# **Design of Heat Exchangers**

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Heat exchangers are used in chemical and refining processes to

- Exchange heat between two process streams (called Crossexchangers)
- Heat process streams (called *heaters* and *furnaces*)
- Cool process streams (called *coolers*)
- Vaporize liquids (called *vaporizers*)
- Condense vapors (called *condensers*)
- Remove the heat of exothermic reactions

• Provide the heat of endothermic reactions

And other similar services. As shown in the list, exchangers are given different names for different services, but the principle of operation is basically the same for all of them, the transfer of heat from a hot stream to a colder stream. *Heat* is by definition *energy in transit due to a difference in temperature*.

#### 1. Enthalpy Balances on a Heat Exchanger

These notes will deal with the transfer of heat from and to flowing streams, that is, in open systems. Recall from your course in thermodynamics that energy balances in open systems are simplified by using the enthalpy H of the streams instead of the internal energy E. The reason is that the enthalpy includes the flow work pv, that is, the product of the pressure p times the specific volume of the stream v. This is the work performed on and by the stream as it flows in and out of the open system.

In heat exchangers the two streams exchanging heat are separated by a metal wall. Consider the exchanger surrounded by a box into and out of which the hot and cold streams flow:



#### Figure 1. Heat exchanger

An enthalpy balance on the exchanger of Figure 1, neglecting heat losses, yields:

$$q = m_c (H_{cb} - H_{ca}) = m_h (H_{ha} - H_{hb})$$
(1)

Where q = heat transfer rate, or the exchanger heat duty

 $m_c$ ,  $m_h$  = flows of the cold and hot streams, respectively

 $H_{ca}$ ,  $H_{cb}$  = cold stream inlet and outlet specific enthalpies

 $H_{ha}$ ,  $H_{hb}$  = hot stream inlet and outlet specific enthalpies The notation to be used in these notes is that subscript cdenotes the cold stream, h denotes the hot stream, a denotes the inlet condition and b denotes the outlet condition. Notice that as the hot and cold streams do not mix, the flow of each stream into the exchanger is the same as the flow of that stream out of the exchanger.

Equation (1) is really two equations with seven variables, so its solution requires five specifications. In a design problem the flow and enthalpies of one stream are specified; from these the heat duty q can be determined. For the other stream,

- If the inlet and outlet enthalpies are specified, the flow of the stream can be determined;
- If the flow and inlet enthalpy are specified, the outlet enthalpy can be determined.

## 2. Calculation of the Enthalpy

The enthalpy of a stream is usually a function of temperature and of the state of the stream, liquid or vapor. Suppose that a stream enters an exchanger as a vapor and the vapor is condensed to its saturation temperature, then completely condensed to a liquid and finally the liquid is cooled to the exit temperature. The total change in enthalpy for the stream is then:

$$H_{ha} - H_{hb} = C_{pv} (T_{ha} - T_{sat}) + \lambda_{sat} + C_{pL} (T_{sat} - T_{hb})$$
(2)

Where  $T_{ha}$ ,  $T_{sat}$ ,  $T_{hb}$  = inlet, saturation, and outlet temperatures  $c_{pv}$ ,  $c_{pL}$  = specific heats of the vapor and liquid

 $\lambda_{sat}$  = latent heat at the saturation temperature When there on phase change for the stream in an exchanger there is no latent heat term and there is only one sensible heat term, either the vapor or the liquid.

#### **Example 1. Enthalpy Balance on an Exchanger.**

A process produces 55,000 kg/hr of liquid aniline product at a temperature of 136°C. It is desired to cool the product stream to 35°C for storage. Cooling water at 30°C is used in the exchanger to cool the aniline with a maximum allowed rise in temperature of 10°C. Determine the heat duty of the exchanger and the flow of cooling water required in liters/min.

Solution.



To determine the enthalpy change of the aniline we need its specific heat. From McCabe, Smith and Harriott,  $7^{\text{th}}$  ed., Appendix 15, at the average temperature of  $(136 + 35)/2 = 85.5^{\circ}$ C or  $186^{\circ}$ F,  $c_{pL} = 0.55$  cal/gm-°C. The heat duty is:

 $q = (55,000) (0.55) (136 - 35) = 3.06 \times 10^{6} \text{ kcal/hr}$ 

For the calculation of the required flow of cooling water we must decide on the outlet temperature of the water. The maximum temperature rise of 10°C should be used whenever possible because it results in a smaller flow of water. Using a water specific heat of 1.0 cal/gm-°C and density of 1 kg/liter:

$$m_c = \frac{q}{c_{pc}(T_{cb} - T_{ca})} = \frac{3.06 \times 10^6}{1.0(10^\circ C)} \frac{liter}{kg} \frac{hr}{60min} = 5100 \frac{liter}{min}$$

Notice that if we had chosen a smaller water temperature rise, for example  $5^{\circ}$ C, it would have required twice the flow of water.

## 3. Sizing the Exchanger

The purpose of a heat exchanger is to provide enough area between the hot and the cold streams for heat to be transferred between them. The area is that of the metal that separates the two streams, either a tube or a plate. It is very important for you to visualize in your mind the hot and cold fluids on each side of the separating surface and the heat flowing through the surface from the hot fluid to the cold fluid. When you visualize this you can better understand that the rate at which heat is transferred through the surface is proportional to the area of the separating surface and also to the temperature **DIFFERENCE** between the hot stream and the cold stream.

It is critically important that you never confuse the temperature <u>difference</u> between the hot and cold streams with the <u>change</u> in the temperature of a stream as it goes through the exchanger. To facilitate the distinction we will reserve the term  $\Delta T$  for the difference in temperature between the hot and cold streams and never use it for the change in the temperature of a stream (although this is often done in thermodynamic courses). It is the temperature difference between the hot and cold streams that drives the heat from the hot stream to the cold stream.

The design of an exchanger consists of determining the area of the exchanger that is necessary to transfer the required heat

for the conditions of the exchanger. The basic design equation is given by:

$$q = U_o A \Delta T \tag{3}$$

where q = heat duty of the exchanger  $U_o$  = over-all heat transfer coefficient A = heat transfer area  $\Delta T$  = temperature difference between the hot and cold streams

In order to calculate the heat transfer area A from Equation (3) it is necessary to determine the heat transfer coefficient U and the temperature difference  $\Delta T$ . We will look at these next. Recall that the heat duty q is obtained from the enthalpy balance on the exchanger, Equation (1).

# 4. Heat Transfer Coefficient

Consider a hot and a cold stream separated by a plate as in Figure 2.



Figure 2. Hot and cold streams separated by a plate

Assume that both streams are flowing in turbulent flow past the plate so that, away from the plate they are perfectly mixed and the temperature of each stream is uniform,  $T_h$  for the hot stream and  $T_c$  for the cold stream. Recall from your fluid dynamics course that, due to viscous forces, the fluid near the wall is slowed down and flows in a thin laminar film called the *viscous sublayer*. The resistance of this film causes the temperature of

the hot stream to decrease from  $T_h$  to the temperature at the wall on the hot side,  $T_{wh}$ . The resistance of the wall causes a further decrease in temperature from  $T_{wh}$  to the wall temperature on the cold side,  $T_{wc}$ . The laminar film on the cold side causes further decrease in temperature to the bulk temperature of the cold fluid  $T_c$ .

The two fluid films and the plate wall represent three resistances to the flow of heat from the hot stream to the cold stream. The resistances of the films depend on the thickness of the films and the thermal conductivity of the fluids, and the resistance of the wall is a function of the wall thickness  $x_w$  and the thermal conductivity of the wall material.

The thermal conductivity is a property of the material that indicates how good a conductor of heat it is. Metals have the highest thermal conductivities and gases the lowest. Appendices 10-13 of McCabe, Smith and Harriott, 7<sup>th</sup> ed., give the thermal conductivities of a number of substances, and Appendix 6 has the thermal conductivity of water.

As the heat transfer rate per unit area through the plate of Figure 2 is the same through all the resistances, we can write the following expression:

$$\frac{q}{A} = U_o(T_h - T_c) = h_h(T_h - T_{wh}) = \frac{k_w}{x_w}(T_{wh} - T_{wc}) = h_c(T_{wc} - T_c)$$
(4)

Where  $h_h$ ,  $h_c$  = film coefficients on the hot and cold sides  $k_w$  = thermal conductivity of the wall

and the temperatures are defined in Figure 2. Now the over-all temperature difference is the sum of the three individual temperature differences:

$$T_h - T_c = (T_h - T_{wh}) + (T_{wh} - T_{wc}) + (T_{wc} - T_c)$$
(5)

Substituting Equation (4) into Equation (5) and simplifying:

$$\frac{1}{U_o} = \frac{1}{h_h} + \frac{x_w}{k_w} + \frac{1}{h_c}$$
(6)

Equation (6) is a most important equation. To understand it you must understand the following:

- The over-all heat transfer coefficient  $U_o$  represents the over-all conductance of the exchanger to the transfer of heat. Its reciprocal is then the total resistance to heat transfer.
- The film coefficients of heat transfer  $h_h$  and  $h_c$  are the *conductances* of the films on the hot and cold sides to heat transfer (see Figure 2). Their reciprocals are the *resistances* of the films to heat transfer.
- The term  $x_w/k_w$  is the *resistance* of the plate wall to heat transfer.

Therefore, what Equation (6) says is that the total resistance to heat transfer is the sum of the resistances of the films on each side of the plate plus the resistance of the plate wall. As we shall see later, in the detailed design of the exchanger we must consider a fourth resistance to heat transfer caused by the deposits on the heat transfer surface. This resistance is called the *fouling resistance*. As the total resistance is the sum of the individual resistances, the largest resistance is called the *controlling resistance*. This is because it has the greatest effect on the size of the exchanger.

#### **Example 2. Estimate of the Over-all Heat Transfer Coefficient**

A process heater uses steam to heat a hydrocarbon liquid in a plate exchanger made of carbon steel. The film coefficient of heat transfer for the condensing steam is 20,000 kJ/hr-m<sup>2</sup>-°C, and that for heating the hydrocarbon is 3,000 kJ/hr-m<sup>2</sup>-°C. The separating wall of the separating plate is 0.200 in. Determine the over-all heat transfer coefficient for the heater.

Solution. From McCabe, Smith and Harriott,  $7^{th}$  ed., Appendix 10, the thermal conductivity of 1% carbon steel is about (26 Btu/hr-ft-°F)(1.055 kJ/Btu)(ft/0.3048 m)(1.8 °F/°C) = 162 kJ/hr-m-°C. The three resistances are then:

$$1/h_{h} = 1/20,000 = 5.0 \times 10^{-5} \text{ hr} - \text{m}^{2} - \text{°C/kJ}$$

$$x_{w}/k_{w} = (0.200 \text{ in}) (\text{ft}/12 \text{ in}) (0.3048 \text{ m/ft}) / (162 \text{ kJ/hr} - \text{m}^{-}\text{°C})$$

$$= 3.1 \times 10^{-5} \text{ hr} - \text{m}^{2} - \text{°C/kJ}$$

$$1/h_{c} = 1/3,000 = 33.3 \times 10^{-5} \text{ hr} - \text{m}^{2} - \text{°C/kJ}$$

The total resistance is then:  $(5.0 + 3.1 + 33.3) \times 10^{-5} = 41.4 \times 10^{-5} \text{ hr} - \text{m}^2 - ^{\circ}\text{C/kJ}$ And the over-all heat transfer coefficient is:  $U_0 = 1/(41.4 \times 10^{-5}) = 2,400 \text{ kJ/hr} - \text{m}^2 - ^{\circ}\text{C}$ 

Notice that the plate wall resistance is less than 8% of the total resistance. So, if we were to consider making the heater wall out of gold, copper, or silver, with thermal conductivities from 6 to 9 times greater than carbon steel, we would obtain a minor increase the over-all heat transfer coefficient at the expense of a much higher exchanger cost. Notice also that the resistance of the condensing steam side is smaller than that of the hydrocarbon side. In general condensing steam represents one of the smaller resistances to heat transfer in an exchanger.

## 5. Heat Transfer Coefficients for Cylindrical Tubes

In most heat exchangers one of the streams flows inside of one or more cylindrical tubes and the other one outside the tubes. The area of a cylinder is proportional to its diameter and to its length. (This is because heat is transferred through the side of the cylinder, not though its end, where the cross sectional area is proportional the square of the diameter. Make sure you can visualize the cylinder and understand this.) The area of heat transfer is then greater on the outside of the tube than on the inside, because of the thickness of the tube wall, as Figure 3 shows.



Figure 3. Heat transfer in a cylindrical tube

The side area of a cylinder is  $\pi DL$  and, as the figure shows, the area is larger on the outside of the tube because the diameter is larger. The heat transfer rate through the tube wall is then

$$q = U_o \pi D_o L(T_h - T_c) = h_o \pi D_o L(T_h - T_{wo}) = \frac{k_w}{x_w} \pi D_M L(T_{wo} - T_{wi}) = h_i \pi D_i L(T_{wi} - T_c) \quad (7)$$

Where  $h_o$ ,  $h_i$  = film coefficients on the outside and inside And the other variables are marked on Figure 3. Diameter  $D_M$  on the wall resistance term is a mean diameter between the outside and the inside diameter because the diameter varies through the wall of the tube. As the area is proportional to the diameter, it can be shown that the mean diameter is the logarithmic mean between the inside and the outside diameter:

$$D_M = \frac{D_o - D_i}{\ln\left(\frac{D_o}{D_i}\right)} \tag{8}$$

As with Equations (5) and (6), the over-all temperature difference is the sum of the three temperature differences and, solving from Equation (7) and simplifying, we obtain:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{D_o x_W}{D_M k_W} + \frac{D_o}{D_i h_i}$$
(9)

As Equation (6), Equation (9) says that the total heat transfer resistance in tube exchangers is the sum of the individual resistances, the difference being the correction for the inside and outside diameters.

Notice that the over-all heat transfer coefficient  $U_o$  is based on the outside area of the tube. This is the common practice because the outside diameter of a pipe or tube is kept

the same for the tube of a nominal size when the tube wall thickness is increased to withstand a higher pressure.

#### **5.1 Determination of the Wall Temperature**

From Equation (7) we can determine the temperature differences across each film resistance and from them the temperature of the wall on the inside and the outside of the tube:

$$T_h - T_{wo} = \frac{U_o}{h_o} (T_h - T_c) \qquad T_{wi} - T_c = \frac{U_o D_o}{h_i D_i} (T_h - T_c)$$
(10)

#### Example 3. Estimate of Heat Transfer Coefficient in a Pipe Exchanger

The heater of Example 2 is designed as a pipe exchanger instead of a plate exchanger. The pipe is 1-in Schedule 40 standard steel pipe. Estimate the over-all heat transfer coefficient for the same film coefficients of Example 2.

Solution. From McCabe, Smith and Harriott,  $7^{th}$  ed., Appendix 3, the pipe dimensions are:

$$D_o = 1.315$$
 in  $x_w = 0.133$  in  $D_i = 1.049$  in

The log mean diameter is: 
$$D_M = \frac{1.315 - 1.049}{ln\left(\frac{1.315}{1.049}\right)} = 1.177$$
 in

From Example 2, assuming the steam condenses outside the pipe, the film coefficients are:

 $h_o = 20,000 \text{ kJ/hr-m}^2\text{-}^{\circ}\text{C} \qquad h_i = 3,000 \text{ kJ/hr-m}^2\text{-}^{\circ}\text{C}$ and  $k_w = 162 \text{ kJ/hr-m}\text{-}^{\circ}\text{C}$ . The resistances are:

$$1/h_o = 1/20,000 = 5.0 \times 10^{-5} \text{ hr} - \text{m}^2 - \text{°C/kJ}$$

$$D_o x_w / D_M k_w = (1.315 \text{ in}) (0.133 \text{ in}) (\text{m}/39.4 \text{ in}) / [(1.177 \text{ in}) (162)]$$

$$= 2.3 \times 10^{-5} \text{ hr} - \text{m}^2 - \text{°C/kJ}$$

$$D_o / D_i h_i = (1.315 \text{ in}) / [(1.049 \text{ in}) (3,000)]$$

$$= 41.8 \times 10^{-5} \text{ hr} - \text{m}^2 - \text{°C/kJ}$$

The total resistance is:

 $(5.0 + 2.3 + 41.8) \times 10^{-5} = 49.1 \times 10^{-5} \text{ hr-m}^2 \text{-}^\circ\text{C/kJ}$  The over-all heat transfer coefficient is

$$U_{o} = 1/49.1 \times 10^{-5} = 2,040 \text{ kJ/hr-m}^{2} - ^{\circ}\text{C}$$

The total resistance is higher than in Example 2 because the inside resistance, which is the largest of the three, operates on the inside diameter which is smaller than the outside diameter. You should check that if the steam condenses on the inside of the pipe the total resistance is smaller.

## 6. The Temperature Difference

Having the heat duty q from the enthalpy balance and the overall heat transfer coefficient  $U_o$ , the third term required to estimate the required heat transfer area A from Equation (3) is the temperature difference  $\Delta T$ . As the temperature of either stream or both often change as heat is transferred to or from it, a mean temperature difference is usually required. The calculation of the mean temperature difference depends on the *temperature profile* for the exchanger which consists of a plot of the stream temperatures with heat flow. It is very important for you to understand how the temperature profile affects the calculation of the mean temperature difference. We will see later that the temperature profile also affects which heat

exchanger model must be used in a process simulator such as HySys.

#### **6.1 Linear Temperature Profiles**

When the enthalpy of both streams changes by sensible heat, that is, without phase change or chemical reaction, the temperature profile of each fluid is approximately linear with heat flow (because of the variation of the specific heat of the streams with temperature the profile is slightly curved, but the curvature can be neglected for design calculations). There are two ways to arrange the flow of the streams through the heat exchanger, *parallel* and *countercurrent* flow. We will look at each in turn.

**Parallel Flow.** In parallel flow the hot and cold streams enter at one end of the exchanger and exit at the other end, as in Figure 4.



Figure 4. Parallel flow in a heat exchanger

If the enthalpy change of both streams is due to sensible heat, the temperature profile for the exchanger of Figure 4 is given in Figure 5. The figure shows how the temperature of the hot stream drops linearly as heat is transferred to the cold stream and similarly the temperature of the cold stream increases linearly.

It can be shown that the mean temperature difference for the linear profiles of Figure 5 is given by the *logarithmic mean* of the differences at the two ends of the exchanger:

$$\Delta T_M = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{11}$$

Where  $\Delta T_1 = T_{ha} - T_{ca}$  and  $\Delta T_2 = T_{hb} - T_{cb}$ .



Figure 5. Linear temperature profiles for parallel flow

Once again we remind you that you are not to confuse the temperature differences between the hot and cold streams with the temperature changes of the streams. The temperature changes in Figures 4 and 5 are  $T_{ha} - T_{hb}$  for the hot stream (a drop in temperature) and  $T_{cb} - T_{ca}$  for the cold stream (a rise in temperature).

**Countercurrent Flow.** In countercurrent flow the hot and cold streams enter the exchanger at opposite ends and flow in opposite direction to each other, as in Figure 6. The temperature profile for countercurrent flow is given in Figure 7.



Figure 6. Countercurrent flow in a heat exchanger



Figure 7. Linear temperature profiles for countercurrent flow

As the temperature profiles are linear, the mean temperature difference is the logarithmic mean given by Equation (11), but the temperature differences at the two ends are now

$$\Delta T_1 = T_{ha} - T_{cb}$$
 and  $\Delta T_2 = T_{hb} - T_{ca}$ 

Countercurrent flow is usually preferred to parallel flow because it is more efficient. Parallel flow is used when it is necessary to protect the cold stream from the high inlet temperature of the hot stream because of possible decomposition of the components.

## **Example 4. Comparison of Parallel vs. Countercurrent Flow.**

Calculate the mean temperature difference for the conditions of Example 1 for both parallel and countercurrent flow.

Solution. In Example 1 the hot stream is the aniline and the cold stream is the cooling water. The temperatures are:

$$T_{ha} = 136^{\circ}C$$
  $T_{hb} = 35^{\circ}C$   $T_{ca} = 30^{\circ}C$ 

And using the maximum temperature rise of the water,

$$T_{cb} = 30 + 10 = 40$$
 °C

For parallel flow the temperature differences at the ends are:

$$\Delta T_1 = T_{ha} - T_{ca} = 136 - 30 = 106^{\circ}C$$

$$\Delta T_2 = T_{hb} - T_{cb} = 35 - 40 = -5^{\circ}C$$

This shows it is not possible to design the exchanger for these conditions because heat cannot be transferred from a low temperature, 35°C, to a higher temperature, 40°C. There are two options to resolve this problem. One is to cool the aniline down to a higher temperature, say 45°C instead of 35°C, and the other is to let the cooling water exit at a lower temperature, say 32°C. The first option requires storing the aniline at a higher temperature than desired by 10°C. This could raise some safety concerns. The second option requires five times more cooling water than originally intended (you should check this out by solving Example 1 for a water exit temperature of 32°C). Let us calculate the mean temperature difference for this option:

$$\Delta T_2 = 35 - 32 = 3^{\circ} C$$
  $\Delta T_M = \frac{106-3}{\ln(\frac{106}{3})} = 28.9^{\circ} C$ 

A much better option is to run the exchanger in countercurrent flow. The temperature differences for countercurrent flow are:

$$\Delta T_1 = T_{ha} - T_{cb} = 136 - 40 = 96^{\circ}C$$
$$\Delta T_2 = T_{hb} - T_{ca} = 35 - 30 = 5^{\circ}C$$

And the mean temperature difference is:  $\Delta T_M = \frac{96-5}{\ln\left(\frac{96}{5}\right)} = 30.8^\circ C$ 

This example has demonstrated why countercurrent flow is more efficient than parallel flow in heat exchangers. It also shows that the mean temperature difference is controlled by the smaller of the two end temperatures. Notice that although the larger difference is higher for parallel flow (106°C vs. 96°C), and the smaller difference is only 2°C higher for countercurrent flow (5°C vs 3°C) the mean temperature difference is higher for countercurrent flow. The higher the mean temperature difference the smaller the required exchanger area (see Equation 3).

#### **6.3 Constant Temperature Profile**

There are a number of heat exchange situations in which the temperature of a stream can be assumed constant as the heat is transferred. These are some of them:

 Condensation of a pure component at approximately constant pressure. Even when there is a small de-superheating of the vapor or sub-cooling of the condensed liquid, the bulk of the heat is transferred at the constant saturation temperature of the component.

- Vaporization of a pure refrigerant at approximately constant pressure. The heat of vaporization is transferred at the constant boiling temperature.
- Cooling of an exothermic (or heating of an endothermic) continuous stirred tank reactor. The heat of reaction is transferred at the constant temperature of the reactor.
- Condensing of vapors from a distillation column. The heat of condensation is assumed to be transferred at the constant saturation temperature (bubble-point) of the distillate.
- Generation of vapors in a reboiler of a distillation column. The heat of vaporization is assumed to be transferred at the constant saturation temperature (bubble point) of the bottoms product.

Steam, which is commonly used for heating process stream, falls in the first category. Figure 7 shows the schematic of a heater with steam as the heating medium and the corresponding temperature profile. As the temperature profiles are linear the mean temperature difference is the logarithmic mean of Equation (11) with the following temperature differences at the ends:

 $\Delta T_1 = T_h - T_{ca}$  and  $\Delta T_2 = T_h - T_{cb}$ 



Figure 8. Schematic and temperature profile of a heater

When both the hot and cold stream profiles are constant, as when steam condenses to heat a distillation reboiler, the men temperature difference is just the difference between the constant hot and cold temperatures:

$$\Delta T_M = T_h - T_c$$

#### **6.4 Nonlinear Temperature Profiles**

When one of the stream temperature profiles is not linear, the logarithmic mean of the two end differences is not the correct mean temperature difference. This happens in the following situations:

- When there is both sensible heat and latent heat transfer in one of the streams.
- Condensation of a mixture of components where the condensation temperature varies non-linearly with heat transferred as the composition of the vapors and their corresponding saturation temperature (dew point) varies.

To determine the heat transfer area of an exchanger with a nonlinear profile the exchanger must be divided into sections so that the temperature profile is linear in each section. The area of each section is calculated and the sum of the areas of all the sections becomes the total required area of the exchanger.

#### Example 5. Sizing an Exchanger with a Nonlinear Temperature Profile

A vapor stream containing 10,000 kg/hr of dichloromethane (DCM) at 320°C is to be cooled and completely condensed at atmospheric pressure in a heat exchanger. Cooling water is available at 30°C with a maximum allowed temperature rise of 10°C. Determine the mean temperature difference. Compare it with the mean temperature difference if linear temperature profiles were assumed.

Solution. From Reid, Prausnitz and Sherwood,  $3^{rd}$  ed., Appendix, the molecular weight of DCM is 84.9, its saturation temperature at atmospheric pressure (normal boiling point) is 39.9°C, and latent heat is 6,690 kcal/kmole. From the correlation in Perry's, 8<sup>th</sup> ed., Table 2-156, the specific heat of the vapors at the average temperature of  $(320 + 39.9)/2 = 180^{\circ}$ C is 63.3 kJ/kmole-°C.

From the enthalpy balance the heat duties for cooling the vapors (sensible heat) and condensing them (latent heat) are:

Sensible: 
$$q_S = \frac{10,000 \text{kg}}{\text{hr}} \frac{\text{kmole}}{84.9 \text{kg}} \frac{66.3 \text{kJ}}{\text{kmole*}^{\circ}\text{C}} (320^{\circ}\text{C} - 39.9^{\circ}\text{C}) = 2.09 \times 10^{6} \frac{\text{kJ}}{\text{hr}}$$

Latent:  $q_L = \frac{10,000 \text{ kg}}{\text{hr}} \frac{\text{kmole}}{84.9 \text{ kg}} \frac{6,690 \text{ kcal}}{\text{kmole}} \frac{4.187 \text{ kJ}}{\text{kcal}} = 3.30 \times 10^6 \frac{\text{kJ}}{\text{hr}}$ 

The water temperature varies from  $30^{\circ}$ C at the inlet to the outlet of  $30 + 10 = 40^{\circ}$ C, using the maximum rise. From the enthalpy balance, the required flow of water is:

$$m_c = \frac{q_S + q_L}{c_{pw(T_{cb} - T_{ca})}} = (2.09 + 3.30)10^6 \frac{\text{kJ}}{\text{hr}} \frac{\text{kg*°C}}{4.187 \text{kJ}(40 - 30)^{\circ}\text{C}} = 129,000 \frac{\text{kg}}{\text{hr}}$$

The temperature of the water at the point the vapors start condensing is:

$$T_{cm} = T_{ca} + \frac{q_L}{m_c c_{pw}} = 30^{\circ}\text{C} + \frac{3.30 \times 10^6 \text{kJ}}{\text{hr}} \frac{\text{hr}}{129,000 \text{kg}} \frac{\text{kg*}^{\circ}\text{C}}{4.187 \text{kJ}} = 36.1^{\circ}\text{C}$$

Figure 9 shows the temperature profile for the exchanger.



Figure 9. Temperature profile for Example 5

The three temperature differences are:

 $\Delta T_1 = T_{ha} - T_{cb} = 320 - 40 = 280^{\circ}C$ 

$$\Delta T_m = T_{hb} - T_{cm} = 39.9 - 36.1 = 3.8$$
°C

$$\Delta T_2 = T_{hb} - T_{ca} = 39.9 - 30 = 9.9^{\circ}C$$

Because of the nonlinear profile we must divide the exchanger into two sections with linear profiles in each section, the sensible heat section and the latent heat section. The mean temperature differences are the logarithmic mean of the to end temperatures for each section:

Sensible: 
$$\Delta T_S = \frac{\Delta T_1 - \Delta T_m}{\ln(\frac{\Delta T_1}{\Delta T_m})} = \frac{280 - 3.8}{\ln(\frac{280}{3.8})} = 64.0^{\circ} \text{C}$$

Latent: 
$$\Delta T_L = \frac{\Delta T_m - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} = \frac{3.8 - 9.9}{\ln(\frac{3.8}{9.9})} = 6.3^{\circ}\text{C}$$

Had the over-all heat transfer coefficients been specified we would now calculate the area of each section, but as they are not, we now calculate the *UA* product for each section; from Equation (3):

Sensible: 
$$(UA)_S = \frac{q_S}{\Delta T_S} = \frac{2.09 \times 10^6 \text{kJ/hr}}{64.0^\circ \text{C}} = 32.6 \times 10^3 \frac{\text{kJ}}{\text{hr*}^\circ \text{C}}$$

Latent: 
$$(UA)_L = \frac{q_L}{\Delta T_L} = \frac{3.30 \times 10^6 \text{kJ/hr}}{6.3^\circ \text{C}} = 524 \times 10^3 \frac{\text{kJ}}{\text{hr}^{*\circ \text{C}}}$$

At this point we could estimate a different over-all coefficient U for each section and calculate the required heat transfer area A for each section. For now we will calculate the mean temperature difference for the combined two sections, it is:

Mean temperature difference: 
$$\Delta T_M = \frac{q_S + q_L}{(UA)_S + (UA)_L} = \frac{(2.09 + 3.30)10^6}{(32.6 + 524)10^3} = 9.7^{\circ}C$$

Had we assumed a linear temperature profile the mean temperature difference would have been,

$$\frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{280 - 9.9}{\ln\left(\frac{280}{9.9}\right)} = 80.7^{\circ}C$$

If this mean temperature difference were to be used to size the exchanger it would be undersized by a factor of 8. The reason is that the logarithmic mean of the two end temperature differences assumes a linear drop in temperature. You should be able to see by inspection of Figure 9 that under these conditions the temperature difference between the hot and cold streams would be much higher than it actually is everywhere in the exchanger, resulting in a higher mean temperature difference.

#### 6.5 Minimum Approach Temperature

In design engineers must adjust to a number of premises arrived by the experience of past designs. One such premise is the establishment of a *minimum approach temperature*. The approach temperature is the lowest temperature difference in an exchanger. When the temperature profiles are linear the approach temperature occurs at one of the ends of the exchanger, but this is not the case with nonlinear temperature profiles, as in Example 5 where the approach temperature takes place at the point where the vapors start condensing.

As pointed out in Example 4, the logarithmic mean temperature difference is controlled by the smallest of the two temperature differences. This means that the mean temperature difference and therefore the size of the exchanger are highly dependent on the temperature approach. For this reason it is not a good idea to specify small approaches.

In this course we will use a minimum temperature approach of  $5^{\circ}C$  (10°F).

You may ask, why specify the minimum approach? The answer is that it depends on how the approach affects the objectives of the over-all process. For example, in Example 1 the minimum approach is specified because it is safer to store the aniline at the lowest possible temperature. At any rate you should be aware that there is usually a minimum approach when deciding the temperatures in your

process design. You must also be aware that the smaller the approach the larger and more costly the exchanger.

#### 7. Heat Exchanger Simulation

Process simulators such as HySys, Aspen Plus, Process and others can perform very accurate enthalpy and equilibrium calculations based on extensive data banks of component physical properties. This section presents some guidelines on how to simulate heat exchangers. It is important to realize that the simulator is just a tool; you, the engineer, must still do the thinking.

**Inlet Streams.** The flow, composition, and temperature or other enthalpy parameter (e.g., fraction vaporized, saturated liquid) must either be defined or be outlet streams from other operations such as mixers, reactors or distillation columns.

**Pressure Drop.** The pressure drop required for the stream to flow through the exchanger must be specified. The following are common design premises for the pressure drop:

- 70 kPa (10 psi) for all liquid streams being heated or cooled.
- 35 kPa (5 psi) for vapor or partially vaporized streams.
- Negligible pressure drop when the stream is totally condensed or totally vaporized. For example, condensers and reboilers in distillation columns, condensing steam, vaporizing refrigerant.

Understand that in open (flowing) systems the pressure must always drop when the stream flows through the exchanger, even when the stream is being heated.

#### 7.1 Simple Exchanger Models

Simulators offer a simple exchanger model (called "Heater" or "Cooler" in HySys and "One-sided Exchanger" in Aspen Plus). These models compute only the heat duty of the exchanger and are used when there is only one process stream and the other stream is a utility such as cooling water, steam or refrigerant. Furnaces are also simulated with these models.

The only specification required is either the temperature or the enthalpy condition of the exit stream. The enthalpy condition is either the vapor fraction, saturated liquid (vapor fraction = 0) or saturated vapor (vapor fraction = 1). When the enthalpy condition is specified the simulator calculates the temperature and the enthalpy of the exit stream and that is what is needed to calculate the heat duty of the exchanger. The exit temperature must only be specified when the exit stream is not saturated, that is, either a subcooled liquid or a superheated vapor.

The simple exchanger model may be used

- when it is not necessary to size the exchanger from the results of the simulation, or
- when the temperature profile of the process stream is linear (no phase change). This is because the area of the exchanger can then be calculated from the logarithmic mean of the two end temperature differences.

## 7.2 Rigorous Heat Exchanger Models

Simulators also offer a rigorous heat exchanger model that requires the specification of both streams, even when one of them is a utility. The rigorous model must be used when

- the exchanger is a cross-exchanger between two process stream, or
- the temperature profile of the process stream is nonlinear, as in Example 5.

In the second case the utility stream, steam, cooling water or refrigerant, must be defined.

The specification of the rigorous model is either the outlet temperature or vapor fraction of one of the two process streams in a cross-exchanger. When one stream is a utility its outlet temperature or vapor fraction should be specified; the simulator then calculates the required utility stream flow. Do these instructions make sense to you? They should if you understand the enthalpy balance, Equation (1), and realize that is how the simulator calculates the heat duty.

"Weighted" Exchanger. When the temperature profile is nonlinear the "weighted" option of the rigorous heat exchanger model must be specified if the results of the simulation are to be used to size the exchanger. The other option is the "End Point" which calculates the mean temperature difference as the logarithmic mean of the two end differences. (Even the simulator can result if you calculate the incorrect don't specify it correctly.) The weighted option divides the exchanger into sections of linear temperature profiles, as in Example 5, as many sections as you specify, but normally five sections by default. The result of the model is then the total UA for the exchanger which can be used to determine the area A from the over-all heat transfer coefficient U.

# 8. Film Coefficients of Heat Transfer

We have seen earlier in these notes that the over-all heat transfer coefficient  $U_o$  is a function of the film coefficients on each side of the wall separating the hot and cold fluids, and of the resistance of the wall (Equations 6 and 9). This section presents the correlations to estimate the film coefficients for different flow situations. One thing to keep in mind is that the formulas we use are correlations of experimental data and not fundamental equations. Their accuracy is of the order of  $\pm 20$ %.

## 8.1 Turbulent Flow inside Conduits without Phase Change

Turbulent flow is more efficient for heat transfer than laminar flow because the bulk of the fluid away from the wall is kept at a uniform temperature by the mixing eddies so that the resistance to heat transfer is limited to a narrow film around the wall (see Figure 2). A popular correlation to estimate the film coefficient of heat transfer for turbulent flow without phase change is the Sieder-Tate equation:

$$\frac{hD_e}{k} = 0.023 \left(\frac{D_e\rho u}{\mu}\right)^{0.8} \left(\frac{c_p\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{12}$$

Where 
$$h =$$
 the film coefficient of heat transfer  
 $D_e =$  the equivalent diameter (to be defined shortly)  
 $k =$  the thermal conductivity of the fluid  
 $\rho =$  the density of the fluid  
 $u =$  the average velocity of the fluid  
 $\mu =$  the viscosity of the fluid  
 $c_p =$  the specific heat of the fluid  
 $\mu_w =$  the viscosity of the fluid at the wall temperature

The fluid properties are evaluated at the arithmetic mean of the inlet and outlet temperatures of the fluid, and the wall temperature is determined from Equation (10) using the arithmetic mean temperatures of the cold and hot fluids.

Equation (12) involves four dimensionless numbers:

- $hD_e/k$  = the Nusselt number
- $D_e \rho u / \mu$  = the Reynolds number
- $c_p \mu/k$  = the Prandtl number
- $\mu/\mu_{\rm W}$  = the viscosity at the wall correction.

The equation is only valid for Reynolds number greater than 6,000 and does not apply to molten metals.

The viscosity at the wall correction term  $(\mu/\mu_w)^{0.14}$  takes into consideration that the viscosity near the wall is what controls the thickness of the laminar sub-layer and therefore the resistance of the film. The term is greater than unity for the cold liquid and less than unity for the hot liquid because the viscosity of liquids decreases with temperature. The reverse is true for gases. As the exponent is small, the correction may be neglected for fluids with low viscosity such as water.

#### **8.2 The Equivalent Diameter**

The equivalent diameter  $D_e$  in the Nusselt and Reynolds numbers is defined as:

$$D_e = 4\frac{S}{L_w} \tag{13}$$

Where S is the cross sectional area of the conduit and  $L_w$  is the wetted perimeter. For example, for flow inside a cylindrical pipe or tube,

$$S = \pi D_i^2 / 4 \qquad \qquad L_w = \pi D_i \qquad \qquad D_e = D_i$$

Where  $D_i$  is the inside diameter of the pipe. Some exchangers are built with one pipe inside another larger pipe with one fluid flowing through the smaller pipe and the other one through the annular space between the two pipes, as in Figure 10. These are called *double-pipe heat exchangers*.



 $S = \pi (D_1^2 - D_2^2) / 4$  $L_w = \pi (D_1 + D_2)$  $D_e = D_1 - D_2$ 

Figure 10.End view of double-pipe heat exchanger

Where  $D_1$  is the inside diameter of the outer pipe and  $D_2$  is the outside diameter of the inner pipe. Make sure you understand the equivalent diameter is used only in the Nusselt and Reynolds numbers. The cross sectional area is not  $S = \pi D_e^2/4$  and the side area is not  $A = \pi D_e L$ .

**Example 6. Size Aniline Cooler** 

Design a double-pipe heat exchanger for the aniline cooler of Example 1. Use a 1-in Schedule 40 steel pipe inside a 2-in Schedule 40 pipe.

Solution. Select countercurrent flow with the aniline flowing through the inner pipe and the water through the annulus between the pipes.



#### Figure 11. Double pipe heat exchanger for Example 6

```
      From Example 1:

      Aniline: m_h = 55,000 \text{ kg/hr}
      T_{ha} = 136^{\circ}\text{C}
      T_{hb} = 35^{\circ}\text{C}

      Water: m_c = 306,000 \text{ kg/hr}
      T_{ca} = 30^{\circ}\text{C}
      T_{cb} = 40^{\circ}\text{C}

      Heat duty: q = (3.06 \times 10^6 \text{ kcal/hr}) (4.187 \text{ kJ/kcal}) = 12.8 \times 10^6 \text{ kJ/hr}

      And, from Example 4: \Delta T_M = 30.8^{\circ}\text{C}

      From McCabe, Smith and Harriott, 6^{\text{th}} ed. Appendix 3:

      Inner pipe, 1-in Sch. 40:

      D_i = 1.049 in, D_o = 1.315 in, S_h = 0.00600 \text{ ft}^2, x_w = 0.133 in

      Outer pipe, 2-in Sch. 40:
      D_{oi} = 2.067 in

      For the annulus:

      Equivalent diameter: 2.067 - 1.315 = 0.752 in
```

Cross sectional area:  $\pi(2.067^2 - 1.315^2)/4 = 2.00 \text{ in}^2 = 1.29 \times 10^{-3} \text{m}^2$ Properties of aniline, from McCabe, Smith and Harriott, 7<sup>th</sup> ed.: Average temperature:  $(136 + 35)/2 = 85.5^{\circ}\text{C}$  (186°F) Viscosity: 0.91 cP = 3.28 kg/m-hr (Appendix 9) Thermal conductivity:  $0.100 \text{ Btu/ft-hr-}^{\circ}\text{F} = 0.622 \text{ kJ/m-hr-}^{\circ}\text{C}$  (Appendix 13) Specific heat:  $0.55 \text{ kcal/kg-}^{\circ}\text{C} = 2.30 \text{ kJ/kg-}^{\circ}\text{C}$  (Appendix 15)

Specific gravity: 1.022 (Perry's, Table 2-2)

Prandtl number:

$$\left(\frac{c_p\mu}{k}\right)_i = \frac{2.30 \text{kJ}}{\text{kg*}^\circ \text{C}} \frac{3.28 \text{kg}}{\text{m*hr}} \frac{\text{m*hr*}^\circ \text{C}}{0.622 \text{kJ}} = 12.1$$

Properties of water, from McCabe, Smith & Harriott, Appendix 6: Average temperature: (30 + 40)/2 = 35°C (95°F) Viscosity: 0.723 cP = 2.60 kg/m-hr Thermal conductivity: 0.360 Btu/ft-hr-°F = 2.24 kJ/m-hr-°C Specific heat: 1.0 kcal/kg-°C = 4.187 kJ/kg-°C Specific gravity: 1.0

Prandtl number:

$$\left(\frac{c_p\mu}{k}\right)_o = \frac{4.187 \text{kJ}}{\text{kg*}^{\circ}\text{C}} \frac{2.60 \text{kg}}{\text{m*hr}} \frac{\text{m*hr*}^{\circ}\text{C}}{0.173 \text{kJ}} = 4.86$$

Calculate the average velocities:

Aniline:  $u_i = \frac{55,000 \text{kg}}{\text{hr}} \frac{1}{0.00600 \text{ft}^2} \left(\frac{\text{ft}}{0.3048 \text{m}}\right)^2 \frac{\text{m}^3}{1022 \text{kg}} \frac{\text{hr}}{3,600 \text{s}} = 26.8 \frac{\text{m}}{\text{s}}$  or 88 ft/s This velocity is too high. We must use several exchangers in parallel to bring the velocity down. Let us use 50 exchangers in parallel:  $u_i = 26.8/50 = 0.536 \text{ m/s}$ . Water velocity:  $u_o = \frac{1}{50} \frac{306,000 \text{kg}}{\text{hr}} \frac{1}{1.29 \times 10^{-3} \text{m}^2} \frac{\text{m}^3}{1000 \text{kg}} \frac{\text{hr}}{3.600 \text{s}} = 1.316 \frac{\text{m}}{\text{s}}$ 

Next calculate the Reynolds numbers: Aniline:  $\frac{D_i \rho_i u_i}{\mu_i} = \frac{1.049in}{39.4in/m} \frac{1022kg}{m^3} \frac{0.536m}{s} \frac{m*hr}{3.28kg} \frac{3,600s}{hr} = 16,000$ 

Water: 
$$\frac{D_e \rho_0 u_o}{\mu_o} = \frac{0.752 in}{39.4 in/m} \frac{1000 \text{kg}}{m^3} \frac{1.316 \text{m}}{\text{s}} \frac{\text{m*hr}}{2.60 \text{kg}} \frac{3,600 \text{s}}{\text{hr}} = 34,800$$

The flow is definitely turbulent, so we can use the Sieder-Tate correlation. The film coefficients are then, neglecting the viscosity at the wall correction:

Aniline:

$$h_i = 0.023 \frac{0.622 \text{kJ}}{\text{m} \cdot \text{hr} \cdot ^\circ \text{C}} \frac{39.4 \text{in}/\text{m}}{1.049 \text{in}} (16,000)^{0.8} (12.1)^{1/3} = 2,850 \frac{\text{kJ}}{\text{hr} \cdot \text{m}^2 \cdot ^\circ \text{C}}$$

Water:

$$h_o = 0.023 \frac{2.24 \text{kJ}}{\text{m*hr*°C}} \frac{39.4 \text{in/m}}{0.752 \text{in}} (34,800)^{0.8} (4.86)^{1/3} = 19,600 \frac{\text{kJ}}{\text{hr*m}^2 \text{*°C}}$$

For the wall resistance, the thermal conductivity of steel is 26  $Btu/ft-hr-^{F}$  (McCabe, Smith & Harriott, 7<sup>th</sup> ed., Appendix 10). So the resistances are:

Aniline: 
$$D_o/D_ih_i = (1.315/1.049)/2,850 = 4.4 \times 10^{-4} \text{ hr}-\text{m}^2-^{\circ}\text{C/kJ}$$
  
Water:  $1/h_o = 1/$  19,600 =  $0.5 \times 10^{-4} \text{ hr}-\text{m}^2-^{\circ}\text{C/kJ}$ 

Wall: 
$$D_M = \frac{D_o - D_i}{\ln\left(\frac{D_o}{D_i}\right)} = \frac{1.315 - 1.049}{\ln\left(\frac{1.315}{1.049}\right)} = 1.177 \text{ in}$$

$$\frac{x_w D_o}{k_w D_M} = \frac{0.133 \text{in}}{\frac{39.4 \text{in}}{\text{m}}} \frac{\text{hr} * \text{ft} * {}^\circ\text{F}}{26 \text{Btu}} \frac{\text{Btu}}{1.055 \text{kJ}} \frac{0.3048 \text{m}}{\text{ft}} \frac{{}^\circ\text{C}}{1.8 {}^\circ\text{F}} \frac{1.315 \text{in}}{1.177 \text{in}} = 0.2 \times 10^{-4} \frac{\text{hr} * \text{m}^2 * {}^\circ\text{C}}{\text{kJ}}$$

The aniline resistance is controlling. Over-all heat transfer coefficient:

The total required heat transfer area is:

$$A_o = \frac{q}{U_o \Delta T_M} = \frac{12.8 \times 10^6}{(1,960)(30.8)} = 212 \, m^2 \qquad (2,300 \, \text{ft}^2)$$

Length of each exchanger:  $A_o = 50 \pi D_o L$ 

$$L = \frac{212m^2}{50} \frac{39.4\text{in/m}}{\pi(1.315\text{in})} = 40.4 \text{ m}$$
 (132 ft)

So, fifty exchangers each longer than 40 m are required. Obviously this will not be a good design. A shell-and-tube heat exchanger should be used. These will be introduced shortly.

## **8.3 Laminar Flow**

Although laminar flow is not as efficient as turbulent flow to transfer heat, sometimes it cannot be avoided when the fluid is very viscous. The following correlation can be used when the Reynolds number is less than 6,000:

$$\frac{h_i D_e}{k} = 2.0 \left(\frac{mc_p}{N_t kL}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{14}$$

Where m = the mass flow

 $N_t$  = the number of tubes in parallel

L = the tube length

Notice that  $m/N_t$  is the mass flow per tube. The rest of the terms were defined with Equation (12).

#### 8.4 Condensing Vapors on Vertical Tubes

The model for vapors condensing on vertical tubes is that the resistance to heat transfer is presented by a film of condensate flowing down the tube. The thickness of this film increases down the tube. The flow may be laminar or turbulent. The correlation uses the following form of the Reynolds number:

$$N_{Re} = \frac{4m_h}{\mu_f N_t \pi D_o} \tag{15}$$

Where  $N_t$  is the number of tubes. When this Reynolds number is less than 2,100, the following correlation applies:

$$h_o = 1.47 N_{Re}^{-1/3} \left( \frac{k_f^3 \rho_f^2 g}{\mu_f^2} \right)^{1/3} \tag{16}$$

And when the Reynolds number is greater than 2,100:

$$h_o = 0.0076 N_{Re}^{0.4} \left(\frac{k_f^3 \rho_f^2 g}{\mu_f^2}\right)^{1/3} \tag{17}$$

The subscript f on the physical properties indicates that they must be evaluated at the *film temperature*, defined by

$$T_f = T_h - 0.75 (T_h - T_{wh}) \tag{18}$$

As condensing steam is commonly used for heating process streams in industry, the group on the right of Equations (16) and (17) and later on Equation (20), is given as a function of temperature for water in Appendix 6 of McCabe, Smith and Harrriott, 7<sup>th</sup> edition.

#### **8.5 Condensing Vapors on Horizontal Tubes**

The model for condensation on horizontal tubes is that the condensate from each tube drains to the tubes below so that the film thickness is larger on the lower tubes. The correlation uses the following form of the Reynolds number:

$$N_{Re} = \frac{4m_h}{\mu_f N_t L} \tag{19}$$

Where  $N_t$  is the number of tubes. When this Reynolds number is less than 2,100, the following correlation applies:

$$h_o = 1.51 \frac{N_{Re}^{-1/3}}{N_{stack}^{1/4}} \left(\frac{k_f^3 \rho_f^2 g}{\mu_f^2}\right)^{1/3}$$
(20)

Where  $N_{stack}$  is the number of tubes stacked in a vertical row.

## **8.6 Boiling Liquids**

There are no generally accepted correlations for estimating the film coefficients in boiling liquids. Experimental determined over-all coefficients are used in the design of evaporators and reboilers.

One important characteristic of boiling liquids is that there are several regimes of boiling that depend on the temperature difference between the heating surface and the boiling liquid. The regimes are,

- Natural Convection. At low temperature differences (AT < 5°C) heat is transferred at very low rates by natural convection.</li>
- Nucleate Boiling. As the temperature difference increases  $(5^{\circ}C < \Delta T < 25^{\circ}C)$  vapor bubbles start forming and as they rise to the surface create forceful mixing of the liquid promoting efficient heat transfer.

• Film Boiling. As the temperature difference continues to increase ( $\Delta T > 25^{\circ}$ C) the bubbles form so rapidly that a film of vapor covers the heating surface creating a higher resistance to heat transfer. This is undesirable.

You may observe these regimes by boiling a pot of water on a stove and observing the boiling process, at least the first two regimes. The generation of the vapor bubbles should let you appreciate why heat transfer is more efficient in the nucleate boiling regime.

The most efficient heat transfer is obtained around the upper range of the nucleate boiling regime, 25°C. Study of Figures 13.4 and 13.5 of McCabe, Smith and Harriott, 7<sup>th</sup> ed. and the description in the text will help you understand the basic concepts of designing evaporators and reboilers.

## 9. Shell and Tube Heat Exchangers

Shell and tube heat exchangers consist of a bundle of tubes enclosed in a cylindrical shell. One stream flows through the tubes and the other one flows through the shell outside the tubes. These exchangers provide a high density of heat transfer area, up to about 500 m<sup>2</sup> (5,000 ft<sup>2</sup>) in a single shell. The results of Example 6 showed us that it is impractical to build a large double-pipe heat exchanger. The same can be said of coils and jackets to cool and heat reactors and other vessels, and of plate heat exchangers. When large areas are required to transfer heat, shell and tube heat exchangers are normally used.

Shell and tube exchangers can have a single tube pass, as in Figure 12, and multi-pass exchangers, as in Figure 13. When a multi-pass exchanger is used the number of tube passes is always and even number.



Figure 12. Single tube-pass shell and tube exchanger

In the figures the subscript T refers to the stream flowing inside the tubes and S is for the stream flowing in the shell. Either stream can be the hot or the cold stream.

The *baffles* on the shell side are intended to force the shell side streams to flow perpendicular to the tubes to promote higher turbulence and also to increase the velocity of the stream. The turbulence and the higher velocity increase the film coefficient on the shell side. One design parameter is the baffle pitch *P* which is adjusted by selecting the number of baffles and determines the velocity of the shell side stream. Baffles are not necessary when the shell stream is either a condensing vapor or a vaporizing liquid.

The disadvantage of the single tube pass exchanger is that the shell cannot be removed to clean the outside of the tubes when solids deposit on them. Inspection of Figure 13 shows that the shell in a multi-pass exchanger can be removed to clean the tubes, as long the number of tube passes is even. (The baffles are not attached to the shell.)



Figure 13. Multiple tube-pass shell and tube exchanger

Notice that in the multi-pass exchanger the shell stream should enter at the end of the exchanger where the tube stream enters and leaves. Otherwise there may be a temperature cross, that is, a portion of the exchanger where the cold stream temperature is higher than the hot stream temperature. Surprisingly enough, in theory, the mean temperature difference is the same independent of the entrance end of the shell stream. In other words, in theory it does not matter if there is a temperature cross in a multi-pass exchanger, but it should be avoided in practice anyway.

The heat transfer area of a shell and tube exchanger based on the outside area of the tubes is:

$$A_{o} = N_{T} \pi D_{o} L \tag{21}$$

Where  $N_T$  = the total number of tubes in the shell

 $D_{\rm o}$  = the outside diameter of the tubes

L = the length of the tubes

Most chemical plants and refineries require a specific tube diameter and length on their heat exchangers to reduce the inventory of spare tubes that must be kept in stock. The number of tubes in the shell depends on the diameter of the shell, the diameter, layout and spacing of the tubes, and the number of tube passes. The Tubular Exchanger Manufacturers Association (TEMA) has prepared tables giving the number of tubes per shell for what they have defined as standard shells. These tables can be found on the web.

#### 9.1 Shell-side Film Coefficient without Phase Change

between the baffle and the shell is:

The film coefficient of heat transfer for the shell side can be estimated with the following correlation, known as the Donohue equation:

$$\frac{h_o D_o}{k} = 0.2 \left(\frac{D_o G_e}{\mu}\right)^{0.6} \left(\frac{c_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{22}$$

Where  $D_o$  = outside diameter of the tubes

 $G_e = \rho u_e$  = mean mass velocity

A mean mass velocity must be computed because the stream flows between the baffles part of the time and through the opening left by the baffle the other part of the time. The area of flow

$$S_b = f_b \frac{\pi}{4} (D_s^2 - N_T D_o^2) \tag{23}$$

Where  $S_b$  = area of flow between the baffle a the shell  $D_s$  = inside diameter of the shell

When the baffle length is  $\frac{3}{4}$  of the shell diameter,  $f_b = 0.196$ . The area of flow between baffles is determined at the center of the shell:

$$S_c = PD_s \left( 1 - \frac{D_o}{p_T} \right) \tag{24}$$

Where P = the baffle pitch or spacing between the baffles

 $p_T$  = tube pitch or distance between the tube centers. The mean mass velocity is then determined with the geometric mean of the two flow areas:

$$G_e = \frac{m_s}{\sqrt{S_b S_c}} \tag{25}$$

Where  $m_S$  is the mass flow of the shell stream.

From the correlation of Equation (22) the shell-side film coefficient proportional to the mass velocity to the 0.6 power. When the shell-side coefficient is too small and the controlling resistance, it can be increased by increasing the number of baffles thus reducing the area of flow between the baffles and increasing the mass velocity. However, when the velocity is increased the pressure drop through the exchanger also increases imposing a limit on the number of baffles. The pressure drop is proportional to the square of the velocity and to the number of

baffles. To keep the pressure drop reasonable the velocity should not exceed 2 m/s (7 ft/s) for liquids or 20 m/s (70 ft/s) for vapors.

#### 9.2 Selecting the Number of Tube Passes

In a multi-pass exchanger the number of passes can be selected as an even number (2, 4, 6...). The velocity inside the tubes is determined by the number of tubes per pass:

$$G_T = \rho u_T = \frac{m_T}{N_p \frac{\pi}{4} D_i^2} \tag{26}$$

Where  $G_T$  = mass velocity in the tubes  $u_T$  = average velocity in the tubes  $m_T$  = mass flow of the stream flowing in the tubes  $N_p$  = number of tubes per pass  $D_i$  = inside diameter of the tubes.

For the film coefficients inside the tubes the Sieder-Tate correlation, Equation (12), applies. The film coefficient is proportional to the velocity raised to the 0.8 power. So, if the inside film coefficient is small and the controlling resistance, it can be increased by increasing the number of tube passes which in turn increases the velocity. So going from 2 to four passes the velocity doubles and the inside film coefficient is increased by a factor of  $2^{0.8} = 1.74$  for a 74% increase.

As with the number of baffles on the shell side, increasing the number of tube passes increases the pressure drop through the tubes, but unlike the baffles the number of tube passes can take only even numbers. The pressure drop through the tubes is

proportional to the velocity squared and to the number of tube passes so, if the number of tube passes is doubled, the pressure drop increases by a factor of  $2^2 \times 2 = 8$ . So, for example, if the initial pressure drop is small such as 1 psi (7 kPa), increasing the number of passes from 2 to 4 increases the pressure drop to 8(1) = 8 psi (56 kPa) which is acceptable, but increasing the number of passes to 6 increases the pressure drop to  $3 \times 3^2(1) = 27$  psi (186 kPa) which is excessive. Recall that the desired pressure drops in exchangers are typically from 5 to 10 psi (35 to 70 kPa). To maintain reasonable pressure drops the velocity should not exceed 2 m/s (7ft/s) for liquids or 20 m/s (70 ft/s) for vapors.

#### 9.3 Correction of Mean Temperature Difference for Multi-pass Exchanger

exchanger is The flow in a multi-pass not parallel or countercurrent, but a combination of the two. As Figure 13 shows, the stream in the tubes moves first in parallel with the shell-side stream and then in countercurrent. If the temperature of one of the two streams is constant, the mixed flow does not affect the mean temperature difference which is the logarithmic mean of the end differences. Figure 14 shows the profile as a plot of temperature versus the position in the exchanger for two tube passes. (The profiles versus position are not linear while the profiles of temperature versus heat transferred are linear.)



Figure 14. Temperature vs. position for two-tube pass exchanger

The mean temperature difference for a multi-pass exchanger is determined by using a correction factor to the logarithmic mean temperature difference for the counter-current exchanger:

$$\Delta T_M = F_G \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{27}$$

Where the correction factor  $F_G$  for a single shell pass and any even number of tube passes is:

$$F_{G} = \frac{\left(\frac{\sqrt{Z^{2}+1}}{Z-1}\right) \ln\left(\frac{1-\eta_{H}}{1-\eta_{H}Z}\right)}{\ln\left[\frac{\frac{2}{\eta_{H}}-1-Z+\sqrt{Z^{2}+1}}{\frac{2}{\eta_{H}}-1-Z-\sqrt{Z^{2}+1}}\right]}$$
(28)

Where 
$$Z = \frac{T_{ha} - T_{hb}}{T_{cb} - T_{ca}}$$
  $\eta_H = \frac{T_{cb} - T_{ca}}{T_{ha} - T_{ca}}$ 

The correction factor is the same whether the cold or the hot stream flow through the tubes. A plot of Equation (28) can be found in McCabe, Smith and Harriott,  $7^{th}$  ed., Figure 15.6(a).

#### 9.4 Correction of Mean Temperature Difference for Two Shell Passes

When the correction factor  $F_G$  for one shell pass is too low (< 0.9), two shell passes are used. Two shell passes usually consists of two identical shells with multiple tube passes, with the total number of tube passes being a multiple of 4 (4, 8, 12,...). The two shells must be arranged in countercurrent, as in Figure 15. Although two shells are used, the exchanger is sized (and simulated) as a single exchanger.



Figure 15. Arrangement for two shell passes (4 tube passes)

The correction factor for two shell passes and 4, 8, 12, etc. tube passes is:

$$F_{G} = \frac{\left(\frac{\sqrt{Z^{2}+1}}{2(Z-1)}\right) \ln\left(\frac{1-\eta_{H}}{1-\eta_{H}Z}\right)}{\ln\left[\frac{\frac{2}{\eta_{H}}-1-Z+\frac{2}{\eta_{H}}\sqrt{(1-\eta_{H})(1-\eta_{H}Z)}+\sqrt{Z^{2}+1}}{\frac{2}{\eta_{H}}-1-Z+\frac{2}{\eta_{H}}\sqrt{(1-\eta_{H})(1-\eta_{H}Z)}-\sqrt{Z^{2}+1}}\right]}$$
(29)

The factor is used in Equation (27) to compute the mean temperature difference with two shells. Keep in mind that the two shells must be identical. A plot of Equation (29) can be found in McCabe, Smith and Harriott, 7<sup>th</sup> ed., Figure 15.6(b).

The following applies to Equations (28) and (29):

- If the temperature of one of the two streams is constant (condensing pure vapor or boiling liquid)  $F_G = 1.0$ .
- It does not matter which stream, the hot or the cold, flows in the tubes.
- It does not matter in which direction the shell fluid flows.

## 9.5 Cross Flow

In some exchangers, typically air-cooled exchangers, the flow of the two streams is perpendicular to each other. This is called *cross flow*. For the correction factor of the mean temperature difference in cross flow see McCabe, Smith and Harriott, 7<sup>th</sup> ed., Figure 15.7. We will not discuss the design of air-cooled exchangers in these notes.

# 10. Quick Design of Heat Exchangers

In situations such as the preliminary design of a process there are many heat exchangers to design. It is then desirable to have a quick design approach that saves the time of having to do the detailed design of each heat exchanger. The quick design procedure consists of calculating the heat duty and mean temperature difference and estimate the heat transfer area using typical over-all coefficients for the type of exchanger under consideration. The detailed design of each exchanger is then done at a later stage of the design of the process.

To estimate the heat transfer coefficient, resistances for typical heat transfer situations are used, such as those shown in Table 1.

#### Table 1. Typical Film Resistances for Quick Design

Fluid	No phase	Boiling	Condensing
	change	liquid	vapor
Fixed gases (N <sub>2</sub> , O <sub>2</sub> , etc.)	2.2×10 <sup>-3</sup>		
Light hydrocarbon gases	1.7×10 <sup>-3</sup>		
Aromatic liquids	0.35×10 <sup>-3</sup>	0.15×10 <sup>-3</sup>	0.35×10 <sup>-3</sup>
Light hydrocarbon liquids	0.2×10 <sup>-3</sup>	0.15×10 <sup>-3</sup>	0.2×10 <sup>-3</sup>
Chlorinated hydrocarbons	0.2×10 <sup>-3</sup>	0.15×10 <sup>-3</sup>	0.2×10 <sup>-3</sup>
Steam	2.2×10 <sup>-3</sup>		0.05×10 <sup>-3</sup>
Boiler water	0.35×10 <sup>-3</sup>	0.15×10 <sup>-3</sup>	
Cooling tower water	0.35×10 <sup>-3</sup>		

In  $hr-m^2-°C/kJ$  (multiply by 20 to get  $hr-ft^2-°F/Btu$ )

At this level of detail the wall and dirt resistances may be neglected and the over-all coefficient estimated by

$$\frac{1}{U_o} = r_h + r_c \tag{30}$$

Where  $r_h$  and  $r_c$  are the resistances of the hot and cold streams from Table 1.

#### Example 7. Estimate the area required for the aniline cooler of Example 1

From the results of Example 1, Heat duty:  $q = (3.06 \times 10^6 \text{ kcal/hr}) (4.187 \text{ kJ/kcal}) = 12.8 \times 10^6 \text{ kJ/hr}$ From the results of Example 3, for countercurrent flow, Mean temperature difference:  $\Delta T_M = 30.8^{\circ}\text{C}$ 

Solution.	From	Table	1	the	resista	nces	are:		
Cooling a:	romati	ic liqu	iid	ds:	$r_h =$	0.35	×10 <sup>-3</sup>	$hr-m^2-°C$	
Cooling to	ower w	vater:			$r_h =$	0.35	×10 <sup>-3</sup>	hr-m <sup>2</sup> -°C	

Over-all coefficient:  $U_{\circ} = 1/(0.35+0.35) \times 10^{-3} = 1,430 \text{ kJ/hr-m}^2-^{\circ}\text{C}$ 

Heat transfer area:  $A_o = (12.8 \times 10^6) / [(30.8) (1, 430)] = 290 \text{ m}^2$ 

Or 3,100  $ft^2$ . Compare with 212  $m^2$  for the detailed design of a double-pipe heat exchanger in Example 6. The disagreement is within the expected accuracy of the values in Table 1. The purpose of the table is just to obtain an approximate exchanger size with a quick calculation. Compare also the complexity of the calculations of Example 6 with those in this example.

## **11.** Quick Estimate of Exchanger Cost

Correlations have been developed to estimate the cost of an exchanger based on the heat transfer area. As would be expected, the accuracy of these correlations is of the order of  $\pm 30\%$ . The following correlations are adapted from the ones on Douglas, *Conceptual Design of Chemical Processes*, McGraw-Hill, 1989, Appendix E.

$$IC_{exch} = \left(\frac{M\&S}{280}\right) \$480 \left(\frac{A_o}{m^2}\right)^{0.65} (F_c + 2.29) \tag{31}$$

Where  $IC_{exch}$  = the installed cost of the exchanger

M&S = the Marshal and Swift equipment inflation index

 $A_o$  = the heat transfer area in m<sup>2</sup>

And  $F_c$  is a cost factor which is 1.0 for a floating head exchanger in carbon steel with a design pressure of less than 1000 kPa gage (150 psig).

The correlation of Equation (31) is valid for exchangers of up to 500 m<sup>2</sup> (5,000 ft<sup>2</sup>). When the area is larger than the correlation limit the cost is best estimated by assuming more than one exchanger all with the same area less than the limit, e.g., for a 900 m<sup>2</sup> exchanger assume three exchangers of 300 m<sup>2</sup> each.

For other exchanger types, design pressures and materials of construction, the cost factor is

$$F_c = F_m (F_d + F_p) \tag{32}$$

Most exchangers are of the floating head type for which  $F_d = 1.00$ , U-tube type,  $F_d = 0.85$ , or fixed head (one tube pass),  $F_d =$ 

0.80. The factors for material of construction and the design pressure are:

Material	CS/CS	CS/Brass	CS/MO	CS/SS	SS/SS	CS/Monel	Monel/Monel	CS/Ti	Ti/Ti
$F_m$	1.00	1.30	2.15	2.81	3.75	3.10	4.25	8.95	13.05
CS = Ca	rbon S	Steel; S	S = St	cainle	ss Ste	eel; Ti	= Titanium	l <b>.</b>	

Cost factors for material of construction, shell/tube

Cost factors for design pressures up to

kPa gage	1030	2070	2760	5510	6900
psig	150	300	400	800	1000
$F_p$	0.00	0.10	0.25	0.52	0.55

#### **Example 8. Estimate the cost of the heat exchanger of Example 1**

From the results of Example 7,  $A_o = 290 \text{ m}^2$ .

Solution. Assuming a floating head exchanger in carbon steel and low pressure, from Equation (31), using a current inflation index of 2,000:

Installed cost: 
$$IC_{exch} = \frac{2,000}{280} 480(290)^{0.65}(1.0 + 2.29) = $450 \times 10^3$$

With the area from Example 6, 212  $m^2$ , the cost would be \$370×10<sup>3</sup>, which for preliminary design purposes is the same (within 20%).

# **12.Detailed Design of a Heat Exchanger**

The detailed design of a shell and tube exchangers goes beyond the estimate of the heat transfer area  $A_o$ . It involves the determination of

• The shell diameter

- The number of tubes
- The number of tube passes
- The number of baffles
- A more accurate estimation of the over-all heat transfer coefficient  $U_o$  based on the velocities and properties of the hot and cold streams.

Before discussing the design procedure we must present the concept of the *fouling resistance*.

#### **12.1 Fouling Resistance**

When an exchanger is new or has just been cleaned there is no dirt deposited on its heat transfer surfaces and the over-all coefficient is the highest it can be for that exchanger. This coefficient is called the *clean coefficient*. As the exchanger is put in service dirt deposits on its surfaces and the coefficient decreases reducing the capacity of the exchanger to transfer heat. Eventually the coefficient is so low that the exchanger is said to be "fouled up." It must then be taken out of service and cleaned. If the exchanger is not sufficiently over-sized the period between cleanings is short requiring frequent shut downs for cleaning. This is costly and inconvenient. Therefore exchangers must be over-sized by having the heat transfer area required by the clean coefficient. larger than that The oversizing is done by allowing for the resistance of the dirt, called the fouling allowance.

The fouling allowance is calculated as a resistance which is the difference between the resistance of the design coefficient and the clean coefficient:

$$ff = \frac{1}{U_{oD}} - \frac{1}{U_{oC}} \tag{33}$$

Where ff = fouling allowance  $U_{oD}$  = design over-all coefficient  $U_{oC}$  = clean over-all coefficient

A properly designed exchanger must allow sufficient fouling resistance for the service of the exchanger so that the period between cleanings is long, of the order of several months to a year. For most services the fouling allowance should be of the order of  $0.1 \times 10^{-3}$  hr-m<sup>2</sup>-°C/kJ (0.002 hr-ft<sup>2</sup>-°F/Btu).

#### **12.2 Detailed Design Procedure**

The following is the procedure for the detailed design of a shell and tube heat exchanger:

- 1. Calculate the heat duty q and exit temperature or flow from the enthalpy balance.
- 2. Calculate the mean temperature difference  $\Delta T_M$ . Note: If one shell pass does not work must use two shell passes.
- 3. Estimate the over-all heat transfer coefficient  $U_o$  from typical resistances such as those in Table 1.
- 4. Calculate the estimated heat transfer area,  $A_o = q/U_o \Delta T_M$ .
- 5. Determine the number of tubes  $N_T$  using the company-specified tube diameter and length,  $N_T = A_o/\pi D_o L$ .
- 6. Select the number of tube passes  $N_p$  to obtain a reasonable velocity inside the tubes,  $u_i = m_i N_p / (N_T \rho_i \pi {D_i}^2 / 4)$ .

- 7. Select a TEMA standard shell diameter containing a number of tubes  $N_T$  slightly higher than the one calculated in step 5. Correct the tube side velocity from step 6.
- 8. Select the number of baffles  $N_b$  to produce a reasonable velocity of the shell stream,  $u_o = m_o/[PD_s\rho_o(1-D_o/p_T)]$ ,  $P = L/(N_b + 1)$ . Note: The minimum baffle pitch P is  $D_s/5$  or 2 in, whichever is larger.
- 9. Calculate the design coefficient,  $U_{oD} = q/(N_T \pi D_o L \Delta T_M)$ .
- 10. Calculate the clean coefficient  $U_{oC}$  from the film coefficients and the resistance of the wall and check that the allowed fouling resistance is large enough.
- 11. If the allowed fouling resistance is too small or negative, select a larger shell diameter and repeat the calculations from step 7. Otherwise the detailed design of the exchanger is complete.

This procedure is demonstrated in the following example.

#### **Example 9. Detailed design of aniline cooler of Example 1**

The client requires the exchanger tubes to be 5/8-in OD on 15/16-in square pitch and 20 ft long.

Solution. Have the aniline flow through the tubes and the water through the shell. From Example 1:

$m_h = 55,000  \text{kg/hr}$	$T_{ha} = 136 {}^{\circ}\mathrm{C}$	$T_{cb} = 35^{\circ}C$
$m_c = 306,000  \text{kg/hr}$	$T_{ca} = 30^{\circ} C$	$T_{cb} = 40^{\circ} C$

1. Enthalpy balance (from Example 1):  $q = 12.8 \times 10^6 \text{ kJ/hr}$ 

2. Mean temperature difference, withy multi-pass correction. The one shell correction factor tells us that two shells are needed. So, the correction factor for two shells is

$$\begin{split} \eta_{\rm H} &= (40 - 30) \,/ \,(136 - 30) \,= \,0.0943 \\ F_G &= \frac{\frac{\sqrt{10.1^2 - 1}}{2(10.1 - 1)} \ln\left(\frac{1 - 0.0943}{1 - (0.0943)(10.1)}\right)}{\ln\left(\frac{2}{0.0943} - 1 - 10.1 + \frac{2}{0.0943} \sqrt{(1 - 0.0943)(1 - (0.0943)(10.1)]} + \sqrt{10.1^2 - 1}\right)}{\frac{2}{0.0943} - 1 - 10.1 + \frac{2}{0.0943} \sqrt{(1 - 0.0943)(1 - (0.0943)(10.1)]} - \sqrt{10.1^2 - 1}\right)} = 0.948 \\ \varDelta T_M &= 0.948 \frac{(136 - 40) - (35 - 30)}{\ln\left(\frac{136 - 40}{35 - 30}\right)} = 29.2^{\circ}C \end{split}$$

Z = (136 - 35) / (40 - 30) = 10.1

- 3. Estimate the over-all heat transfer coefficient. From Table
  - 1, the estimated resistances are:

$$r_{h} = 0.35 \times 10^{-3} \text{ hr} - \text{m}^{2} - \text{°C/kJ} \qquad (\text{aromatic liquid})$$

$$r_{c} = 0.35 \times 10^{-3} \text{ hr} - \text{m}^{2} - \text{°C/kJ} \qquad (\text{cooling tower water})$$

$$U_{0} = 1/(0.035 + 0.035) \times 10^{-3} = 1,430 \text{ kJ/hr} - \text{m}^{2} - \text{°C}$$

4. Estimate the heat transfer area.

 $A_{\circ} = (12.8 \times 10^{6}) / [(1, 430) (29.2)] = 307 \text{ m}^{2}$ 

5. Determine the number of tubes. For 5/8-in OD 14 BWG tubes, 20 ft long, from McCabe, Smith and Harriott, 7<sup>th</sup> ed., Appendix 4:

> $D_o = 5/8 = 0.625$  in  $D_i = 0.459$  in  $S_i = 0.00115$  ft<sup>2</sup> = 1.07×10<sup>-4</sup> m<sup>2</sup>  $x_w = 0.083$  in

And L = 20 ft = 6.10 m

$$N_T = \frac{A_o}{\pi D_o L} = \frac{307 \text{m}^2}{\pi (0.625 \text{in})(6.10 \text{m})} \frac{39.4 \text{in}}{m} = 1,010 \text{ tubes}$$

6. Select the number of tube passes for a reasonable tube-side velocity. The density of aniline, from Perry's Table 2.2, is  $1,022 \text{ kg/m}^3$ .

For 4 tube passes:  $u_i = \frac{4(55,000 \text{kg/hr})}{(1010)(1,022 \text{kg/m}^3)(1.07 \times 10^{-4} \text{m}^2)} \frac{\text{hr}}{3,600 \text{s}} = 0.55 \frac{\text{m}}{\text{s}}$ This velocity is too low, so use 8 tube passes, for a velocity of 1.10 m/s. 7. Select a TEMA standard shell with the number of tubes. From <u>http://www.globalspec.com/reference/38650/203279/Appendix-</u> <u>C-Tube-Count-Tables</u>, for 5/8-in OD tubes on 13/16-in square pitch:

> $D_s = 23.25$  in = 0.590 m  $N_T = 2(536) = 1072$  tubes  $p_T = 13/16 = 0.812$  in = 0.0206 m Correct the tube-side velocity:

$$u_i = \frac{8(55,000 \text{ kg/hr})}{(1072)(1,022 \text{ kg/m}^3)(1.07 \times 10^{-4} \text{ m}^2)} \frac{\text{hr}}{3,600 \text{s}} = 1.044 \frac{\text{m}}{\text{s}}$$

8. Select the number of baffles for a reasonable shell-side velocity. Try 20 baffles:

P = L/(20 + 1) = 6.10/21 = 0.290 m

The density of water is  $1,000 \text{ kg/m}^3$ .

$$u_o = \frac{m_c}{PD_s\rho_c \left(1 - \frac{D_o}{p_T}\right)} = \frac{306,000 \text{ kg/hr}}{(0.29 \text{ m})(0.590 \text{ m})(1,000 \text{ kg}/m^3) \left(1 - \frac{0.625 \text{ in}}{0.812 \text{ in}}\right)} \frac{\text{hr}}{3,600 \text{ s}} = 2.15 \frac{\text{m}}{\text{s}}$$

This is above the maximum velocity, try 18 baffles:

$$u_o = 2.15(18 + 1)/(20 + 1) = 1.95 \text{ m/s}$$

$$P = 6.10/19 = 0.321 \text{ m}$$

9. Calculate the heat transfer area and the design coefficient.

$$A_o = N_T \pi D_o L = (1072) \pi (0.625/39.4) (6.10) = 326 \text{ m}^2$$

$$U_{oD} = \frac{q}{A_o \Delta T_M} = \frac{12.8 \times 10^6 \text{kJ/hr}}{(326 \text{m}^2)(29.2^\circ\text{C})} = 1,344 \frac{\text{kJ}}{\text{hr} * \text{m}^{2} \circ\text{C}}$$

10. Calculate the clean over-all coefficient from the film coefficients. The physical properties were determined in Example 6.

Aniline: 
$$\mu_h$$
 = 3.28 kg/m-hr  $k_h$  = 0.622 kJ/hr-m<sup>2</sup>-°C  
 $N_{Prh}$  = 12.1

Water:  $\mu_c = 2.6 \text{ kg/m-hr}$   $k_c = 2.24 \text{ kJ/hr-m}^2-^{\circ}\text{C}$  $N_{Prh} = 4.86$ 

Tube side:

$$\begin{split} N_{Rei} &= \frac{D_i \rho_h u_i}{\mu_h} = \frac{(0.459in)(1.022kg/m^3)(1.044m/s)}{(\frac{3.28kg}{m*hr})} \frac{m}{39.4in} \frac{3.600s}{hr} = 13,700 \\ h_i &= 0.023 \frac{0.622kJ}{hr*m^{2}c} \frac{39.4in/m}{0.459in} (13,700)^{0.8} (12.1)^{1/3} = 5,750 \frac{kJ}{hr*m^{2}c} \\ \text{Shell side } (f_b &= 0.196 \text{ for }^{3}4 \text{ baffle cut}): \\ S_b &= f_b \frac{\pi}{4} \left( D_s^2 - \frac{N_T}{2} D_o^2 \right) = 0.196 \frac{\pi}{4} \left( 0.590^2 - \frac{1.072}{2} \left( \frac{0.625in}{39.4in/m} \right)^2 \right) = 0.0329 \text{ m}^2 \\ S_c &= PD_s \left( 1 - \frac{D_o}{p_T} \right) = (0.321m)(0.590m) \left( 1 - \frac{0.6251in}{0.812in} \right) = 0.0437 \text{ m}^2 \\ \text{Mean mass velocity: } G_e &= \frac{m_c}{\sqrt{s_b S_c}} = \frac{306,000 \text{ kg/hr}}{\sqrt{(0.0329m^2)(0.0437m^2)}} = 8.06 \times 10^6 \frac{\text{kg}}{\text{hr*m}^2} \\ N_{Reo} &= \frac{D_o G_e}{\mu_c} = \frac{(0.625in)(8.06 \times 10^6 \text{ kg/hr*m}^2)}{(\frac{2.60kg}{m*hr})} \frac{m}{39.4in} = 49,100 \\ h_o &= 0.2 \frac{2.24kJ}{hr*m^{2}c} \frac{39.4in/m}{0.625in} (49,100)^{0.6} (4.86)^{1/3} = 31,200 \frac{\text{kJ}}{\text{hr*m}^{2}c} \\ \text{Tube wall: } D_M &= \frac{D_o - D_i}{\ln \left(\frac{D_o}{D_i}\right)} = \frac{0.625in - 0.459in}{\ln \left(\frac{0.625in}{0.459in}\right)} = 0.538 \text{ in} \\ \text{McCabe, Smith & Harriott, 7^{th} ed., Appendix 6: \\ k_W &= 26 \text{ Btu/ft-hr-}^{\circ} \text{F} = 162 \text{ kJ/m-hr-}^{\circ} \text{C} \\ \end{split}$$

Resistances:

Wall: 
$$r_w = \frac{D_o}{D_M} \frac{x_w}{k_w} = \frac{0.625 \text{in}}{0.538 \text{in}} \frac{0.083 \text{in}}{39.4 \text{in}/\text{m}} \frac{\text{m} \cdot \text{hr} \cdot \text{s}^\circ \text{C}}{162 \text{kJ}} = 15 \times 10^{-6} \frac{\text{hr} \cdot \text{m}^2 \cdot \text{s}^\circ \text{C}}{\text{kJ}}$$

Tube side: 
$$r_i = \frac{D_o}{D_i} \frac{1}{h_i} = \frac{0.625 \text{in}}{0.459 \text{in}} \frac{\text{m}^2 \cdot \text{hr} \cdot \text{°C}}{5.750 \text{kJ}} = 237 \times 10^{-6} \frac{\text{hr} \cdot \text{m}^2 \cdot \text{°C}}{\text{kJ}}$$

Shell side: 
$$r_o = \frac{1}{h_o} = \frac{m^2 * hr * {}^{\circ}C}{31,200 kJ} = 32 \times 10^{-6} \frac{hr * m^2 * {}^{\circ}C}{kJ}$$

Clean coefficient:  $U_{oC} = \frac{1}{r_i + r_w + r_o} = \frac{1}{(237 + 15 + 32) \times 10^{-6}} = 3,519 \frac{\text{kJ}}{\text{hr} \cdot \text{m}^2 \cdot \text{cC}}$ 

11. Check allowance for fouling:

$$ff = \frac{1}{U_{oD}} - \frac{1}{U_{oC}} = \frac{1}{1,344} - \frac{1}{3,519} = 460 \times 10^{-6} \frac{\text{hr} * \text{m}^2 * ^{\circ}\text{C}}{\text{kJ}}$$

The allowance is much higher than the typical value of  $100 \times 10^{-6}$  hr-m<sup>2</sup>-°C/kJ. So the exchanger is oversized. Probably two 19.25-in shells with 352 tubes each would be a better design.

## **Summary**

These notes have presented the design of heat exchangers. The design of a heat exchanger consists of determining the area required to transfer heat which depends on

- The heat duty, determined from the enthalpy balance on the exchanger
- The mean temperature difference which is highly dependent on the temperature profile which is a plot of the temperatures versus the heat transferred
- The over-all heat transfer coefficient, dependent on the resistances of the films on both sides of the heat transfer wall and the resistance of the wall.

A quick exchanger design procedure and two detailed design procedures have been presented, one for a double-pipe heat exchanger and one for a shell and tube heat exchanger. The simulation of exchangers and estimation of their cost has also been presented.

## **Review Questions**

- 1. Why are enthalpy balances performed on heat exchangers?
- 2. What is the objective of sizing a heat exchanger?
- 3. Define the over-all heat transfer coefficient.
- 4. Define the film coefficients of heat transfer.
- 5. Which are the three major resistances to heat transfer in a heat exchanger?
- 6. What is *parallel flow* in a heat exchanger? What is *countercurrent flow*?

- 7. When is the logarithmic mean of the two temperature differences at the end of the exchanger the mean temperature difference?
- 8. Under what conditions are the temperature profiles linear? Under what conditions is the profile constant?
- 9. Which is the controlling resistance in a heat exchanger?
- 10. Cite the three regimes of boiling a liquid? Which one is the most efficient for heat transfer?
- 11. Define the equivalent diameter of a conduit.
- 12. Describe a shell and tube heat exchanger. What is the major advantage? What are the baffles?
- 13. Why use multiple tube passes in a shell and tube heat exchanger?
- 14. How is the number of tube passes determined?
- 15. How is the number of baffles determined?
- 16. When are multiple shell passes required?
- 17. What is the fouling resistance? How is it determined?

# **Problems**

- 1. Quick Design of a Cross-Exchanger. A double pipe heat exchanger is to be used to cool nitrobenzene flowing at 1,200 kg/hr from 80°C to 40°C by cross-exchanging it with aniline flowing at 4,800 kg/hr end entering the exchanger at 30°C. Using the resistances from Table 1, determine the length of the exchanger for (a) countercurrent flow, and (b) parallel flow. You may neglect the resistance of the wall and the fouling resistance.
- 2. Detailed Design of Double-pipe Heat Exchanger. Design the cross exchanger of Problem 1 if the exchanger consists of

<sup>34</sup>-in Sch. 40 steel pipe surrounded by a 1<sup>1</sup>/<sub>2</sub>-in Sch. 40 pipe. Calculate the inside and outside film coefficients and include the wall. Then use these values to determine the required length of the exchanger for both countercurrent and parallel flow.

- 3. Design of a Process Heater. A shell-and-tube exchanger consists of a 194-in. ID shell with 1-in. OD 16 BWG tubes, 20 ft long, in a  $1\frac{1}{4}$ -in. triangular pitch. It is to be used to heat 80,000 lb/h of benzene (sp.gr. 0.879) from 75°F to 245°F using saturated steam at 20 psig. The benzene flows inside the tubes and the steam condenses outside the tubes. There are eight tube passes and the exchanger is horizontal. Determine the steam consumption rate, and the available fouling resistance. Neglect the correction for the viscosity at the wall for benzene, and assume the condensing film temperature is essentially that of the condensing steam. Note: from TEMA tube count tables, the exchanger contains 128 tubes.
- 4. Quick Design of a Process Vaporizer. An exchanger is to be designed to heat and vaporize a process fluid at the rate of 22,000 kg/h using saturated steam at 10 psig. The process fluid enters as a liquid at 20°C and is heated to its boiling point of 75°C, and then it is completely vaporized at a constant temperature of 75°C. The liquid has a specific heat of 0.455 cal/gm-°C and its latent heat at 75°C is 625 cal/gm. Determine the steam consumption rate and the heat transfer area required to heat the liquid and to vaporize it. Use the resistances from Table 1.
- 5. Rating of a cross-exchanger. A shell-and-tube heat exchanger consisting of a 25-in ID shell with 422 <sup>34</sup>-in OD by BWG 16 tubes on a 1-in triangular pitch is being

considered to cross exchange two process streams. One of the streams is the feed to a reactor and consists of 200,000 lb/h of oil with a sp.gr. of 0.79, a specific heat of 0.567 BTU/lb-°F, a thermal conductivity of 0.15 BTU/hrft-°F, and a viscosity of 0.7 cP. This stream is to be heated from 150°F to 450°F and flows in the shell side of the exchanger. The other stream is the outlet from the reactor, enters at 850°F, and has essentially the same mass flow and properties as the inlet stream. The exchanger tubes are 24 ft long and are arranged in four tube passes. There is one shell pass with a baffle pitch of 6 inches. Determine if the exchanger is suitable for this service. The required allowance for fouling resistance is 0.006 hrft<sup>2</sup>-°F/BTU. You may neglect the correction for the viscosity at the wall for both fluids.

6. Design of a Process Vaporizer. An exchanger is to be designed to vaporize and superheat a process fluid at the rate of 42,000 kg/h. The process fluid enters as a liquid at its boiling point of 55°C and is completely vaporized at a constant temperature of 55°C, then it is superheated as much as possible with saturated steam at 30 psig and an approach temperature of 6°C. The vapor has a specific heat of 0.345 kcal/kg-°C and the latent heat of vaporization at 125°C is 525 kcal/kg. Determine the steam consumption rate and the heat transfer area required to vaporize the liquid and superheat the vapors. You may assume the outside film coefficient for condensing steam is 7,500 W/m<sup>2</sup>-°C, that the film coefficient of heat transfer for heating the vapor is 350 W/m<sup>2</sup>-°C, and for vaporization is 2,800 W/m<sup>2</sup>-°C. Neglect the resistance of the tube wall and assume that the film

coefficients given include fouling and the diameter ratio correction.

7. Quick Design of a Process Cooler. Aniline flowing at 28,000 kg/hr is to be cooled from 100°C to as low a temperature as possible using cooling water entering at 20°C in a shell-and-tube heat exchanger. The minimum allowed approach temperature is 5°C and the maximum allowed temperature rise of the water is 10°C. The tubes are 1-in. 16 BWG, 8 m long. Determine the number of tubes required if the flow is countercurrent, and the required flow of cooling water. Use the resistances from Table 1.





8. Air Cooler. The figure depicts a cooling system for three identical *heat loads* of 5 million Btu/hr each. Water is circulated through the loads and through an *air cooler;* the water absorbs the heat from the heat loads and releases it

to the air in the cooler. Heat losses in the tank and the lines may be neglected. The pump can deliver a constant flow of water of 2,800 gpm (gallons per minute) and the air cooler has an effective heat transfer area of 12,000  $ft^2$ . The supply temperature to the heat loads, *Tw*, is controlled by the valve that by-passes water around the air cooler. Although the air cooler was designed to remove the heat supplied by all three loads, because of fouling and damage to the fins in the exchanger, its capacity to cool has been reduced.

With only two heat loads in operation and half the total water flowing through each load--you find the following conditions:

- Temperature of the water to the heat loads:  $T_w = 145^{\circ}F$
- Temperature of the water out of the air cooler:  $T_{hb} = 138^{\circ}F$
- Average temperature of the air in the air cooler:  $T_c = 115^{\circ}F$
- a) Determine, for these conditions, the over-all heat transfer coefficient for the air cooler. Determine also what fraction of the water must be by-passed around the air cooler for these conditions to apply.
- b) Assuming the over-all heat transfer coefficient of the air cooler remains the same--because it is controlled by the air-side film coefficient--and that the average air temperature also remains the same, determine how to change the other variables to be able to handle all three heat loads, with one third of the water flowing through each load.

- 9. Quick Design of a Condenser. A single-pass countercurrent exchanger is to be designed to cool and completely condense a vapor stream at the rate of 15,000 kg/h using cooling water entering at 30°C and exiting at 40°C. The process stream enters as a superheated vapor at 250°C and is cooled to its saturation temperature of 50°C and then completely condensed at a constant temperature of 50°C. The vapor has a specific heat of 0.385 kcal/kg-°C and its latent heat at 50°C is 385 kcal/kg. Determine the heat transfer area required to cool the vapor and the area required to condense it and the required total flow of cooling water. Use the resistances of Table 1.
- 10. Design of a Double-pipe Cooler. An exchanger consisting of a <sup>3</sup>4-in Sch. 40 steel pipe inside a 1<sup>1</sup>/<sub>2</sub>-in schedule 40 pipe is to be used to cool 2,000 kg/h of heptane (sp.gr. 0.684) from 85°C to 25°C using cooling water entering at 20°C and exiting at 30°C. The flow is countercurrent with the heptane flowing inside the inner pipe and the water flowing in the annulus between the two pipes. Determine the required length of the exchanger. You may neglect the correction for the viscosity at the wall for both fluids.