Lesson
22
Condensers & Evaporators

The specific objectives of this lesson are to:

1. Discuss general aspects of evaporators and condensers used in refrigeration systems (*Section 22.1*)
2. Introduce refrigerant condensers (*Section 22.2*)
3. Classify refrigerant condensers based on the external fluid used, based on constructional details etc. (*Section 22.3*)
4. Compare air cooled condensers with water cooled condensers (*Section 22.3.4*)
5. Present analysis and design aspects of refrigerant condensers, estimation of heat transfer coefficients on external fluid side on refrigerant side for different configurations (*Section 22.4*)
6. Discuss briefly the effect of presence of air and other non-condensible gases in refrigerant condensers (*Section 22.5*)
7. Discuss briefly the concept of optimum condensing pressure for lowest running cost of a refrigeration system (*Section 22.6*)

At the end of the lecture, the student should be able to:

1. Classify and describe refrigerant condensers based on the external fluid used, based on the external fluid flow and based on constructional aspects
2. Compare air-cooled condensers with water-cooled condensers
3. Perform condenser design calculations using various correlations presented for estimating heat transfer coefficients on external fluid and refrigerant side and estimate the required condenser area for a given refrigeration system
4. Explain the effect of presence of non-condensible gases on condenser performance
5. Explain the concept of optimum condenser pressure

22.1. Introduction:

Condensers and evaporators are basically heat exchangers in which the refrigerant undergoes a phase change. Next to compressors, proper design and selection of condensers and evaporators is very important for satisfactory performance of any refrigeration system. Since both condensers and evaporators are essentially heat exchangers, they have many things in common as far as the design of these components is concerned. However, differences exists as far as the heat transfer phenomena is concerned. In condensers the refrigerant vapour condenses by rejecting heat to an external fluid, which acts as a heat sink. Normally, the external fluid does not undergo any phase change, except in some special cases such as in cascade condensers, where the external fluid (another refrigerant) evaporates. In evaporators, the liquid refrigerant evaporates by extracting heat from an external fluid (low temperature heat source). The external fluid may not undergo phase change, for example if the system is used for sensibly cooling water, air or some other fluid. There are many refrigeration and air conditioning applications, where the external fluid also undergoes phase change. For example, in a typical summer air conditioning system, the moist air is dehumidified by condensing water vapour and then, removing the condensed liquid water. In many low temperature refrigeration applications freezing or frosting of evaporators takes place. These aspects have to be considered while designing condensers and evaporators.

22.2. Condensers:

As already mentioned, condenser is an important component of any refrigeration system. In a typical refrigerant condenser, the refrigerant enters the condenser in a superheated state. It is first de-superheated and then condensed by rejecting heat to an external medium. The refrigerant may leave the condenser as a saturated or a sub-cooled liquid, depending upon the temperature of the external medium and design of the condenser. Figure 22.1 shows the variation of refrigeration cycle on T-s diagram. In the figure, the heat rejection process is represented by 2-3'-3-4. The temperature profile of the external fluid, which is assumed to undergo only sensible heat transfer, is shown by dashed line. It can be seen that process 2-3' is a de-superheating process, during which the refrigerant is cooled sensibly from a temperature T_2 to the saturation temperature corresponding condensing pressure, $T_{3'}$. Process 3'-3 is the condensation process, during which the temperature of the refrigerant remains constant as it undergoes a phase change process. In actual refrigeration systems with a finite pressure drop in the condenser or in a system using a zeotropic refrigerant mixture, the temperature of the refrigerant changes during the condensation process also. However, at present for simplicity, it is assumed that the refrigerant used is a pure refrigerant (or an azeotropic mixture) and the condenser pressure remains constant during the condensation process. Process 3-4 is a sensible, sub cooling process, during which the refrigerant temperature drops from T_3 to T_4.

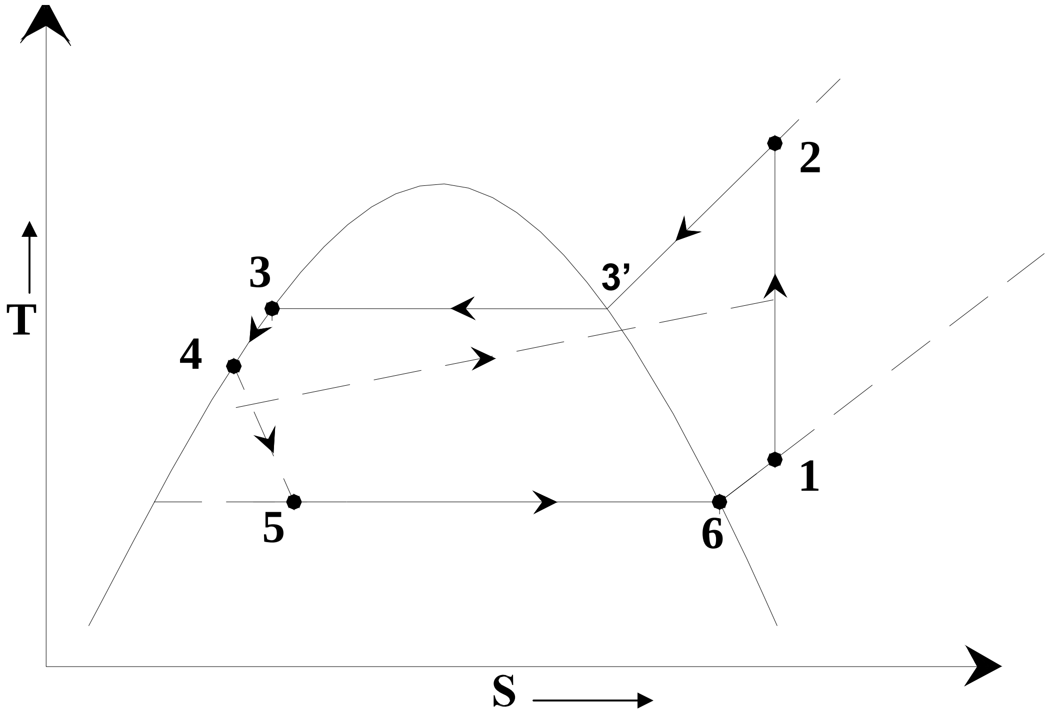


Fig.22.1: *Refrigeration cycle on T-s diagram*

22.3. Classification of condensers:

Based on the external fluid, condensers can be classified as:

a) Air cooled condensers
b) Water cooled condensers, and
c) Evaporative condensers

22.3.1. Air-cooled condensers:

As the name implies, in air-cooled condensers air is the external fluid, i.e., the refrigerant rejects heat to air flowing over the condenser. Air-cooled condensers can be further classified into natural convection type or forced convection type.

Natural convection type:

In natural convection type, heat transfer from the condenser is by buoyancy induced natural convection and radiation. Since the flow rate of air is small and the radiation heat transfer is also not very high, the combined heat transfer coefficient in these condensers is small. As a result a relatively large condensing surface is required to reject a given amount of heat. Hence these condensers are used for small capacity refrigeration systems like household refrigerators and freezers. The natural convection type condensers are either plate surface type or finned tube type. In plate surface type condensers used in small refrigerators and freezers, the refrigerant carrying tubes are attached to the outer walls of the refrigerator. The whole body of the refrigerator (except the door) acts like a fin. Insulation is provided between the outer cover that acts like fin and the inner plastic cover of the refrigerator. It is for this reason that outer body of the refrigerator is always warm. Since the surface is warm, the problem of moisture condensation on the walls of the refrigerator does not arise in these systems. These condensers are sometimes called as flat back condensers.

The finned type condensers are mounted either below the refrigerator at an angle or on the backside of the refrigerator. In case, it is mounted below, then the warm air rises up and to assist it an air envelope is formed by providing a jacket on backside of the refrigerator. The fin spacing is kept large to minimize the effect of fouling by dust and to allow air to flow freely with little resistance.

In the older designs, the condenser tube (in serpentine form) was attached to a plate and the plate was mounted on the backside of the refrigerator. The plate acted like a fin and warm air rose up along it. In another common design, thin wires are welded to the serpentine tube coil. The wires act like fins for increased heat transfer area. Figure 22.2 shows the schematic of a wire-and-tube type condenser commonly used in domestic refrigerators. Regardless of the type, refrigerators employing natural convection condenser should be located in such a way that air can flow freely over the condenser surface.

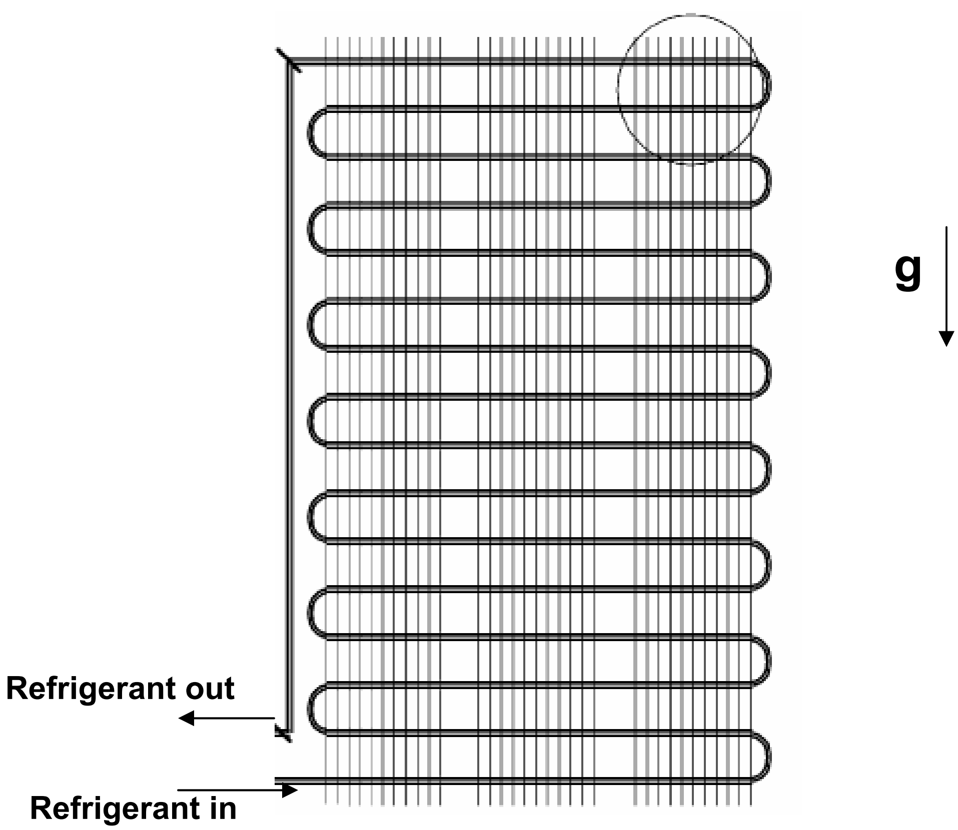


Fig.22.2: *Schematic of a wire-and-tube type condenser used in small refrigeration systems*

Forced convection type:

In forced convection type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. Figure 22.3 shows the schematic of a plate-fin type condenser. Forced convection type condensers are commonly used in window air conditioners, water coolers and packaged air conditioning plants. These are either chassis mounted or remote mounted. In chassis mounted type, the compressor, induction motor, condenser with condenser fan, accumulator, HP/LP cut- out switch and pressure gauges are mounted on a single chassis. It is called condensing unit of rated capacity. The components are matched to condense the required mass flow rate of refrigerant to meet the rated cooling capacity. The remote mounted type, is either vertical or roof mounted horizontal type. Typically the air velocity varies between 2 m/s to 3.5 m/s for economic design with airflow rates of 12 to 20 cmm per ton of refrigeration (TR). The air specific heat is 1.005 kJ/kg-K and density is 1.2 kg/m^3. Therefore for 1 TR the temperature rise Δt_a = 3.5167/(1.2x1.005 x 16/60) = 10.9^{o}C for average air flow rate of 16 cmm. Hence, the air temperature rises by 10 to 15^{o}C as compared to 3 to 6^{o}C for water in water cooled condensers.

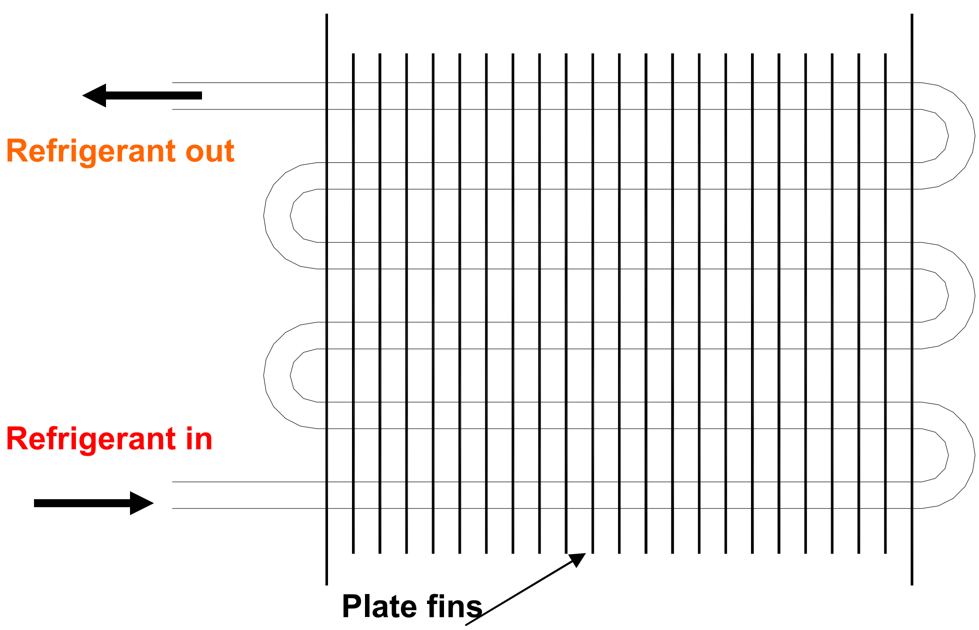


***Fig.22.3**: Forced convection, plate fin-and-tube type condenser*

The area of the condenser seen from outside in the airflow direction is called face area. The velocity at the face is called face velocity. This is given by the volume flow rate divided by the face area. The face velocity is usually around 2m/s to 3.5 m/s to limit the pressure drop due to frictional resistance. The coils of the tube in the flow direction are called rows. A condenser may have two to eight rows of the tubes carrying the refrigerant. The moist air flows over the fins while the refrigerant flows inside the tubes. The fins are usually of aluminum and tubes are made of copper. Holes of diameter slightly less than the tube diameter are punched in the plates and plates are slid over the tube bank. Then the copper tubes are pressurized which expands the tubes and makes a good thermal contact between the tube and fins. This process is also known as bulleting. For ammonia condensers mild steel tubes with mild steel fins are used. In this case the fins are either welded or galvanizing is done to make a good thermal contact between fin and tube. In case of ammonia, annular crimpled spiral fins are also used over individual tubes instead of flat-plate fins. In finned tube heat exchangers the fin spacing may vary from 3 to 7 fins per cm. The secondary surface area is 10 to 30 times the bare pipe area hence; the finned coils are very compact and have smaller weight.

22.3.2. Water Cooled Condensers:

In water cooled condensers water is the external fluid. Depending upon the construction, water cooled condensers can be further classified into:

1. Double pipe or tube-in-tube type
2. Shell-and-coil type
3. Shell-and-tube type

Double Pipe or tube-in-tube type:

Double pipe condensers are normally used up to 10 TR capacity. Figure 22.4 shows the schematic of a double pipe type condenser. As shown in the figure, in these condensers the cold water flows through the inner tube, while the refrigerant flows through the annulus in counter flow. Headers are used at both the ends to make the length of the condenser small and reduce pressure drop. The refrigerant in the annulus rejects a part of its heat to the surroundings by free convection and radiation. The heat transfer coefficient is usually low because of poor liquid refrigerant drainage if the tubes are long.

Shell-and-coil type:

These condensers are used in systems up to 50 TR capacity. The water flows through multiple coils, which may have fins to increase the heat transfer coefficient. The refrigerant flows through the shell. In smaller capacity condensers, refrigerant flows through coils while water flows through the shell. Figure 22.5 shows a shell-and-coil type condenser. When water flows through the coils, cleaning is done by circulating suitable chemicals through the coils.

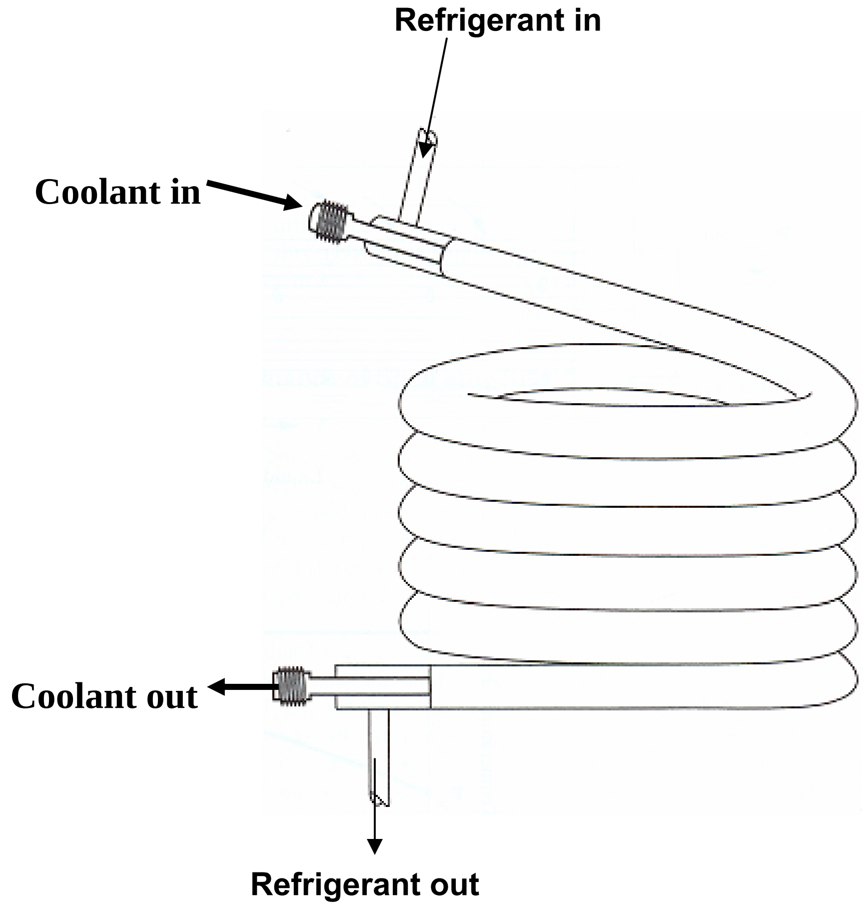


Fig.22.4: Double pipe (tube-in-tube) type condenser

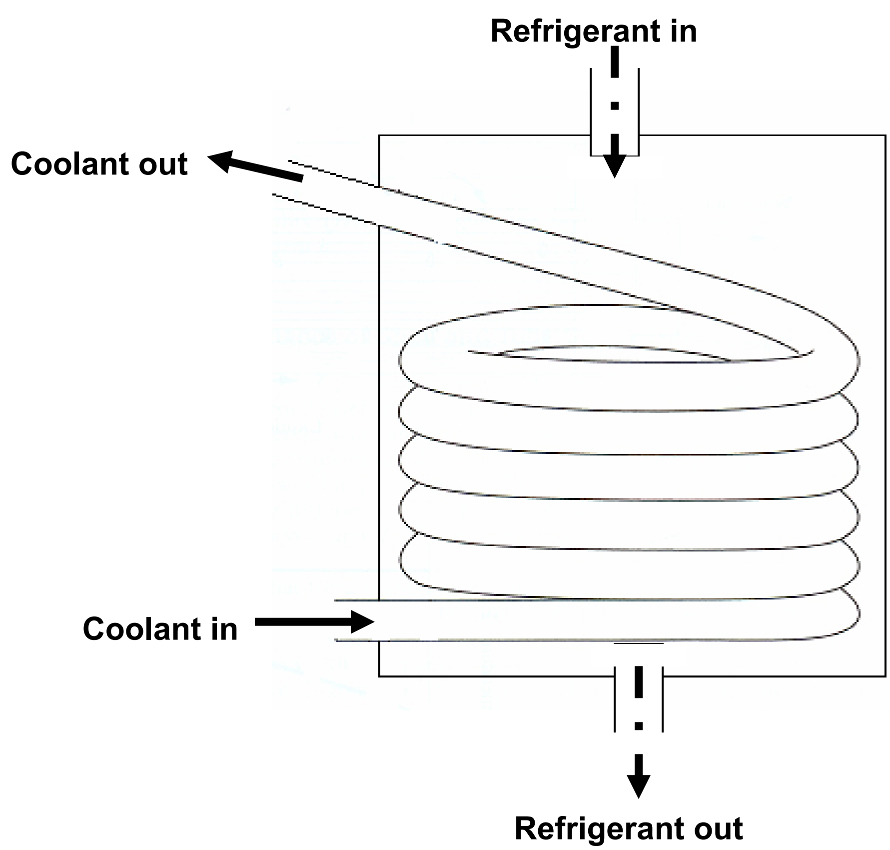


Fig.22.5: Shell-and-coil type condenser

Shell-and-tube type:

This is the most common type of condenser used in systems from 2 TR upto thousands of TR capacity. In these condensers the refrigerant flows through the shell while water flows through the tubes in single to four passes. The condensed refrigerant collects at the bottom of the shell. The coldest water contacts the liquid refrigerant so that some subcooling can also be obtained. The liquid refrigerant is drained from the bottom to the receiver. There might be a vent connecting the receiver to the condenser for smooth drainage of liquid refrigerant. The shell also acts as a receiver. Further the refrigerant also rejects heat to the surroundings from the shell. The most common type is horizontal shell type. A schematic diagram of horizontal shell-and-tube type condenser is shown in Fig. 22.6.

Vertical shell-and-tube type condensers are usually used with ammonia in large capacity systems so that cleaning of the tubes is possible from top while the plant is running.

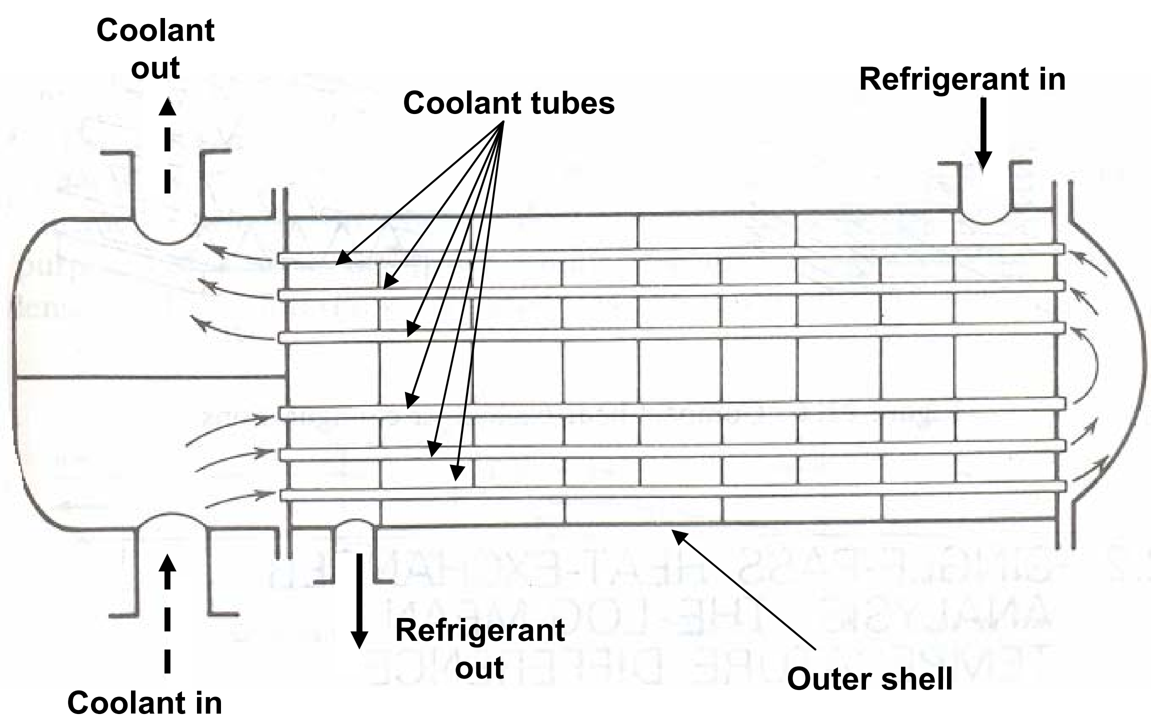


***Fig.22.6**: A two-pass, shell-and-tube type condenser*

22.3.3. Evaporative condensers:

In evaporative condensers, both air and water are used to extract heat from the condensing refrigerant. Figure 22.7 shows the schematic of an evaporative condenser. Evaporative condensers combine the features of a cooling tower and water-cooled condenser in a single unit. In these condensers, the water is sprayed from top part on a bank of tubes carrying the refrigerant and air is induced upwards. There is a thin water film around the condenser tubes from which evaporative cooling takes place. The heat transfer coefficient for evaporative cooling is very large. Hence, the refrigeration system can be operated at low condensing temperatures (about 11 to 13 K above the wet bulb temperature of air). The water spray countercurrent to the airflow acts as cooling tower. The role of air is primarily to increase the rate of evaporation of water. The required air flow rates are in the range of 350 to 500 m^3/h per TR of refrigeration capacity.

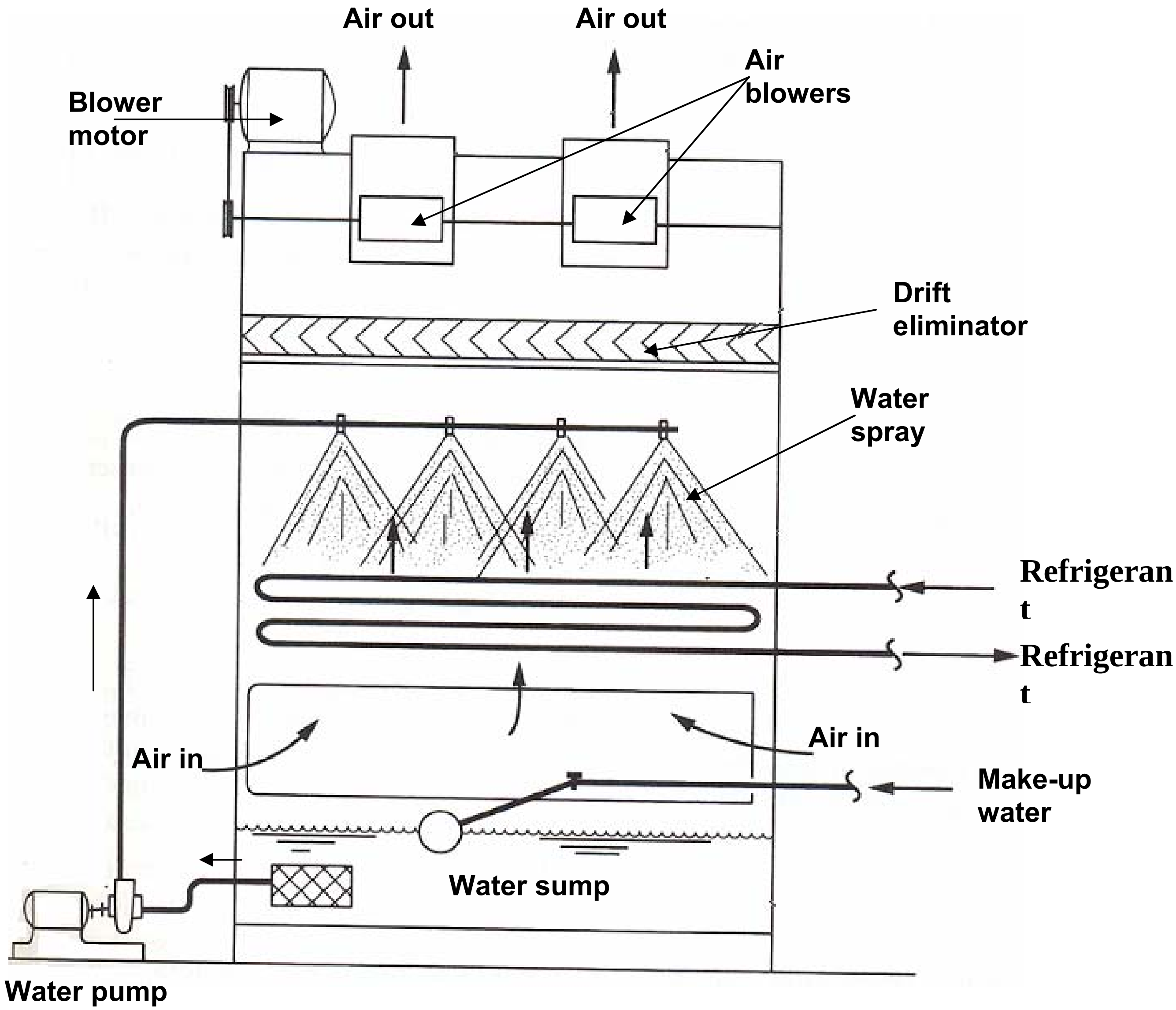


Fig.22.7: *Schematic of an evaporative condenser*

Evaporative condensers are used in medium to large capacity systems. These are normally cheaper compared to water cooled condensers, which require a separate cooling tower. Evaporative condensers are used in places where water is scarce. Since water is used in a closed loop, only a small part of the water evaporates. Make-up water is supplied to take care of the evaporative loss. The water consumption is typically very low, about 5 percent of an equivalent water cooled condenser with a cooling tower. However, since condenser has to be kept outside, this type of condenser requires a longer length of refrigerant tubing, which calls for larger refrigerant inventory and higher pressure drops. Since the condenser is kept outside, to prevent the water from freezing, when outside temperatures are very low, a heater is placed in the water tank. When outside temperatures are very low it is possible to switch-off the water pump and run only the blowers, so that the condenser acts as an air cooled condenser.

Another simple form of condenser used normally in older type cold storages is called as atmospheric condenser. The principle of the atmospheric condenser is similar to evaporative condenser, with a difference that the air flow over the condenser takes place by natural means as no fans or blowers are used. A spray system sprays water over condenser tubes. Heat transfer outside the tubes takes by both sensible cooling and evaporation, as a result the external heat transfer coefficient is relatively large. The condenser pipes are normally large, and they can be either horizontal or vertical. Though these condensers are effective and economical they are being replaced with other types of condensers due to the problems such as algae formation on condenser tubes, uncertainity due to external air circulation etc.

22.3.4. Air cooled vs water cooled condensers:

The Salient features of air cooled and water cooled condensers are shown below in Table 22.1. The advantages and disadvantages of each type are discussed below.

Parameter	Air cooled	Water cooled
Temperature difference, $T_C - T_{coolant}$	6 to 22^o C	6 to 12^o C
Volume flow rate of coolant per TR	12 to 20 m^3/min	0.007 to 0.02 m^3/min
Heat transfer area per TR	10 to 15 m^2	0.5 to 1.0 m^2
Face Velocity	2.5 to 6 m/s	2 to 3 m/s
Fan or pump power per TR	75 to 100 W	negligible

Table 22.1: Comparison between air cooled and water cooled condensers

Advantages and disadvantages:

Air-cooled condensers are simple in construction since no pipes are required for air. Further, the disposal of warm air is not a problem and it is available in plenty. The fouling of condenser is small and maintenance cost is low. However, since the specific heat of air is one fourth of that of water and density is one thousandth of that of water, volume flow rates required are very large. The thermal conductivity is small; hence heat transfer coefficient is also very small. Also, air is available at dry-bulb temperature while water is available at a lower temperature, which is 2 to 3 ^{o}C above the wet-bulb temperature. The temperature rise of air is much larger than that of water, therefore the condenser temperature becomes large and COP reduces. Its use is normally restricted to 10 TR although blower power goes up beyond 5 TR. In systems up to 3 TR with open compressors it is mounted on the same chassis as the compressor and the compressor motor drives the condenser fan also. In middle-east countries where is shortage of fresh water these are used up to 100 TR or more.

The air-cooled condensers cost two to three times more than water-cooled condensers. The water-cooled condenser requires cooling tower since water is scarce in municipality areas and has to be recycled. Water from lakes and rivers cannot be thrown back in warm state since it affects the marine life adversely. Increased first cost and maintenance cost of cooling tower offsets the cost advantage of water-cooled condenser. Fouling of heat exchange surface is a big problem in use of water.

22.4. Analysis of condensers:

From Fig.22.1, the total heat rejected in the condenser, Q_c is given by:

$$\mathbf{Q_c = \dot{m}(h_2 - h_4) = \dot{m}_{ext}\, C_{p,ext}(T_{ext,o} - T_{ext,i})} \qquad (22.1)$$

where $\dot{m}$ is the mass flow rate of refrigerant

h_2, h_4 are the inlet and exit enthalpies of refrigerant

$\dot{m}_{ext}$ is the mass flow rate of the external fluid

$C_{p,ext}$ is an average specific heat of the external fluid, and

$T_{ext,i}$ and $T_{ext,o}$ are the inlet and exit temperatures of the external fluid

The required condenser area is then given by the equation:

$$\mathbf{Q_c = U.A.\Delta T_m} \qquad (22.2)$$

where U is the overall heat transfer coefficient

A is the heat transfer area of the condenser, and

ΔT_m is mean temperature difference between refrigerant and external fluid

In a typical design problem, the final objective is to find the heat transfer area A required from given input. From the above equation it can be seen that to find heat transfer area, one should know the amount of heat transfer rate across the condenser (Q_c), the overall heat transfer coefficient (U) and the mean temperature difference. The heat transfer rate in the condenser depends on the refrigeration capacity of the system and system COP. The overall heat transfer coefficient depends on the type and design of condenser. The mean temperature difference depends on the operating temperature of the refrigeration system, type of the condenser and the external fluid. In a typical rating problem, the objective is to find the rate of heat transfer when other parameters are fixed.

22.4.1. Condenser Heat Rejection Ratio (HRR):

The heat rejection ratio (HRR) is the ratio of heat rejected to the heat absorbed (refrigeration capacity), that is,

$$\mathbf{HRR} = \frac{\mathbf{Q_c}}{\mathbf{Q_e}} = \frac{\mathbf{Q_e} + \mathbf{W_c}}{\mathbf{Q_e}} = 1 + \frac{1}{\mathrm{COP}} \qquad (22.3)$$

For a fixed condenser temperature, as the evaporator temperature decreases the COP decreases and heat rejection ratio increases. For fixed evaporator temperature as the condenser temperature increases the COP decreases hence the heat rejection ratio increases. At a given evaporator and condenser temperatures, the HRR of refrigeration systems using hermetic compressors is higher than that of open compressor systems. As discussed in earlier chapters, this is due to the additional heat rejected by motor and compressor in hermetic systems. These characteristics are shown in Fig.22.8. Such curves can be drawn for all refrigerants so that the condenser heat rejection can be determined for given T_e, T_c and TR.

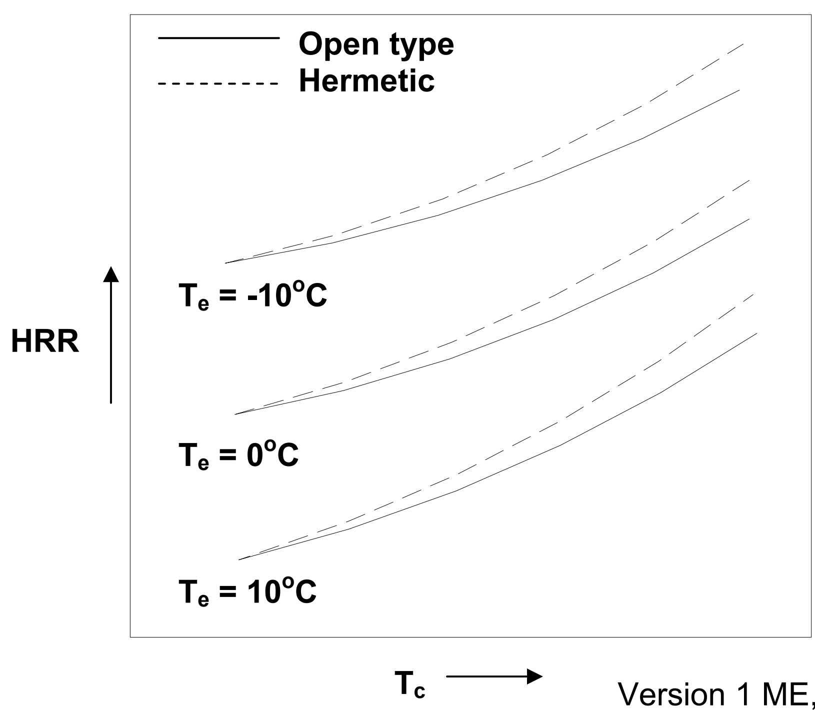


***Fig.22.8**: Variation of heat rejection ratio (HRR) with evaporator and condenser temperatures*

22.4.2. Mean temperature difference:

In a refrigerant condenser, the mean temperature difference ΔT_m, between the refrigerant and the external fluid varies continuously along the length as shown in Fig.22.9. However, the heat transfer coefficient on the refrigerant side, h_r is small during de-superheating (2-3) in vapour phase but temperature difference between refrigerant and coolant ΔT is large, while during condensation (3-3') the heat transfer coefficient on refrigerant side is large and the temperature difference is small. As a result, the product $h_r\Delta T$ is approximately same in both the regions; hence as an approximation one may design the condenser by assuming that condensation occurs throughout the condenser. This implies that the refrigerant temperature is assumed to remain constant at condensing temperature throughout the length of the condenser. As mentioned, this is an approximation, and is considered to be adequate for rough estimation of condenser area. However, for accurate design of condenser, one has to consider the de-superheating, condensation and subcooling regions separately and evaluate the area required for each region, and finally find the total area.

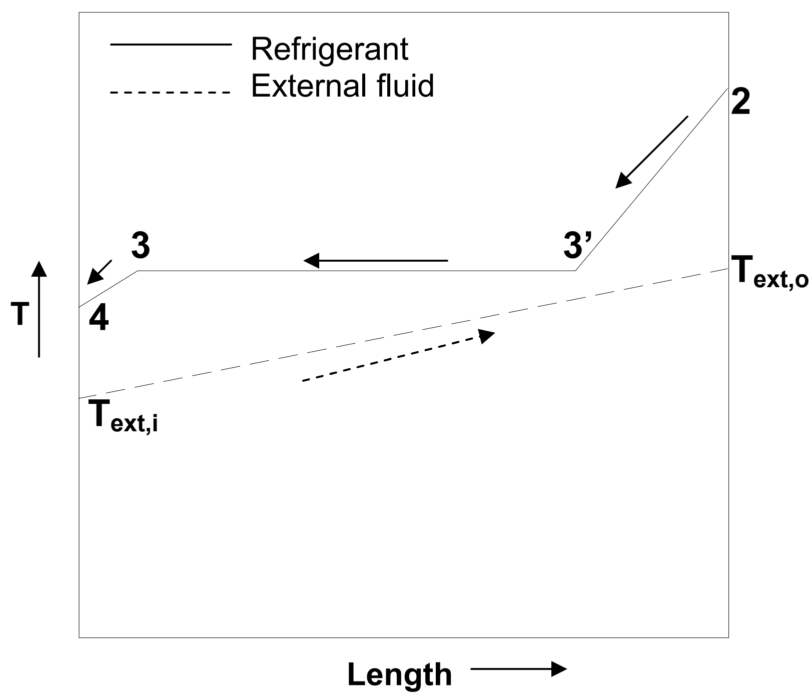


Fig.22.9: *Variation of refrigerant and external fluid temperature in a condenser*

If we assume condensation throughout the length of the condenser and also assume the pressure drop to be negligible, then the mean temperature difference is given by the Log Mean Temperature Difference (LMTD):

$$\mathrm{LMTD} = \frac{(T_{ext,o} - T_{ext,i})}{\ln\left(\frac{T_c - T_{ext,i}}{T_c - T_{ext,o}}\right)} \tag{22.4}$$

In the above equation, $T_{ext,i}$ and $T_{ext,o}$ are the inlet and outlet temperatures of the external fluid, and T_c is the condensing temperature.

22.4.3. Overall heat transfer coefficient:

Evaluation of overall heat transfer coefficient, U is an important step in the design of a condenser. The overall heat transfer coefficient can be based on either internal area (A_i) or external area (A_o) of the condenser. In general we can write:

$$UA = U_i A_i = U_o A_o = \frac{1}{\sum_{i=1}^{n} R_i} \tag{22.5}$$

where R_i is the heat transfer resistance of i^{th} component

A general expression for overall heat transfer coefficient is given by:

$$\frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{[h(A_f \eta_f + A_b)]_o} + \frac{\Delta x}{k_w A_m} + \frac{1}{[h(A_f \eta_f + A_b)]_i} + \frac{R''_{f,o}}{A_o} + \frac{R''_{f,i}}{A_i} \tag{22.6}$$

In the above expression, h is the convective heat transfer coefficient, A_f and A_b are the finned and bare tube areas of the heat exchanger, respectively, η_f is the fin efficiency. Subscripts “i” and “o” stand for inner and outer sides, Δx is the thickness of the wall separating the refrigerant from external fluid, k_w and A_m are the thermal conductivity and mean area of the wall. R''_f is the resistance due to fouling.

The fouling due to deposition of scale on the fin side of an air cooled condenser usually has little effect since $1/h_{co}$ is rather large. In some cases an allowance may be made for imperfect contact between the fins and the tubes, however it is difficult to evaluate. It is negligible for good construction. The fouling resistance for the inside of the tube is not negligible and must be included. For an externally finned tube condenser, the overall heat transfer coefficient based on the external area, U_o is given by:

$$U_o = \frac{1}{\frac{A_o}{h_i A_i} + \frac{R''_{f,i} A_o}{A_i} + \frac{A_o}{A_i}\frac{r_i \ln(r_o / r_i)}{k_w} + \frac{A_o}{[h_o(A_f \eta_f + A_b)]_o}} \quad (22.7)$$

In the above expression A_o is the total external area (A_f+A_b), h_i and h_o are the inner and outer convective heat transfer coefficients, respectively and r_i, r_o are the inner and outer radii of the tube, respectively.

For water-cooled condensers without fins, the expression for overall heat transfer coefficient simplifies to:

$$U_o = \frac{1}{\frac{A_o}{h_i A_i} + \frac{R''_{f,i} A_o}{A_i} + \frac{A_o}{A_i}\frac{r_i \ln(d_o / d_i)}{k_w} + \frac{1}{h_o}} \quad (22.8)$$

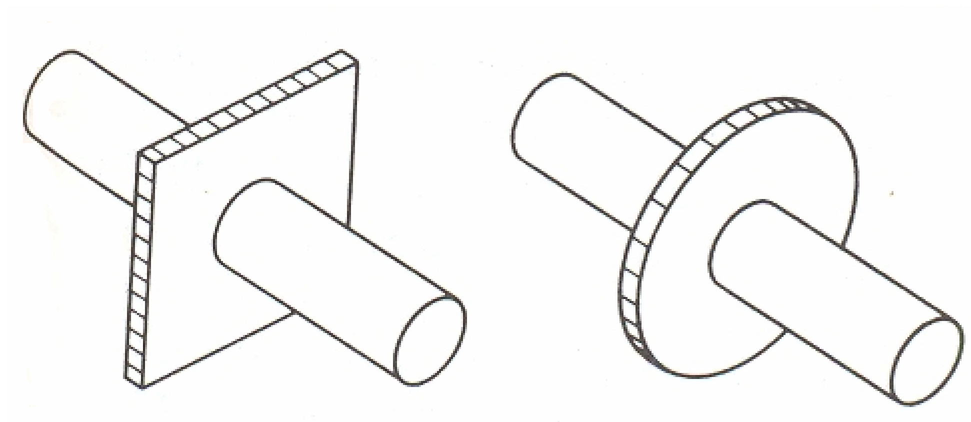
The condensation heat transfer coefficient is of the order of 7000 W/m^2-K for ammonia. However it is of the order of 1700 W/m^2-K for synthetic refrigerants such as R 12 and R 22, whereas the waterside heat transfer coefficient is high in both the cases for turbulent flow. Hence it is advisable to add fins on the side where the heat transfer coefficient is low. In case of R 12 and R 22 condensers the tubes have integral external fins to augment the heat transfer rate. This is easily seen if the overall heat transfer coefficient is written in terms of inside area as follows.

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{r_i \ln(d_o / d_i)}{k_w} + \frac{1}{h_o}\frac{A_i}{A_o} + R''_{f,i} \quad (22.9)$$

It can be observed that by increasing the area ratio A_o/A_i ,that is the outside surface area the overall heat transfer coefficient can be increased.

Fin efficiency:

In finned tube condensers, the fin efficiency depends on the type and material of the fin and on fluid flow characteristics. Expressions for fin efficiency can be derived analytically for simple geometries, however, for complex geometries, the fin efficiency has to be obtained from actual measurements and manufacturers' catalogs. The most commonly used fin configuration is the plate-fin type as shown in Fig. 22.3. The plate-fin is often approximated with an equivalent annular fin as shown in Fig.22.10. This is done as analytical expressions and charts for the efficiency of annular fin have been obtained. Figure 22.11 shows a typical efficiency chart for annular fins. In the figure, r_o and r_i are the outer and inner radii of the annular fin, h_o is the external heat transfer coefficient, k is the thermal conductivity of fin material and t is the thickness of the fin.



Rectangular plate-fin segment **Equivalent annular fin**

Fig.22.10: *Approximating a plate-fin with an equivalent annular fin*

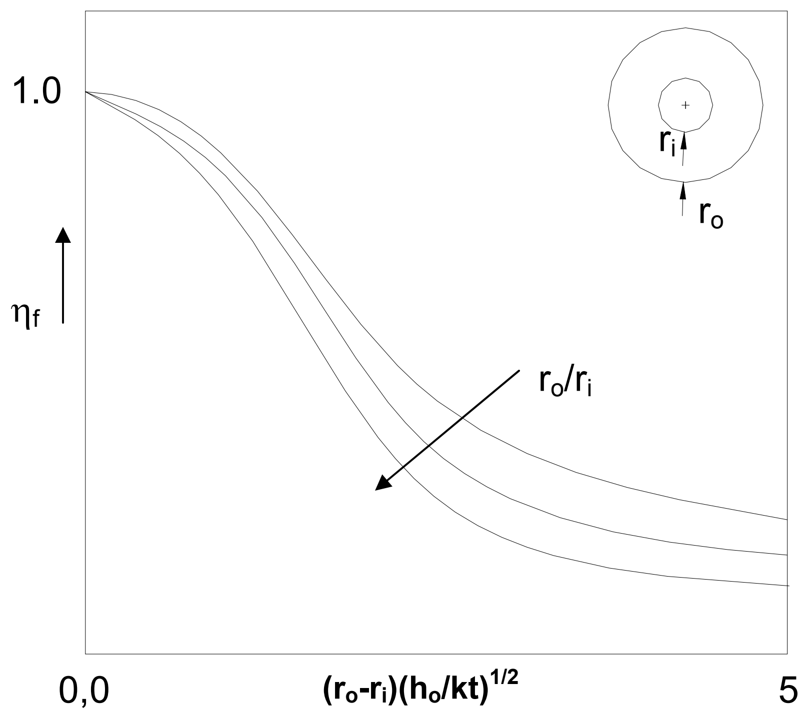


Fig.22.11: Fin efficiency curves for an annular fin

As shown in Fig.22.3, if the spacing between the tubes is *B* units within a row and *C* units between rows. Then the area of the fin is given by ($B \times C - \pi r_1^2$). Now the outer radius (r_2) of an equivalent annular fin is obtained by equating the fin areas, i.e.,

$$\boldsymbol{B \times C - \pi r_1^2 = \pi(r_2^2 - r_1^2) \;\therefore\; r_2 = \sqrt{(B \times C/\pi)}} \qquad (22.10)$$

Then the efficiency of the rectangular plate-fin is obtained from the efficiency of an equivalent annular fin having an inner radius of r_1 and outer radius of r_2 (= $\sqrt{(B \times C/\pi)}$).

22.4.4. Heat transfer areas in finned tube condensers:

Figures 22.3 shows the schematic diagram of a condenser or a cooling coil with tubes and fins. The air flows through the passages formed by the fins. Figure 22.12 shows a section of the plate fin-and-tube condenser and its side view.

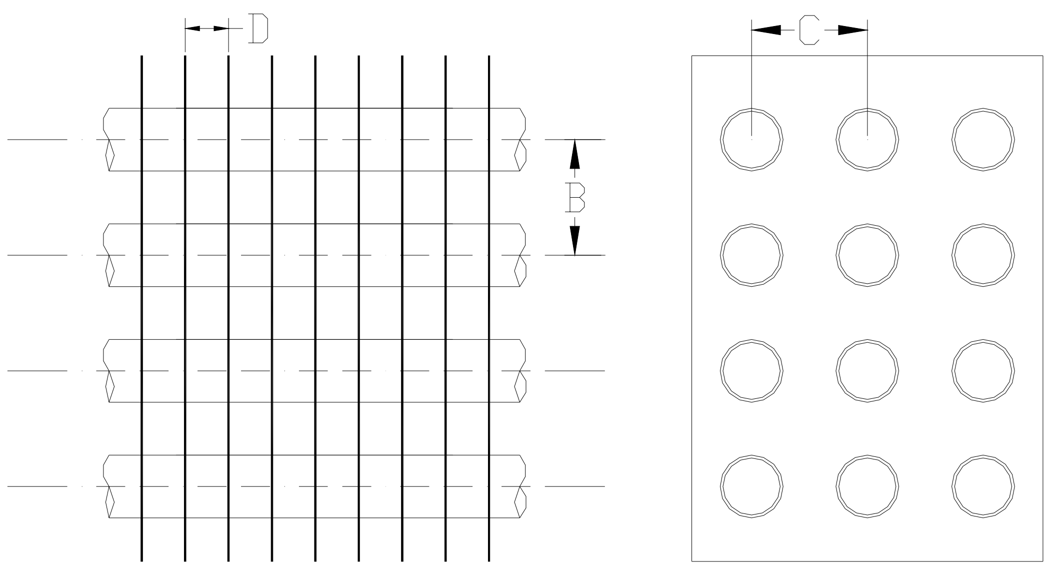


***Fig.22.12**: A portion of a plate fin-and-tube type condenser and its side view*

The heat transfer takes place from the fins and the exposed part of the tube. Hence heat transfer occurs from following areas

1.□. Bare tube area between the consecutive fins, A_b
b) Area of the fins, A_f

These areas are expressed in terms per m^2 of face area and per row. Face area A_{face} is the area of condenser seen from outside, the actual flow area is less than the face area since fins have finite thickness. Further, as air flows through it, it has to pass between the narrow passage between the tubes. The flow area is minimum at these locations. This will be denoted by A_c. To find these areas we consider condenser of 1.0 m height and 1.0 m width as shown in Fig.22.12, so that the face area is 1 m^2. All the dimensions are in mm. Following nomenclature is used.

B: Vertical spacing between the tubes in a row, mm
C: Spacing between the tube in different rows, mm
t: Thickness of the fins, mm
D: Centre-to center spacing between the fins, mm
d_o: Outer diameter of the tubes, mm
d_i : Inner diameter of the tubes, mm

No. of tubes per m height = (1000/*B*) (tubes per m^2 face area per row)
No. of fin passages per m width = (1000/*D*) (no. of passages per m^2 face area)
No. of fins per m^2 face area = 1 + 1000/*D* ≈ 1000/*D*
Width of each passage = (*D* – *t*) /1000 (in meters)

Then the various areas are as follows:

Bare tube area, A_b = (tube perimeter) x (number of fin passages) x (number of tubes) x (width of each passage) = ($\pi\ d_o$/1000) (1000/*D*) (1000/*B*) (*D* –*t*)/1000

$$\mathbf{A_b = \frac{D-t}{DB}\pi d_o} \qquad m^2 \text{ per } m^2 \text{ face area per row} \qquad (22.11)$$

Fin Area, A_f = (number of fins) (two sides of fins){width of fin per row – number of tubes x area of cross section of each tube)} = (1000/D)(2){1 x C/1000 – (1000/B) $\pi(d_o/1000)^2/4$]

$$\mathbf{A_f = \frac{2}{D}\left[C - \frac{\pi d_o^2}{4B}\right]} \qquad m^2 \text{ per } m^2 \text{ face area per row} \qquad (22.12)$$

Minimum flow area, A_c = (number of fin passages) x (width of each passage) x (height – number of tubes per row x diameter of tube) = (1000/*D*){(*D* – *t*)/1000}{1 – (1000/*B*)(d_o/1000)}

$$\mathbf{A_c = \frac{D-t}{D}\left[1 - \frac{d_o}{B}\right]} \qquad m^2 \text{ per } m^2 \text{ face area per row} \qquad (22.13)$$

Total heat transfer area A_o = Bare tube area + Fin area

$$\mathbf{A_o = A_b + A_f} \qquad m^2 \text{ per } m^2 \text{ face area per row} \qquad (22.14)$$

Wetted Perimeter, P = total heat transfer area/length in flow direction

$$\mathbf{P = A_o/(C/1000)} \qquad (22.15)$$

Hydraulic diameter, D_h = 4 A_c/wetted perimeter

$$D_h = \frac{4\,C\,A_c}{1000\,A_o} \qquad (22.16)$$

The Reynolds number and the Nusselt numbers are based upon hydraulic diameter.

Inside heat transfer area, A_i = (πd_i/1000) x (Number of tubes) = $\pi d_i/B$

$$\mathbf{A_i = \pi d_i/B} \qquad (22.17)$$

22.4.5. Estimation of heat transfer coefficients:

1.. **Air side heat transfer coefficients in air cooled condensers:**

1.. **Flow over finned surfaces:**

As discussed before, in these condensers, the refrigerant flows through the tubes, while air flows over the finned tubes. The forced convection heat transfer coefficient for the air-side depends upon, the type of fins, fin spacing, fin thickness tube diameters etc. It can be evaluated experimentally for particular fin and tube arrangement. Kays and London (1955) have carried out extensive measurements on different types of fin and tube arrangements. They have presented the data in the forms of plot of Colburn j-factor (St.$Pr^{2/3}$) vs. Reynolds number (Re) for various geometries. On the average, following correlation is a good fit to their data for various geometries.

$$\mathbf{Nu = 0.117 Re^{0.65}\, Pr^{1/3}} \qquad (22.17)$$

The Nusselt number and Reynolds numbers are based upon hydraulic diameter defined earlier in Eqn.(22.16).

Another simple expression has been proposed Air conditioning and Refrigeration Institute, Arlington Va.(1972) , which is as follows

$$\mathbf{h_o = 38\, V_f^{\,0.5}} \qquad (22.18)$$

Where, V_f is the face velocity in m/s and h_o is in W/m^2.K

b) Correlations for Pressure drop

Rich (1974) has carried out extensive measurements over the fin-tube heat exchangers and has given pressure drop plots. A correlation fitted to his data is given in Table 22.2 for various fin spacing for pressure drop in Pa per row. The velocity is the face velocity in m/s

Number of fins/m	315	394	472	531
Δp (Pa per row)	7.15 $V^{1.56}$	8.5$V^{1.56}$	9.63 $V^{1.56}$	11 $V^{1.56}$

***Table 22.2**: Pressure drop correlations for various fin spacings (Rich,1974)*

ii. Flow over tube banks:

a) Heat transfer

Grimson has given correlations for average heat transfer coefficient for forced convection from tube banks in cross flow for staggered as well as in-line arrangement of tubes as shown in Fig. 22.13. As mentioned earlier, face area A_f of the heat exchanger is the area seen from the flow direction and Q_f is the volume flow rate of flow then face velocity V_f is given by:

$$\mathbf{V_f = Q_f/A_f} \qquad (22.19)$$

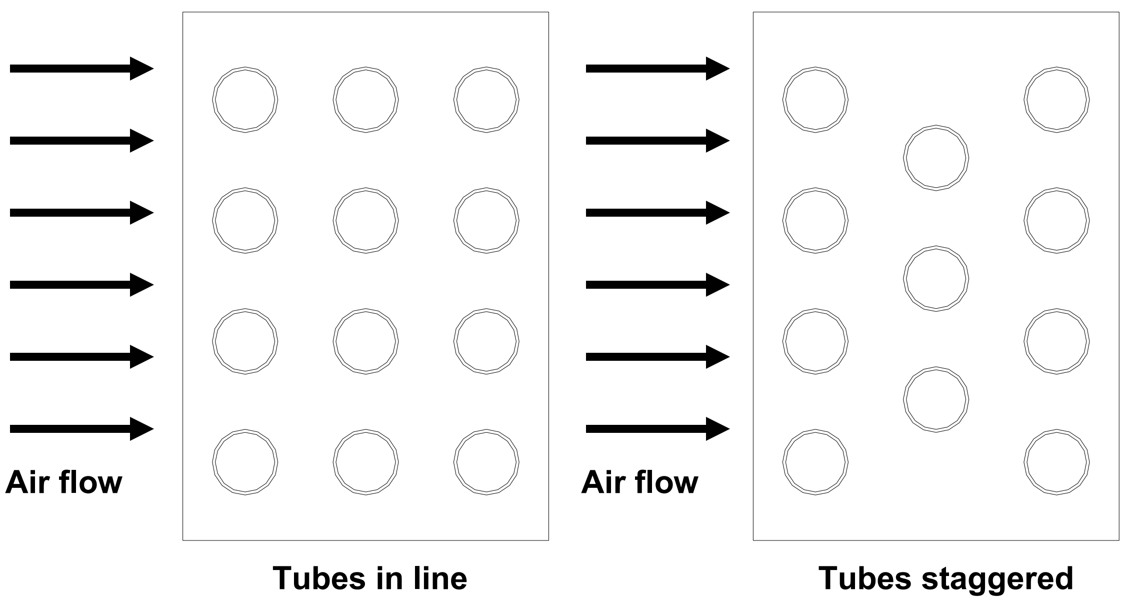


***Fig.22.13**: Schematic diagram of plate find-and-tube condenser with Tubes-in-line and tubes staggered*

The maximum velocity occurs between the tubes since the tubes block a part of the flow passage. If B is the spacing between tubes in the face and C is the tube spacing between rows, and d_o is the tube diameter then maximum velocity is given by

$$V_{max} = V_f B/(B - d_o) \tag{22.20}$$

The Reynolds and Nusselt number are defined as follows for this case:

$$Re = \frac{\rho V_{max} d_o}{\mu} \quad \text{and} \quad Nu = \frac{h d_o}{k} \tag{22.21}$$

The Grimson's correlation is as follows

$$Nu = C\, Re^{n}\, Pr^{1/3} \tag{22.22}$$

Where the constants *C* and *n* are dependent upon Reynolds number and are given in Table 22.3.

Reynolds number, Re	Constant C	Constant n
0.4 to 4	0.989	0.33
4 to 40	0.911	0.385
40 to 4000	0.683	0.466
4000 to 40000	0.193	0.618
40000 to 400000	0.0266	0.805

Table 22.3*: Values of constants C and 'n' used in Eqn.(22.22)*

b) Pressure drop

O.L. Pierson and E.C. Huge have given the correlation for pressure drop for flow over tube banks as follows:

$$\Delta p = fNV^{2}/2 \tag{22.23}$$

Where, *f* is the friction factor and *N* is the number of rows. The friction factor is given by

$$f = Re^{-0.15}\left[0.176 + \frac{0.32\,b}{(a-1)^{0.43} + 1.13/b}\right] \quad \textbf{for tubes in-line}$$

$$f = Re^{-0.16}\left[1.0 + \frac{0,47}{(a-1)^{1.08}}\right] \quad \textbf{for staggered tubes} \tag{22.24}$$

where, $a = B/d_o$ and $b = C/d_o$

iii. Free convection over hot, vertical flat plates and cylinders:

<u>Constant wall temperature:</u>

$$\text{Average Nusselt number}, \overline{Nu}_L = \left(\frac{\bar{h}_c L}{k_f}\right) = c\,(Gr_L\,Pr)^{n} = cRa_L^{\,n} \tag{22.25}$$

where c and n are 0.59 and ¼ for laminar flow ($10^4 < Gr^L.Pr < 10^9$) and 0.10 and ⅓ for turbulent flow ($10^9 < Gr^L.Pr < 10^{13}$)

In the above equation, Gr_L is the average Grashoff number given by:

$$\text{Average Grashoff Number } Gr_L = \frac{g\beta\ (T_w - T_\infty)\ L^3}{\upsilon^2} \qquad (22.26)$$

where g is the acceleration due to gravity, β is volumetric coefficient of thermal expansion, T_w and T_∞ are the plate and the free stream fluid temperatures, respectively and ν is the kinematic viscosity. Correlations for other conditions are presented in Chapter 7.

b) Water side heat transfer coefficients in water cooled condensers:

In water cooled condensers, the water flows through the tubes. The water flow is normally turbulent, hence one can use Dittus-Boelter equation given by:

$$\mathbf{Nu_d = 0.023\ Re_d^{0.8}\ Pr^{0.4}} \qquad (22.27)$$

If the viscosity variation is considerable, then one can use Seider-Tate equation given by:

$$\mathbf{Nu_d = 0.036\ Re_d^{0.8}\ Pr^{1/3}\ (\mu/\mu_w)^{0.14}} \qquad (22.28)$$

If the Reynolds number on water side is less than 2300, then the flow will be laminar, hence one has to use the correlations for laminar flow. For example, if the flow is laminar and not fully developed, then one can use Hausen's correlation given by:

$$\overline{Nu}_d = 3.66 + \frac{0.0668(D_i/L)Pe}{1 + 0.04[(D_i/L)\,Pe]^{2/3}} \qquad (22.29)$$

where Pe is the Peclet number = $Re_d.Pr$

1.. Condensation heat transfer coefficient:

When refrigerant vapour comes in contact with the surface whose temperature is lower than the saturation temperature of refrigerant at condenser pressure, the refrigerant condenses. Depending upon the type of the surface, condensation can be filmwise or dropwise. Even though dropwise condensation yields higher heat transfer coefficients compared to filmwise condensation, normally design calculations are based on filmwise condensation. This is due to the reason that it is difficult to maintain dropwise condensation continuously as the surface characteristics may undergo change with time. In filmwise condensation, the condensed refrigerant liquid forms a film over the condensing surface. This liquid film resists heat transfer, hence, for high condensation heat transfer rates, the thickness of the liquid film should be kept as small as possible. This requires continuous draining of condensed liquid so that the vapour has better contact with the heat transfer surface of the condenser. Since the rate at which condensed liquid is drained depends among other factors on the orientation of the surface, the condensation heat transfer coefficients vary widely with orientation.

Outside Horizontal Tubes

A typical correlation known as Nusselt's correlation for film-wise condensation outside a bank of horizontal tubes is as follows:

$$\mathbf{h_0 = 0.725\left[\frac{k_f^3 \rho_f (\rho_f - \rho_g) g h_{fg}}{N D_0 \mu_f \Delta t}\right]^{0.25}} \qquad (22.30)$$

The density of liquid is much more than that of vapour hence this may be approximated by

$$\mathbf{h_o = 0.725\left[\frac{k_f^3 \rho_f^2 g h_{fg}}{N D_o \mu_f \Delta t}\right]^{1/4}} \qquad (22.31)$$

This expression is exactly valid for still vapour. In this expression subscript *f* refers to the properties of saturated liquid, which are evaluated at mean film temperature of (t_{wo} + t_r)/2. D_0 is the outer diameter of the tube and N is the average number of tubes per column.

Some of the features of this correlation are as follows:

i. As thermal conductivity k_f increases, the heat transfer coefficient increases since conduction thermal resistance of the condensate film decreases.
ii. Similarly a decrease in viscosity or increase in density will offer less frictional resistance and cause rapid draining of the condensate, thereby causing an increase in heat transfer coefficient.
iii. A high value of latent heat h_{fg} means that for each kW of heat transfer there will be smaller condensate thickness and higher heat transfer coefficient.
iv. An increase in diameter means larger condensate thickness at the bottom and hence a smaller heat transfer coefficient.
v. A large value of temperature difference will lead to more condensation and larger condensate thickness and will lead to a smaller heat transfer coefficient
vi. An increase in number of tubes will lead to larger condensate thickness in the lower tubes leading to smaller heat transfer coefficient

In actual practice the vapour will not be still but it will move with some velocity and the condensate will splash and ripples will be caused which may lead to larger value of heat transfer coefficient. Hence the above equation gives a very conservative estimate of condensation heat transfer coefficient.

Outside Vertical Tube *:*

For laminar flow the average heat transfer coefficient by Nusselt's Correlation for condensation over a vertical tube is as follows

$$\mathbf{h_0 = 1.13\left[\frac{k_f^3 \rho_f(\rho_f - \rho_g) g h_{fg}}{L\mu_f \Delta t}\right]^{0.25}} \quad \text{where L is the tube length} \quad (22.32)$$

This may be used in laminar flow up to Re_f = 1800, where $Re_f = 4\dot{m}/(\pi\mu_f D)$
Kirkbride has rearranged this in terms of *condensation number Co*, which is defined as follows:

$$\mathbf{Co = h_0\left[\frac{\mu_f^2}{k_f^3 \rho_f^2 g}\right]^{1/3} = 1.514\,Re_f^{-1/3} = 1.514\,R_{ef}^{-1/3}} \qquad (22.33)$$

For turbulent flow : Re_f > 1800 , the Kirkbride Correlation is as follows:

$$\mathbf{Co = h_0\left[\frac{\mu_f^2}{k_f^3 \rho_f^2 g}\right]^{1/3} = 0.0077\,Re_f^{0.4}} \qquad (22.34)$$

Condensation Inside Tubes

Condensation heat transfer inside tube causes a reduction in the area of condensation due to liquid collecting in the bottom of the tubes. The draining of the condensate may retard or accelerate the vapour flow depending upon whether it flows in same direction as the vapour or in opposite direction. Here flow rate of vapour considerable influences the heat transfer coefficient.

1.□. *Chaddock and Chato's Correlation*

Chaddock and Chato suggested that condensation heat transfer coefficient inside tubes is 0.77 times that of Nusselt's heat transfer coefficient outside the tubes particularly if the vapour Reynolds number $Re_g = 4\dot{m}/(\pi\mu_g D_i) < 35000$. This gives the average value of heat transfer coefficient over the length of the tube.

$$\mathbf{H_{TP} = 0.77\, h_0} \qquad (22.35)$$

$$\mathbf{h_{TP} = 0.555\left[\frac{k_f^3 \rho_f(\rho_f - \rho_g) g h'_{fg}}{D_i\mu_f \Delta t}\right]^{0.25}} \qquad (22.36)$$

Where the modified enthalpy of evaporation is defined as $h'_{fg} = h_{fg} + 3\, C_{pf}\, \Delta t/8$, Δt *is the difference between the temperature of condensing refrigerant and temperature of the surface.*

(b) *Cavallini Zecchin Correlation*

This correlation represents the condensation heat transfer coefficient in a manner similar to Dittus-Boelter equation for turbulent flow heat transfer inside tubes. The constant is different from that equation and an equivalent Reynolds number is used to take care of two-phase flow and incomplete condensation. The local values of heat transfer coefficient can also be found if the quality distribution is known.

$$h_{TP} = 0.05\, Re_{eq}^{0.8}\, Pr_f^{0.33}\, k_f / D_i$$

$$Re_{eq} = Re_f\,(1-x) + x\left(\frac{\mu_g}{\mu_f}\right)\left(\frac{\rho_f}{\rho_g}\right)^{0.5} Re_g$$

$$\text{Where, } Re_g = \frac{4\dot{m}}{\pi D_i \mu_g} \text{ and } Re_f = \frac{4\dot{m}}{\pi D_i \mu_f} \qquad (22.37)$$

© *Traviss et al. Correlation*

This correlation uses Lockhart-Martinelli parameter, which takes into account incomplete condensation. This can also be used for evaluation of local heat transfer coefficient if the quality of mixture is known. The correlation covers a wide range of Reynolds numbers defined as $Re_l = (1- x)\, Re_f$, where Re_f is the Reynolds number if all the refrigerant flows in liquid phase.

$$Nu = \left[\frac{Pr_f\, Re_l^{0.9}}{F_2}\right] F_{tt} \quad : \text{for } 0.15 < F_{tt} < 15 \qquad \mathbf{(22.38)}$$

$$F_{tt} = 0.15\,[X_{tt}^{-1} + 2.85\, X_{tt}^{-0.467}] \quad \text{and}$$

$$F_2 = 0.707\, Pr_f\, Re_l \text{ for } Re_l < 50 \text{ where, } Re_l = (1-x)\, Re_f$$

$$F_2 = 5 Pr_f + 5 \ln\,[1 + Pr_f\,(0.09636\, Re_l^{0.585} - 1)] \qquad : 50 < Re_l < 1125$$

$$F_2 = 5 Pr_f + 5 \ln[1 + 5 Pr_f] + 2.5 \ln[0.00313\, Re_l^{0.812}] \quad : Re_l > 1125$$

$$X_{tt} = [(1-x)/x]^{0.9}\, (\rho_g/\rho_f)^{0.5}\, (\mu_f/\mu_g)^{0.1} \quad = \text{Lockhart - Martinelli parameter}$$

1.□. *Shah's Correlation*

This correlation takes into account the pressure of the refrigerant also in addition to the quality of the mixture. This can also be used to find the local condensation heat transfer coefficient. The heat transfer coefficient is a product of heat transfer coefficient given by Dittus-Boelter equation and an additional term.

$$h_{TP} = h_L \left[(1-x)^{0.8} + \frac{3.8\,x^{0.76}\,(1-x)^{0.04}}{p_r^{0.38}} \right]$$

$$\text{where}, p_r = p/p_{critical} = \text{reduced pressure}$$

$$h_L = 0.023\,Re_f^{0.8}\,Pr_f^{0.4}\,k_f/D_i \qquad (22.40)$$

$$\bar{h}_{TP} = h_{TP}[0.55 + 2.09/p_r^{0.38}] : \text{avg value of h.t.coeff. at } x = 0.5$$

1.□. *Akers, Dean and Crosser Correlation*

Akers, Dean and Crosser have proposed following correlation when the rate of condensation or the length is very large. This is very similar to Dittus-Boelter correlation for turbulent heat transfer in tubes, except the constant is different.

$$\frac{hD_i}{k_f} = 5.03\,Re_m^{1/3}\,Pr_f^{1/3} \qquad : Re_g < 5x10^4$$

$$= 0.0265\,Re_m^{0.8}\,Pr_f^{1/3} \qquad : Re_g > 5x10^4$$

$$\text{where } Re_m = Re_f\,[1 + (\rho_f/\rho_g)^{0.5}] \qquad (22.41)$$

In this correlation the heat transfer coefficient is independent of temperature difference and it increases with the increase in liquid Reynolds number, Re_f. Sometimes, it overestimates the heat transfer coefficient.

Fouling Factor

The condenser tubes are clean when it is assembled with new tubes. However with usage some scale formation takes place in all the tubes and the value of overall heat transfer coefficient decreases. It is a standard practice to control the hardness of water used in the condenser. Even then it is good maintenance practice to de-scale the condenser once a year with 2% HCl or muric acid solution. Stoecker suggests the following values of deposit coefficients.

$R''_{f.}$ = 0.00009 m^2.K/W for R12 and R-22 with copper tubes
$R''_{f.}$ = 0.000178 m^2.K/W for steel tubes with ammonia

22.5. Effect of air and non-condensables:

This is usually a problem with high boiling point refrigerants such as R 11, R 113 and R718 (water), which operate under vacuum leading to air leakage into the system. In addition, some air may be left behind before the system is evacuated and charged with refrigerant. If some non-condensable gases or air enters the system, it will collect in the condenser where they affect performance in two ways:

1. Condensation takes place at saturation pressure corresponding to condenser pressure, which will be the partial pressure of refrigerant in mixture of refrigerant and air in this case. The air will have its partial pressure proportional to its amount in the condenser. The total pressure will be the sum of these two partial pressures, which will be high and the compressor has to work against this pressure ratio hence the work requirement will increase.

2. Non-condensable gases do not diffuse throughout the condenser as the refrigerant condenses. They cling to the tubes and reduce the precious heat transfer area. The reduction in heat transfer area causes the temperature difference between cold water and refrigerant to increase. This raises the condenser temperature and the corresponding pressure thereby reducing the COP.

22.6. Optimum condenser pressure for lowest running cost

The total running cost of a refrigeration system is the sum of costs of compressor power and the cost of water. The cost of water can be the cost of municipal water or the cost of running a cooling tower. The compressor power increases as the condenser temperature or the pressure increases for fixed evaporator temperature. The water from a cooling tower is usually available at a fixed temperature equal to wet-bulb temperature of air plus the approach of the cooling tower. As the condenser temperature increases the overall *log mean temperature difference* increases, as a result lower mass flow rate of cooling water is required. This reduces the cost of water at higher condenser temperatures. Figure 22.14 shows the general trend of the total running cost of a refrigeration system. It is observed that there is a condenser pressure at which the running cost is minimum and it is recommended that the system should be run at this pressure. A complete analysis of the cost should actually be carried out which should include the first cost of the whole system, the interest on capital, the depreciation, the maintenance cost the operator cost etc. The final selection of the system and operating conditions should be such that the cost is the least over the running life of the system.

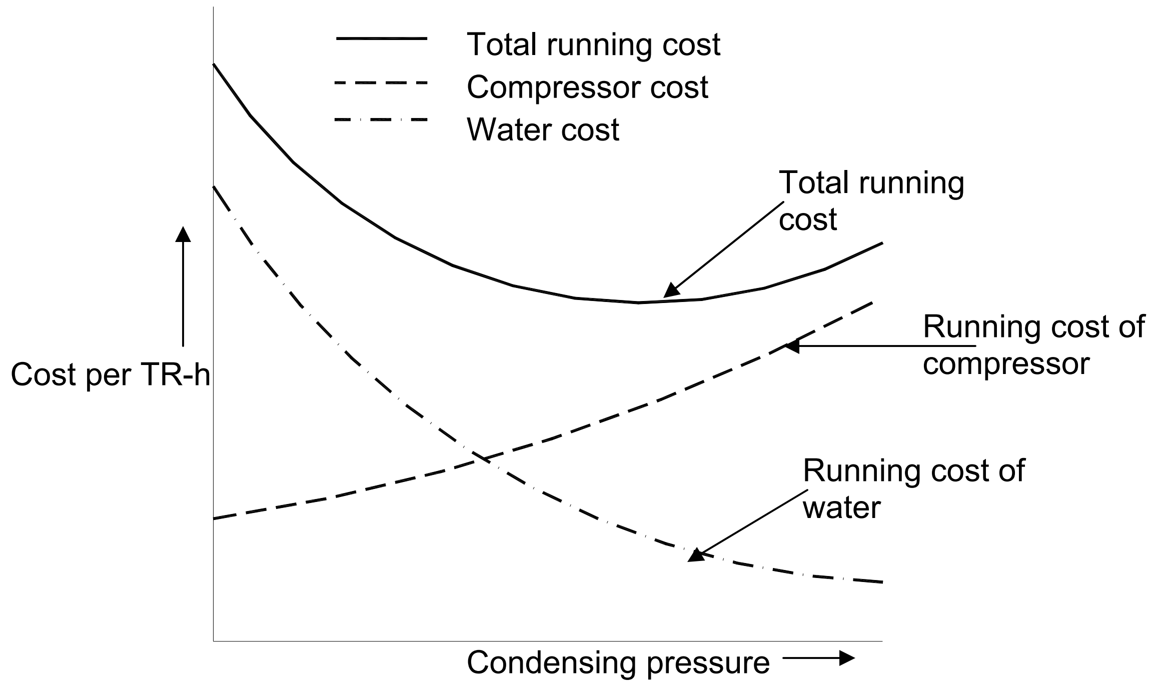


Fig.22.14*: Variation of total running cost of a refrigeration system with condensing pressure*

Questions & answers:

1. Which of the following statements are TRUE?

a) Natural convective type condensers are used in small capacity systems as the overall heat transfer coefficient obtained is small
b) Compared to natural convection type, forced convection type condensers have smaller weight per unit capacity
c) Evaporative condensers are normally used in small capacity systems
d) Compared to water-cooled condensers, the water consumption is high in evaporative condensers

Ans.: a) and b)

2. Which of the following statements are TRUE?

a) Compared to water cooled condensers, the maintenance cost is low in air cooled condensers
b) Normally, systems with water cooled condensers operate at lower condensing temperature as compared to systems with air cooled condensers
c) The initial cost of water cooled condenser is high compared to air cooled condenser
d) All of the above
Ans.: d)

3. Which of the following statements are TRUE?

a) Heat Rejection Ratio increases as evaporator temperature increases and condenser temperature decreases
b) Heat Rejection Ratio increases as evaporator temperature decreases and condenser temperature increases
c) For the same evaporator and condenser temperatures, Heat Rejection Ratio of open type compressors is small compared to hermetic compressors
d) The required size of condenser increases as Heat Rejection Ratio decreases

Ans.: b) and c)

4. The approximation of constant temperature in a condenser generally holds good as:

a) The heat transfer coefficient in de-superheating zone is larger than that in condensing zone
b) The heat transfer coefficient in de-superheating zone is smaller than that in condensing zone
c) The temperature difference between refrigerant and external fluid in de-superheating zone is large compared to condensing zone
d) The temperature difference between refrigerant and external fluid in de-superheating zone is small compared to condensing zone

Ans.: b) and c)

5. Which of the following statements is TRUE?

a) In water-cooled condensers using ammonia, fins are used on refrigerant side due to low condensing heat transfer coefficient
b) In water-cooled condensers using synthetic refrigerants, fins are used on refrigerant side due to low condensing heat transfer coefficient
c) Fouling resistance on external fluid side is negligible in water-cooled condensers
d) Fouling resistance on external fluid side is negligible in air-cooled condensers

Ans.: b) and d)

6. Presence of non-condensible gases in a condenser:

a) Increases the condenser pressure
b) Decreases condenser pressure
c) Increases resistance to heat transfer
d) Decreases COP

Ans.: a), b) and d)

7. The average condensing heat transfer coefficient for a refrigerant condensing on a single horizontal tube is found to be 4000 W/m^2.K. Now another tube is added directly below the first tube. Assuming everything else to remain constant, what will be the new average condensing heat transfer coefficient?

Ans.: From Nusselt's correlation for condensation heat transfer coefficient on the outside of a horizontal tube, we find that when everything else remains constant:

$$\mathbf{h_o \propto \left[\frac{1}{N}\right]^{1/4}}$$ where N is the number of tubes in a vertical row.

From the above equation, the ratio of condensing heat transfer coefficient with 1 tube and 2 tubes is given by:

$$\mathbf{\frac{h_{o,2}}{h_{o,1}} = \left[\frac{1}{2}\right]^{1/4} = 0.8409}$$

$$\Rightarrow \mathbf{h_{o,2} = h_{o,1} \times 0.8409 = 3363.6\ W/m^2.K} \qquad \textbf{(Ans.)}$$

8. A refrigeration system of 55 kW cooling capacity that uses a water-cooled condenser has a COP of 5.0. The overall heat transfer coefficient of the condenser is 450 W/m^2.K and a heat transfer area of 18 m^2. If cooling water at a flow rate of 3.2 kg/s enters the condenser at a temperature of 30^{o}C, what is the condensing temperature? Take the specific heat of water as 4.18kJ/kg.K.

Ans.:

The Heat Rejection Ratio of the system is equal to:

HRR = 1 + 1/COP = 1.2

Hence condenser heat rejection rate, $\mathbf{Q_c}$

$\mathbf{Q_c}$ **= Refrigeration capacity x HRR = 66 kW**

Hence the LMTD of the condenser is equal to:

LMTD = $\mathbf{Q_c/(U.A) = 8.148^oC}$

The exit temperature of water, $\mathbf{T_{w,e} = T_{w,i} + Q_c/(m_w x c_p) = 34.93^oC}$

From the expression for LMTD; **LMTD =** $\mathbf{(T_{w,e}-T_{w,i})/[ln(T_C-T_{w,i})/(T_C-T_{w,e})]}$

We find condensing temperature, $\mathbf{T_c = 40.86^oC}$ **(Ans.)**

9. Find the length of tubes in a two pass 10 TR Shell-and-Tube R-22 based, water-cooled condenser with 52 tubes arranged in 13 columns. The Heat Rejection Ratio (HRR) is 1.2747. The condensing temperature is 45^oC. Water inlet and outlet temperature are 30^oC and 35^oC respectively. The tube outer and inner diameters are 14.0 and 16.0 mm respectively.

Ans.: Average properties of R 22 and water are:

Water	R 22
μ_w = 7.73 x 10^{-4} kg/m-s	μ_f = 1.8 x 10^{-4} kg/m-s
k_w = 0.617 W/m-K	k_f = 0.0779 W/m-K
ρ_w = 995.0 kg/m^3	ρ_f = 1118.9 kg/m^3
C_{pw} = 4.19 kJ/kg-K	h_{fg} = 160.9 kJ/kg
Pr_w = 5.25	

The fouling resistance on water side and thermal conductivity of copper are:

$R''_{f,i}$ = 0.000176 m^2-K/W $\qquad$ k_{cu} = 390 W/m-K

•Heat transfer rate in condenser, Q_C

$$\mathbf{Q_C = HRR.Q_e = 1.2747 \times 10 \times 3.5167 = 44.83\ kW}$$

•Required mass flow rate of water, m_w

$$\mathbf{Q_C = m_w C_{p,w}(T_{w,o}-T_{w,i})}$$

$$\mathbf{m_w = Q_C/C_{p,w}(T_{w,o}-T_{w,i}) = 44.83/4.19X5 = 2.14\ kg/s}$$

Since it is a 2-pass condenser with 52 tubes, water flow through each tube is given by:

$$\mathbf{m_{w,i} = m_w/26 = 0.0823\ kg/s}$$

Reynolds number for water side, Re_w

$$\mathbf{Re_w = 4m_{w,i}/(\pi d_i \mu_w) = 4682.6} \quad (\Rightarrow \text{Turbulent flow})$$

Heat transfer coefficient on water side, h_i

•From Dittus-Boelter Equation:

$$\mathbf{Nu_W = (h_i d_i/k_W) = 0.023 Re_W^{0.8} Pr_W^{0.4} = 68.96}$$

$$\mathbf{h_i = Nu_W \times k_W/d_i = 3039\ W/m^2.K}$$

<u>Condensation heat transfer coefficient, h_o</u>

Nusselt's correlation will be used to estimate h_o:

Number of tubes per row, N = 52/13 = **4**

Substituting the above and other property values in Nusselt's correlation, we obtain:

$$\mathbf{h_o = 2175/\Delta T^{0.25}}$$

$\Delta T = T_{ref}-T_S$ is not known a priori, hence, a trial-and-error method has to be used

For water-cooled condensers without fins; the overall heat transfer coefficient is given by:

$$U_o = \frac{1}{\dfrac{A_o}{h_i A_i} + \dfrac{R''_{f,i} A_o}{A_i} + \dfrac{A_o}{A_i}\dfrac{r_i \ln(d_o/d_i)}{k_w} + \dfrac{1}{h_o}}$$

Substituting the values of various parameters, we obtain:

$$\frac{1}{U_o} = 0.0005781 + \frac{1}{h_o}$$

<u>First trial: Assume $\Delta T = 5^oC$</u>

Then condensation heat transfer coefficient,

$$\mathbf{h_o = 2175/\Delta T^{0.25} = 1454.5\ W/m^2.K}$$

Then the overall heat transfer coefficient is given by:

$$\mathbf{(1/U_o) = 0.0005781+(1/h_o) = 0.0012656\ m^2K/W}$$

Hence, U_o = **790.2 W/m²K**

$$\mathbf{Q_C = U_O A_O LMTD = 44.83\ kW}$$

$$\mathbf{LMTD = (T_{w,o}-T_{w,i})/[ln(T_C-T_{w,i})/(T_C-T_{w,o})] = 12.33\ K}$$

Therefore, $\mathbf{A_O = 4.6\ m^2}$

Now we have cross-check for the initially assumed value of $\Delta T = 5^oC$:

$$\mathbf{\Delta T = Q_C/(h_O.A_O)}$$

•Substituting the value; $\mathbf{\Delta T_{calc} = 6.7\ K}$

Since the calculated value is not equal to the assumed value, we have to repeat the calculation with **ΔT = 7 K (Second trial)**

Repeating the above calculations with ΔT of 7K, we obtain **ΔTcalc = 6.96 K**

Since, this value is sufficiently close to the 2^{nd} guess value of 7K, it is not necessary to repeat the calculations.

For 7 K temperature difference, we obtain the value of **U_O to be 754 W/m^2.K**
From the values of U_O, LMTD and Qc, we obtain;

$$\mathbf{A_O = 4.82\ m^2}$$

$$\mathbf{Now,\ A_O = 56\pi d_O L}$$

Hence, length of each tube, L = 1.713 m (Ans.)

10. Determine the required face area of an R 12 condenser for **5 TR** refrigeration plant. The condensing temperature is 40^oC, the system COP is **4.9** and refrigeration effect is **110.8 kJ/kg**. Air at an inlet temperature of **27^oC** flows through the condenser with a face velocity of **2.5 m/s.** The inside and outside diameters of the tubes are **11.26 and 12.68 mm**, respectively. Fin efficiency is **0.73**. Other dimensions with reference to Fig. 22.12 are:

B = 43 mm; C = 38 mm, D = 3.175 mm, t = 0.254 mm

Ans.: Various heat transfer areas are:

1.Bare area, A_b: (m^2 per row per m^2 face area)

$$A_b = \frac{D - t}{BD} \pi d_o = \frac{3.175 - 0.254}{43x3.175} 3.14159(12.68) = 0.8523$$

2. Fin area, A_f: (m^2 per row per m^2 face area)

$$A_f = \frac{2}{D}\left[C - \frac{\pi d_o^2}{4B}\right] = 22.087$$

3. Min.flow area, A_c:(m^2 /row per m^2 face area)

$$A_c = \frac{D - t}{D}\left[1 - \frac{d_o}{B}\right] = 0.6487$$

Total area, A_o: (m^2/row/m^2 face area)

$$A_o = A_b + A_f = 22.94$$

Internal area, A_i: (m^2/row/m^2 face area)

$$A_i = \pi d_i / B = 0.82266$$

•Hydraulic diameter, D_h: (m)

$$D_h = \frac{4 C A_c}{1000 A_o} = \frac{4(38)0.6487}{1000(22.9393)} = 4.2984 X 10^{-3} \text{ m}$$

Area ratios:

$$A_o / A_i = 27.885$$

$$A_b / A_f = 0.03859$$

Condenser heat rejection rate, Q_c:

$$Q_c = HRR.Q_e = (1+1/COP).Q_e = 21.17 \text{ kW}$$

Mass flow rate of refrigerant, m_r:

$$m_r = Q_e/\text{refrigeration effect} = 0.15869 \text{ kg/s}$$

Condensation Heat Transfer Coefficient:

From the properties of R12 at 40°C:

We find:

Prandtl number, $Pr_f = 3.264$

Reynolds number of vapour, $Re_g = 1385X10^3$

Reynolds number of liquid, $Re_f = 74.8X10^3$

To find condensation heat transfer coefficient inside tubes, we use Dean, Ackers and Crosser's correlation, which assumes complete condensation and uses a modified Reynolds number Re_m

Substituting various property values and Re_f,
We obtain:

Reynolds number, $Re_m = 431383$

The Nusselt number is found to be, Nu = 1265.9

Then the Condensation heat transfer coefficient, h_i is

$$h_i = 8206.7 \text{ W/m}^2\text{.K}$$

<u>Air side heat transfer coefficient, h_o:</u>

$$u_{max} = 2.5/A_c = 3.854 \text{ m/s}$$

Reynolds number, **$Re = U_{max}D_h/\nu = 983.6$**

$$Nu = h_o D_h/k = 0.117\, Re^{0.65}\, Pr^{1/3} = 7.835$$

Heat transfer coefficient, **$h_o = 51.77$ W/m^2-K**

Overall heat transfer coefficient, U_o:

$$U_o = \frac{1}{\frac{A_o}{h_i A_i} + \frac{R''_{f,i} A_o}{A_i} + \frac{A_o}{A_i}\frac{r_i \ln(r_o/r_i)}{k_w} + \frac{A_o}{[h_o(A_f\eta_f + A_b)]_o}}$$

Substituting the values; **$U_o = 31.229$ W/m^2-K**

•Since outlet temperature of air is not given, assume this value to be 35^oC; then

$$LMTD = \frac{(T_{ext,o} - T_{ext,i})}{\ln\left(\frac{T_c - T_{ext,i}}{T_c - T_{ext,o}}\right)} = \frac{(35-27)}{\ln\left(\frac{40-27}{40-35}\right)} = 8.3725^o C$$

Hence, total heat transfer area, A_{ot} is

$$A_{ot} = Q_C/(U_o.LMTD) = 21.17 \text{ X } 1000/(31.229 \text{ X } 8.3725) = 80.967 \text{ m}^2$$

Taking the number of rows to be 4;

$$A_{ot} = A_{face} \text{ x number of rows x } A_o$$
$$A_{face} = 80.967/(22.94 \text{ x } 4) = 0.882 \text{ m}^2$$

•Mass flow rate of air is given by:

$$m_{air} = \rho A_{face}.V = 1.1774 \text{ x } 0.8824 \text{ x } 2.5 = 2.5973 \text{ kg/s}$$

Check for guess value of air outlet temperature (35^{o}C):

$$Q_C = m_{air} C_p \Delta T$$

$$\Rightarrow \Delta T = 21.17/(2.5973 \text{x} 1.005) = 8.11\ ^oC$$

$$\Rightarrow T_{air,out} = 35.11^oC$$

Since the guess value (35^{o}C) is close to the calculated value (35.11^{o}C), we may stop here. For better accuracy, calculations may be repeated with 2nd guess value of 5.1^{o}C (say). The values obtained will be slightly different if other correlations are used for h_i.