately evaluate operating expenses that prevail at all of the *off-design* meteorological conditions.

Steady increasing fuel costs, constrains on permissible environmental inputs and limitations on water availability in addition to the national objective of efficient use of fuel and water make it increasingly necessary to optimize the whole heat rejection system of the power plant and to make dry-wet cooling system more economical than wet system alone.

18.10. WATER DISTRIBUTION SYSTEMS USED IN COOLING TOWERS

The water distribution system in cooling tower should distribute the water uniformly over the packing in the tower. The water droplets formed should be minimum in diameter because it exposes maximum surface area for the given water quantity. But simultaneously, the droplet of the water should not be too small because otherwise there is excessive carry-over.

The water is distributed over packing either by gravity flow or under pressure through the nozzles provided.

1. Gravity Flow. This type of distribution is commonly used on cross-flow induced draft tower. The water collecting basins are set at the top of the tower, completely open for inspection and maintenance. Water is pumped in the basin and then it flows through the plastic orifices in the floor of the basin over the packings. The water coming out of each orifice strikes the diffusion deck just below and spreads evenly over the packing. The arrangement of the system is shown in Fig. 18.33.

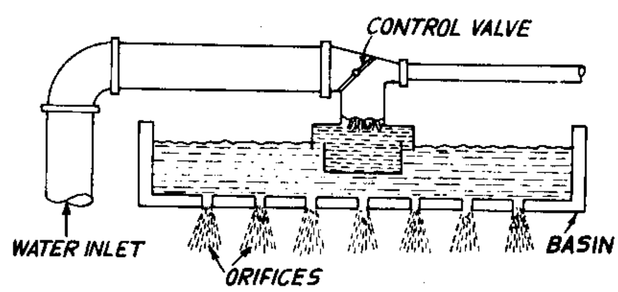


Fig. 18.33. Gravity distribution.

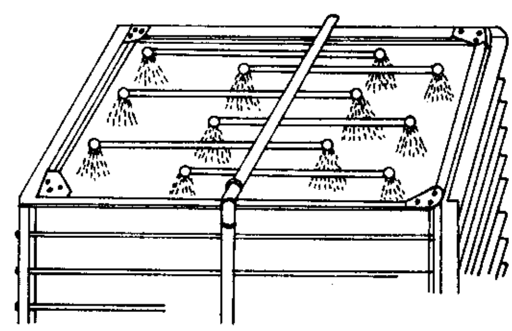


Fig. 18.34. Spray nozzles.

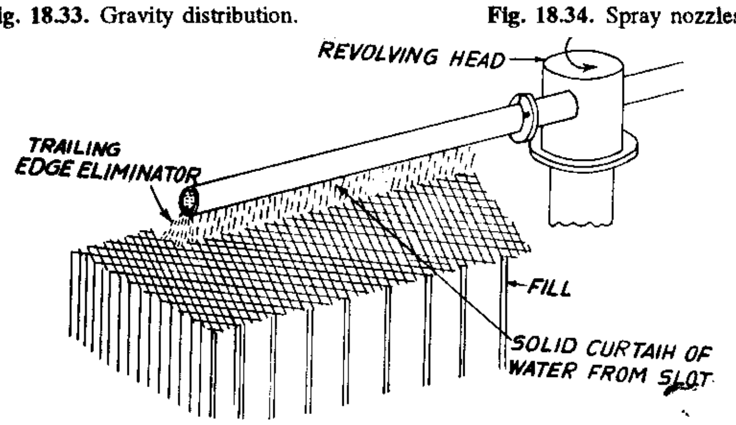


Fig. 18.35. Rotating distribution arms.

2. Pressure System. In forced and induced draft towers, the water is passed through the nozzles under pressure for better atomization. In this system, hollow cone sprays are formed which help to prevent the logging of the nozzle. This type of water distribution gives higher efficiency and better control under variable load conditions. The arrangement of the system is shown in Fig. 18.34.

3. Rotating Distributor System. The arrangement of this distribution system is shown in Fig. 18.35. This is relatively a new concept of water distribution. Two slotted distributor arms (only one is shown in figure) are attached to a central hub and are supported on ball bearings. The reaction force of long thin curtain of water passing under pressure through a slotted opening as shown in figure gives the rotation to the distributor arms. A speed of 30 r.p.m. can be given only with a water head of 60 cm. This can also be varied by adjusting slots to the proper angle. The rotating arms discharge a curtain of water evenly over top of the tower packing.

This gives better efficiencey of tower but it is not in common use due to constructional and operating difficulties.

18.11. PREVENTION OF CARRYOVER LOSSES AND FOG FORMATION FROM COOLING TOWERS

The carryover of a cooling tower is defined as amount of water which is carried away by the air through the top of the tower in form of fine droplets. This loss is generally expressed as percentage of water flowing through the circuit. The water losses by carry-over may vary from 0.2 to 0.6% of the water flow in the circuit and it depends upon the design and type of cooling tower used, the velocity of the air adopted and atmospheric conditions.

The carryover losses are undesirable because :

(1) It does not take any part in heat transfer and cooling purposes therefore it is a mere loss without purpose. The make-up water required also increases with an increase in carry-over loss.

(2) The carry-over loss results in corrosion of fan blades (induced draft type) and causes unbalancing in normal services therefore it is necessary to rebalance the blades after every 3 to 4 years.

(3) The air carries the water in the form of fine water droplets (50 to 100 μ size) and forms the fog around or nearby tower. This fog creates an obstruction of vision along a roadway adjacent to tower. It also forms the ice during cold winter on the walk-ways and stair-ways.

It is so obvious that the carry-over is always undesirable for the reasons mentioned above therefore it is necessary to reduce the carryover losses and to prevent the formation of fog. The carryover losses are considerably reduced by using draft eliminators. Basically, drift eliminators force the leaving air-stream to make a sudden change in direction of flow and therefore the resulting centrifugal force separates the drops of moisture from the air and throws them against the baffle. A thin film of water is formed and flows back into the cooling tower.

The drift eliminators one, two or three baffles depending on the needs of each tower as shown in Fig. 18.36 are used.

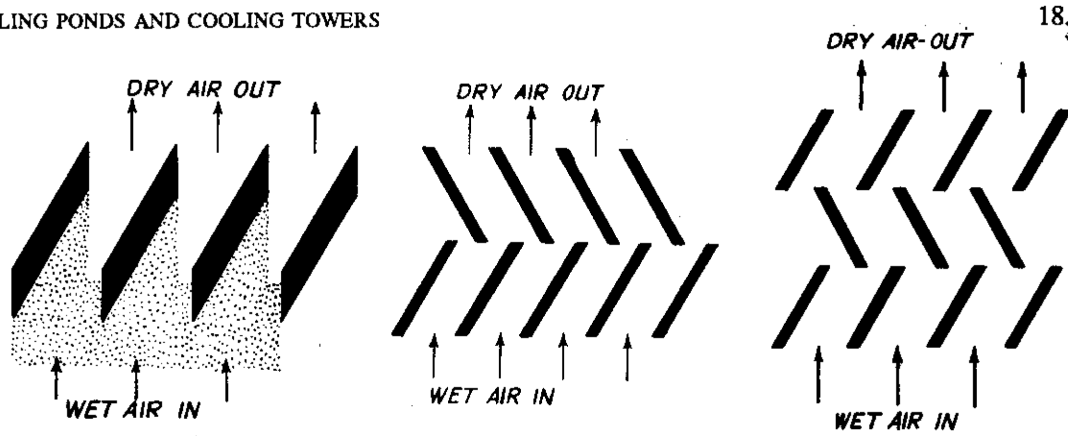
The common materials used for eliminators are treated wood, aluminium and fibreglass.

Fog-formation and its Control. Cooling towers often emit large plumes or fog when operated during cold weather. The plume leaving the tower can create serious visibilty or icing problems.

Figure 18.37 (*a*) shows that as the air coming out of tower is cooled in the atmosphere, the R.H. exceeds 100% and a plume begins to form. Therefore to prevent formation of plume, the R.H. must be reduced so that air is cooled without crossing saturation line.

To reduce the fog formation around the tower, the sensible heat must be given to air before it leaves the tower. This is done by passing the hot water through the tubes and passing the air over them just before leaving the tower or it is heated by gas burners placed inside the stack. Both these methods require energy input.

Figure 18.37 (*a*) shows the cooling of exhaust air to atmospheric temperature after leaving the tower on psychrometric chart. The R.H. exceeds 100% ; as the air is cooled and plume begins to form. This can be avoided by heating the tower exhaust air either by hot water or gas burner as mentioned earlier. This

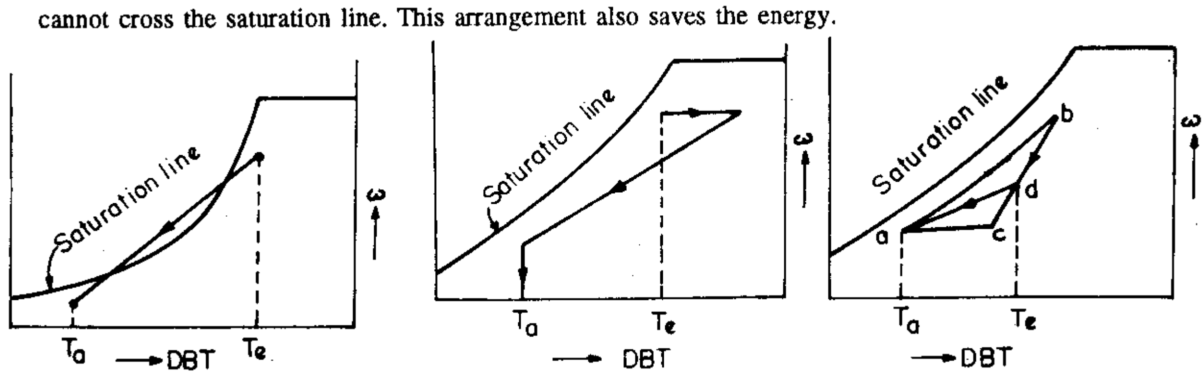


(*a*) Single Row Eliminator. (*b*) Two-Rows Eliminator. (*c*) Three Rows Eliminator.

Fig. 18.36. Different Types of Eliminators.

heating is shown in Fig. 18.35 (*b*). As the cooling of air does not cross the saturation line, there is no danger of plume formation. But this method requires some energy and it costs to prevent the plume formation.

A new approach to eliminate plume formation without increasing operating cost is to incorporate wet and dry cooling in same film fill pack. In such arrangement, warm saturated air represented by point *b* on Fig. 18.37 (*c*) is mixed with dry hot air represented by the point *c*. The condition of the exhaust air is the mixture of air at point *b* and *c*, represented by the point *d*. The line *da* represents the cooling of the tower exhaust air. In this case also, there is hardly any possibility of forming the plume as the cooling line *da* cannot cross the saturation line. This arrangement also saves the energy.



(*a*) (*b*) (*c*)

Fig. 18.37.

T_a = Ambient air temperature.

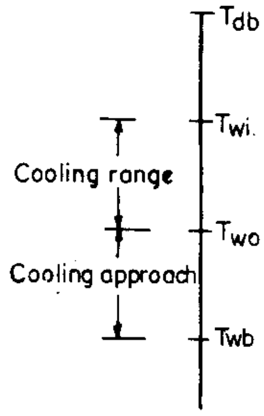
T_c = Exhaust air temperature from tower.

18.12. PERFORMANCE OF COOLING TOWER AND METHODS TO IMPROVE IT

The cooling tower performance is always referred to WBT of the incoming air. This is the lowest temperature that the outgoing water can be cooled. The finite dimensions of a tower and the limited time in which water and air contact with each other make it impossible to achieve this ideal cooling.

The principal performance factor of a tower is its approach to the wet-bulb temperature ; this is the difference between the cooled water temperature leaving the tower and WBT of the entering air. The smaller the approach, the more efficient the tower. Another important performance factor is the cooling range. This is the difference between the hot water temperature entering the tower and cold water temperature leaving the tower. These are represented as shown in Fig. 18.38.

For a given heat load, approach depends mainly on design wet-bulb. cooling range and type of tower selected. Close approach requires a larger and more expensive tower than a long approach. For example, cost of a tower for 3°C approach at a given heat load is about 60 to 70% more than for a 5°C approach tower.



T_{db} = Dry-bulb temp. of entering air.

T_{wb} = Wet-bulb temp. of entering air = 30

T_{wi} = Temp. of water entering the tower = 45

T_{wo} = Temp. of water leaving the tower = 33

Fig. 18.38. Representation of cooling approach and cooling range.

Theoretically, water in the cooling tower can be cooled to WBT of the air. But it is not desirable to cool water to WBT (30°C) when design WBT for tower is specified as 33°C. It is not usually economical to provide an approach closer than 3°C.

Total heat content of the air is related with WBT for given DBT. Higher WBT (or higher R.H.) gives higher heat content. Thus WBT of a particular location will vary with season and the weather conditions. Therefore, it is necessary to select design WBT for an economical tower size. Generally the design WBT is close to the average maximum WBT for summer months at that location.

18.13. COOLING TOWER ENVIRONMENTAL EFFECTS

The hot condenser cooling water is circulated through the cooling tower and cooling is achieved by evaporation and loss of sensible heat as the water falls through the tower packing. A small part of splashing over the packing will be suspended in the air as droplets, some of which are carried out of the tower, as either unevaporated or partially evaporated droplets. These droplets are known as *drift*.

The evaporated water vapour that leaves the tower is partially or totally condensed as it is cooled by ambient air. These condensed water droplets also evaporate as they mix with the ambient dry air forming *water vapour plumes* as shown in Fig. 18.39.

The drift droplets, but not the plume droplets, contain salts which are dissolved in the condenser cooling water. The drift droplets are much larger than the condensed plume droplets and thus fall to the ground much closer to the cooling tower as shown in the figure.

Due to the small fall velocities of the plume droplets, no accumulation of the soft ice is expected on the horizontal surfaces, such as highways. Icing on horizontal and vertical surfaces can occur from the drift and can cause troubles on the highways and aerodromes.

The drift droplets have the same chemical impurities as the circulating water and the deposition of these chemicals on objects in the vicinity of the cooling tower makes it a potential environmental concern. The use of blackish water cooling towers makes the drift problem of more concern because of its effect on vegetation, soil, ground water and as a corrosive agent.

Drift rates are presently controlled by drift eliminators to 0.002 to 0.03% of the total circulating water. Recent experience with tower manufacturers indicates that they will guarantee drift elimination to as low as 0.003% but lower values only at considerable cost.

In recent years, several closed cycle coastal power plant units have been constructed and proposed in USA, primarily because of siting restrictions and environmental regulations. The problem of salt drift

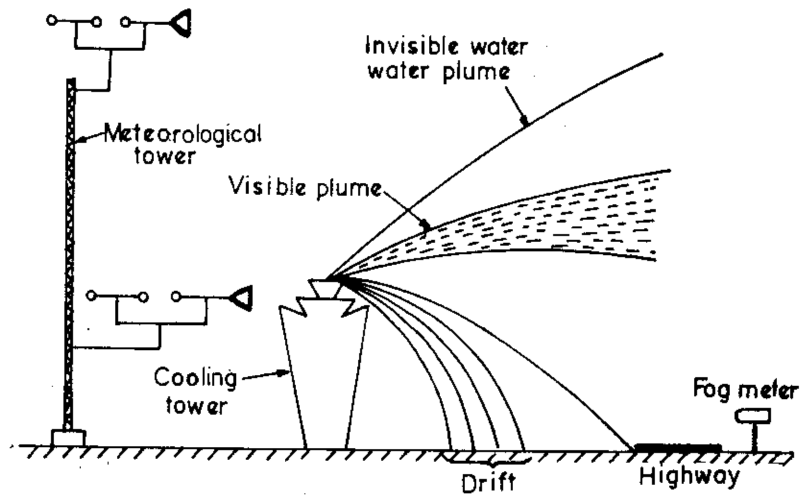


Fig. 18.39. Cooling tower and environmental parameters.

from such power plant is more serious as the sea water contains 2600 to 3600 ppm salt. Assuming a drift rate of 0.002% from cooling towers using sea water as cooling medium in closed system, emits 400 kg of salt per hour if a plant capacity is 2 × 1150 MW, against the permitted emission is only 15 kg/hr. The sodium and chloride salts in the sea water cause injury to vegetation, through soil salinization. Figure 18.40 shows the potential salt drift damage to vegetation from deposition of salt on the ground. The salt drift levels are much higher for mechanical tower compared with natural draft tower.

The transmission interruptions and outages were experienced at stations within a range of 500 m distance between the cooling system and transmission facilities. The salt drift deposition of 2 kg/acre/month is considered critical.

Another major effect of salt drift is the *pothole effect* (damage to concrete structure due to chloride compounds penetrated into the cracks).

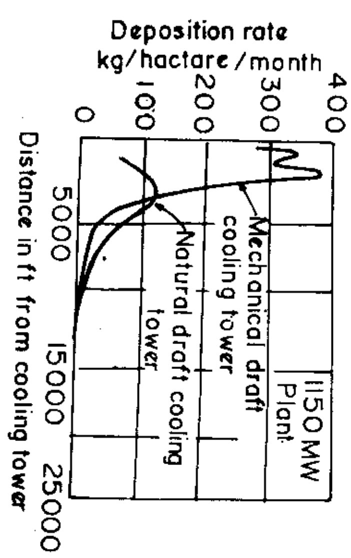


Fig. 18.40.

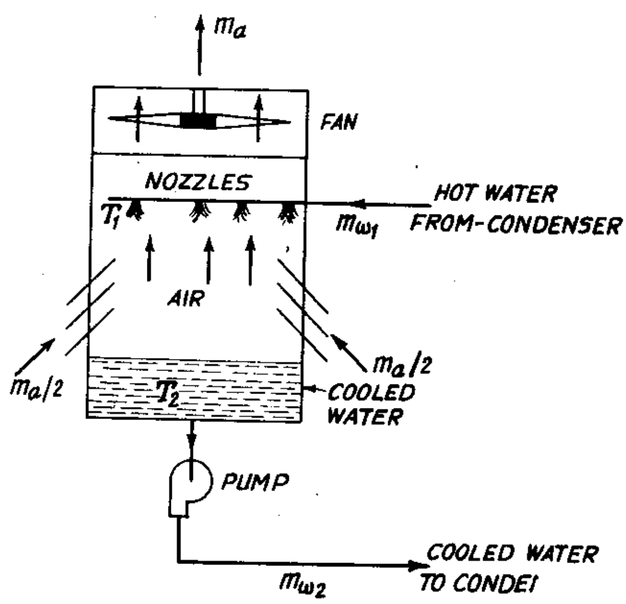


Fig. 18.41.

Atlantic City Electric Co. *England station* power plant in U.S. went on line in 1975 as the first sea water natural draft cooling tower system where 63500 gallons of water per minute are circulated.

18.14. ANALYSIS OF COOLING TOWERS

The arrangement of the components of induced draft cooling tower is shown in Fig. 18.41.

The hot water coming out from the condenser is sprayed into the cooling tower through the nozzles as shown in figure. The air from the atmosphere is drawn into the cooling tower in opposite direction of water with the help of induced fan. The water is cooled by losing the heat to the water evaporated and carried with the air.

The following assumptions are made for the analysis of cooling tower :

1. The system is considered an adiabatic system, therefore total heat lost by the water is equal to the total heat gained by the air.

2. No water particles (unevaporated) are carried with air.

∴ The conditions of air at the entry and exit of the cooling tower are represented on psychrometric chart as shown in Fig. 18.42. The suffix 1 and 2 represent the condition of air as well as condition of water entering and leaving the cooling tower.

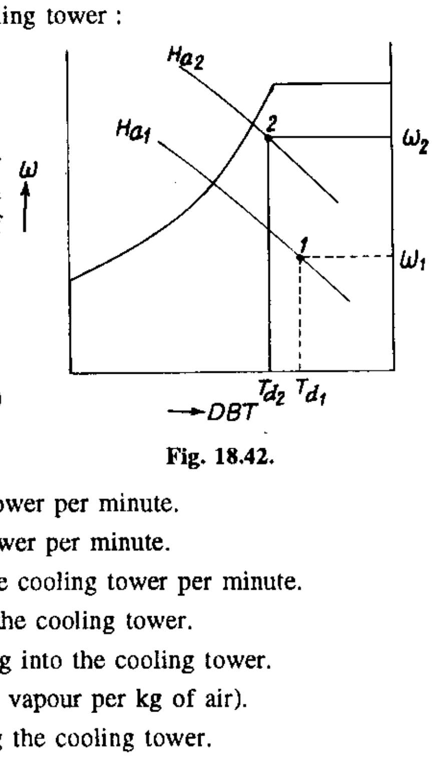


Fig. 18.42.

As per the assumptions,

Heat lost by water = Heat gained by air.

$$\therefore \quad m_{w1}h_{w1} - m_{w2}\, h_{w2} = \frac{V}{v_{s1}}\,(H_{a2} - H_{a1}) \qquad \text{...(18.17)}$$

where

m_{w1} = Mass of water entering the cooling tower per minute.
m_{w2} = Mass of water leaving the cooling tower per minute.
V = Volume of air in m^3 entering into the cooling tower per minute.
v_{s1} = Specific volume of air entering into the cooling tower.
h_{w1} = Sensible heat per kg of water entering into the cooling tower.
ω = Specific humidity of air (kg of water vapour per kg of air).
h_{w2} = Sensible heat per kg of water leaving the cooling tower.
H_{a1} = Enthalpy per kg of air entering into the cooling tower.
H_{a2} = Enthalpy per kg of air leaving the cooling tower.
C_p = Specific heat of water which is unity.

$$\therefore \quad m_{w1}h_{w1} - \left[m_{w1} - \frac{V}{v_{s1}}\,(\omega_2 - \omega_1) \right] h_{w2} = \frac{V}{v_{s1}}\,(H_{a2} - H_{a1})$$

$$\therefore \quad m_{w1}(h_{w1} - h_{w2}) + \frac{V}{v_{s1}}\,(\omega_2 - \omega_1)\, h_{w2} = \frac{V}{v_{s1}}\,(H_{a2} - H_{a1})$$

$$\therefore \quad m_{w1}C_{pw}(T_1 - T_2) + \frac{V}{v_{s1}}\,(\omega_2 - \omega_1) \times C_{pw}T_2 = \frac{V}{v_{s1}}\,(H_{a2} - H_{a1}).$$

$$\therefore \quad m_{w1}C_{pw}(T_1 - T_2) = \frac{V}{v_{s1}} \left[(H_{a2} - H_{a1}) - (\omega_2 - \omega_1)\, C_{pw}\, T_2 \right] \qquad \text{...(18.18)}$$

where T_1 and T_2 are the inlet and outlet temperatures of water and $T_1 > T_2$.

All the required values in the above equations can be calculated with the help of psychrometric chart if initial and exit conditions of water and air are known. The air conditions can be measured by measuring DBT and WBT of air and water conditions can be known by measuring the water temperatures at inlet and outlet.

18.15. WATER TREATMENT SYSTEM FOR COOLING POND AND COOLING TOWER

The cooling water system used for closed system presents special problems for its treatment. Because, the cooling of water in cooling towers is affected by partly evaporating the water and therefore salt concentration in the remaining water goes on increasing. The water coming out of condenser can be effectively cooled, therefore, more rise of cooling water temperatures is allowed in the condenser to reduce the size of condenser. Both situations are in favour of scale formation.

In closed cooling system, the water comes in close contact with air, acid, gases and oxygen as it is passed through the cooling towers. The soluble gases and the air dissolve and concentrate in the recirculating water. In large cities or industrial areas, the air may be highly polluted with acidic gases. The result is highly acidic water in cooling tower which badly corrodes the metal sections in a very short time.

In addition, air borne dirt, debris, algae or slime may clog condensers. These impurities may also cause the water to foam and interfere with proper water recirculation. Therefore the chemical treatment must take care of acid gases and other damaging impurities in the surrounding air.

If enough acid gases are present in the air, the pH of the recirculating water rapidly drops to the range of 3 to 4. Such a low pH means a very corrosive water that can ruin iron and copper equipments in a very short time. Therefore, careful regulation of pH to avoid any acid conditions in recirculating water is absolutely necessary. The desirable pH value of cooling water lies between 7.5 to 8.5. To maintain pH value in a desirable range, a caustic soda is added. The amount added is determined by trial and error rather than by calculation.

As closed cooling system is continuously aerated, therefore it always contains plenty of dissolved oxygen. Since oxygen is the main cause of corrosion in closed cooling system, chemical inhibitors are used to counteract the attack. Chromate is the most effective and most commonly used corrosion inhibitor. A minimum of 200 ppm of $Na_2\,CrO_4$ gives enough corrosion protection. A pH of 6.5 or higher is satisfactory for good corrosion protection when chromate is used as inhibitor.

Polyphosphates, silicates and nitrites are also used as inhibitors to prevent corrosion. Although they are not as effective as chromates, but they are not as expensive as chromates. If cost is the major factor, then they can be used in place of chromate.

Little water (0.1 to 0.2%) is lost in the cooling tower by evaportion and therefore concentration containing dissolved impurities enters into the condenser water circuit to replace the pure vapour which has been evaporated from the cooling tower leaving impurities behind.

If sufficient make-up water supply is available to permit a blowdown equal to or greater than the evaporation from the tower, the trouble of scale formation can be reduced. Continuous blowdown permits much better control of water conditions than intermittent blowdowns. The blowdown controls the level of dissolved solids in the cooling water and thus avoids the possibility of scale formation. The defects of blowdown system used to control the level of dissolved salts is, the chemicals added to the system are also carried with the blowdown which cannot be recovered or refused again.

18.16. ALGAE GROWTH AND ITS PREVENTION

Efficient operation of condenser requires that the water contact surfaces must be cleaned at intervals or preferably they should be kept clean by suitable preventive treatment. Preventive treatment should be used to maintain continuous operation at optimum efficiency.

Cooling water used either in open or closed system invariably contains vegetable matter and micro-organisms known as algae. These organisms grow very rapidly in hot and dump weather as they create their food in the presence of sunlight and CO_2 only. The conditions, water available either at cooling tower or at condenser are highly favourable for the algae growth. Cooling tower systems where sewage effluent is used for make up is also highly conducive for the growth of algae.

A slimy lining of algae is formed on the condenser tube surfaces and it further helps to deposit the suspended particles in the cooling water and reduce the heat transfer efficiency in a short time.

The deposition of the algae on the heat transfer surfaces and in the pipes carrying the water and its further growth in favourable atmosphere not only reduces the heat transfer rates but many times stops the flow of water through the pipes. Such trouble was experienced at one station in England from a fungus-like growth. It was found that the trouble was bacterial and due to the effluent from a sugar beet factory. In some seaboard plants, mussels cause difficulty by accumulating on the walls of intake structures to the point of obstructing water flow and increasing the pumping power. The mussels are killed by water temperature above 40°C. The Redondo power station of Southern California Edison Company takes the advantage of this to kill the mussels and prevents its growth. The treating of cooling water with a small amount of copper-sulphate has proved effective in reducing the growth of green matter and maintaining the condenser tubes free from algae.

The common method which is universally used for killing algae to prevent slime coating of condenser tubes is chlorination of the cooling water. The chlorination has established a satisfactory experience record as effective treatment in controlling the accumulation of biological slimes.

The chlorine treatment consists of injecting liquid chlorine or chlorine compounds into the water. The injecting liquid chlorine or chlorine compound solution combines with water to form a chlorine solution. The solution is fed into the condensing water system, usually at the tower outlet or circulating pump inlet. Nearly all forms of algae are killed by exposure to 1 to 2 parts of free chlorine per million. Intermittent feed of chlorine is usually as effective as a continuous feed and is more economical also.

The chlorination process is also used to sea water system (open type) to prevent mussel growth in pipes and condenser circuits. But it is not always possible to use chlorination on river systems (open system) owing to the fishing rights held by river authorities.

A chlorinating plant is always designed to prevent the accumulation of low heat transfer biological deposits on condenser tube surfaces which arise from the cooling water, thereby ensuring maintenance of optimum vacuum and economy in fuel consumption and permitting the generating plant to run continuously as per operating schedule by obviating the need for periodic shutdown for condenser cleaning.

When algae forms at intervals, it is better to use a shock treatment that kills algae and sterilizes the tower surfaces. Common treatments are copper sulphate and strong chlorine compounds as mentioned above. Both can cause fast corrosion of metal parts and the chlorine will attack wooden parts also. Therefore, when using those shock treatments, it is necessary to do the job quickly and drain the treating liquid and thoroughly wash out the tower and piping to remove all dead algae and treatment residue. Do the job within 12 hours and never leave treatment in overnight.

A line diagram of chlorination plant used for medium size power plant is shown in Fig. 18.43. The diagram is self-explanatory.

Another way of controlling algae and slime is to use inhibitors. Mogul Division of the Dexter Corporation of Ohio has developed a number of inhibitors to be used in the cooling system. They are highly effective in killing and preventing the growth of algae and slimes. These inhibitors are harmless to metal gaskets, packing and valves in the cooling system.

It is well recognised that most chemicals used against algae and slime can accomplish a better kill at a given treatment level in waters of low pH and low hardness.

In general, algae are most resistant in waters of high alkalinity and many chemicals now used form precipitates in high hardness water thus limiting their effectiveness. Mogul microbiocides are less susceptible to these problems. These formulas are compatible with waters of a board hardness range and their extra killing power makes them more effective over a wide alkalinity range.

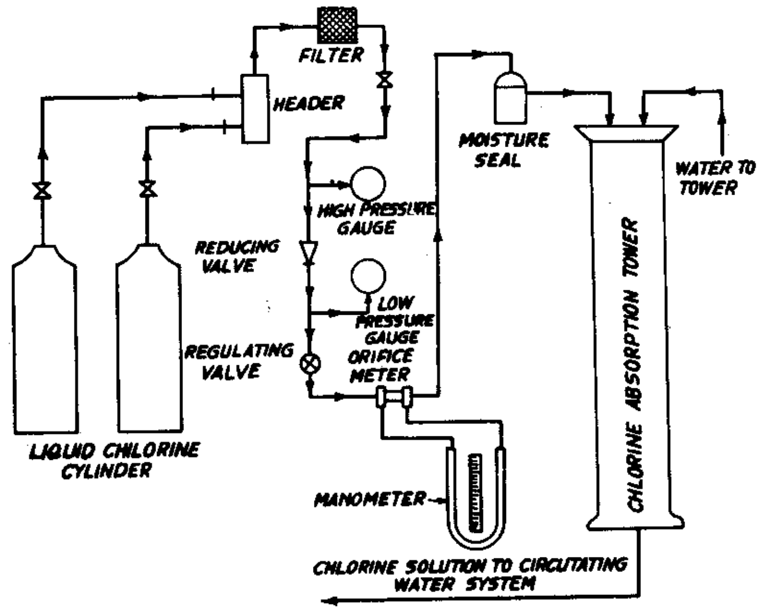


Fig. 18.43. Chlorination plant.

Mogul microbiocides also provide longer and more effective inhibition of algae and slime in water holding systems. They kill most species of micro-organisms effectively. In addition to all the above-mentioned advantages, the cost of these inhibitors are also competitive.

18.17. PROBLEM OF MINERALS CONCENTRATION AND ITS PREVENTION

The evaporation process necessary for cooling the water also acts to concentrate the minerals that are normally present in the water supply. All the calcium and magnesium remain behind when the water evaporates.

It does not take many hours of operation before the mineral concentration becomes so great that it can no longer be held in suspension in the water. Then the minerals begin to separate out in scale form on the nearest available surfaces. This scaling begins first on the warm heat exchange surfaces in the condenser. Scale reduces heat transfer rate and also reduces water flow through tubes and pipes. Therefore, it is necessary to prevent the building of mineral concentration above a particular level.

Constant Bleed. A constant bleed is a valved line from the pump discharge above tower water level to a drain. The idea behind this is to bleed off as much water as you evaporate and carry off the excess minerals in the bleed off water.

All cooling tower manufacturers stress the need for constant bleed in their operating manuals. There are no water supplies available that will not build up scale forming mineral concentrations. In fact there are some water supplies that require chemical treatment in addition to a constant bleed.

When you stop to think about the additional energy it takes to operate at higher head pressures with a scaled condenser and of the great expense to clean or replace condenser when it stops operating, then it does not make any sense to operate without a constant bleed.

Bleed-off. Cooling tower operation manuals contain graphs and tables for the recommended amount of bleed-off from their towers. It is given as a percentage of the circulated water to be bled-off. To use this percentage, one must know the litres/min circulated by the pump. If this is not known, another way to find it is to measure suction and discharge pressure on the pump to find the total head on the pump and then refer to the pump operating curves. It may be found many times, many pumps are operating at negative suction pressures.

It is also necessary to know the cooling range of the tower. Multiply the *gallons* per minute pumped by the percentage of bleed-off for the tower cooling range. The following table (taken from Baltimore Aircoil Manual) gives the bleed-off against the cooling range.

Cooling Range	5	10	15	20	25	30	35	40
Bleed off %	0.005	0.010	0.015	0.020	0.025	0.030	0.035	0.040

Higher cooling ranges (above 15°C) is not used for air-conditioning as 15°C is the highest range allowed in cooling tower. Above this range, the coolers are used for industrial processes and engine heat rejection.

All condenser water is pumped in a closed circuit through a coil in the condenser and that is cooled by open circuit water pumped through the tower and evaporated. Then it is heated by the closed circuit tubes. The mineral concentration is in the tower only. The scale that forms is on the inner surface of the tubes of condenser. This type of scale is much harder to remove and even it cannot be acid cleaned without damaging the tubes. Therefore, a constant bleed is more imperative to save the life of condenser as well as cooling tower.

The effect of constant bleed is shown in Fig. 18.44 and it is also shown by calculation that it maintains the concentration in cooling water constant.

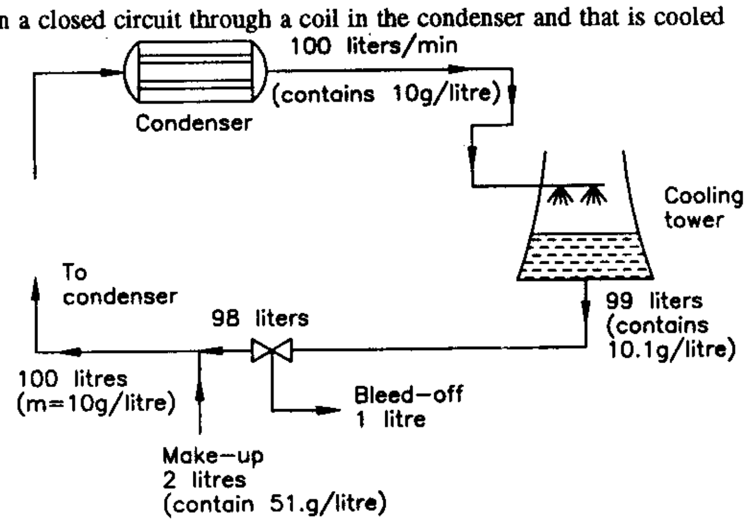


Fig. 18.44.

$$\therefore \quad 100\,m = 98 \times 10.1 + 2 \times 5.1$$

$$\therefore \quad m = \frac{989.8 + 10.2}{100} = 10 \text{ gms}$$

The above Fig. and calculation show that the concentration can be maintained by constant bleed and compensating outer water having low concentration.

SOLVED PROBLEMS

Problem 18.1. *The cooling water used in a power plant consists of 10 big fans. The quantity of cooling water circulated through the tower is 1000 kg per minute and it is cooled from 35°C to 30°C. The atmospheric conditions are 35°C DBT and 25°C WBT. The air leaves the tower at 30°C and 90% R.H. Find out*

(*a*) *the quantity of air handled per fan per minute ; and*

(*b*) *the quantity of make-up water per hour. Neglect the loss due to carry-over and heat losses.*

Solution. The heat transfer between water and air in the cooling tower is given by the following equation

$$m_{w1} C_{pw} (T_1 - T_2) = \frac{V}{v_{s1}} [(H_{a2} - H_{a1}) - (\omega_2 - \omega_1)\, C_{pw} T_2]$$

The conditions of air at inlet and outlet are represented on psychrometric chart as shown in Fig. Prob. 18.1 (*b*).

From psychrometric chart

$$H_{a1} = 76.4 \text{ kJ/kg},\ H_{a2} = 94.5 \text{ kJ/kg}$$
$$\omega_1 = 19 \text{ grams/kg}\ \omega_2 = 24.4 \text{ grams/kg}$$
$$v_{s1} = 0.895 \text{ m}^3\text{/kg}$$

The water temperatures at inlet and outlet are given as

$$T_1 = 35°\text{C and } T_2 = 30°\text{C}.$$

Substituting the values in the above equation

$$1000 \times 4.2\ (35 - 30) = \frac{V}{0.895}\left[(94.5 - 76.4) - \left(\frac{24.4 - 16}{1000}\right) \times 4.2 \times 30\right]$$

$\therefore \qquad V = 1103 \text{ m}^3\text{/min}$

$\therefore$ Capacity of each fan

$$= \frac{1103}{10} = \mathbf{110.3\ m^3/min}$$

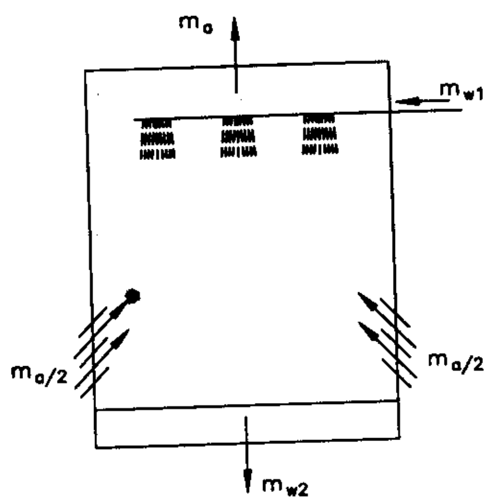


Fig. Prob. 18.1. (*a*)

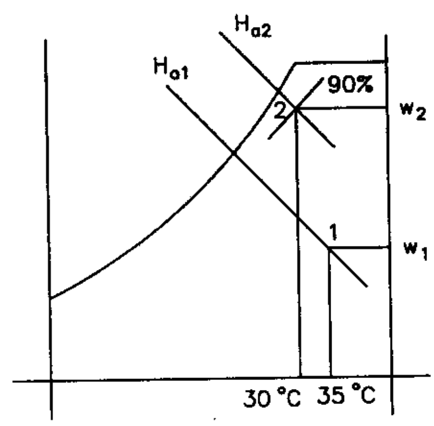


Fig. Prob. 18.1. (*b*)

The quantity of make-up

= loss of water due to evaporation in the cooling tower and carried with air in vapour form

$$= \frac{V}{v_{s2}} \frac{(\omega_2 - \omega_1)}{1000} \times 60 = \frac{110.3}{0.895} \times \frac{(24.4 - 16)}{1000} \times 60 = \mathbf{621\ kg/hr.}$$

Problem 18.2. *A small size cooling tower is designed to cool 400 kg of water per minute. The water enters the tower at 43.5°C. The air enters the tower at 18.5°C and 60% relative humidity, and it leaves the tower at 27°C and in saturated condition. The motor driven fan forces 600 m³ of air per minute through the tower and power absorbed is 4 kW. Find (a) temperature of water coming out of the tower ; and (b) make-up required per hour.*

Solution. The conditions of air entering and leaving the cooling tower are shown in Fig. Prob. 18.2 as represented on psychrometric chart.

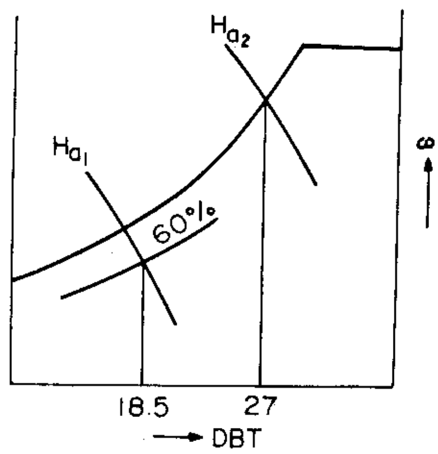


Fig. Prob. 18.2.

Assume that the mass of water and air entering the tower are m_w and m_a then.

Total heat of air at inlet + total heat of water at inlet + heat dissipated by motor = Total heat of air at outlet + total heat of water at outlet.

$$m_a H_{a1} + m_{w1} h_1 + Q$$

$$= m_a H_{a2} + \left[m_{w1} - m_a \left(\frac{\omega_2 - \omega_1}{1000} \right) \right] h_{w2}$$

$$\therefore \quad m_a \left[H_{a2} - H_{a1} - \left(\frac{\omega_2 - \omega_1}{1000} \right) h_{w2} \right]$$

$$= m_{w1} (h_{w2} - h_{w1}) + Q \qquad ...(a)$$

From psychrometric chart

$H_{a1} = 38.87$ kJ/kg, $H_{a2} = 84.85$ kJ/kg

$\omega_1 = 7.8$ grams/kg, $\omega_2 = 22.6$ grams/kg

$v_{s1} = 0.836$ m^3/kg.

$$\therefore \quad m_a = \frac{600}{0.836} = 717 \text{ kg/min}$$

$$Q = 4 \times 60 = 240 \text{ kJ/min.}$$

Substituting the values in the equation (*a*)

$$717 \left[(84.85 - 38.87) - \left(\frac{22.6 - 7.8}{1000} \right) \times 4.2 \times T_2 \right] = 400 \times 4.2\ (43.5 - T_2) + 240$$

$$\therefore \quad 32982 - 444.5\ T_2 = 73080 - 1680\ T_2 + 240$$

$$\therefore \quad 1235.5\ T_2 = 40338$$

$$\therefore \quad T_2 = \mathbf{32.65°C}$$

$$\text{Make-up required} = m_a \left(\frac{\omega_2 - \omega_1}{1000} \right)$$

$$= 717 \left(\frac{22.6 - 7.8}{1000} \right) = \mathbf{10.6\ kg/min.}$$

Problem 18.3. *Calculate the make-up water required for a cooling tower system circulating the water of 5000 m^3/hr. The cooling range is 12°C and allowable concentration ratio is 3.*

Assume that an evaporation loss is 1% of the recirculating water quantity for every 6°C of cooling range and drift and windage loss is 0.2%. The blowdown is given by $B = \frac{E}{C-1} - W$ *where C is permissible concentration ratio (total dissolved solids in the circulating water/total dissolved solids in make-up water).*

Solution. The make-up water (*M*) is given by

$$M = E + W + B$$

where E, W and B are evaporation losses, drift and windage losses and blowdown rate respectively.

Now the given data is

$$E = \frac{2}{100} \times 5000 = 100 \text{ m}^3/\text{hr}.$$

$$W = \frac{0.2}{100} \times 5000 = 10 \text{ m}^3/\text{hr}.$$

$$B = \frac{E}{C-1} - W = \frac{100}{3-1} - 10 = 40 \text{ m}^3/\text{hr}.$$

$$\therefore \quad M = 100 + 10 + 40 = 150 \text{ m}^3/\text{hr}.$$

EXERCISES

18.1. What is the function of a cooling tower in a modern steam power plant ? Describe with a neat sketch the working of a mechanical type cooling tower.

18.2. What is the necessity of cooling the surface condenser water when a closed system is used ?

18.3. What do you understand by mixed type cooling system ? When it is more preferable ?

18.4. Draw a neat diagram of condenser water cooling system which operates as open system during rainy season and operates as closed system during summer season ? What are the economical aspects considered in the design of this type of system ?

18.5. What is the mechanism of evaporative cooling ? When spray cooling ponds are preferred compared with cooling towers ?

18.6. Draw the neat diagram of hyperbolic cooling tower and discuss its merits and demerits.

18.7. Explain the working of 'forced draft' and 'induced draft' cooling towers with neat sketches. Discuss the merits and demerits of each over other.

18.8. Mechanical draft (forced or induced) cooling towers work better in hot and dry weather whereas hyperbolic tower work better in cold and wet weather. Discuss.

18.9. The cooling tower gives better performance at Koradi Thermal Station at Nagpur than the cooling tower of the same capacity at Tata Thermal Power Station at Bombay during summer season. Discuss.

18.10. Explain the different methods of water distribution system in cooling towers with neat sketches. Discuss the merits and demerits also.

18.11. What do you understand by carry-over losses ? Explain the different methods used to reduce these losses.

18.12. Why air-cooled (or dry-type cooling) systems are preferred over water-cooling system in modern power plants ?

18.13. Explain the difference between indirect type dry-cooling system and direct type dry-cooling system with neat sketches.

18.14. Prove that the quantity of air circulated through a cooling tower for the cooling of given water through a given temperature when air inlet and outlet conditions are known is given by

$$\frac{V}{v_{s1}}\left[(H_{a2} - H_{a1}) - (\omega_2 - \omega_1)\, C_{pw} T_2\right] = m_{w1} C_{pw}\,(T_1 - T_2)$$

where $T_1 > T_2$

18.15. Prove that the mass of air circulated through an evaporative condenser for taking the condenser load is given by

$$m_a = \frac{m_w(h_{w2} - h_{w1})}{[(H_{a2} - H_{a1}) - (\omega_2 - \omega_1)\, h_{w1}]}$$

18.16. What are the causes of corrosion and scale formation is condenser tubes ? Discuss the different methods used for their prevention.

18.17. Why the scale formation is a more serious problem in closed cooling system compared with once through system ? Discuss the different methods used to reduce the scale formation and resistance to corrosion.

18.18. Why an algae growth is major problem in closed cooling system ? What factors are responsible for its growth ? Explain the common method used with neat sketch to prevent the algae growth in the cooling system.

18.19. A 900 kg of water is cooled in an induced draft cooling tower from 35.5°C to 30.5°C. Determine the amount of air to be handled and quantity of make-up water required. Air enters the tower at 35°C DBT and 25.5°C WBT and leaves the tower at 32.2°C DBT and 90% saturated. Assume there is no entertainment loss.

18.20. In a forced draft cooling tower, 1000 kg of condenser water per minute is cooled from 26°C to 12°C. Air enters into the tower at 15°C DBT and 55% R.H. and leaves the tower at 20°C and in saturated conditions. Find out the quantity of air supplied in cu.m. per minute and make-up water per hour.

18.21. 5000 kg of condenser cooling water is supplied to the forced draught cooling tower at 40°C and is cooled to 3°C of approach temperature while falling through the tower. The air entering the tower is at 38°C DBT and 25°C WBT and leaves the tower at 38°C DBT and 50% R.H. Find out the quantity of air in cu.m. per minute handled by the fan. Also find out the make-up water required.

18.22. Find out the loss in kg/hr due to evaporative cooling in a cooling tower when the water supplied is 2000 kg/min. The water is cooled from 92°C to 47°C. Air enters the tower at 27°C DBT and 21°C WBT and leaves the tower at 57°C DBT and 95% saturated.

□□□