Chapter III

THEORY

3.1 Convection Heat Transfer

Convection is the most important and also the most difficult of the modes of transport which comprise the field of heat transfer. The classifications used in organizing the field of convection result from, and are controlled by fluid mechanics considerations. There are two fundamental subdivisions. When the fluid motion results from a forcing condition, such as fluid motion forced by a pump or fan, the process is called forced convection. When the fluid motion results from buoyancy effects caused by a temperature-created density difference in the fluid, as in the region around an object cooling in air or in a pan of water being heated on a stove the process is called natural convection.

The heat transfer rate of convection processes is convenient to relate to the temperature difference causing the heat flow. For a surface of area A, a convection coefficient h, is defined as follows:

$$q = h A \Delta t \qquad \text{.............(3-1)}$$

Where Δt is the temperature difference for the convection processes. In English units the convection coefficient is usually expressed in terms of Btu/hr ft^2 ^oF.

3.2 Heat Transfer Coefficients

3.2.1 Individual heat transfer coefficients

In the field of heat transfer, it is not convenient to measure the thickness of the fluid film or the temperature at the interface between the film and the main body of the fluid, and since both conduction and convection are involved, the differential rate of heat flow between fluid and solid is calculated from the equation.

$$dq = h \, dA \, \Delta t \qquad \text{.........(3-2)}$$

and the observed value of h is called the individual coefficient, "film coefficient", or surface coefficient, and includes the thermal resistances of the laminar film, "buffer" layer between film and core, and turbulent core. The coefficient h is determined by dividing the known rate of heat flow per unit surface of the wall by the difference between the temperatures of the fluid and surface. The reciprocal of h is called the resistance.

3.2.2 Over all heat transfer coefficients

In testing commercial heat transfer equipment, it is not convenient to measure tube temperatures, and hence the oven all performance is expressed as an over all coefficient of heat transfer U based on a convenient area dA

which may be dA_i, dA_o or an average of dA_i and dA_o; whence, by definition,

$$dq = U\, dA\, t \qquad ....(3\text{-}3)$$

U is called the "over all coefficient of heat transfer", or merely "over all coefficient". The reciprocal of U is called over all resistances. Here A_i is the inner surface area of the tube and A_o is the outer surface area of the tube.

3.3 The Effects of Convective Heat Transfer Coefficients

3.3.1 Resistances to heat transfer

In the heat transfer processes, the effect of the resistances is to cause a significant decrease in the heat transfer rate. The major resistances consist of the resistance of fluid layers and surfaces.

Generally, the relation between the over all resistance and the over all coefficient is expressed as

$$U = \frac{1}{R} \qquad ....(3\text{-}4)$$

The resistance of surfaces is the sum of the resistances of various components of a barrier. The resistance of a conduction layer is $\frac{x}{k}$ and of a surface process is $1/h$. If the resistances are reduced the heat transfer rate will be higher.

Another resistance is the boundary film or the thermal boundary layer. For a heat transfer system which heat is transfered between a fluid and the wall of a pipe in which it is flowing; if the fluid enters at a uniform temperature t_o and the pipe wall is at some lower temperature t_s, the development of a thermal boundary layer is as depicted in Figure 3.1

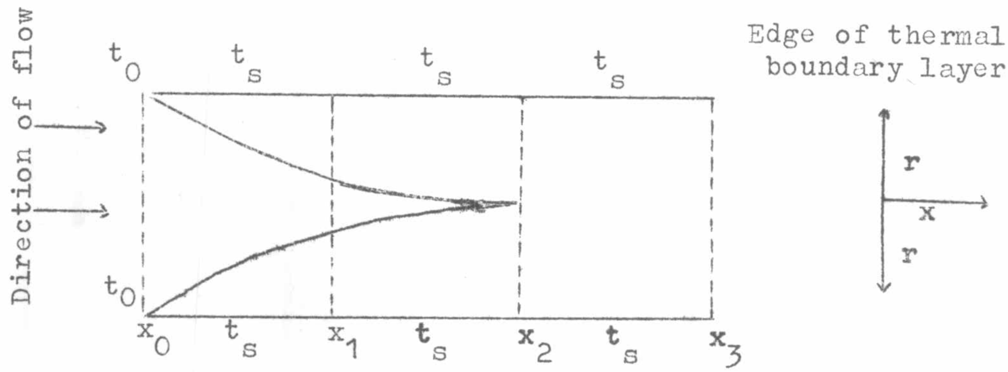


Fig 3.1 Development of a thermal boundary layer for flow in a pipe

The thickness of thermal boundary layer increases with the increase of the distance from the pipe inlet. The layer finally comes together at the center of the pipe. The axial distance from the pipe inlet to this point x_2 is known as the thermal-entrance length.

Beyond this point the temperature profile becomes flatter; if the pipe is long enough, the profile becomes completely flat at a uniform temperature t_s.

The boundary layer thickness results from convection effects and conduction effects. The classification of the boundary layers as laminar boundary layer and turbulent boundary layer is as follows:

i) The fluid enters the pipe with a uniform velocity profile at a rate such that $Re < 2100$. Under these conditions a laminar boundary layer will build up, starting from the leading edge, until it fills the pipe at some distance down-stream. The heat transfer coefficient will be infinite at the inlet but will continue to diminish even after the point of developed flow has been reached.

ii) The fluid enters the pipe in laminar flow with a uniform velocity profile at rate such $Re > 2100$. At the leading edge a laminar boundary layer develop which, at a critical distance, breaks down into a turbulent boundary layer. This turbulent boundary layer increases in thickness with increasing down stream distance until it fills the pipe with a turbulent core and a laminar sublayer at the wall. Downstream from this point the system is identical in all respects with the system that would develop if the flow had been turbulent from the entrance. This flow behavior is reflected in the values of the local heat transfer coefficients which decrease from infinity at the inlet to some minimum value at the critical point where the laminar boundary layer changes into a turbulent boundary layer. Near this point the heat transfer coefficient increases in magnitude for a short distance.

In the turbulent transport, the velocity distribution will form turbulent diffusivities that reduce the thickness of film which is the major resistance in convection, so the heat transfer rate increases.

3.3.2 The temperature differences

From the general equation of convective heat transfer, the heat transfer rate can also be increased by increasing the temperature gradient between the bulk fluid and surface temperatures.

Conventionally, the total heat transfer rate of an exchanger is equal to the product of U, the total exchange area A and a temperature difference Δt,

$$q = U A \Delta t \qquad ....(3\text{-}5)$$

The relation between the exchanger mean temperature

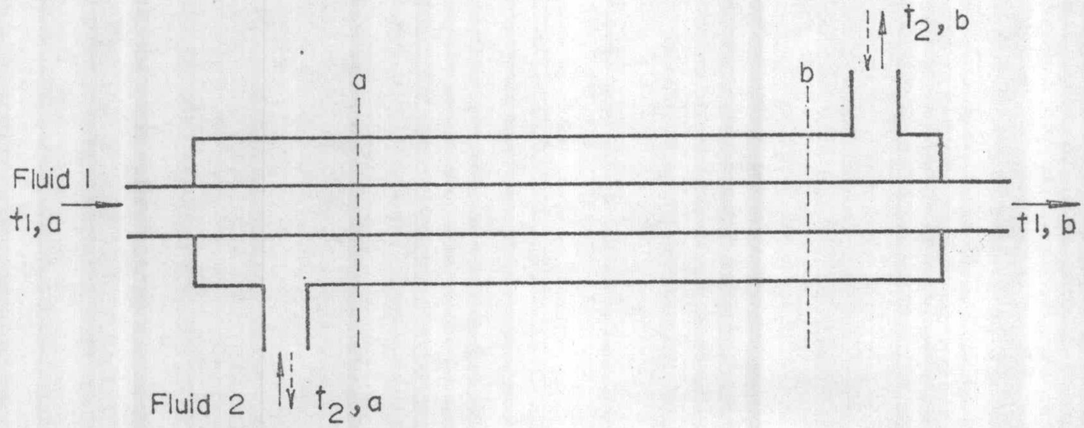


Fig. 3.2 a) Double pipe heat exchanger.

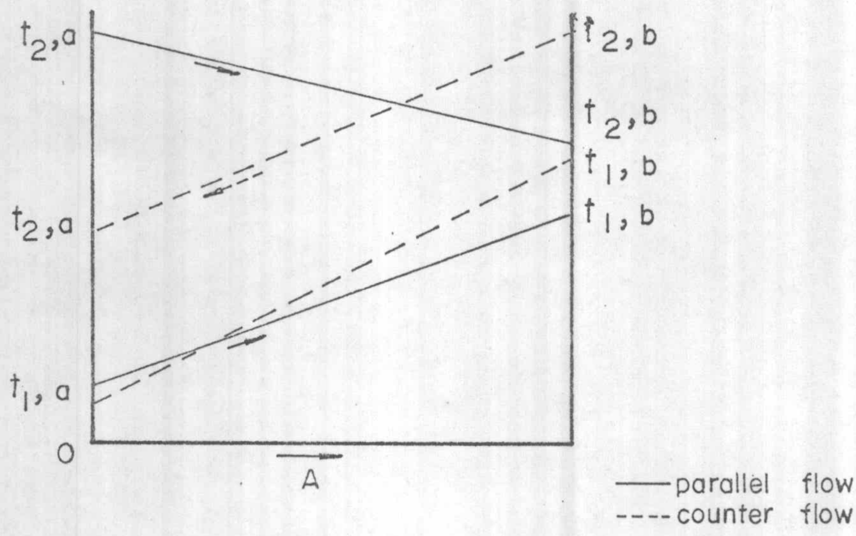


Fig. 3.2 b) Temperature difference along a double pipe H.E.

difference, and the fluid temperature distributions is considered for several important arrangements. There are two typical cases; the fluids may flow in the same directiion, parallel flow; or in opposite directions, counter flow.

3.4 The Effects of Controlled Cycling on Heat Transfer Coefficients

3.4.1 Resistances to heat transfer

The major resistances in heat transfer processes usually are those of the boundary layers. The controlled cycling operation is expected to reduce the thickness of boundary layer by forming eddy motions in the film. The bulk of the liquid is perfectly mixed by its composite turbulent movement. The resistance to heat flow is located in a thin layer close to the wall.

Controlled cycling heat transfer is an unsteady state process. Its model is similar to the heat transfer in a scraped heat exchanger, in which the liquid is scraped from the surface and renewed alternately. So, the resistance to heat transfer of controlled-cycling will be lower than convdntional one.

3.4.2 The temperature differences

Heat transfer between a solid surface and a fluid phase is generally expressed by equation (3-1)

$$q = h A \Delta t = h A (t_s - t_b) \qquad \text{..........(3-1)}$$

Where t_s is the surface temperature and t_b is the bulk temperature of fluid. For the application at a point on the surface, equation (3-2) is used:

$$dq = h\ A\ dt = h(t_s - t_b)\ dA \qquad \text{..........(3-2)}$$

Because the quantities h, $t_s - t_b$ and dq can often be expressed as functions of temperature, the integral form of Eq.(3-2) is written as:

$$\int_o^q \frac{dq}{h(t_s - t_b)} = \int_o^A dA \qquad \text{...(3-6)}$$

According to the Fourier's law, the heat transfer at the interface of the system shown in Figure 3.3, is expressed as:

$$q = k\ A\ \left(\frac{dt}{dy}\right)_{y = o} \qquad \text{......(3-7)}$$

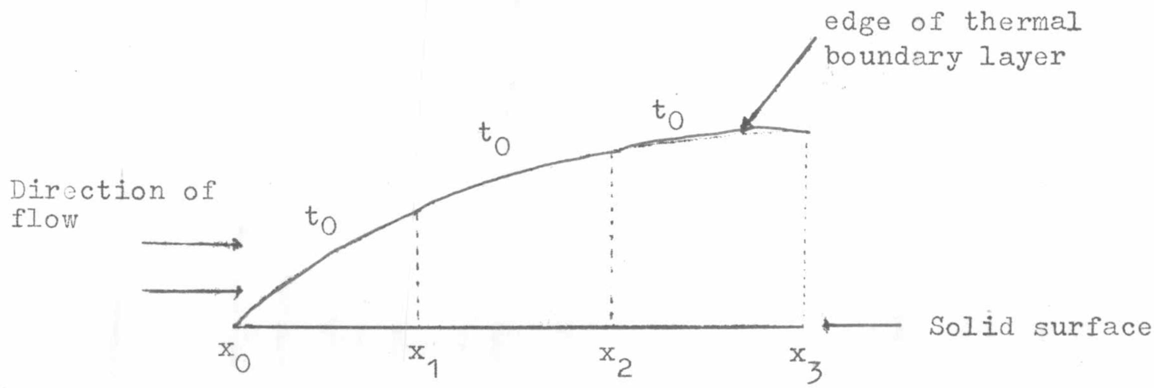


Fig 3.3 Development of a thermal boundary layer

In considering conduction at the interface with known k for the fluid, $(\frac{dt}{dy})_{y=o}$ is unknown. Furthermore the flowing fluid at values of $y > o$, energy is also being carried by convection transport rather than the simple co conduction. It is not practical to evaluate heat transfer rate q using equation (3-1). Hence, equation (3-1) is generally used in calculation q.

In differential form, eq (3-7) is written as

$$dq = k \, d A \, (\frac{dt}{dy})_{y=o}$$

$$= h \, (t_o - t_s) \, dA \qquad \text{.......(3-8)}$$

Equation (3-8) can be arranged to give

$$h = \frac{k}{t_o - t_s} \, (\frac{dt}{dy})_{y=o} \qquad \text{.......(3-9)}$$

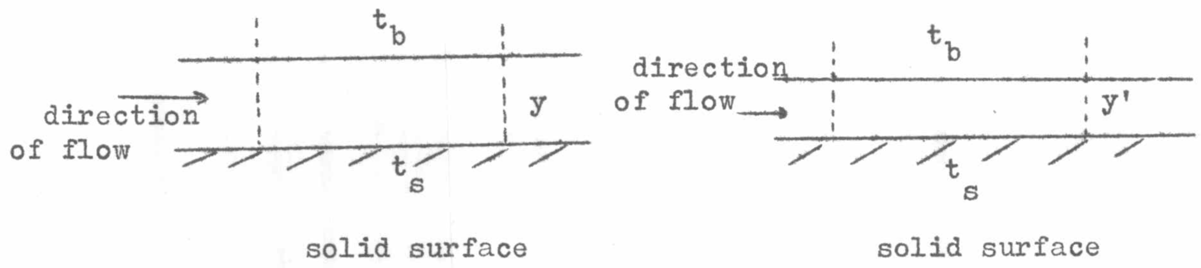
or in an equivalent form

$$h = k \left\{ \frac{d \left[t - t_s \right] / \left[t_o - t_s \right]}{dy} \right\}_{y=o} \qquad \text{..(3-10)}$$

Equation (3-10) is written in terms of the dimensionless unaccomplished temperature change from the surface to the edge of thermal boundary layer.

It is apparent from thes relations that any factor which would cause an increase in the temperature gradient at the wall, would also cause an increase in the value of the heat transfer coefficient.

When controlled cycling is applied to a two-phase heat transfer system, the boundary layer thickness is reduced, as compared to the conventional operation and the temperature gradient at the wall surface (y=0) is expected to be increased. Therefore the heat transfer coefficient should also be increased.



a) in a conventional manner b) in controlled cycling manner

Fig 3-4 Boundary Layer thickness