

Lesson

23

Condensers & Evaporators

The specific objectives of this lesson are to:

1. Classify refrigerant evaporators as natural convection or forced convection type, flooded or dry type, refrigerant flow inside the tubes or outside the tubes (*Section 23.1*)
2. Discuss salient features of natural convection coils (*Section 23.2*)
3. Discuss salient features of flooded evaporators (*Section 23.3*)
4. Discuss salient features of shell-and-tube type evaporators (*Section 23.4*)
5. Discuss salient features of shell-and-coil evaporator (*Section 23.5*)
6. Discuss salient features of double pipe evaporators (*Section 23.6*)
7. Discuss salient features of Baudelot evaporators (*Section 23.7*)
8. Discuss salient features of direct expansion fin-and-tube type evaporators (*Section 23.8*)
9. Discuss salient features of plate surface evaporators (*Section 23.9*)
10. Discuss salient features of plate type evaporators (*Section 23.10*)
11. Discuss thermal design aspects of refrigerant evaporators (*Section 23.11*)
12. Discuss enhancement of boiling heat transfer (*Section 23.12*)
13. Discuss the concept of Wilson's plot (*Section 23.13*)

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

1. Classify refrigerant evaporators and discuss the salient features of different types of evaporators
2. Perform thermal design calculations on refrigerant evaporators using various heat transfer correlations presented in the lecture
3. Use Wilson's plots and determine external and internal heat transfer coefficients from given experimental data and specifications of evaporators and condensers

Introduction:

An evaporator, like condenser is also a heat exchanger. In an evaporator, the refrigerant boils or evaporates and in doing so absorbs heat from the substance being refrigerated. The name evaporator refers to the evaporation process occurring in the heat exchanger.

23.1. Classification

There are several ways of classifying the evaporators depending upon the heat transfer process or refrigerant flow or condition of heat transfer surface.

23.1.1. Natural and Forced Convection Type

The evaporator may be classified as *natural convection* type or *forced convection* type. In forced convection type, a fan or a pump is used to circulate

the fluid being refrigerated and make it flow over the heat transfer surface, which is cooled by evaporation of refrigerant. In natural convection type, the fluid being cooled flows due to natural convection currents arising out of density difference caused by temperature difference. The refrigerant boils inside tubes and evaporator is located at the top. The temperature of fluid, which is cooled by it, decreases and its density increases. It moves downwards due to its higher density and the warm fluid rises up to replace it.

23.1.2. Refrigerant Flow Inside or Outside Tubes

The heat transfer phenomenon during boiling inside and outside tubes is very different; hence, evaporators are classified as those with flow inside and outside tubes.

In natural convection type evaporators and some other evaporators, the refrigerant is confined and boils inside the tubes while the fluid being refrigerated flows over the tubes. The direct expansion coil where the air is directly cooled in contact with the tubes cooled by refrigerant boiling inside is an example of forced convection type of evaporator where refrigerant is confined inside the tubes.

In many forced convection type evaporators, the refrigerant is kept in a shell and the fluid being chilled is carried in tubes, which are immersed in refrigerant. Shell and tube type brine and water chillers are mainly of this kind.

23.1.3. Flooded and Dry Type

The third classification is flooded type and dry type. Evaporator is said to be *flooded type* if liquid refrigerant covers the entire heat transfer surface. This type of evaporator uses a float type of expansion valve. An evaporator is called *dry type* when a portion of the evaporator is used for superheating the refrigerant vapour after its evaporation.

23.2. Natural Convection type evaporator coils

These are mainly used in domestic refrigerators and cold storages. When used in cold storages, long lengths of bare or finned pipes are mounted near the ceiling or along the high sidewalls of the cold storages. The refrigerant from expansion valve is fed to these tubes. The liquid refrigerant evaporates inside the tubes and cools the air whose density increases. The high-density air flows downwards through the product in the cold storage. The air becomes warm by the time it reaches the floor as heat is transferred from the product to air. Some free area like a passage is provided for warm air to rise up. The same passage is used for loading and unloading the product into the cold storage.

The advantages of such natural convection coils are that the coil takes no floor space and it also requires low maintenance cost. It can operate for long

periods without defrosting the ice formed on it and it does not require special skill to fabricate it. Defrosting can be done easily (e.g. by scraping) even when the plant is running. These are usually welded at site. However, the disadvantage is that natural convection heat transfer coefficient is very small hence very long lengths are required which may cause excessive refrigerant side pressure drops unless parallel paths are used. The large length requires a larger quantity of refrigerant than the forced convection coils. The large quantity of refrigerant increases the time required for defrosting, since before the defrosting can start all the liquid refrigerant has to be pumped out of the evaporator tubes. The pressure balancing also takes long time if the system trips or is to be restarted after load shedding. Natural convection coils are very useful when low air velocities and minimum dehumidification of the product is required. Household refrigerators, display cases, walk-in-coolers, reach-in refrigerators and obviously large cold storages are few of its applications. Sufficient space should be provided between the evaporator and ceiling to permit the air circulation over the top of the coil. Baffles are provided to separate the warm air and cold air plumes. Single ceiling mounted is used for rooms of width less than 2.5 m. For rooms with larger widths more evaporator coils are used. The refrigerant tubes are made of steel or copper. Steel tubes are used for ammonia and in large capacity systems.

23.3.Flooded Evaporator

This is typically used in large ammonia systems. The refrigerant enters a surge drum through a float type expansion valve. The compressor directly draws the flash vapour formed during expansion. This vapour does not take part in refrigeration hence its removal makes the evaporator more compact and pressured drop due to this is also avoided. The liquid refrigerant enters the evaporator from the bottom of the surge drum. This boils inside the tubes as heat is absorbed. The mixture of liquid and vapour bubbles rises up along the evaporator tubes. The vapour is separated as it enters the surge drum. The remaining unevaporated liquid circulates again in the tubes along with the constant supply of liquid refrigerant from the expansion valve. The mass flow rate in the evaporator tubes is $f \cdot \dot{m}$ where \dot{m} is the mass flow rate through the expansion valve and to the compressor. The term f is called recirculation factor. Let x_4 be the quality of mixture after the expansion valve and x be the quality of mixture after boiling in the tubes as shown in Figure 23.1. In steady state mass flow rate from expansion valve is same as the mass flow rate to the compressor hence mass conservation gives

$$x_4 \cdot \dot{m} + x \cdot f \cdot \dot{m} = \dot{m} \quad (23.1)$$

$$\therefore f = \frac{(1 - x_4)}{x} \quad (23.2)$$

For $x_4 = x = 0.25$, for example, the circulation factor is 3, that is mass flow rate through the evaporator is three times that through the compressor. Since, liquid refrigerant is in contact with whole of evaporator surface, the refrigerant side heat transfer coefficient will be very high. Sometimes a liquid refrigerant pump may also be used to further increase the heat transfer coefficient. The lubricating oil tends to accumulate in the flooded evaporator hence an effective oil separator must be used immediately after the compressor.

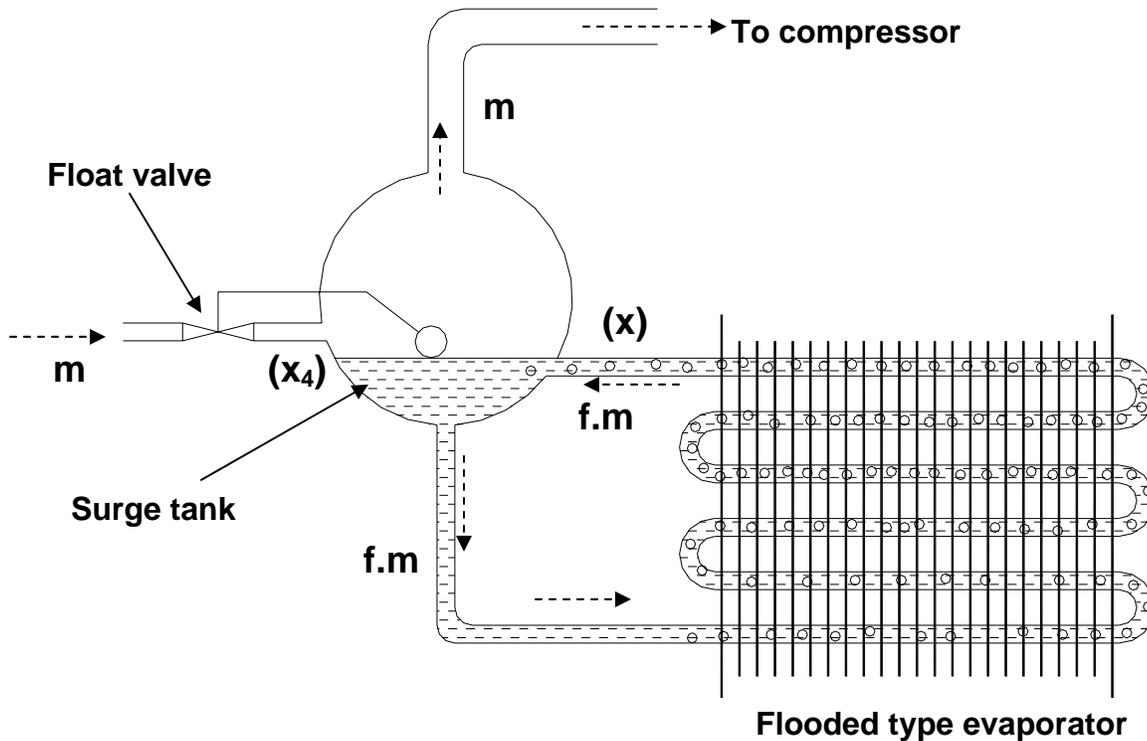


Fig.23.1. Schematic of a flooded evaporator

23.4. Shell-and-Tube Liquid Chillers

The shell-and-tube type evaporators are very efficient and require minimum floor space and headspace. These are easy to maintain, hence they are very widely used in medium to large capacity refrigeration systems. The shell-and-tube evaporators can be either dry type or flooded type. As the name implies, a shell-and-tube evaporator consists of a shell and a large number of straight tubes arranged parallel to each other. In dry expansion type, the refrigerant flows through the tubes while in flooded type the refrigerant is in the shell. A pump circulates the chilled water or brine. The shell diameters range from 150 mm to 1.5 m. The number of tubes may be less than 50 to several thousands and length may be between 1.5 m to 6 m. Steel tubes are used with ammonia while copper tubes are used with freons. Ammonia has a very high heat transfer coefficient while freons have rather poor heat transfer coefficient hence fins are used on the refrigerant side. Dry expansion type uses fins inside the tubes while flooded type uses fins outside the tube. Dry-expansion type require less charge of refrigerant and have positive lubricating oil return. These are used for small and medium capacity refrigeration plants with capacity ranging from 2 TR to 350 TR. The flooded type evaporators are available in larger capacities ranging from 10 TR to thousands of TR.

23.4.1 Flooded Type Shell-and-Tube Evaporator

Figure 23.2 shows a flooded type of shell and tube type liquid chiller where the liquid (usually brine or water) to be chilled flows through the tubes in double pass just like that in shell and tube condenser. The refrigerant is fed through a float valve, which maintains a constant level of liquid refrigerant in the shell. The shell is not filled entirely with tubes as shown in the end view of Fig. 27.2. This is done to maintain liquid refrigerant level below the top of the shell so that liquid droplets settle down due to gravity and are not carried by the vapour leaving the shell. If the shell is completely filled with tubes, then a surge drum is provided after the evaporator to collect the liquid refrigerant.

Shell-and-tube evaporators can be either single pass type or multipass type. In multipass type, the chilled liquid changes direction in the heads. Shell-and-tube evaporators are available in vertical design also. Compared to horizontal type, vertical shell-and-tube type evaporators require less floor area. The chilled water enters from the top and flows downwards due to gravity and is then taken to a pump, which circulates it to the refrigeration load. At the inlet to tubes at the top a special arrangement introduces swirling action to increase the heat transfer coefficient.

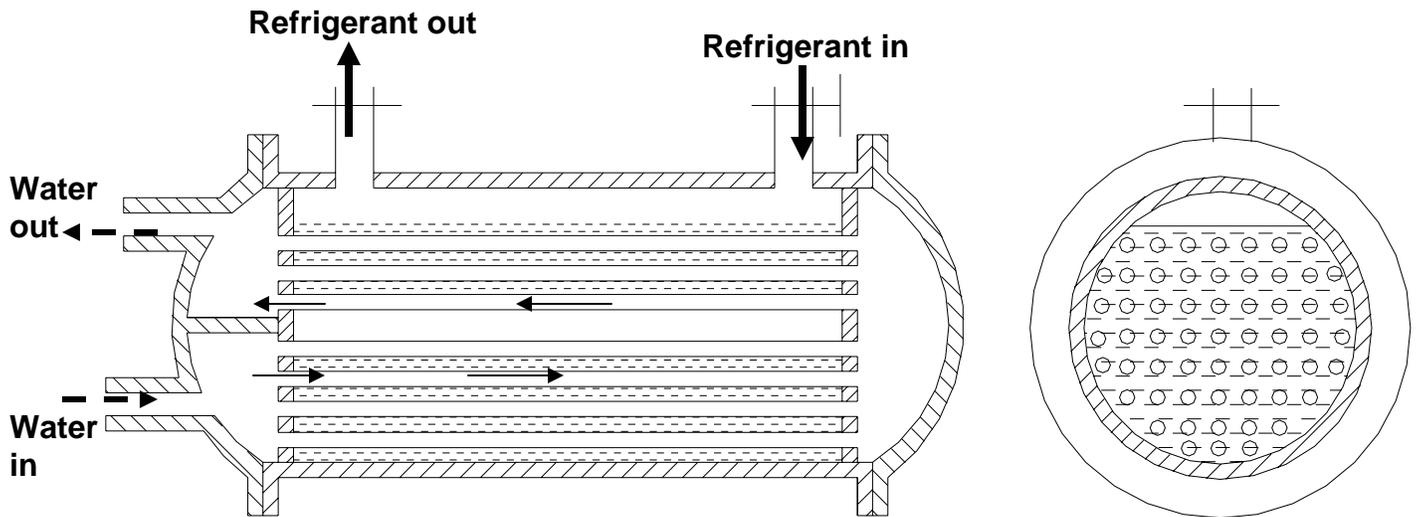


Fig.23.2: Schematic of a flooded type shell-and-tube evaporator

23.4.2. Direct expansion type, Shell-and-Tube Evaporator

Figure 23.3 shows a liquid chiller with refrigerant flowing through the tubes and water flowing through the shell. A thermostatic expansion valve feeds the refrigerant into the tubes through the cover on the left. It may flow in several passes through the dividers in the covers of the shell on either side. The liquid to be chilled flows through the shell around the baffles. The presence of baffles turns the flow around creating some turbulence thereby increasing the heat transfer coefficient. Baffles also prevent the short-circuiting of the fluid flowing in the shell. This evaporator is of dry type since some of the tubes superheat the vapour. To maintain the chilled liquid velocity so as to obtain good heat transfer coefficient, the length and the spacing of segmental baffles is varied. Widely spaced baffles are used when the flow rate is high or the liquid viscosity is high. The number of passes on the refrigerant side are decided by the partitions on the heads on the two sides of the heat exchanger. Some times more than one circuit is also provided. Changing the heads can change the number of passes. It depends upon the chiller load and the refrigerant velocity to be maintained in the heat exchanger.

23.5. Shell-and-Coil type evaporator

These are of smaller capacity than the shell and tube chillers. These are made of one or more spiral shaped bare tube coils enclosed in a welded steel shell. It is usually dry-expansion type with the refrigerant flowing in the tube and chilled liquid in the shell. In some cases the chiller operates in flooded mode also with refrigerant in the shell and chilled water flowing thorough the spiral tube. The water in the shell gives a large amount of thermal storage capacity called *hold-up*

capacity. This type is good for small but highly infrequent peak loads. It is used for cooling drinking water in stainless steel tanks to maintain sanitary conditions. It is also used in bakeries and photographic laboratories.

When the refrigerant is in the shell that is in flooded mode it is called instantaneous liquid chiller. This type does not have thermal storage capacity, the liquid must be instantaneously chilled whenever required. In the event of freeze up the water freezes in the tube, which causes bursting of the tubes since water expands upon freezing. When water is in the shell there is enough space for expansion of water if the freezing occurs. The flooded types are not recommended for any application where the temperature of chilled liquid may be below 3°C.

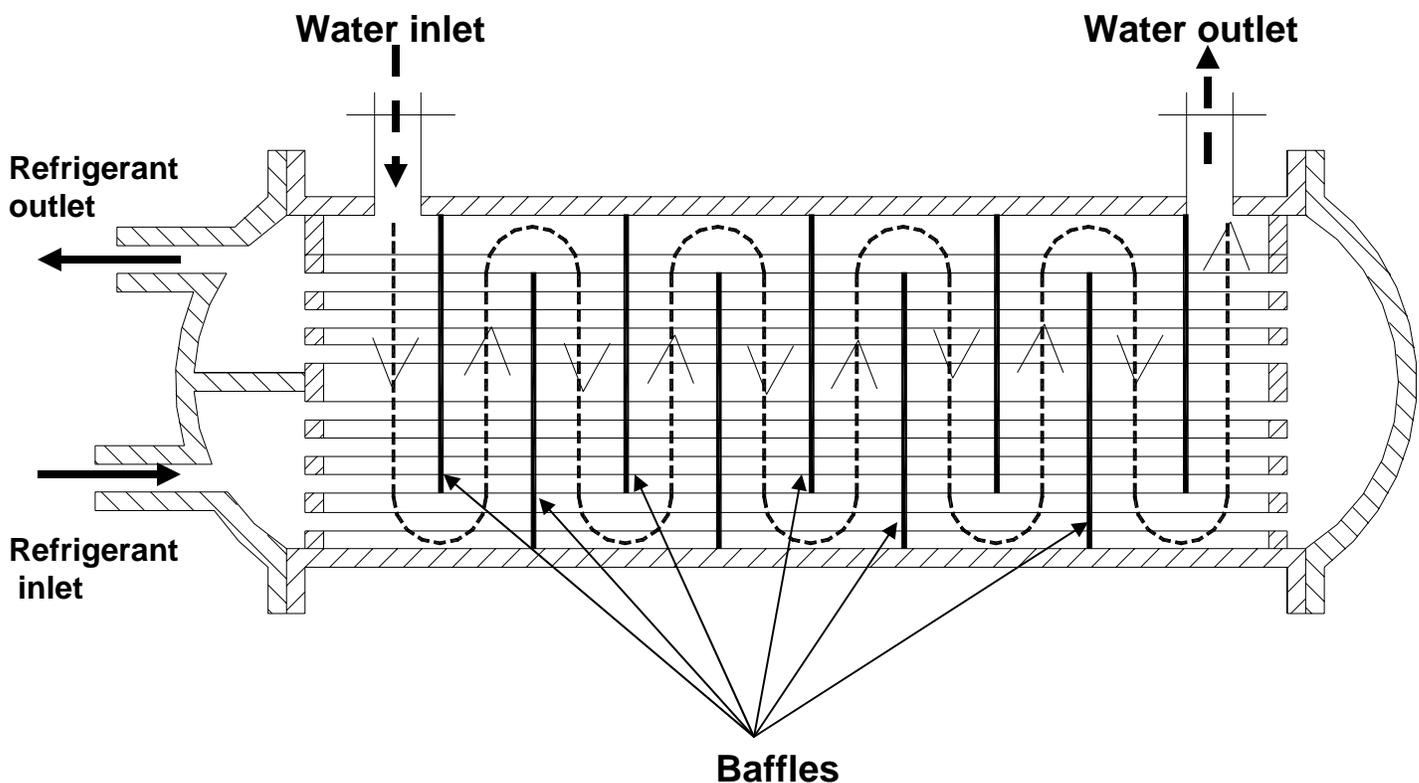


Fig.23.3: Schematic of a direct expansion type, Shell-and-Tube evaporator

23.6. Double pipe type evaporator

This consists of two concentric tubes, the refrigerant flows through the annular passage while the liquid being chilled flows through the inner tube in counter flow. One design is shown in Fig. 23.4 in which the outer horizontal tubes are welded to vertical header tubes on either side. The inner tubes pass through the headers and are connected together by 180° bends. The refrigerant side is welded hence there is minimum possibility of leakage of refrigerant. These may

be used in flooded as well as dry mode. This requires more space than other designs. Shorter tubes and counter flow gives good heat transfer coefficient. It has to be insulated from outside since the refrigerant flows in the outer annulus which may be exposed to surroundings if insulation is not provided.

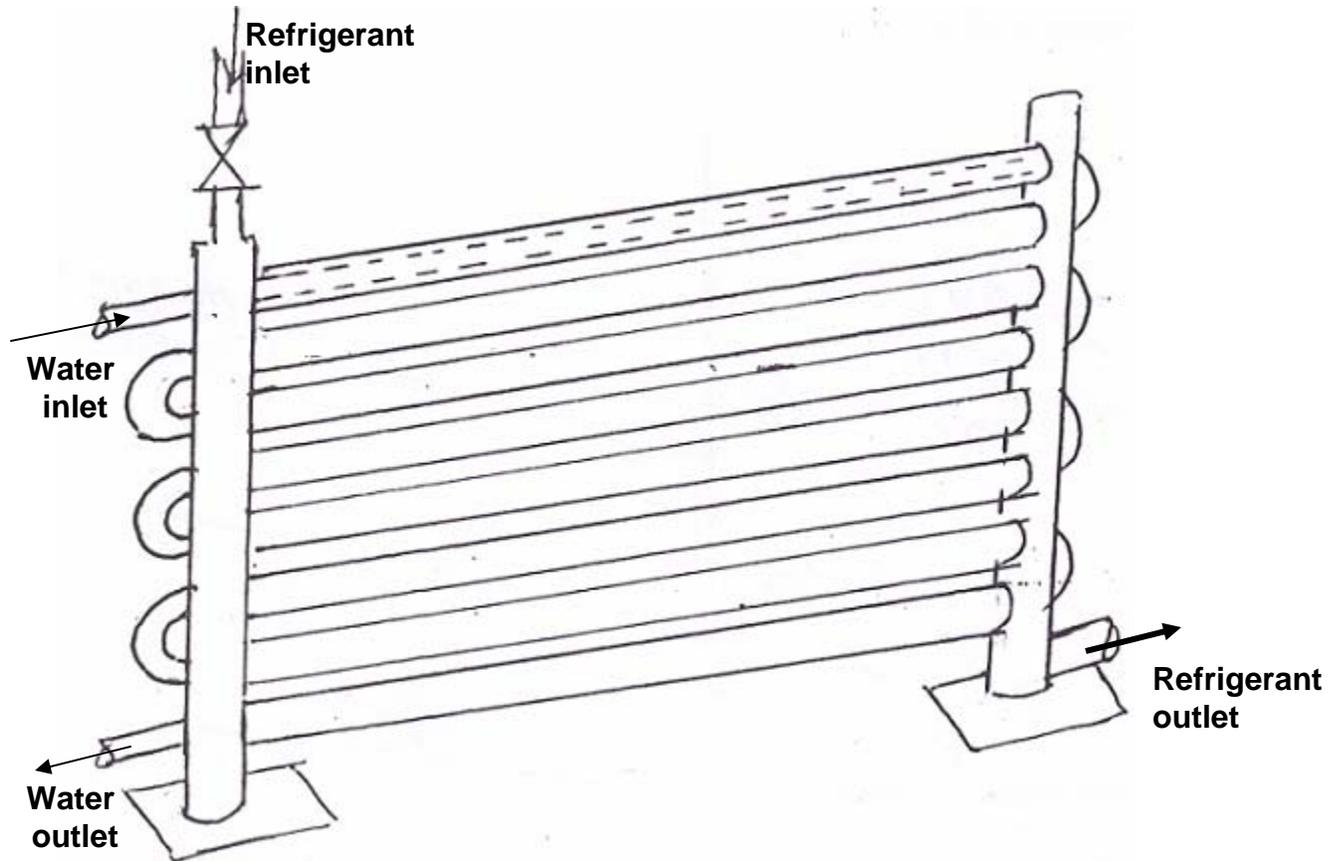


Fig.23.4: Schematic of a double pipe type evaporator

23.7. Baudelot type evaporators

This type of evaporator consists of a large number of horizontal pipes stacked one on top of other and connected together to by headers to make single or multiple circuits. The refrigerant is circulated inside the tubes either in flooded or dry mode. The liquid to be chilled flows in a thin layer over the outer surface of the tubes. The liquid flows down by gravity from distributor pipe located on top of the horizontal tubes as shown in Figure 23.5. The liquid to be chilled is open to atmosphere, that is, it is at atmospheric pressure and its aeration may take place during cooling. This is widely used for cooling milk, wine and for chilling water for carbonation in bottling plants. The liquid can be chilled very close to its freezing temperature since freezing outside the tubes will not damage the tubes. Another advantage is that the refrigerant circuit can be split into several parts, which

permit a part of the cooling done by cold water and then chilling by the refrigerant.

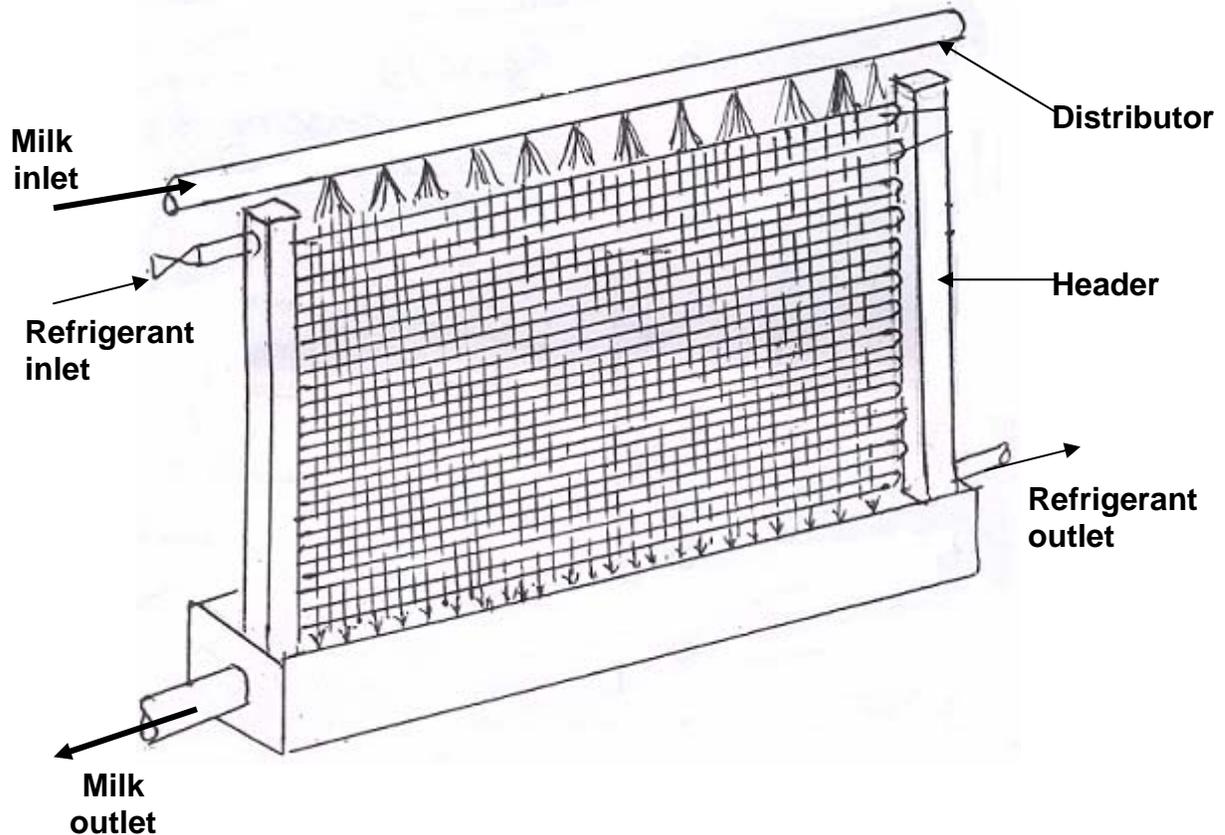


Fig.23.5: Schematic of a Baudelot type evaporator for chilling of milk

23.8. Direct expansion fin-and-tube type

These evaporators are used for cooling and dehumidifying the air directly by the refrigerant flowing in the tubes. Similar to fin-and-tube type condensers, these evaporator consists of coils placed in a number of rows with fins mounted on it to increase the heat transfer area. Various fin arrangements are used. Tubes with individual spiral straight fins or crimped fins welded to it are used in some applications like ammonia. Plate fins accommodating a number of rows are used in air conditioning applications with ammonia as well as synthetic refrigerants such as fluorocarbon based refrigerants.

The liquid refrigerant enters from top through a thermostatic expansion valve as shown in Fig. 23.6. This arrangement makes the oil return to compressor better rather than feeding refrigerant from the bottom of the coil. When evaporator is close to the compressor, a direct expansion coil is used

since the refrigerant lines are short, refrigerant leakage will be less and pressure drop is small. If the air-cooling is required away from the compressor, it is preferable to chill water and pump it to air-cooling coil to reduce the possibility of refrigerant leakage and excessive refrigerant pressure drop, which reduces the COP.

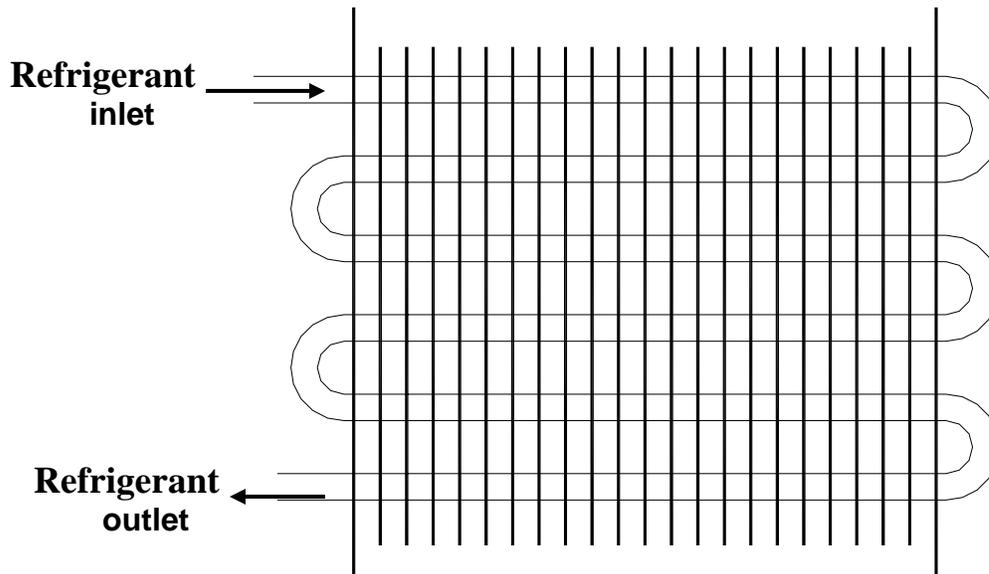


Fig.23.6: Schematic of a direct expansion fin-and-tube type

The fin spacing is kept large for larger tubes and small for smaller tubes. 50 to 500 fins per meter length of the tube are used in heat exchangers. In evaporators, the atmospheric water vapour condenses on the fins and tubes when the metal temperature is lower than dew point temperature. On the other hand frost may form on the tubes if the surface temperature is less than 0°C . Hence for low temperature coils a wide spacing with about 80 to 200 fins per m is used to avoid restriction of flow passage due to frost formation. In air-conditioning applications a typical fin spacing of 1.8 mm is used. Addition of fins beyond a certain value will not increase the capacity of evaporator by restricting the airflow. The frost layer has a poor thermal conductivity hence it decreases the overall heat transfer coefficient apart from restricting the flow. Therefore, for applications in freezers below 0°C , frequent defrosting of the evaporator is required.

23.9. Plate Surface Evaporators

These are also called *bonded plate or roll-bond type evaporators*. Two flat sheets of metal (usually aluminum) are embossed in such a manner that when these are welded together, the embossed portion of the two plates makes a passage for refrigerant to flow. This type is used in household refrigerators. Figure 23.7 shows the schematic of a roll-bond type evaporator.

23.10. Plate type evaporators:

Plate type evaporators are used when a close temperature approach (0.5 K or less) between the boiling refrigerant and the fluid being chilled is required. These evaporators are widely used in dairy plants for chilling milk, in breweries for chilling beer. These evaporators consist of a series of plates (normally made of stainless steel) between which alternately the milk or beer to be cooled and refrigerant flow in counterflow direction. The overall heat transfer coefficient of these plate type evaporators is very high (as high as $4500 \text{ W/m}^2\text{K}$ in case of ammonia/water and $3000 \text{ W/m}^2\text{K}$ in case of R 22/water). In addition they also require very less refrigerant inventory for the same capacity (about 10 percent or even less than that of shell-and-tube type evaporators). Another important advantage when used in dairy plants and breweries is that, it is very easy to clean the evaporator and assemble it back as and when required. The capacity can be increased or decreased very easily by adding or removing plates. Hence these evaporators are finding widespread use in a variety of applications. Figure 23.9 shows the schematic of a plate type evaporator.

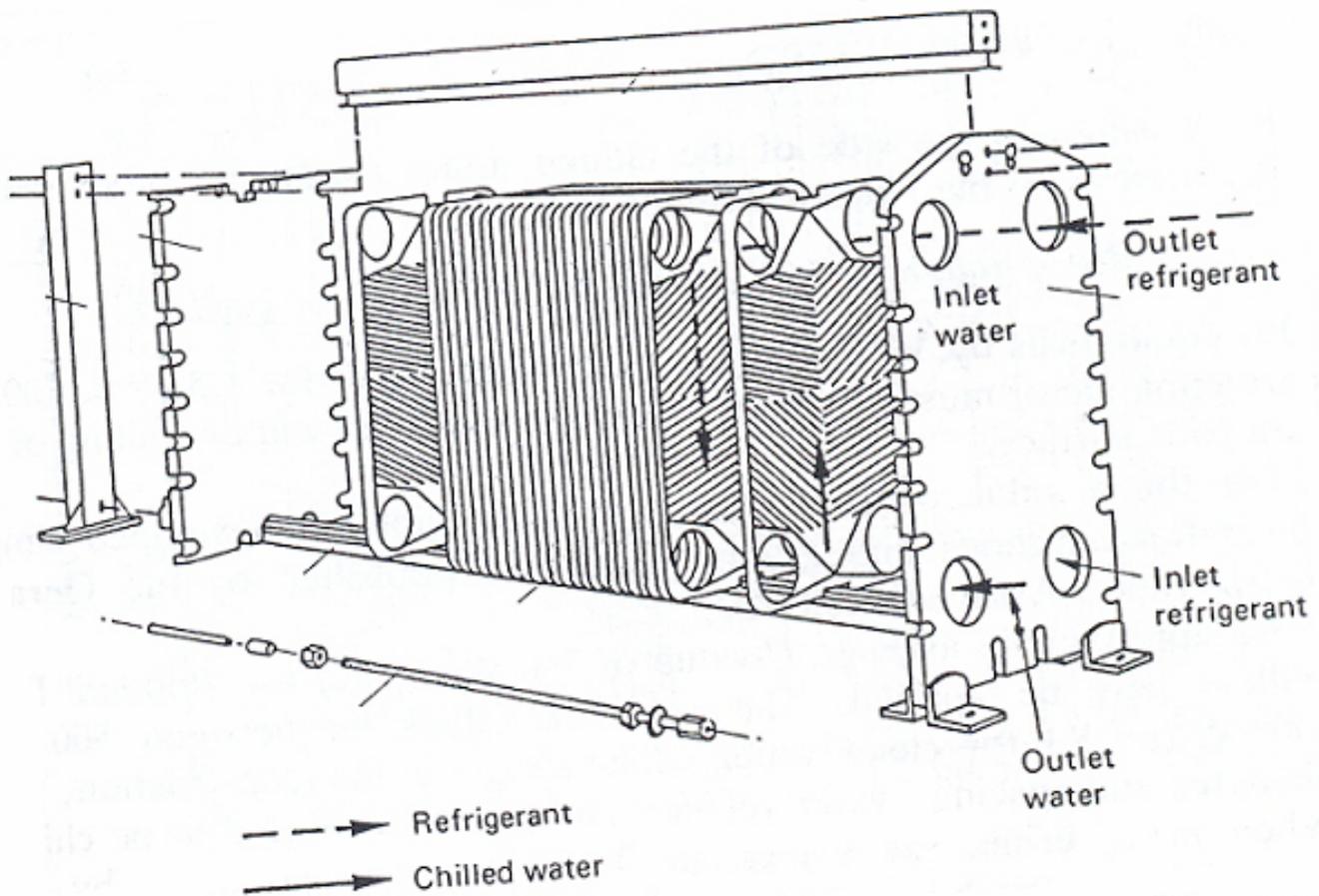


Fig.23.9: Schematic of a plate type evaporator

23.11. Thermal design of evaporators:

Compared to the design of refrigerant condensers, the design of refrigerant evaporators is more complex. The complexity arises due to the following factors:

- a) On the refrigerant side, the heat transfer coefficient varies widely when evaporation takes place in tubes due to changing flow regimes. Accurate estimation of heat transfer coefficient is thus difficult
- b) On the external fluid side, if the external fluid is air (as in air conditioning and cold storage applications), in addition to sensible heat transfer, latent heat transfer also takes place as moisture in air may condense or even freeze on the evaporator surface. The evaporator surface may be partly dry and partly wet, depending upon the operating conditions. Hence, mass transfer has to be considered in the design. If frost formation due to freezing of moisture takes place, then heat transfer resistance varies continuously with time.
- c) The lubricating oil gets separated in the evaporator tubes due to low miscibility of oil at evaporator temperature and pressure. The separation of oil affects both heat transfer and pressure drop characteristics. A minimum refrigerant velocity must be provided for oil carry over in direct expansion type evaporators.
- d) Compared to condenser, refrigerant pressure drop in evaporator is more critical as it has significant influence on the performance of the refrigeration system. Hence, multiple circuits may have to be used in large systems to reduce pressure drops. Refrigerant velocity has to be optimized taking pressure drop and oil return characteristics into account.
- e) Under part-load applications, there is a possibility of evaporator flooding and compressor slugging. This aspect has to be considered at the time of evaporator design.

Estimation of heat transfer area and overall heat transfer coefficients

For plate fin type evaporators, the expressions of various heat transfer areas are similar to those given for the air-cooled condensers. The expression for overall heat transfer coefficient is also similar to that of condenser as long as no phase change (e.g. moisture condensation or freezing) takes place. However, as mentioned in air-cooled evaporators the possibility of moisture condensing/freezing on the evaporator surface must be considered unlike in condensers where the heat transfer on airside is only sensible. This requires simultaneous solution of heat and mass transfer equations on the airside to arrive at expressions for overall heat transfer coefficient and mean temperature difference. The efficiency of the fins will also be affected by the presence of condensed layer of water or a frozen layer of ice. Expressions have been derived for overall heat transfer coefficient, mean temperature difference and fin efficiency of fin-and-tube type evaporators in which air undergoes cooling and

dehumidification. The analysis of cooling and dehumidification coils requires knowledge of psychrometry and is obviously much more complicated compared to evaporators in which the external fluid does not undergo phase change. In this lecture, only the evaporators wherein the external fluid does not undergo any phase change are considered. Readers should refer to advanced books on refrigeration for the design aspects of cooling and dehumidifying coils.

Estimation of heat transfer coefficients:

a) Air side heat transfer coefficients in fin-and-tube type evaporators:

If air undergoes only sensible cooling as it flows over the evaporator surface (i.e., dry evaporator), then the correlations presented for air cooled condensers for heat transfer coefficients on finned (e.g. Kays & London correlation) and bare tube surface (e.g. Grimson's correlation) can be used for air cooled evaporator also. However, if air undergoes cooling and dehumidification, then analysis will be different and correlations will also be different. These aspects will be discussed in a later chapter.

b) Liquid side heat transfer coefficients:

Liquid flowing in tubes:

When liquids such as water, brine, milk etc. flow through tubes without undergoing any phase changes, the correlations presented earlier for condensers (e.g. *Dittus-Boelter*, *Sieder-Tate*) can be used for evaporator also.

Liquid flowing in a shell:

In direct expansion type, shell-and-tube evaporators refrigerant flows through the tubes, while water or other liquids flow through the shell. Analytical prediction of single phase heat transfer coefficient on shell side is very complex due to the complex fluid flow pattern in the presence of tubes and baffles. The heat transfer coefficient and pressure drop depends not only on the fluid flow rate and its properties, but also on the arrangement of tubes and baffles in the shell. Several correlations have been suggested to estimate heat transfer coefficients and pressure drops on shell side. A typical correlation suggested by Emerson is given below:

$$\text{Nu} = \frac{hd}{k_f} = C \text{Re}_d^{0.6} \text{Pr}^{0.3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (23.3)$$

where constant C depends on the geometry, i.e, on the arrangement of the tubes, baffles etc.

In the above expression the Reynolds number Re_d is defined as:

$$Re_d = \frac{Gd}{\mu} \quad (23.4)$$

where G is the mass velocity which is equal to the mass flow rate divided by the characteristic flow area ($\text{kg}/\text{m}^2 \cdot \text{s}$). From the expression for Nusselt number, it can be seen that the heat transfer coefficient is proportional to the 0.6 power of the flow rate as compared to 0.8 power for flow through tubes.

The pressure drop of liquid flowing through the shell is also difficult to predict analytically. Normally the pressure drop on shell side is obtained from experimental measurements and is provided in the form of tables and charts for a particular type of shell-and-tube heat exchanger.

c) Boiling Heat Transfer Coefficients:

Pool boiling vs flow boiling:

In evaporators boiling of refrigerant may take place outside tubes or inside tubes. When boiling takes place outside the tubes it is called as pool boiling. In pool boiling it is assumed that the tube or the heat transfer surface is immersed in a pool of liquid, which is at its saturation temperature. Figure 23.10 shows a typical boiling curve, which shows the variation of surface heat flux with temperature difference between the surface and the saturation temperature for different regimes. For a small temperature difference, the heat transfer from the surface is by free convection (regime 1). As the temperature difference increases, bubbles start to form at selected nucleation sites. The bubbles grow in size as heat is transferred and the evaporation of liquid occurs. After achieving a critical diameter depending upon the surface tension and other factors, the bubbles get detached from the surface and rise to the free surface where the vapour inside the bubbles is released. During the detachment process, the surrounding liquid rushes towards the void created and also during the bubble motion upwards convection heat transfer increases from its free convection value at smaller temperature differences. This region is known as *individual bubble regime* (regime 2). As the temperature difference increase further, more and more bubbles are formed and it is the columns of bubbles, which rise up increasing the heat transfer drastically. This regime is known as *column bubble regime* (regime 3).

As the temperature difference increases further, more and more bubbles are formed, and columns of bubbles rise to the free surface. The heat transfer rate increases rapidly. As the bubble columns move upwards they entrain some liquid also that rises upwards to the free surface. The vapour in the bubbles escapes at the free surface but the liquid returns to the bottom because of its lower temperature and higher density. A given surface can accommodate only a few such rising columns of bubbles and descending columns of relatively colder

liquid. Hence, the heat transfer rate cannot increase beyond a certain value. It becomes maximum at some temperature difference. The maximum heat transfer rate is called *critical heat transfer rate*.

If temperature difference is increased beyond this value, then a blanket of film forms around the heat transfer surface. This vapour film offers conduction thermal resistance; as a result the heat transfer rate decreases. The film however is unstable and may break at times. This regime is called *unstable film regime (regime 4)*.

If temperature difference is increased further it becomes so high that radiation heat transfer becomes very important and heat transfer rate increases because of radiation component. This regime is called *stable film boiling regime (regime 5)*. After this, due to the high surface temperature, radiation effects become important (*regime 6*).

As the temperature difference is increased, the temperature of the surface t_w continues to increase since conduction thermal resistance of the film becomes larger as the film thickness increases. All the heat from the surface cannot be transferred across the film and surface temperature increases. Ultimately the temperature may approach the melting point of the metal and severe accident may occur (if these are the tubes of nuclear power plant). This point is referred to as burnout point.

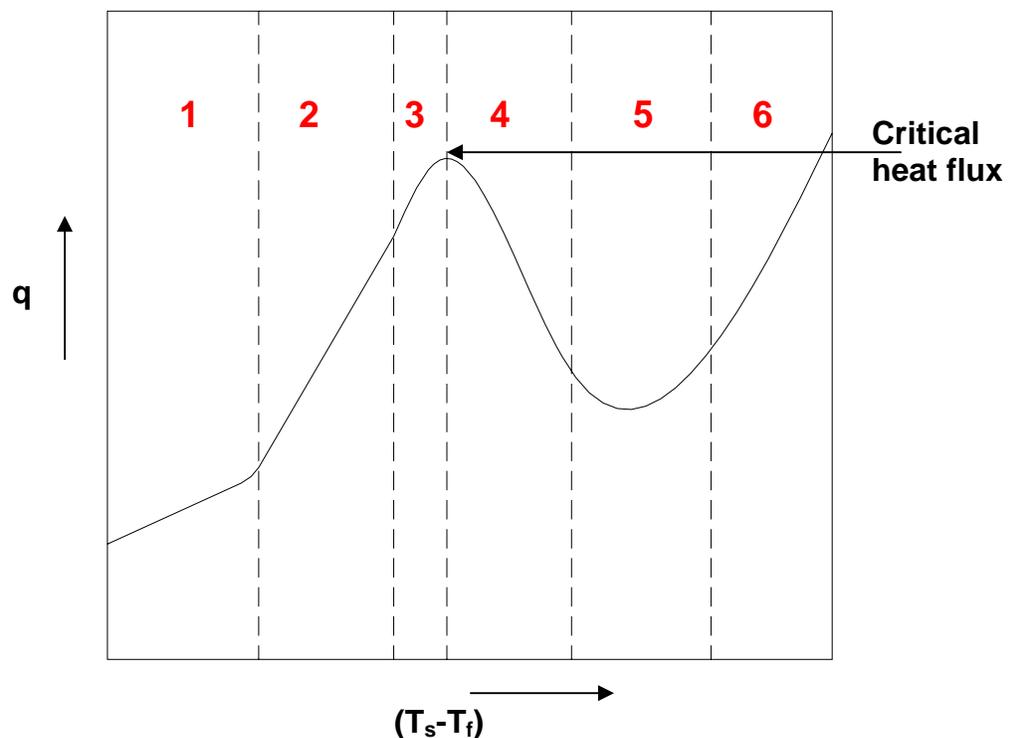


Fig.23.10: A typical pool boiling curve showing different regimes, 1 to 6

Boiling inside tubes is called as flow boiling. Flow boiling consists of nucleate boiling as well as convective heat transfer. As the liquid evaporates, more vapour is formed which increases the average velocity and the convective heat transfer rate. The flow pattern changes continuously as boiling takes place along the tube. For example in a horizontal tube, the flow can be stratified flow, wavy flow, slug flow, annular flow, mist flow etc. The flow pattern will be different if it takes place in an inclined or vertical tube. The heat transfer coefficient depends upon fraction of vapour present and parameters of forced convection heat transfer. In general, prediction of boiling heat transfer coefficients during flow boiling is much more complex than pool boiling. However, a large number of empirical correlations have been developed over the years to predict boiling heat transfer coefficients for both pool as well as flow boiling conditions. The following are some of the well-known correlations:

Nucleate Pool Boiling

Normally evaporators are designed to operate in nucleate pool boiling regime as the heat transfer coefficients obtained in this regime are stable and are very high. Various studies show that in nucleate pool boiling region, the heat transfer coefficient is proportional to the 2 or 3 power of temperature difference between the surface and the boiling fluid, i.e.,

$$h_{nb} = C(T_s - T_f)^{2 \text{ to } 3} \quad (23.5)$$

the value of C depends upon type of the surface etc. The exponent can be as high as 25 on specially treated surfaces for enhancement of boiling.

Rohsenow's Correlation for nucleate pool boiling: This correlation is applicable to clean surfaces and is relatively independent of shape and orientation of the surface.

$$\frac{C_f \Delta T_x}{h_{fg}} = C_{sf} \left[\frac{Q/A}{\mu_f h_{fg}} \sqrt{\frac{\sigma}{g(\rho_f - \rho_g)}} \right]^{0.33} Pr_f^s \quad (23.6)$$

where:

C_f = Specific heat of liquid

ΔT_x = Temperature difference between surface and fluid

h_{fg} = Latent heat of vaporization

σ = Surface Tension

C_{sf} = constant which depends on the surface-fluid combination, e.g. 0.013 for halocarbons boiling on copper surface

Q/A = heat flux

μ_f = Viscosity of fluid

ρ_f, ρ_g = Density of saturated liquid and saturated vapour, respectively

Pr_f = Prandtl number of saturated liquid

s = constant, 1 for water and 1.7 for halocarbons

All the fluid properties are calculated at saturation temperature corresponding to the local pressure.

Forced Convection Boiling inside tubes:

Rohsenow and Griffith suggested that flow boiling in tubes be analyzed as a combination of pool boiling and forced convection. The total heat flux (q_{total}) is the sum of heat flux due to nucleate pool boiling (q_{nb}) and forced convection (q_{fc}), i.e.,

$$q_{total} = q_{nb} + q_{fc} \quad (23.7)$$

Heat flux due to nucleate pool boiling (q_{nb}) is calculated by using nucleate pool boiling correlations and heat flux due to forced convection (q_{fc}) can be calculated by using standard forced convection correlations, such as Dittus-Boelter correlation.

Some of the other correlations suggested for flow boiling are given below:

(a) Bo Pierre's Correlation : This correlation gives average heat transfer coefficients and is valid for inlet quality $x_{inlet} \approx 0.1$ to 0.16.

$$\bar{Nu}_f = 0.0009 \left(Re_f^2 K_f \right)^{1/2} \quad \text{: for incomplete evaporation and } x_{exit} < 0.9$$

$$\bar{Nu}_f = 0.0082 \left(Re_f^2 K_f \right)^{1/2} \quad \text{: for complete evaporation} \quad (23.8)$$

In the above equations, Re_f and Nu_f are liquid Reynolds and Nusselt numbers, respectively. K_f is the load factor, defined as:

$$K_f = \frac{\Delta x h_{fg}}{L} \quad (23.9)$$

where L is the length of the tube.

(b) Chaddock-Brunemann's Correlation:

$$h_{TP} = 1.91 h_L \left[Bo \cdot 10^4 + 1.5 (1/X_{tt})^{0.67} \right]^{0.6} \quad (23.10)$$

$$Bo = \text{Boiling Number} = \frac{Q/A}{h_{fg} (\dot{m}/A)}$$

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} (\rho_g / \rho_f)^{0.5} (\mu_f / \mu_g)^{0.1} \quad \text{Lockhart - Martinelli Parameter}$$

(c) Jung and Radermacher Correlation:

$$h_{TP} = N_1 h_{sa} + F_1 h_L \quad (23.11)$$

where h_L is the single phase (liquid) heat transfer coefficient as predicted by Dittus-Boelter equation, and h_{sa} is given by:

$$h_{sa} = 207 \frac{k_f}{bd} \left(\frac{q}{k_f} \cdot \frac{bd}{T_{sat}} \right)^{0.745} \left(\frac{\rho_g}{\rho_f} \right)^{0.581} Pr_f^{0.533}$$

$$bd = 0.0146 \beta \left[\frac{2\sigma}{g(\rho_f - \rho_g)} \right]^{-0.5} \quad : \beta = 35^\circ \quad (23.12)$$

$$N_1 = 4048 X_{tt}^{1.22} Bo^{1.13} \quad : \text{for } X_{tt} \leq 1$$

$$N_1 = 2.0 - 0.1 X_{tt}^{-0.28} Bo^{-0.33} \quad : \text{for } 1 < X_{tt} \leq 5$$

$$F_1 = 2.37 (0.29 + 1/X_{tt})^{0.85}$$

In nucleate boiling, the heat transfer coefficient is mainly dependent on the heat flux and is a very weak function of mass flux. However, in flow boiling the heat transfer coefficient depends mainly on mass flux and is a weak function of heat flux. Studies show that for boiling inside tubes, initially when the vapour fraction (quality) is low, then nucleate boiling is dominant and the heat transfer coefficient depends on heat flux. However, as the fluid flows through the tubes, the vapour fraction increases progressively due to heat transfer and when it exceeds a critical vapour fraction, convective boiling becomes dominant. As mentioned, in this region, the heat transfer coefficient depends mainly on the mass flux and is almost independent of heat flux. As a whole, the heat transfer coefficient due to boiling increases initially reaches a peak and then drops towards the end of the tube. Thus accurate modeling of evaporators requires estimation of heat transfer coefficient along the length taking into account the complex physics.

Horizontal vs Vertical tubes: As mentioned before, boiling heat transfer coefficients in vertical columns will be different from that in a horizontal tube. In a vertical tube, due to hydrostatic head, the evaporation temperature increases, which in turn reduces the driving temperature difference, and hence, the heat transfer rate.

Effect of oil in evaporator: Studies on R 12 evaporators show that the boiling heat transfer coefficient inside tubes increases initially with oil concentration upto a value of about 4 percent and then decreases. The initial increase is attributed to the greater wetting of the tube surface due to the presence of oil. The subsequent reduction is due to the rapid increase in viscosity of the refrigerant-oil mixture as oil is more viscous than refrigerant. For the estimation of heat transfer

coefficient, the presence of oil may be neglected as long as its concentration is low (less than 10 percent).

23.12. Enhancement of heat transfer coefficients:

The overall heat transfer coefficient of a heat exchanger depends mainly on the component having the largest resistance to heat transfer. When air is used as an external fluid, the heat transfer coefficient on the air side is small, hence to obtain a high overall heat transfer coefficient, the air side heat transfer is augmented by adding fins. When liquid water is used as the external fluid, then the heat transfer coefficient on the water side will be high, when the flow is turbulent (which normally is the case). Hence to further improve the overall heat transfer coefficient, it may become necessary to enhance heat transfer on the refrigerant side. This is especially the case with synthetic refrigerants. The enhancement of boiling heat transfer coefficient can be achieved in several ways such as: increasing the refrigerant velocity by using an external pump in flooded evaporators, by using integrally finned tubes, by using treated surfaces, by using turbulence promoters etc. These methods improve the refrigerant side heat transfer coefficient and hence the overall heat transfer coefficient significantly leading to compact and lightweight evaporators. However, it should be kept in mind that normally any heat transfer enhancement technique imposes a penalty by means of increased pressure drop, hence it is essential to optimize the design so that the total cost is minimized.

23.13. Wilson's plot:

The concept of Wilson's plot was introduced way back in 1915 by Wilson to determine individual heat transfer coefficients from the experimental data on heat transfer characteristics of heat exchangers. This is sometimes applied to determine the condensing or boiling heat transfer coefficients of condensers and evaporators respectively.

For example, in a water-cooled condenser a number of tests are conducted by varying the flow rate of water and measuring the inlet and outlet water temperatures. The total heat transfer rate is determined from

$$Q = \dot{m}_w C_{pw} (t_{wo} - t_{wi}) = U_o A_o (\text{LMTD}) \quad (23.13)$$

From measured temperatures, LMTD is calculated. From the heat transfer rate Q , area of the heat exchanger (A_o) and LMTD, the overall heat transfer coefficient for a given flow rate is calculated using Eqn.(23.13).

Then the overall heat transfer coefficient U_o is equated to the following equation (for clean tubes are clean with negligible scale formation)

$$\frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{A_o r_i \ln(d_o/d_i)}{A_i k_w} + \frac{1}{h_o} \quad (23.14)$$

If the water temperature does not vary very significantly during these tests, then properties of water remain nearly constant. Since during these tests no changes are made on the refrigerant side, it can be assumed that the heat transfer resistance offered by the wall separating the two fluids and the heat transfer coefficient on refrigerant side (h_o) remains constant for all values of water flow rates. Hence, the above equation can be written as:

$$\frac{1}{U_o} = C_1 + \frac{C_2}{h_i} \quad (23.15)$$

where C_1 and C_2 are empirical constants that depend on the specifications of the heat exchangers and operating conditions, and the expressions for these can be obtained by equating Eqns.(23.14) and (23.15).

If flow on water side is turbulent and the variation in thermal properties are negligible, then the waterside heat transfer coefficient can be written as:

$$h_i = C_3 \cdot V^{0.8} \quad (23.16)$$

Substituting the expression in Eqn.(23.15), we obtain:

$$\frac{1}{U_o} = C_1 + \frac{C_4}{V^{0.8}} \quad (23.17)$$

Then a plot of $1/U_o$ vs $1/V^{0.8}$ will be a straight line as shown in Fig. 23.11. This plot is extrapolated to infinitely high velocity, i.e., where $1/V^{0.8}$ tends to zero. When $1/V^{0.8}$ tends to zero, from Eqn.(23.16) $1/h_i$ also tends to zero. Hence, the intercept on the ordinate is $C_1 (=1/h_o + A_o r_i \ln(d_o/d_i)/(A_i k_w))$. The thermal conduction resistance of the tube can be calculated and then the condensation heat transfer coefficient h_o can be calculated. As shown in the figure the term $A_o/(A_i h_i)$ can also be obtained from the figure at any value of velocity.

It should be kept in mind that it is an approximation since drawing a straight line and extending it to meet y-axis means that condensation heat transfer remains constant as the velocity tends to infinity. Wilson plot can be applied to air-cooled condensers also. In this case as the heat transfer coefficient for air over finned surface varies as $V^{0.65}$, hence in this case $1/U_o$ will have to be plotted versus $V^{-0.65}$.

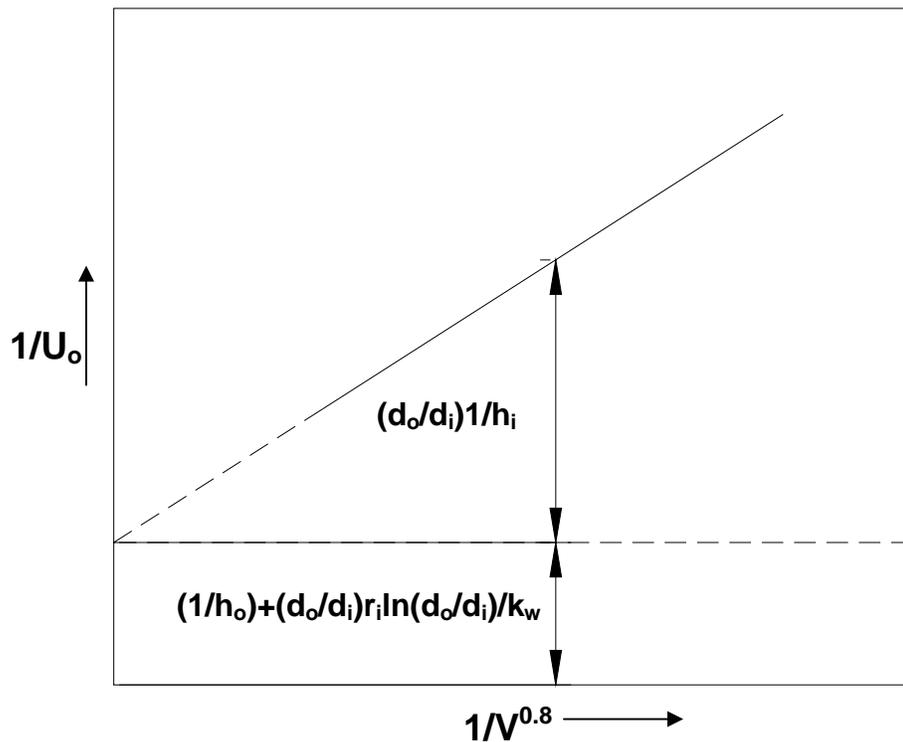


Fig.23.11: Concept of Wilson's plot

Questions and answers:

1. Which of the following statements are TRUE?

- a) In conventional refrigerators, the evaporators are kept at the top as these are natural convection type
- b) Natural convection type coils are useful when the latent loads are very high
- c) Defrosting of evaporators has to be done more frequently in natural convection type coils compared to forced convection evaporator coils
- d) Provision of sufficient free space is very important in natural convection type evaporator coils

Ans.: a) and d)

2. Which of the following statements are TRUE?

- a) Flooded type evaporators are very efficient as the heat transfer coefficient on refrigerant side is very large
- b) In flooded type evaporators, the refrigerant evaporation rate is equal to the refrigerant mass flow rate
- c) An oil separator is always required in flooded evaporators as refrigerant tends to get collected in the evaporator
- d) All of the above

Ans.: a) and c)

3. Which of the following statements are TRUE?

- a) Shell-and-tube evaporators are available in small to very large capacities
- b) In dry expansion type evaporator, refrigerant flows through the shell while the external fluid flows through the tubes
- c) Normally float valves are used expansion devices for flooded type evaporators
- d) In shell-and-coil type evaporators, thermal storage can be obtained by having refrigerant on the shell side

Ans.: a) and c)

4. Which of the following statements are TRUE?

- a) In direct expansion, fin-and-tube type evaporators, the oil return to compressor is better if refrigerant enters at the bottom of the evaporator and leaves from the top
- b) For low temperature applications, the fin spacing of evaporator is kept larger to take care of the frost formation
- c) Double pipe type evaporators are used when close temperature approach is required
- d) Plate type evaporators are used when close temperature approach is required

Ans.: b) and d)

5. Thermal design of evaporators is very complex due to:

- a) Continuous variation of heat transfer coefficient along the length
- b) Possibility of latent heat transfer on the external fluid side also
- c) Presence of lubricating oil affects heat transfer and pressure drop
- d) All of the above

Ans.: d)

6. Which of the following statements are TRUE?

- a) In evaporators using air as an external fluid, fins are frequently required on the refrigerant side
- b) In evaporators using water as an external fluid, fins may be required on the refrigerant side to enhance heat transfer
- c) Flooded type evaporators yield higher heat transfer coefficients compared to direct expansion type evaporators
- d) In general heat transfer enhancement techniques yield more compact heat exchangers, but may also increase pressure drop

Ans.: b), c) and d)

7. Air enters a direct expansion type, fin-and-tube evaporator at a temperature of 17°C and leaves the evaporator at 11°C. The evaporator operates at a constant temperature of 7°C and has total refrigerant side area of 12 m², while the bare tube and finned areas on airside are 10 m² and 212 m², respectively. Find the refrigeration capacity of the evaporator assuming only sensible heat transfer on airside and counterflow type arrangement. Neglect fouling and resistance offered by the tube wall. The fin effectiveness for airside is 0.75. The average heat transfer coefficient on refrigerant and airside are 1700 W/m².K and 34 W/m².K, respectively.

Ans.: Neglecting fouling and resistance of the tube wall, the value of 'UA' of evaporator is given by:

$$\frac{1}{UA} = \frac{1}{[h(A_f \eta_f + A_b)]_o} + \frac{1}{h_i A_i}$$

Substituting the values of airside and refrigerant heat transfer coefficients (h_o and h_i), bare tube (A_b), finned surface (A_f) and refrigerant side areas and fin efficiency ($\eta_f = 0.75$) in the above expression, we obtain:

$$UA = 4483 \text{ W/K}$$

From the values of airside and evaporator temperatures, the LMTD of the evaporator is given by:

$$LMTD = \frac{(17 - 11)}{\ln\left(\frac{17 - 7}{11 - 7}\right)} = 6.55^\circ \text{C}$$

Hence, refrigeration capacity, $Q_e = UA.LMTD = 29364 \text{ W} = 29.364 \text{ kW}$ (Ans.)

8. The following are the values measured on a shell-and-tube ammonia condenser:

Velocity of water flowing through the tubes, V (m/s)	1.22	0.61
Overall heat transfer coefficient, U_o (W/m ² .K)	2300	1570

Water flowed inside the tubes while refrigerant condensed outside the tubes. The tubes were 51 mm OD and 46 mm ID and had a conductivity of 60 W/m.K. Using the concept of Wilson's plot, determine the condensing heat transfer coefficient. What is the value of overall heat transfer coefficient when the velocity of water is 0.244 m/s?

Ans.:

From the data given in the table, the following straight line equation can be obtained:

$$\frac{1}{U_o} = C_1 + \frac{C_4}{V^{0.8}}$$

The values of C_1 and C_4 for the given data are found to be:

$$C_1 = 1.605 \times 10^{-4} \text{ m}^2.\text{K/W} \text{ and } C_4 = 3.223 \times 10^{-4} \text{ m}^{1.2}.\text{K/W}$$

The constant C_1 is equal to:

$$C_1 = \frac{r_o \ln(r_o / r_i)}{k_w} + \frac{1}{h_o} = 1.605 \times 10^{-4}$$

Substituting the values of internal and external radii (r_i and r_o) and the value of thermal conductivity of the tube k_w , we obtain the value of external heat transfer coefficient (condensation heat transfer coefficient, h_o) as:

$$h_o = 8572.9 \text{ W/m}^2.\text{K} \quad (\text{Ans.})$$

The value of overall heat transfer coefficient U_o when the velocity of water is 0.244 m/s is given by:

$$\frac{1}{U_o} = C_1 + \frac{C_4}{V^{0.8}} = 1.605 \times 10^{-4} + \frac{3.223 \times 10^{-4}}{0.244^{0.8}} = 1.1567 \times 10^{-3}$$

$$\Rightarrow U_o = 864.5 \text{ W/m}^2.\text{K} \quad (\text{Ans.})$$