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A BRIEF STUDY OF DIFFERENT HEAT EXCHANGERS WITH SPECIAL REFERENCE TO GASKETED TYPE PLATE HEAT EXCHANGER

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ABSTRACT:

The plate and frame or gasket plate heat exchanger (PHE) consists of number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame. The frame usually has a fixed end cover (headpiece) fitted with connecting ports and a moveable end cover (pressure plate, follower, or tailpiece). In the frame, the plates are suspended from an upper carrying bar and guided by a bottom carrying bar to ensure proper alignment. For this purpose, each plate is notched at the center of its top and bottom edges. The plate pack with fixed and movable end covers is clamped together by long bolts, thus compressing the gaskets and forming a seal. For later discussion., we designate the resulting length of the plate pack as L_{pack} . The carrying bars are longer than the compressed stack, so that when the movable end cover is removed, plates may be slid along the support bars for inspection and cleaning.

Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on sheet metal. On one side of each plate, special grooves are provided along the periphery of the plate and around the ports for a gasket, as indicated by the dark lines in Fig. 1.4

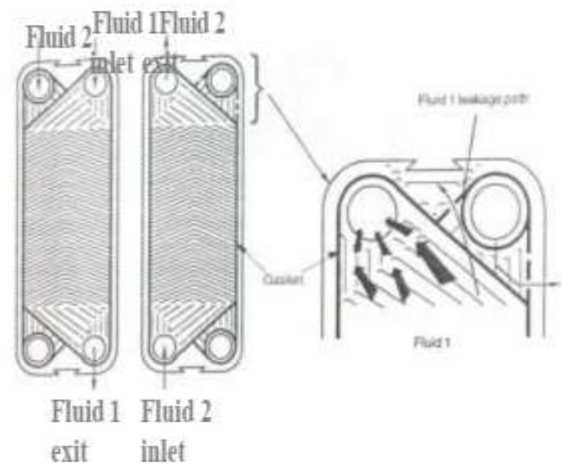


Fig 1.4 Plates showing gaskets around

Alternate plates are assembled such that corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow passages are

Narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing. Plates are designated as hard or soft depending on whether they generate a high or low intensity of turbulence

Extended Surface Heat Exchangers

The tubular and plate-type exchangers described previously are all prime surface heat exchangers, except for a shell-and-tube exchanger with low finned tubing. Their heat exchanger effectiveness is usually 60% or below, and the heat transfer surface area density is usually less than $700m^2/m^3$ (213 ft^2/ft^3). In some applications, much higher (up to about 98%) exchanger effectiveness is essential. and the box volume and mass are limited so that a much more compact surface is mandated. Also, in a heat exchanger with gases or some liquids, the heat transfer coefl1cient is quite lowon one or both fluid sides. This results in a large heat transfer surface area requirement. One of the most common methods to increase the surface area. Addition of fins can increase the surface area by 5 to 12 times the primary surface. Flow area is increased by the use of thin gauge material and sizing the core properly

Plate Fin Heat Exchangers

This type of exchanger has corrugated fins (most commonly having triangular and rectangular cross sections) or spacers sandwiched between parallel plates (referred to as plates or parting sheets) as shown in figure 1.5

Sometimes fins are incorporated in a flat tube with rounded corners,fig.1.6, thus eliminating the need for side bars. If liquid or phase change fluid flows on other side, the parting sheet is replaced by flat tube with or without inserts or webs. (Fig.1.7). other plate fin construction include drawn cup (Fig.1.8) and tube and centre configurations.

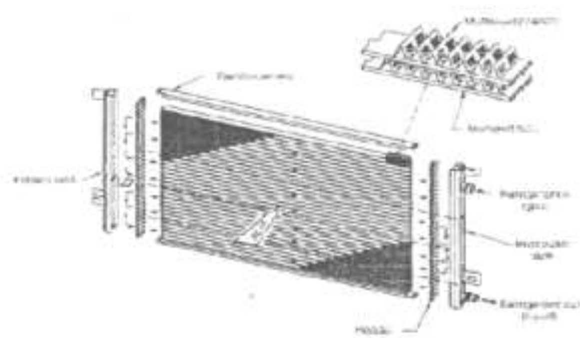


Figure 1.6 Flat webbed tube multilouver fin automotive condensor

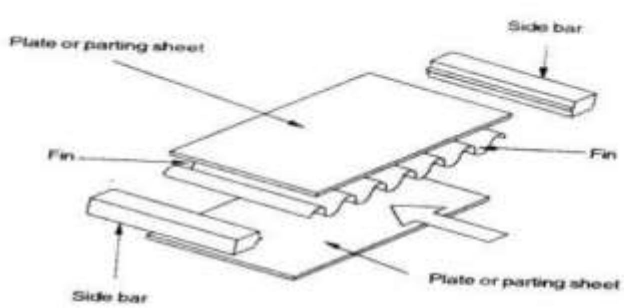


Figure 1.5 Basics components of plate fin heat exchanger

The plates or flat tubes separate the two fluid streams, and the fins form the individual flow passages. Alternate fluid passages arc connect d in parallel by suitable headers to form the two or more fluid sides of the exchanger. Fins are die or roll formed and are attached to the plates by brazing, soldering, adhesive bonding, welding, mechanical fit, or extrusion. Fins may be used on both sides in gas-to-gas heat exchangers. In gas-to-liquid applications, fins are general used only on the gas side, if employed on the liquid side, they are used primarily for structural strength and flow-mixing purposes. Fins are also sometimes used for pressure containment and rigidity. In Europe, a plate-fin exchanger is also referred to as a matrix heat exchanger.

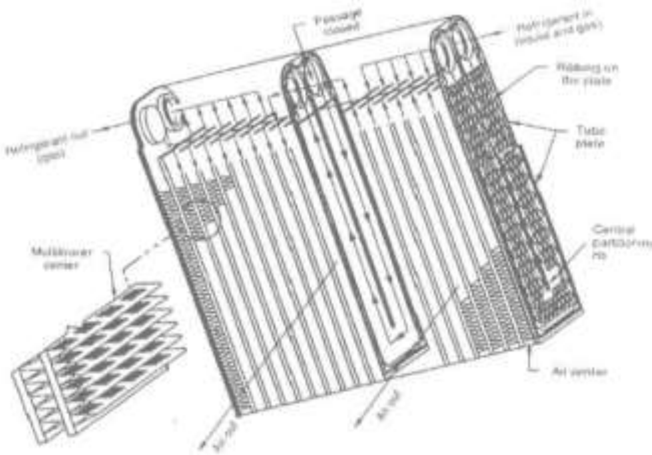


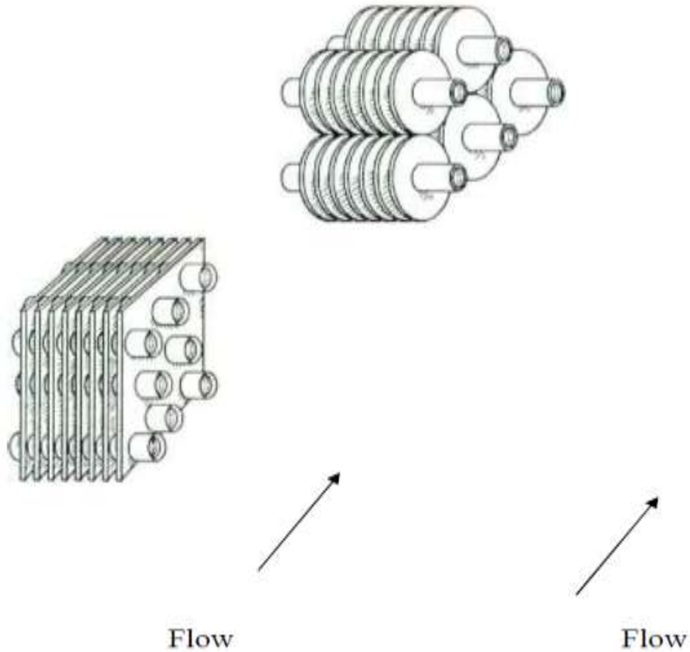
Figure 1.7 U channel ribbed plates and multilouver fin automotive evaporator

Tube-Fin Heat Exchangers

These ,exchangers may be classified as conventional and specialized tube-fin exchangers. In a conventional tube-fin exchanger, heat transfer between the two fluids takes place by conduction through the tube wall. However, in a heat pipe exchanger (a specialized type of tube-fin exchanger), tubes with both ends closed act as a separating wall, and heat transfer between the two fluids takes place through this "separating wall" (heat pipe) by conduction, and evaporation and condensation of the heat pipe fluid. Let us first describe conventional tube-fin exchangers.

Conventional Tube-Fin Exchangers

In a gas-to-liquid exchanger, the heat transfer coefficient on the liquid side is generally one order of magnitude higher than that on the gas side. Hence, to have balanced thermal conductance's (approximately the same hA *)* on both sides for a minimum-size heat exchanger, fins are used on the gas side to increase surface area A*.* This is similar to the case of a condensing or evaporating fluid stream on one side and gas on the other. In addition, if the pressure is high for one fluid. It is generally economical to employ tubes.



In a tube-fin exchanger, round and rectangular tubes are most common, although elliptical tubes are also used. Fins are generally used on the outside, but they may be used on the inside of the tubes in some applications, They are attached to the tubes by a tight mechanical fit, tension winding, adhesivebonding, soldering, brazing, welding, extrusion.

Classification according to Flow Arrangement

The choice of a particular flow arrangement is dependent on the required exchanger effectiveness, available pressure drops, minimum and maximum velocities allowed, fluid flow paths, packaging envelope, allowable thermal stress, temperature levels, piping and plumbing considerations, and other design criteria

Counter flow Exchanger

In a counter flow or countercurrent exchanger, the two fluids flow parallel to each other but in opposite directions. The temperature variation of the two fluids in such an exchanger may be idealized as one-dimensional. The counter flow arrangement is thermodynamically superior to any other flow arrangement. It is the most efficient flow arrangement, producing the highest temperature change in each fluid compared to any other two-fluid flow arrangements for a given overall thermal conductance (UA). Fluid flow rates (actually, fluid heat capacity rates), and fluid inlet temperatures. Moreover, the maximum temperature difference across the exchanger wall thickness (between the wall surfaces exposed on the hot and cold fluid sides) either at the hot- or cold-fluid end is the lowest, and produce minimum thermal stresses in the wall for an equivalent performance compared to any other flow arrangements.

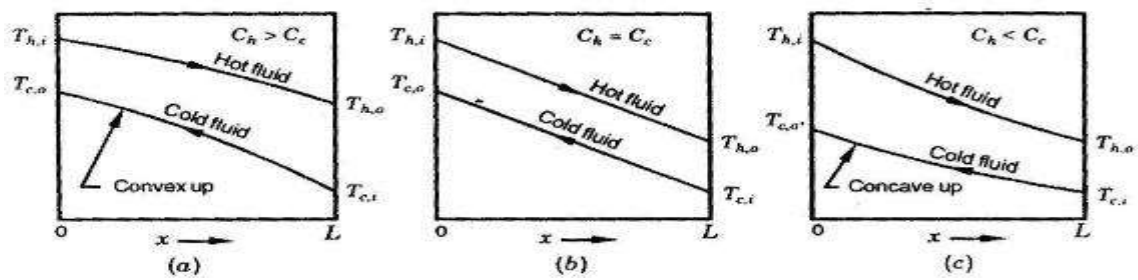


Figure 1.9 temperature distribution in counter flow heat exchanger

Parallel Flow Exchanger.

In a parallel flow exchanger, the fluid streams enter together at one end, flow parallel to each other in the same direction. and leave together at the other end. Fluid temperature variations. idealized as one-dimensional, are shown in Fig 1.11.

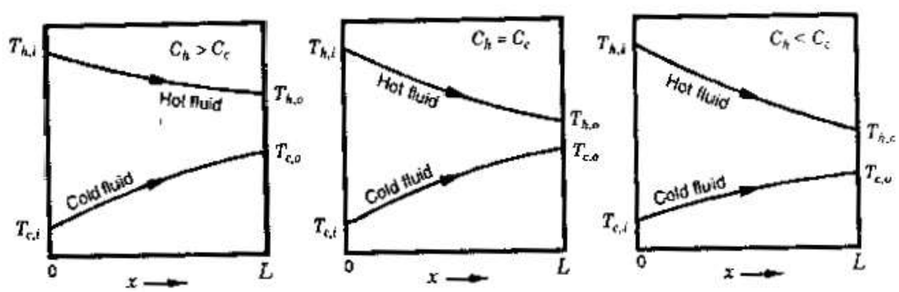


Figure 1.10 temperature distribution in a parallel flow heat exchanger

This arrangement has the lowest exchanger effectiveness among single-pass exchangers for givenoverall thermal conductance (UA) and fluid flow rates and f1uid inlet temperatures: however, some multi pass exchangers may have an even lower effectiveness. However, for low-effectiveness exchangers, the difference in parallel flow and counter flow exchanger effectiveness is small. In a parallel flow exchanger, a large temperature difference between inlet temperatures of hot and cold fluids exit at the inlet side, which may induced High thermal stress in the exchanger wall at the inlet. Although this arrangement is not used for applications requiring high effectiveness.

Cross Flow Heat Exchanger

In this type of exchanger, the two fluids flow in directions normal to each other. Typical fluid temperature variations are idealized as two-dimensional and are shown in Fig.1.12. For the inlet and outlet sections only, thermodynamically, the effectiveness for the cross flow exchanger falls in between that for the counter flow and parallel flow arrangements. The largest structural temperature difference exists at the "corner" of the entering hot and cold fluids, such as point" in Fig. 1.12 This is one of the most common flow arrangements used for extended surface heat exchangers, because it greatly simplifies the header design at the entrance and exit of each fluid. If the desired heat exchanger effectiveness is high (such as greater than 80%), the size penalty for the cross flow exchanger may become excessive. In such a case, acounter flow unit is preferred.

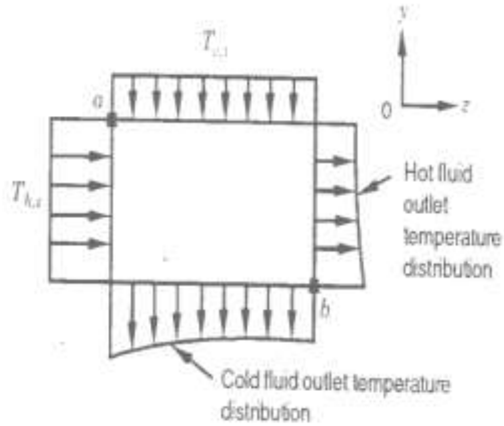


Figure 1.11 temperature distribution at inlets and outlets of an unmixed-unmixed cross flow heat exchanger

Heat Exchanger Design Methods

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing. We will limit ourselves to the design of recuperates only. That is, the design of a two fluid heat exchanger used for the purposes of recovering waste heat. We may write the enthalpy balance on either fluid stream to give:

$Q_c = m_c (h_{c1} - h_{c2})$ (1)

$Q_h = m_h (h_{h1} - h_{h2})$ (2)

For constant specific heats with no change of phase, we may also write $Q_C = (mc_p)_c (T_{c1} - T_{c2})$ (3)

$Q_h = (mc_p)_h (T_{h1} - T_{h2})$ (4)

Now from energy conservation we know that $Q_c = Q_h = Q$, and that we may relate the heat transfer rate Q and the overall heat transfer coe±cientU, to the some meantemperature difference ΔTm by means of

$Q = UA\Delta Tm$ (5)

where A is the total surface area for heat exchange that U is based upon. Later we shall show that

$\Delta Tm = f (T_{h1}, T_{h2}, T_{c1}, T_{c2})$ (6)

It is now clear that the problem of heat exchanger design comes down to obtaining an expression for the mean temperature difference. Expressions for many flow con-figurations, i.e. parallel flow, counter flow, and cross flow, have been obtained in the heat transfer field. Two approaches to heat exchanger design that are the LMTD method and the effectiveness- NTU method.

Each of these methods has particular advantages depending upon the nature of the problem specification.

Overall Heat Transfer Coefficient

A heat exchanger analysis always begins with the determination of the overall heat transfer coefficient. The overall heat transfer coefficient may be defined in terms of Heat Exchangers individual thermal resistances of the system. Combining each ofthese resistances in series gives:

$$\frac{1}{UA} = \frac{1}{(\eta_0 hA)} + \frac{1}{Sk_w} + \frac{1}{(\eta_0 hA)_0} \qquad (7)$$

Where η_0 is the surface efficiency of inner and outer surfaces, h is the heat transfer coefficients for the inner and outer surfaces, and S is a shape factor for the wall separating the two fluids. The surface efficiency accounts for the effects of any extended surface which is present on either side of the parting wall. It is related to the fin efficiency of an extended surface in the following manner:

$$\eta_0 = \left(1 - \left(1 - \eta_f\right)\frac{A_f}{A}\right)$$

The thermal resistances include the inner and outer film resistances, inner and outer extended surface efficiencies, and conduction through a dividing wall which keeps the two fluid streams from mixing. The shape factors for a numberof useful wall configurations are given below in Table 1. Additional results will be presented for some complex doubly connected regions.Equation (7) is for clean or unfouled heat exchanger surfaces. Finally, we should note that

$UA = U_0A_0 = U_iA_i$ (9)

However,

$$U_0 \neq U_i \qquad (10)$$

Finally, the order of magnitude of the thermal resistances in the definition ofthe overall heat transfer coefficient can have a significant influence on the calculation of the overall heat transfer coefficient. Depending upon the nature of the fluids, one or more resistances may dominate making additional resistances unimportant. For example, in Table 2 if one of the two fluids is gas and the other a liquid, then it is easy to see that the controlling resistancewill be that of the gas, assuming that the surface area on each side is equal.

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