

# MODULE 7

## HEAT EXCHANGERS

### 7.1 What are heat exchangers?

Heat exchangers are devices used to transfer heat energy from one fluid to another. Typical heat exchangers experienced by us in our daily lives include condensers and evaporators used in air conditioning units and refrigerators. Boilers and condensers in thermal power plants are examples of large industrial heat exchangers. There are heat exchangers in our automobiles in the form of radiators and oil coolers. Heat exchangers are also abundant in chemical and process industries.

There is a wide variety of heat exchangers for diverse kinds of uses, hence the construction also would differ widely. However, in spite of the variety, most heat exchangers can be classified into some common types based on some fundamental design concepts. We will consider only the more common types here for discussing some analysis and design methodologies.

### 7.2 Heat Transfer Considerations

The energy flow between hot and cold streams, with hot stream in the bigger diameter tube, is as shown in Figure 7.1. Heat transfer mode is by convection on the inside as well as outside of the inner tube and by conduction across the tube. Since the heat transfer occurs across the smaller tube, it is this internal surface which controls the heat transfer process. By convention, it is the outer surface, termed  $A_o$ , of this central tube which is referred to in describing heat exchanger area. Applying the principles of thermal resistance,

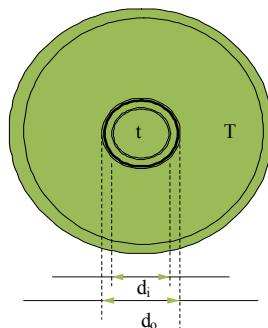


Figure 7.1: End view of a tubular heat exchanger

$$R = \frac{1}{h_o A_o} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi \cdot kl} + \frac{1}{h_i A_i}$$

If we define overall the heat transfer coefficient,  $U_c$ , as:

$$U_c \equiv \frac{1}{RA_o}$$

Substituting the value of the thermal resistance R yields:

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{r_o \ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{A_o}{h_i A_i}$$

Standard convective correlations are available in text books and handbooks for the convective coefficients,  $h_o$  and  $h_i$ . The thermal conductivity,  $k$ , corresponds to that for the material of the internal tube. To evaluate the thermal resistances, geometrical quantities (areas and radii) are determined from the internal tube dimensions available.

### 7.3 Fouling

Material deposits on the surfaces of the heat exchanger tubes may add more thermal resistances to heat transfer. Such deposits, which are detrimental to the heat exchange process, are known as fouling. Fouling can be caused by a variety of reasons and may significantly affect heat exchanger performance. With the addition of fouling resistance, the overall heat transfer coefficient,  $U_c$ , may be modified as:

$$\frac{1}{U_d} = \frac{1}{U_c} + R''$$

where  $R''$  is the fouling resistance.

Fouling can be caused by the following sources:

- 1) *Scaling* is the most common form of fouling and is associated with inverse solubility salts. Examples of such salts are  $\text{CaCO}_3$ ,  $\text{CaSO}_4$ ,  $\text{Ca}_3(\text{PO}_4)_2$ ,  $\text{CaSiO}_3$ ,  $\text{Ca}(\text{OH})_2$ ,  $\text{Mg}(\text{OH})_2$ ,  $\text{MgSiO}_3$ ,  $\text{Na}_2\text{SO}_4$ ,  $\text{LiSO}_4$ , and  $\text{Li}_2\text{CO}_3$ .
- 2) *Corrosion fouling* is caused by chemical reaction of some fluid constituents with the heat exchanger tube material.
- 3) *Chemical reaction fouling* involves chemical reactions in the process stream which results in deposition of material on the heat exchanger tubes. This commonly occurs in food processing industries.

- 4) *Freezing fouling* is occurs when a portion of the hot stream is cooled to near the freezing point for one of its components. This commonly occurs in refineries where paraffin frequently solidifies from petroleum products at various stages in the refining process. , obstructing both flow and heat transfer.
- 5) *Biological fouling* is common where untreated water from natural resources such as rivers and lakes is used as a coolant. Biological microorganisms such as algae or other microbes can grow inside the heat exchanger and hinder heat transfer.
- 6) *Particulate fouling* results from the presence of microscale sized particles in solution. When such particles accumulate on a heat exchanger surface they sometimes fuse and harden. Like scale these deposits are difficult to remove.

With fouling, the expression for overall heat transfer coefficient becomes:

$$\frac{1}{U_d} = \frac{1}{h_i \cdot \left(\frac{r_i}{r_o}\right)} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k} + \frac{1}{h_o} + R''$$

#### 7.4 Basic Heat Exchanger Flow Arrangements

Two basic flow arrangements are as shown in Figure 7.2. Parallel and counter flow provide alternative arrangements for certain specialized applications. In parallel flow both the hot and cold streams enter the heat exchanger at the same end and travel to the opposite end in parallel streams. Energy is transferred along the length from the hot to the cold fluid so the outlet temperatures asymptotically approach each other. In a counter flow arrangement, the two streams enter at opposite ends of the heat exchanger and flow in parallel but opposite directions. Temperatures within the two streams tend to approach one another in a nearly linearly fashion resulting in a much more uniform heating pattern. Shown below the heat exchangers are representations of the axial temperature profiles for each. Parallel flow results in rapid initial rates of heat exchange near the entrance, but heat transfer rates rapidly decrease as the temperatures of the two streams approach one another. This leads to higher exergy loss during heat exchange. Counter flow provides for relatively uniform temperature differences and, consequently, lead toward relatively uniform heat rates throughout the length of the unit.

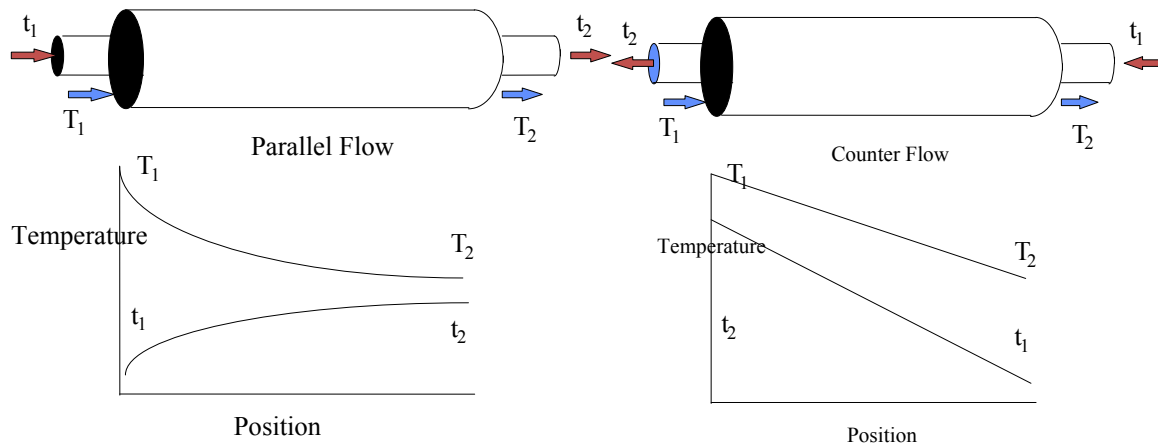


Fig. 7.2 Basic Flow Arrangements for Tubular Heat Exchangers.

### 7.5 Log Mean Temperature Differences

Heat flows between the hot and cold streams due to the temperature difference across the tube acting as a driving force. As seen in the Figure 7.3, the temperature difference will vary along the length of the HX, and this must be taken into account in the analysis.

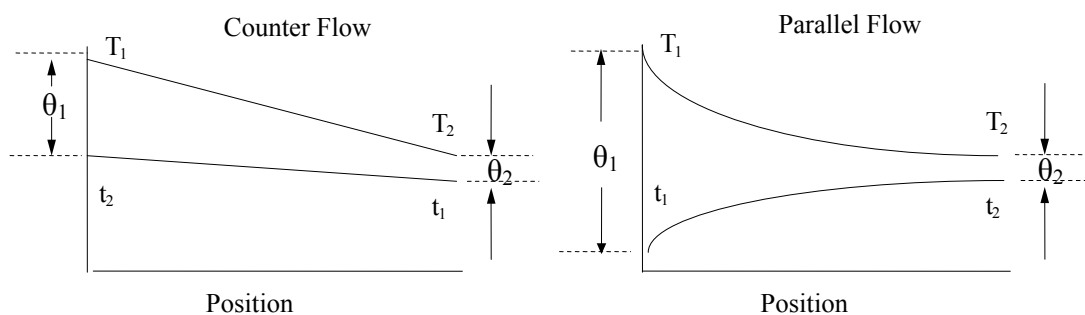


Fig. 7.3 Temperature Differences Between Hot and Cold Process Streams

From the heat exchanger equations shown earlier, it can be shown that the integrated average temperature difference for either parallel or counter flow may be written as:

$$\Delta\theta = LMTD = \frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)}$$

The effective temperature difference calculated from this equation is known as the log mean temperature difference, frequently abbreviated as LMTD, based on the type of mathematical average that it describes. While the equation applies

to either parallel or counter flow, it can be shown that  $\Delta\theta_{\text{eff}}$  will always be greater in the counter flow arrangement.

Another interesting observation from the above Figure is that counter flow is more appropriate for maximum energy recovery. In a number of industrial applications there will be considerable energy available within a hot waste stream which may be recovered before the stream is discharged. This is done by recovering energy into a fresh cold stream. Note in the Figures shown above that the hot stream may be cooled to  $t_1$  for counter flow, but may only be cooled to  $t_2$  for parallel flow. Counter flow allows for a greater degree of energy recovery. Similar arguments may be made to show the advantage of counter flow for energy recovery from refrigerated cold streams.

## 7.6 Applications for Counter and Parallel Flows

We have seen two advantages for counter flow, (a) larger effective LMTD and (b) greater potential energy recovery. The advantage of the larger LMTD, as seen from the heat exchanger equation, is that a larger LMTD permits a smaller heat exchanger area,  $A_o$ , for a given heat transfer,  $Q$ . This would normally be expected to result in smaller, less expensive equipment for a given application.

Sometimes, however, parallel flows are desirable (a) where the high initial heating rate may be used to advantage and (b) where it is required the temperatures developed at the tube walls are moderate. In heating very viscous fluids, parallel flow provides for rapid initial heating and consequent decrease in fluid viscosity and reduction in pumping requirement. In applications where moderation of tube wall temperatures is required, parallel flow results in cooler walls. This is especially beneficial in cases where the tubes are sensitive to fouling effects which are aggravated by high temperature.

## 7.7 Multipass Flow Arrangements

In order to increase the surface area for convection relative to the fluid volume, it is common to design for multiple tubes within a single heat exchanger. With multiple tubes it is possible to arrange to flow so that one region will be in parallel and another portion in counter flow. An arrangement where the tube side fluid passes through once in parallel and once in counter flow is shown in the Figure 7.4. Normal terminology would refer to this arrangement as a 1-2 pass heat exchanger, indicating that the shell side fluid passes through the unit once, the tube side twice. By convention the number of shell side passes is always listed first.

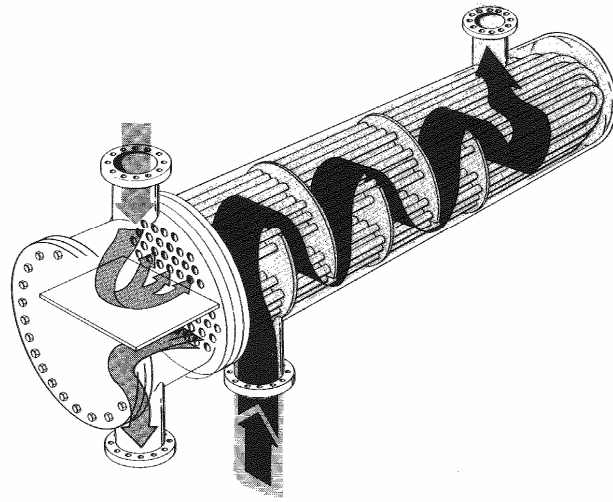


Fig. 7.4 Multipass flow arrangement

The primary reason for using multipass designs is to increase the average tube side fluid velocity in a given arrangement. In a two pass arrangement the fluid flows through only half the tubes and any one point, so that the Reynolds's number is effectively doubled. Increasing the Reynolds's number results in increased turbulence, increased Nusselt numbers and, finally, in increased convection coefficients. Even though the parallel portion of the flow results in a lower effective  $\Delta T$ , the increase in overall heat transfer coefficient will frequently compensate so that the overall heat exchanger size will be smaller for a specific service. The improvement achievable with multipass heat exchangers is substantially large. Accordingly, it is a more accepted practice in modern industries compared to conventional true parallel or counter flow designs.

The LMTD formulas developed earlier are no longer adequate for multipass heat exchangers. Normal practice is to calculate the LMTD for counter flow,  $LMTD_{cf}$ , and to apply a correction factor,  $F_T$ , such that

$$\Delta\theta_{eff} = F_T \cdot LMTD_{CF}$$

The correction factors,  $F_T$ , can be found theoretically and presented in analytical form. The equation given below has been shown to be accurate for any arrangement having 2, 4, 6, .....,  $2n$  tube passes per shell pass to within 2%.

$$F_T = \frac{\sqrt{R^2 + 1} \ln \left[ \frac{1 - P}{1 - R \cdot P} \right]}{(R - 1) \ln \left[ \frac{2 - P(R + 1 - \sqrt{R^2 + 1})}{2 - P(R + 1 + \sqrt{R^2 + 1})} \right]}$$

where the capacity ratio,  $R$ , is defined as:

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

The effectiveness may be given by the equation:

$$P = \frac{1 - X^{1/N_{shell}}}{R - X^{1/N_{shell}}}$$

provided that  $R \neq 1$ . In the case that  $R=1$ , the effectiveness is given by:

$$P = \frac{P_o}{N_{shell} - P_o \cdot (N_{shell} - 1)}$$

where

$$P_o = \frac{t_2 - t_1}{T_1 - t_1}$$

and

$$X = \frac{P_o \cdot R - 1}{P_o - 1}$$

### 7.8 Effectiveness-NTU Method:

Quite often, heat exchanger analysts are faced with the situation that only the inlet temperatures are known and the heat transfer characteristics ( $UA$  value) are known, but the outlet temperatures have to be calculated. Clearly, LMTH method will not be applicable here. In this regard, an alternative method known as the  $\epsilon$ -NTU method is used.

Before we introduce this method, let us ask ourselves following question: How will existing Heat Exchange perform for given inlet conditions? Define effectiveness: The effectiveness,  $\epsilon$ , is the ratio of the energy recovered in a HX to that recoverable in an ideal HX.

$$\epsilon = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}}, \text{ where } \dot{Q}_{max} \text{ is for an infinitely long H.Ex.}$$

$$\text{One fluid } \Delta T \rightarrow \Delta T_{max} = T_{h,in} - T_{c,in}$$

$$\text{and since } \dot{Q} = (\dot{m}c_A)\Delta T_A = (\dot{m}c_B)\Delta T_B = C_A\Delta T_A = C_B\Delta T_B$$

then only the fluid with lesser of  $C_A, C_B$  heat capacity rate can have  $\Delta T_{max}$

$$\text{i.e. } \dot{Q}_{max} = C_{min}\Delta T_{max} \text{ and } \epsilon = \frac{\dot{Q}}{C_{min}(T_{h,in} - T_{c,in})}$$

$$\text{or, } \dot{Q} = \epsilon C_{min}(T_{h,in} - T_{c,in})$$

We want expression for  $\epsilon$  which does not contain outlet T's.

Substitute back into  $\dot{Q} = UA(LMTD)$  .....

$$\epsilon = \frac{1 - \exp\left[\frac{-UA}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \frac{C_{min}}{C_{max}} \exp\left[\frac{-UA}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}$$

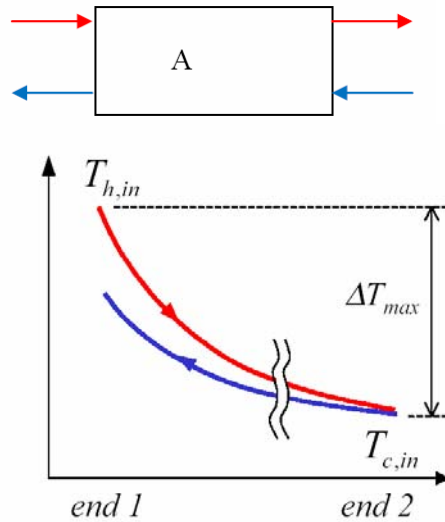


Fig. 7.5 Calculation of effectiveness-NTU

$$\therefore \varepsilon = \varepsilon \left( NTU, \frac{C_{\min}}{C_{\max}} \right)$$

and No. of transfer units (size of HEx.)  $NTU = \frac{UA}{C_{\min}}$

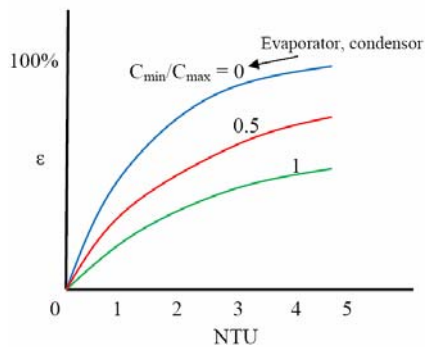
### Charts for each Configuration

#### Procedure:

Determine  $C_{\max}$ ,  $C_{\min}/C_{\max}$

Get  $UA/C_{\min}$ ,  $\rightarrow \varepsilon$  from chart

$$\dot{Q} = \varepsilon C_{\min} (T_{h,in} - T_{c,in})$$



$$NTU_{\max} = \frac{UA}{C_{\min}} \Rightarrow A = \frac{NTU_{\max} C_{\min}}{U}$$



- $NTU_{max}$  can be obtained from figures in textbooks/handbooks  
First, however, we must determine which fluid has  $C_{min}$ .