# EFFECT OF HEAT EXCHANGER TYPES ON HEAT TRANSFER RATE

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## ABSTRACT

This project aims to study the effect of heat exchanger types on heat transfer rate. There were many parameters to study. They were types of heat exchanger (monotube, shell and tube, and plate heat exchanger), flow rate (laminar and turbulent flow) and temperature, direction of flow (co-current and counter-current flow), and efficiency of each type of heat exchanger. It was found that a monotube heat exchanger under counter-current and laminar flow of higher temperature (warm water) and turbulent flow of lower temperature (ambient temperature or cold water) indicated the highest efficiency of heat transfer rate. When a heat exchanger was shielded with glass fiber insulator, the efficiency of heat transfer rate was less significant.

**โครงงานพิเศษ เรื่อง** บทบาทของชนิดเครื่องแลกเปลี่ยนความร้อนต่ออัตราการถ่ายเทความร้อน

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## บทคัดย่อ

โครงงานพิเศษเรื่องนี้ได้ทำการศึกษาบทบาทของชนิดเครื่องแลกเปลี่ยนความร้อนที่มีต่ออัตราการถ่ายเทความร้อน ในโครงงานนี้ได้ศึกษาตัวแปรต่างๆดังนี้ ชนิดของเครื่องแลกเปลี่ยนความร้อน (เครื่องแลก เปลี่ยนความร้อนแบบท่อสองชั้น เครื่องแลกเปลี่ยนความร้อนแบบเชลล์และท่อ และเครื่องแลกเปลี่ยนความร้อนแบบแผ่น) อัตราการไหล (การไหลแบบลามินาร์และการไหลแบบปั่นป่วน) และอุณหภูมิ ทิศทางของการไหล (การไหลแบบไปทางเดียวกันและการไหลแบบสวนทางกัน) และประสิทธิภาพการถ่ายเทความร้อนของเครื่องแลกเปลี่ยนความร้อนแต่ละชนิด หลังจากทำการทดลองพบว่า เครื่องแลกเปลี่ยนความร้อนแบบท่อสองชั้น ที่มีอัตราการไหลของน้ำร้อน (อุณหภูมิของน้ำสูงกว่าอุณหภูมิห้อง) เป็นแบบลามิน่าร์ และน้ำเย็น (อุณหภูมิของน้ำเท่ากับอุณหภูมิห้อง) เป็นแบบปั่นป่วน โดยที่ทิศทางของการไหลเป็นแบบสวนทางกันให้ประสิทธิภาพของการถ่ายเทความร้อนสูงสุด และเมื่อใช้ฉนวนกันความร้อนประเภทใยแก้วหุ้มส่วนต่างๆของเครื่องแลกเปลี่ยนความร้อน ที่สามารถเกิดการสูญเสียความร้อนกับอากาศรอบตัวเครื่อง พบว่ามีผลทำให้ประสิทธิภาพของการแลกเปลี่ยนความร้อนเพิ่มขึ้นเพียงเล็กน้อย

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Mr. Chairat Chadavipoo

Mr. Sutthavee Yacharn

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# CHAPTER 1
# INTRODUCTION

## 1.1 MOTIVATION

Heat exchanger is a broad term used in reference to design devices for exchanging heat. Heat transfer in systems is needed and more essential in small and big scale industrial factories. Most often the heat is transferred from one fluid to another. A fluid that is discharged from a useful process might contain high energy, and it may be desired to recover some of the energy that would ordinarily be discarded. One example of this is in remotely located plants that generate their own electricity via steam turbines.

Heat exchangers can be classified in a number of ways, depending on their construction or on how the fluids move relative to each other through the device. A double pipe heat exchanger consists of two concentric pipes or tubes. One fluid-the warmer one, for example flows through the inner pipe. Another fluid flows through the annulus. Due to a temperature difference between the fluids, heat is transferred. The fluid streams (in the pipe and in the annulus) could be traveling in the same direction (parallel flow) or in opposite directions (counterflow). A shell and tube exchanger consists of a huge outer cylinder (called the shell) within which are contained many tubes. Generally, the shell and tube exchanger can handle fluid flow rates that are many times as large as those in a double pipe exchanger. From the various types of heat exchanger and theirs functions lead to the investigation of their efficiency.

## 1.2 OBJECTIVE

This project is aimed to study effect of heat exchanger types on heat transfer rate. The study was investigated the flow characteristics variables and heat exchanger types including direction of flow and temperature.

## 1.3 SCOPE OF STUDY

i. To study type of heat exchanger (monotube, shell and tube, and plate heat exchanger).
ii. To study flow rate of water (laminar and turbulent) and water temperature.
iii. To study direction of flow (co-current and counter-current).

iv. To compare the efficiency of heat exchanger for each parameter.

v. To study the effect of heat transfer when shielded with fiber glass insulator.

## 1.4 EXPECTED RESULTS

i. To save energy when we use heat exchangers.

ii. To increase the efficiency of heat transfer rate by control temperature, the direction of flow and flow rate of water.

iii. To increase the efficiency of heat transfer rate by insulation technique.

# CHAPTER 2

# LITERATURE REVIEW AND THEORY

## 2.1 LITERATURE REVIEW

Shou-Shing Hsieh, Chihing-Tsung Liuahl and Anthony C. Ku (1987) [1] studied heat transfer coefficients of double pipe heat exchanger with helical type roughened surface this work presents experimental information for single-phase forced convection in double pipe heat exchanger containing a two-dimensional helical fin roughness on the outer surface of the inner tube. The objective is to investigate the effect of augmentation heat transfer on this kind of heat exchanger and to evaluate the possibility of replacing the classical double pipe heat exchanger from the economic viewpoint. The present study experiments with a helical angle ($\alpha$ = 65°), a pitch – to – height ratio (p/e = 1.45), and three aspect ratios (shell side dia. To tube side dia.) of $D_0/D_i$ = 2.68, 3.48 and 5.1 and the corresponding ratios of roughness height to hydraulic dia. ($e/D_H$) of 0.192, 0.13 and 0.08, respectively.

B. Gay, N. V. Mackley and J.D. Jenkins (1976) [2] studied the application of an electrochemical mass-transfer modeling technique to the determination of local shell-side heat-transfer coefficients in a model of a baffled cylindrical shell-and-tube heat exchanger. The validity and accuracy of the electrochemical method are demonstrated by comparison with heat-transfer measurements and with mass-transfer data obtained using the mercury evaporation technique.

Reinhard Wurfel, Nikolai Ostrowski (2003) [3] studied heat transfer and pressure drop during the condensation process within plate heat exchangers of the herringbone-type, the application of compact heat exchangers to the realization of the processes with phase change gaseous – liquid is increasingly significant. However, the state of the knowledge is unsatisfactory for modeling the heat transfer and the pressure drop. Experimental investigations for condensation in channels of corrugated plates of a plate heat exchanger were carried out with the condensing vapors water and n-heptane. The loads of the vapor phase as well as the kind of the plate corrugation were used as experimental parameters. The results to the heat transfer coefficient and friction pressure drop refer to the condition of a complete condensation. An essential influence of the phase load as well as of the corrugation inclination angle is observed in the investigated range of the shear controlled two-phase flow. The intensity of heat transfer increases with a factor of approx. 3-4, compared to the calculation of the Nusselt theory of laminar film.

Jongmin Shin and Samchul Ha (2002) [4] studied on the behavior of water hold-up by condensation on various shapes of fin-and-tube heat exchangers with different surface hydrophilicity, i.e. dynamic contact angle of surface, was conducted. Condensation experiments were conducted, and the amount of water hold-up was measured. Condensation flow patterns on fins with different surface hydrophilicity were visualized. Results showed that the water hold-up of a heat exchanger could be reduced by the enhancement of the surface hydrophilicity and the design of a heat exchanger with a lower number of fins and fins with slant ends.

V.H. Marcos (1988) [5] studied of waste heat recovery shell-and-tube heat exchangers. The exchanger heat duty, overall heat transfer coefficient, effectiveness and tube side friction factor are investigated as functions of the tube surface geometry (plain or dimpled), the flow type ( counter or parallel) , the tube Reynolds number and the shell side heat capacity rate. Water and the exhaust gases of a Diesel engine are passed inside the tube and the shell, respectively.

The heat transfer characteristics increase with an increase in tube Reynolds number and the shell side heat capacity rate, for all the flow types and the surface geometries examined. The counter-flow, shell-and-dimpled-tube heat exchanger, compared with that exchanger having a plain tube, increases the heat duty and the overall heat transfer coefficient by 80%, and the heat exchanger effectiveness increases by 35%. For the parallel-flow, shell-and-dimpled heat exchanger, the heat duty, the overall heat transfer coefficient and the effectiveness increase by 30, 55, and 25%, respectively. At the same time the dimpled tube increases the tube side friction factor by 600% over that of the plain tube. The rate of waste heat recovered from the exhaust gases of the Diesel engine by the counter-flow, shell-and-dimpled-tube heat exchanger is equal to 10% of the maximum brake power of the engine running at 1500 rpm, and the tube Reynolds number equal to 8875.

The shell-and tube heat exchanger is the most widely used type of industrial heat transfer equipment. In order to carry out the thermal-hydraulic design of a shell-and-tube exchanger, pressure drop and heat transfer correlations (or tabulated data) must be available for both the tube side and the shell side. Initially, only plain tubes were used in shell-and-tube exchangers. However, as increasing incentives for more efficient heat exchangers, considerable emphasis has been placed on the development of various augmented, or enhanced, heat transfer surfaces. The use of enhanced surfaces allows the designer to increase the heat duty for a given exchanger, to reduce the size of the exchanger for a given heat duty, to reduce the pumping power, or to reduce the approach temperature difference.

## 2.2 FLUID FLOW PHENOMENA

The behavior of a flowing fluid depends strongly on whether the fluid is under the influence of solid boundaries. In the region where the influence of the wall is small, the shear stress may be negligible and the fluid behavior may approach that of an ideal fluid, one that is incompressible and has zero viscosity. The flow of such an ideal fluid is called potential flow and is completely described by the principles of Newtonian mechanics and conservation of mass. Potential flow has two important characteristics: (1) Neither circulations nor eddies can from within the stream, so that potential flow is also called irrotational flow; and (2) friction cannot develop, so that there is no dissipation of mechanical energy into heat.

Potential flow can exist at distances not far from a solid boundary. A fundamental principle of fluids mechanics, originally stated by Prandtl in 1904, is that, except for fluids moving at low velocities or possessing high viscosities, the effect of the solid boundary on the flow is confined to a layer of the fluid immediately adjacent to the solid wall. This layer is called the boundary layer, and shear and shear forces are confined to this part of the fluid. Outside the boundary layer, potential flow survives. Most technical flow processes are best studied by considering the fluid stream as two parts, the boundary layer and the remaining fluid. In some situations such as flow in a converging nozzle, the boundary layer may be neglected; and in others, such as flow through pipes, the boundary layer fills the entire channel, and there is no potential flow.

Within the current of an incompressible fluid under the influence of solid boundaries, four important effects appear: (1) the coupling of velocity-gradient and shear-stress fields, (2) the onset of turbulence, (3) the formation and growth of boundary layers, layers, and (4) the separation of boundary layers from contact with the solid boundary.

In the flow of compressible fluids past solid boundaries, additional effects appear, arising from the significant density changes that are characteristic of compressible fluids.

**Velocity field**

When a stream of fluid is flowing in bulk past a solid wall, the fluid adheres to the solid at the actual interface between solid and fluid. The adhesion is a result of the force fields at the boundary, which are also responsible for the interfacial tension between solid and fluid. If, therefore, the wall is at rest in the reference frame chosen for the solid-fluid system, the velocity of the fluid at the interface is zero. Since at distances away from the solid the velocity is not zero, there must be variations in velocity from point to point in the flowing stream. Therefore, the velocity at any point is a function of the space coordinates of that point, and a velocity field exists in the space occupied by the fluid. The velocity at a given location may also vary with time. When the velocity at each location is constant, the field is invariant with time and the flow is said to be steady.

**One-dimensional flow**

Velocity is a vector, and in general, the velocity at a point has three components, one for each space coordinate. In many simple situations all velocity vectors in the field are parallel or practically so, and only one velocity component, which may be taken as a scalar, is required. This situation, which obviously is much simpler than the general field, is called one dimensional flow; an example is steady flow through straight pipe. The following discussion is based on the assumptions of steady one-dimensional flow.

### 2.2.1 LAMINAR FLOW, SHEAR RATE, AND SHEAR STRESS

Laminar flow

At low velocities fluid tend to flow without lateral mixing, and adjacent layers slide past one another as playing cards do. There are neither cross-currents nor eddies. This regime is called laminar flow. At higher velocities turbulence appears and eddies from, which, as discussed later, lead to lateral mixing.

**Velocity gradient and of shear**

Consider the steady one-dimensional laminar flow of an incompressible fluid along a solid plane surface. Figure 2.1a shows the velocity profile for such a stream. The abscissa u is the velocity, and the ordinate y is the distance measured perpendicular from the wall and therefore at right angles to the direction of the velocity. At y=0, u=0, and u increases with distance from the wall but at a decreasing rate. Focus attention on the velocities on two nearby planes, plan A and plane B, a distance y apart. Let the velocities along the planes be u and u, respectively, and assume that u >u.Call u = u - u Define the velocity gradient at y, du / dy, by

$$\frac{du}{dy} = \lim_{\Delta y \to 0} \frac{\Delta u}{\Delta y} \tag{2.1}$$

The velocity gradient is clearly the reciprocal of the slope of the velocity profile of Fig.2.1a. The local velocity gradient is also called the shear rate, or time rate of shear. The velocity gradient is usually a function of position in the stream and there fore defines a field, as illustrated in Fig.2.1b.

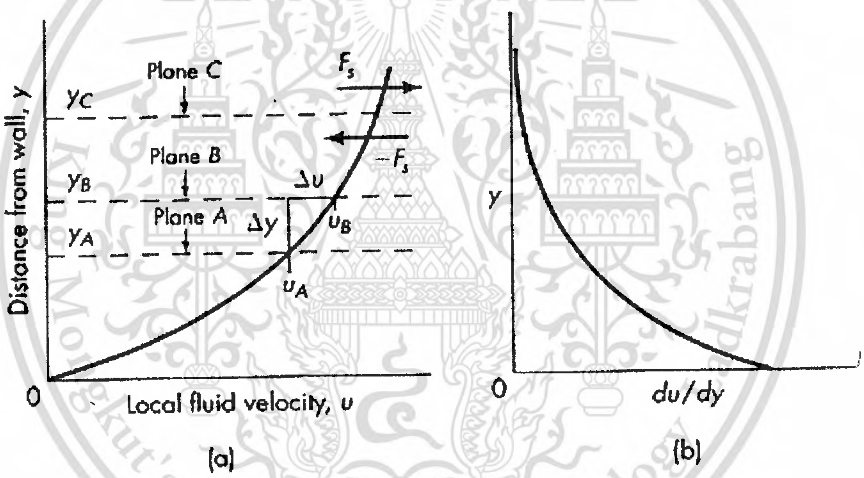



FIGURE 2.1 Profiles of velocity and velocity gradient in layer flow; (a) velocity; (b) velocity gradient or rate of shear. [6]

Shear-stress field

Since an actual resists shear, a shear force must exist wherever there is a time rate of shear. In example, at plane C at distance y from the wall, the shear force F, shown in Fig. 2.1a, acts in the direction shown in the figure. This force is exerted by the fluid above plane C on thew fluid between plane C and the wall. By Newton's third law, an equal and opposite force- F acts on the fluid above plane C from the fluid below plane C. It is convenient to use, not total force F , but the force per unit area of the shearing plane, called the shear stress and denoted by $\tau$, or

$$\tau = \frac{F_s}{A_s} \tag{2.2}$$

Where A is the area of the plane. Since $\tau$ varies with y, the shear stress also constitutes a field. Shear forces are generated in both laminar and turbulent flow. The shear stress arising from viscous or laminar flow is denoted by $\tau_V$.

## 2.2.2 RHEOLOGICAL PROPERTIES OF FLUIDS

**Newtonian and non-newtonian fluids**

The relationships between the shear stress and shear rate in a real fluid are part of the science of rheology. Figure 2.2 shows several examples of the rheological

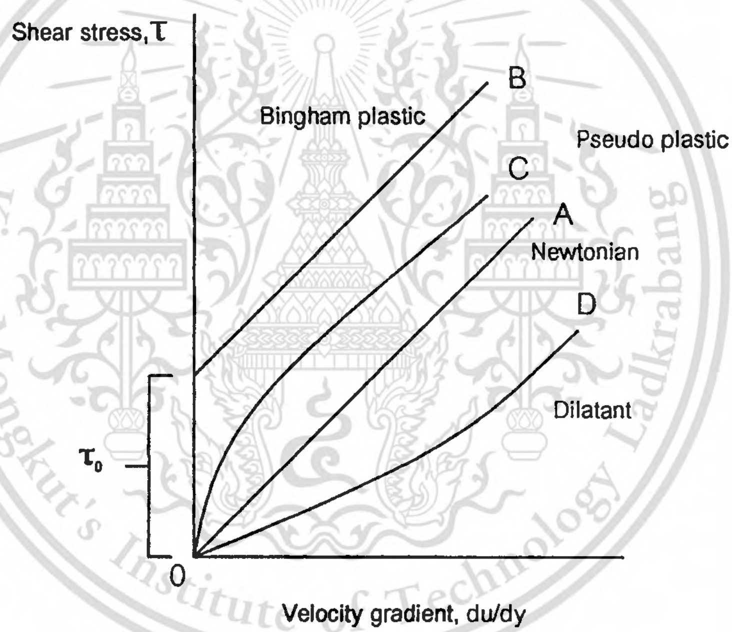



**FIGURE 2.2** Shear stress versus velocity gradient for Newtonian and non-Newtonian fluid behavior of fluids.[6]

The curves are plots of shear stress versus rate of shear and apply at constant temperature and pressure. The simplest behavior is that shown by curve A, which is a straight line passing through the origin. Fluids following this simple linearity are called Newtonian fluids. Gases and most liquids are Newtonian. The other curves in Fig. 2.2 represent the rheological behavior of liquids called non-Newtonian. Some liquids, for example, sewage sludge, do not flow at all until a threshold shear stress, denoted by $\tau_0$ , is attained and then flow linearly, or nearly so, at shear stresses greater than $\tau_0$. Curve B is an example of this relation. Liquids acting this way are called Bingham plastics. Line C represents a pseudoplastic fluid. The curve passes through the origin, is concave downward at low shear, and becomes nearly linear at high shear. Rubber latex is an example of such a fluid. Curve D represents a dilatants fluid. The curve is concaving upward at low shear and almost linear at high shear. Quicksand and some sand-filled emulsions show this behavior. Pseudoplastics are said to be shear rate-thinning and dilatants fluids shear rate-thickening.

Time-dependent flow

None of the curves in Fig. 2.2 depends on the history of the fluid, and a given sample of material shows the same behavior no matter how long the shearing stress has been applied. Such is not the case for some non-Newtonian liquids, whose curves of stress versus rate of shear depend on how long the shear has been active. Thixotropic liquids break down under continued shear and on mixing give lower shear stress for a given shear rate; that is their apparent viscosity decreases with time.

TABLE 2.1 Rheological characteristics of fluids [6]

| Designation | Effect of increasing shear rate | Time-dependent? | Examples[2] |
|---|---|---|---|
| Pseudoplastic | Thins | No | Polymer solutions, starch suspensions, mayonnaise, paints |
| Thixotropic | Thins | Yes | Some polymer solutions, shortening, some paints |
| Newtonian | None | No | Gases, most simple liquids |
| Dilatant | Thickens | No | Corn flour–sugar solutions, wet beach sand, starch in water |
| Rheopectic | Thickens | Yes | Bentonite clay suspensions, gypsum suspensions |

Rheopectic substances behave in the reverse manner, and the shear stress increases with time, as does the apparent viscosity. The original structures and apparent viscosities are usually recovered on standing.

The rheological characteristics of fluids are summarized in Table 2.1.

Viscoelastic fluids

Viscoelastic fluids show both viscous and elastic properties. They exhibit elastic recovery from deformations that occur during flow, but usually only part of the deformation is recovered upon removal of the stress. Examples of viscoelastic fluids are flour dough, napalm and certain polymer melts.

Viscosity

In a Newtonian fluid, the shear stress is proportional to the shear rate, and the proportionality constant is called the viscosity

$$\tau_v = \mu \frac{du}{dy} \tag{2.3}$$

In SI units $\tau_v$ is measured in newtons per square meter and $\mu$ in kilograms permeter-second or pascal-second. In the cgs system, viscosity is expressed in grams per centimeter-second, and this unit is called the poise (P). Viscosity data are generally reported in millipascal-seconds or in centipoises (cP=0.01=1 mPa.s), since most fluids have viscosities much less than 1 pascal-second.

In fps units, viscosity is defined using Newton's law conversion factor $g_c$ , and the units of  are pounds per foot-second or pounds per foot-hour. The defining equation is

$$\tau_v = \frac{\mu}{g_c} \frac{du}{dy} \tag{2.4}$$

Conversion factors among the different systems are given in Table 2.2.

TABLE 2.2 Conversion factors for viscosity [6]

**Conversion factors for viscosity**

| Pa · s | P | cP | lb/ft · s | lb/ft · h |
|---|---|---|---|---|
| 1 | 10 | 1,000 | 0.672 | 2,420 |
| 0.1 | 1 | 100 | 0.0672 | 242 |
| $10^{-3}$ | 0.01 | 1 | $6.72 \times 10^{-4}$ | 2.42 |

Viscosity and momentum flux

Although Eq. (2.3) serves to define to define the viscosity of a fluid, it can be interpreted in terms of momentum flux. The moving fluid a short distance above the wall possesses some momentum, whereas the fluid immediately adjacent to the wall, where the velocity is zero, has none. The moving fluid must therefore acquire momentum from the faster-moving layer above it. Which in turn receives momentum from the next layer up, and so on. Each layer is, in effect, dragged along by the layer above it. In this way x-direction momentum is transferred in the y direction all the way to the wall, where u=0.Since the wall does not move, the momentum is delivered to the wall as a shear force known as wall shear. The shear stress (shear force per unit area) at the wall is denoted by $\tau_w$.

Momentum is thus transferred from a region of high fluid velocity to one of low velocity, much as heat flows from a region of high temperature to one at a lower temperature. The rate of momentum transfer per unit area, or momentum flux, is governed by the velocity gradient du/dy. Equation (2.3) therefore states that the momentum flux normal to the direction of fluid flow is proportional to the velocity gradient, with the viscosity as the proportionality factor. The velocity gradient may be thought of as the "driving force" for momentum transfer. The units of momentum flux are $kg/m.s^2$, the same as the units for $\tau$, since 1N/m equals 1 $kg/m.s^2$.

Momentum transfer is analogous to conductive heat transfer resulting from a temperature gradient, where the proportionality factor between the heat flux and temperature gradient is called the thermal conductivity. This is shown by Fourier's law. In laminar flow, momentum is transferred by viscous action as a result of the velocity gradient, and the viscosity may be regarded as the conductivity of momentum transferred by this mechanism. Momentum transfer is also analogous to the transfer of material by molecular diffusion, where the proportionality factor is the diffusivity of mass. This is summarized in Fick's law.

Viscosites of gases and liquids

The viscosity of a newtonian fluid depends primarily on the temperature and molecular structure and to a minor extent on pressure, except at very high pressures. Gas viscosities at room temperature are generally between 0.005 and 0.02 cP. There is no simple correlation with molecular weight. At 20$^0$C the viscosity is 0.0018 cP for air, 0.014 cP for carbon dioxide, 0.007cP for benzene vapor, and 0.009 cP for hydrogen. Gas viscosities increase with temperature, as predicted by kinetic theory. For approximate calculations the effect of temperature can be estimated by using the exponential equation

$$\frac{\mu}{\mu_0} = \left(\frac{T}{273}\right)^n \qquad (2.5)$$

Where $\mu$ = viscosity at absolute temperature T, K

$\mu_0$ = viscosity at $0^\circ C$ (273 K)

$\eta$ = constant for a particular gas

Exponent n ≈ 0.65 for air; it is approximately 0.9 for carbon dioxide, 0.8 for butane, and 1.0 for steam.

The viscosity of a gas is almost independent of pressure in the region where the ideal gas laws apply. At very high pressures the viscosity increases with pressure, especially in the neighborhood of the critical point.

The viscosities of liquids are generally much greater than those of gases and cover several orders of magnitude. The viscosity usually increases with molecular weight and decreases significantly when the temperature is raised. For example, the viscosity of water falls from 1.79 cP at $0^\circ C$ to 0.28 cP at $100^\circ C$. The viscosity of a liquid increases with pressure, but the effect is generally insignificant at pressures less than 40 atm.

The absolute viscosities of liquids vary over an enormous range of magnitudes, from about 0.1 cP for liquids near their boiling point to as much as $10^6$ P for polymer melts. Most extremely viscous materials are non-newtonian and possess no single viscosity independent of shear rate.

Kinematic viscosity The ratio of the absolute viscosity to the density of a fluid $\mu / \rho$ is often useful. This property is called the kinematic viscosity and designated by $\nu$. In SI, the unit for $\nu$ is square meters per second. In the cgs system, the kinematic viscosity is called the stoke (St), defined as 1 $cm^2$ /s. The fps unit is square feet per second. Conversion factors are

$$1\ m^2/s = 10^4\ St = 10.7639\ ft^2/s$$

For liquids, kinematic viscosities vary with temperature over a somewhat narrower range than absolute viscosities. For gases, the kinematic viscosity increases more rapidly with temperature than does the absolute viscosity.

**Rate of shear versus shear stress for non-newtionian fiuds**

Bingham plastics, like that represented by curve B in Fig. 2.2, follow a rheological equation of the type

$$\tau_v = \tau_0 + K \frac{du}{dy} \tag{2.6}$$

where K is a constant. Over some range of shear rates, dilatant and pseudoplastic fluids often follow a power law, also called the Ostwald-de Waele equation,

$$\tau_v = K' \left(\frac{du}{dy}\right)^{n'} \tag{2.7}$$

where K' and n' are constants called the flow consistency index and the flow behavior index, respectively. Such fluids are known as power law fluids. For pseudoplastics (curve C) n'<1, and for dilatant fluids (curve D) n'>1. Clearly n'=1 for newtonian fluids. Values of n' and K' for some pseudoplastic fluids are given in Table 2.3

TABLE 2.3 Flow property indexes for pseudoplastic [6]

| Fluid | $n'$ | $K' \times 10^{-3}$ |
|---|---|---|
| 1.5% Carboxymethylcellulose in water | 0.554 | 3.13 |
| 3.0% Carboxymethylcellulose in water | 0.566 | 9.31 |
| 4.0% Paper pulp in water | 0.575 | 20.02 |
| 14.3% Clay in water | 0.350 | 0.173 |
| 25% Clay in water | 0.185 | 1.59 |
| Applesauce | 0.645 | 0.500 |
| Banana purée | 0.458 | 6.51 |
| Tomato concentrate | 0.59 | 0.2226 |

## 2.2.3 TURBULENCE

It has long been known that a fluid can flow through a pipe or conduit in two different ways. At low flow rates the pressure drop in the fluid increases directly with the fluid velocity; at high rates it increases much more rapidly, roughly as the square of the velocity. The distinction between the two types of flow was first demonstrated in a classic experiment by Osborne Reynolds, reported in 1883. A horizontal glass tube was immersed in a glass-walled tank filled with water. A controlled flow of water could be drawn through the tube by opening a valve. The entrance to the tube was flared, and provision was made to introduce a fine filament of colored water from the overhand flask into the stream at the tube entrance. Reynolds found that, at low flow rates, the jet of colored water flowed intact along with the mainstream and no cross-mixing occurred. The behavior of the color band showed clearly that the water was flowing in parallel straight lines and that the flow was laminar. When the flow rate was increased, a velocity, called the critical velocity, was reached at which the thread of color became wavy and gradually disappeared, as the dye spread uniformly throughout the entire cross section of the stream of water. This behavior of the colored water showed that the water no longer flowed in laminar motion but moved erratically in the form of cross-currents and eddies. This type of motion is turbulent flow.

**Reynolds number and transition from laminar to turbulent flow**

Reynolds studied the conditions under which one type of flow changes to the other and found that the critical velocity, at which laminar flow changes to turbulent flow, depends on four quantities: the diameter of the tube and the viscosity, density, and average linear velocity of the liquid. Furthermore, he found that these four factors can be combined into one group and that the change in the kind of flow occurs at a definite value of the group. The grouping of variables so found was

$$\mathrm{Re} = \frac{D\bar{V}\rho}{\mu} = \frac{D\bar{V}}{\nu} \tag{2.8}$$

Where D = diameter of tube

V = average velocity of liquid

$\mu$ = viscosity of liquid

$\rho$ = density of liquid

$\nu$ = kinematic viscosity of liquid

The dimensionless group of variables defined by Eq. (2.8) is called the Reynolds number ,Re. It is one of the named dimensionless groups. Its magnitude is independent of the units used, provided the units are consistent.

Additional observations have shown that the transition from laminar to turbulent flow actually may occur over a wide range of Reynolds numbers. In a pipe, flow is always laminar at Reynolds numbers below 2,100, but laminar flow can persist up to Reynolds numbers well above 24,000 by eliminating all disturbances at the inlet. If the laminar flow at such high Reynolds numbers is disturbed, however, say by a fluctuation in velocity, the flow quickly becomes turbulent. Disturbances under these conditions are amplified, whereas at Reynolds numbers below 2,100 all disturbance are damped and the flow remains laminar. At some flow rates a disturbance may be neither damped nor amplified; the flow is then said to be neutrally stable. Under ordinary conditions, the flow in a pipe or tube is turbulent at Reynolds numbers above about 4,000. Between 2,100 and 4,000 a transition region is found where the flow may be either laminar or turbulent, depending upon conditions at the entrance of the tube and on the distance from the entrance.

Reynolds number for non-newtonian fluids

Since non-newtonian fluids do not have a single-valued viscosity independent of shear rate, Eq. (2.8) for the Reynolds number cannot be used. The definition of a Reynolds number for such fluids is somewhat arbitrary; a widely used definition for power law fluids is

$$Re_n = 2^{3-n'}\left(\frac{n'}{3n'+1}\right)^{n'}\frac{D^{n'}\rho\bar{V}^{2-n'}}{K'} \tag{2.9}$$

The onset of turbulence occurs at Reynolds numbers above 2,100 with pseudoplastic fluids, for which $n'<1$.

Nature of turbulence

Because of its importance in many branches of engineering, turbulent flow has been extensively investigated in recent years, and a large literature has accumulated on this subject. Refined methods of measurement have been used to follow in detail the actual velocity fluctuations of the eddies during turbulent flow, and the results of such measurements have shed much qualitative and quantitative light on the nature of turbulence.

Turbulence may be generated in other ways than by flow through a pipe. In general, it can result either from contact of the flowing stream with solid boundaries or from contact between two layers of fluid moving at different velocities. The first kind of turbulence is called wall turbulence and the second kind free turbulence. Wall turbulence appears when the fluid flows through closed or open channels or past solid shapes immersed in the stream. Free turbulence appears in the flow of a jet into a mass of stagnant fluid or when a boundary layer separates from a solid wall and flows through the bulk of the fluid. Free turbulence is especially important in mixing.

Turbulent flow consists of a mass of eddies of various sizes coexisting in the flowing stream. Large eddies are continually formed. They break down into smaller eddies, which in turn evolve still smaller ones. Finally, the smallest eddies disappear. At a given time and in a given volume, a wide spectrum of eddy sizes exists. The size of the largest eddy is comparable with the smallest dimension of the turbulent stream; the diameter of the smallest eddies is 10 to 100 $\mu m$. Smaller eddies than this are rapidly destroyed by viscous shear. Flow within an eddy is laminar. Since even the smallest eddies contain about $10^{12}$ molecules, all eddies are of macroscopic size, and turbulent flow is not a molecular phenomenon.

Any given eddy possesses a definite amount of mechanical energy, much like that of a small spinning top. The energy of the largest eddies is supplied by the potential energy of the bulk flow of the fluid. From an energy standpoint, turbulence is a transfer process in which large eddies, formed from the bulk flow, pass their energy of rotation along a continuous series of smaller eddies. This mechanical energy is not appreciably dissipated into heat during the breakup of large eddies into smaller ones, but is passed along almost quantitatively to the smallest eddies. It is finally converted to heat when the smallest eddies are obliterated by viscous action. Energy conversion by viscous action is called viscous dissipation.

Deviating velocities in turbulent flow

A typical picture of the variations in the instantaneous velocity at a given point in a turbulent flow field is show in Fig. 2.3 This velocity is really a single component of the actual velocity vector, all three components of which vary rapidly in magnitude and direction. Also, the instantaneous pressure at the same point fluctuates rapidly and simultaneously with the fluctuations of velocity. Oscillographs showing these fluctuations provide the basic experimental data on which modern theories of turbulence are based.

Although at first sight turbulence seems to be structureless and randomized, studies of oscillographs like that in Fig. 2.3 show that this is not quite so. The randomness and unpredictability of the fluctuations, which are nonetheless constrained between definite limits, Quantitative characterization of turbulence, however, is commonly done by statistical analysis of the frequency distributions.

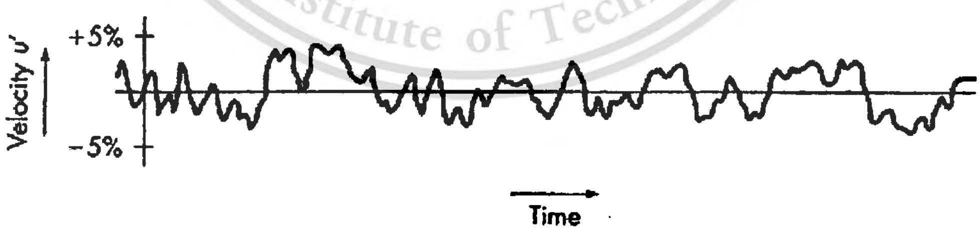



**FIGURE 2.3** Velocity fluctuations in turbulent flow. The percentages are based on the constant velocity. [6]

The instantaneous local velocities at given point can be measured by laser-Doppler anemometers, which are capable of following the rapid oscillations. Local velocities can be analyzed by splitting each component of the total instantaneous velocity into two parts, one a constant part that is the time average, or mean value, of the component in the direction of flow of the stream, and the other, called the deviating velocity, the instantaneous fluctuation of the component around the mean. The net velocity is that measured by ordinary flowmeters, such as a pitot tube, which are too sluggish to follow the rapid variations of the fluctuating velocity. The split of a velocity component can be formalized by the following method. Let the three components (in cartesian coordinates) of the instantaneous velocity in directions x, y, and z, be $u_i$, $v_i$, and $w_i$, respectively. Assume also that the x is oriented in the direction of flow of the stream and that components $v_i$ and $w_i$ are the y and z components, respectively, both perpendicular to the direction of bulk flow. Then the equations defining the deviating velocities are

$$u_i = u + u' \qquad v_i = v' \qquad w_i = w' \tag{2.10}$$

where

$u_i$, $v_i$, $w_i$, = instantaneous total velocity components in x, y, and z directions, respectively

U = constant net velocity of stream in x direction

$u'$, $v'$, $w'$ = deviating velocities in x, y, and z directions, respectively

Terms v and w are omitted in Eqs. (2.10) because there is no net flow in the directions of the y and z axes in one-dimensional flow, and so v and w are zero.

The deviating velocities $u'$, $v'$, and $w'$, all fluctuate about zero as an average. Figure 2.3 is actually a plot of the deviating velocity $u'$; a plot of the instantaneous velocity u, however, would be identical in appearance, since the ordinate would everywhere be increased by the constant quantity u.

For pressure,

$$p_i = p + p' \tag{2.11}$$

where $p_i$ = variable local pressure

p = constant average pressure as measured by ordinary manometers or pressure gauges

$p'$ = fluctuating part of pressure due to eddies

Because of the random nature of the fluctuations, the time averages of the fluctuating components of velocity and pressure vanish when averaged over a time period $t_0$ of the order of a few seconds. Therefore.

$$\frac{1}{t_0}\int_0^{t_0} u'\,dt = 0 \qquad \frac{1}{t_0}\int_0^{t_0} w'\,dt = 0$$

$$\frac{1}{t_0}\int_0^{t_0} v'\,dt = 0 \qquad \frac{1}{t_0}\int_0^{t_0} p'\,dt = 0 \tag{2.12}$$

The reason these averages vanish is that for every positive value of a fluctuation there is an equal negative value, and the algebraic sum is zero.

Although the time averages of the fluctuating components themselves are zero, this is not necessarily true of other functions or combinations of these components.
For example, the time average of the mean square of any one of these velocity component is not zero. This quantity for component u' is defined by

$$\frac{1}{t_0}\int_0^{t^0} (u')^2\,dt = \overline{(u')^2} \tag{2.13}$$

Thus the mean square is not zero, sine u takes on a rapid series of positive and negative values, which, when squared, always give a positive product. Therefore $\overline{(u')^2}$ is inherently positive and vanishes only when turbulence does not exist.

In laminar flow there are no eddies; the deviating velocities and pressure fluctuations do not exist; the total velocity in the direction of flow $u_i$ is constant and equal to u; and $v_i$ and $w_i$ are both zero.

### Statistical nature of turbulence

The distribution of deviating velocities at a single point reveals that the value of the velocity is related to the frequency of occurrence of that value, and that the relationship between frequency and value is gaussian and therefore follows the error curve characteristic of completely random statistical quantities. This result establishes turbulence as a statistical phenomenon, and the most successful treatments of turbulence have been based upon its statistical nature.

By measuring u', v', and w', at different places and over varying time periods, two kinds of data are obtained: (1) The three deviating velocity components at a single point can be measured, each as a function of time, and (2) the values of a single deviating velocity (for example, u' ) can be measured at different positions over the same time period. Figure 2.4 shows values of u' measured simultaneously at two points separated by vertical distance y. Data taken at different values of y show that the correspondence between the velocities at the two stations varies from a very close relationship at very small values of y to complete independence when y is large. This is to be expected, because when the distance between the measurements is small with respect to the size of an eddy, it is a single eddy that is being measured, and the deviating velocities found at the two stations are strongly correlated.

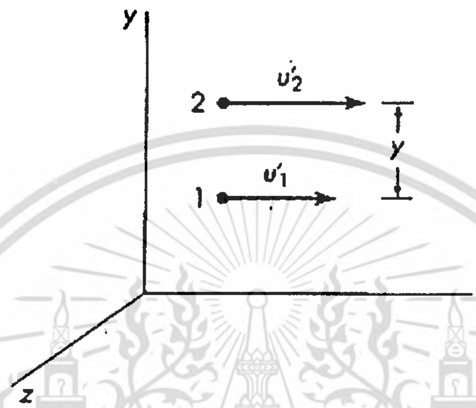



FIGURE 2.4 Fluctuating velocity components in measurement of scale of turbulence [6]

This means that when the velocity at one station changes either in direction or in magnitude, the velocity at the other station acts in practically the same way (or exactly the opposite way). At larger separation distances the measurements are being made on separate eddies, and the correlation disappears.

When the three components of the deviating velocities are measured at the same point, in general any two of them are also found to be correlated and a change in one is accompanied by a change in the other two.

These observations are quantified by defining correlation coefficients. One such coefficient, which corresponds to the situation shown in Fig. 2.4, is defined as follows:

$$R_{u'} = \frac{\overline{u'_1 u'_2}}{\sqrt{\overline{(u'_1)^2}\;\overline{(u'_2)^2}}} \tag{2.14}$$

where $u'_1$ and $u'_2$ are the values of u' at stations 1 and 2, respectively. Another correlation coefficient that applies at a single point is defined by

$$R_{u'v'} = \frac{\overline{u'v'}}{\sqrt{\overline{(u')^2}\,\overline{(v')^2}}} \tag{2.15}$$

Where u' and v' are measured at the same point at the same time.

**Intensity and scale of turbulence**

Turbulent fields are characterized by two average parameters. The first measures the intensity of the field and refers to the speed of rotation of the eddies and the energy contained in an eddy of a specific size. The second measures the size of the eddies. Intensity is measured by the root mean square of a velocity component. It is usually expressed as a percentage of the mean velocity or as

$$100\sqrt{\overline{(u')^2}}/u.$$

Very turbulent fields, such as those immediately below turbulence-producing grids, may reach an intensity of 5 to 10 percent. In unobstructed flow, intensities are less and of the order of 0.5 to 2 percent. A different intensity usually is found for each component of velocity.

The scale of turbulence is based on correlation coefficients such as $R_{u'}$, of $R_{u'}$, as a function of y, the scale L of the eddy in the y direction is calculated by the integral

$$L_y = \int_0^\infty R_{u'}\,dy \tag{2.16}$$

Each direction usually gives a different value of $L_y$, depending upon the choice of velocity components used in the definition. For air flowing in pipes at 12 m/s, the scale is about 10 mm, and this is a measure of the average size of the eddies in the pipe.

**Isotropic turbulence**

Although correlation coefficients generally depend upon the choice of component, in some situations this is not true, and the root-mean-square components are equal for all directions at a given point.

$$\overline{(u')^2} = \overline{(v')^2} = \overline{(w')^2}$$

In this situation the turbulence is said to be isotropic, and nearly isotropic turbulence exists when there is no velocity gradient, as at the centerline of a pipe or beyond the outer edge of a boundary layer. Nearly isotropic turbulence is also fond downstream of a grid placed in the flow. Turbulent flow near a boundary is anisotropy, but the anisotropy occurs action, are practically isotropic.

Reynolds stresses

It has long been known that shear forces much larger than those occurring in laminar flow exist in turbulent flow wherever there is a velocity gradient across a shear plane. The mechanism of turbulent shear depends upon the deviating velocities in anisotropic turbulence. Turbulent shear stresses are called Reynolds stresses. They are measured by the correlation coefficients of the type $R_{u'v'}$, defined in Eq. (2.15).

To relate Reynolds stresses to correlations of deviating velocities, the momentum principle may be used. Consider a fluid in turbulent flow moving in a positive x direction, as shown in Fig. 2.5. Plane S is parallel to the flow. The instantaneous velocity in the plane is $u_i$, and the mean velocity is u. Assume that u increases with y, the positive direction measured perpendicular to the layer S, so that the velocity gradient du/dy is positive. An eddy moving toward the wall has a negative value of V', and its movement represents a mass flow rate $\rho(-V')$ into the fluid below plane S. The velocity of the eddy in the x direction is u , or u + u'; if each such eddy crossing plane S is slowed down to the mean velocity u, the rate of momentum transfer per unit area is $\rho(-V')$ u'. This momentum flux, after time averaging for all eddies, is

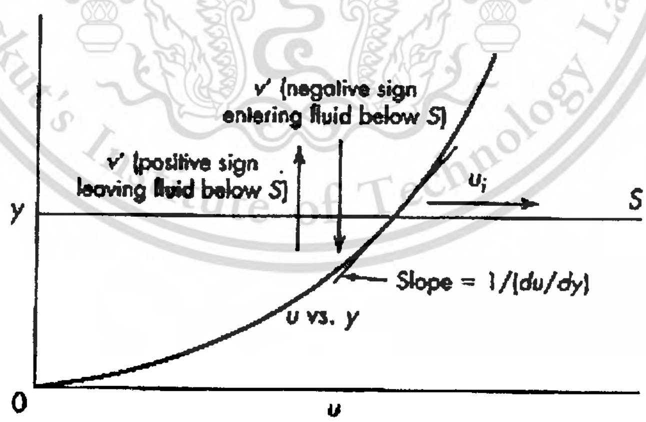



FIGURE 2.5 Reynolds stress. [6]

A turbulent shear stress or Reynolds stress given by the equation

$$\tau_t = \overline{\rho u'v'} \tag{2.17}$$

Eddy viscosity

By analogy with Eq. (2.4), the relationship between shear stress and velocity gradient in a turbulent stream is used to define an eddy viscosity $E_v$ :

$$\tau_t = E_v \frac{du}{dy} \tag{2.18}$$

Quantity $E_v$ is analogous to $\mu$, the absolute viscosity. Also, in analogy with the kinematic viscosity v the quantity $\varepsilon_M$ , called the eddy diffusivity of momentum, is defined as

$$\varepsilon_M = E_v / \rho$$

The total shear stress in a turbulent fluid is the sum of the viscous stress and the turbulent stress, or

$$\tau = (\mu + E_v)\frac{du}{dy} \tag{2.19}$$

$$\tau = (\nu + \varepsilon_M)\frac{d(\rho u)}{dy} \tag{2.20}$$

Although $E_v$ and $\varepsilon_M$ are analogous to $\mu$ and $\nu$, respectively, in that all these quantities are coefficients relating shear stress and the velocity gradient, there is a basic difference between the two kinds of quantities. The viscosities $\mu$ and $\nu$ are true properties of the fluid and are the macroscopic result of averaging motions and momentum of myriad molecules. The eddy viscosity $E_v$ and the eddy diffusivity $\varepsilon_M$ are not just properties of the fluid but depend on the fluid velocity and the geometry of the system. They are functions of all factors that influence the detailed patterns of turbulence and the deviating velocities, and they are especially sensitive to location in the turbulent field and the local values of the scale and intensity of the turbulence. Viscosities can be measured on isolated samples of fluid and presented in tables or charts of physical properties. Eddy viscosities and diffusivities are determined (with difficulty, and only by means of special instruments) by experiments on the flow itself.

### 2.2.4 BOUNDARY LAYERS

Flow in boundary layer

A boundary layer is defined as that part of a moving fluid in which the fluid motion is influenced by the presence of a solid boundary. As a specific example of boundary layer formation, consider the flow of fluid parallel with a thin plate, as shown in Fig.2.6. The velocity of the fluid upstream from the leading edge of the plate is uniform across the entire fluid stream. The velocity of the fluid at the interface between the solid and fluid is zero. The velocity increases with distance from the plate, as shown in Fig 2.6. Each of these curves corresponds to a definite value of x, the distance from the leading edge of the plate. The curves change slope rapidly near the plate; they also show that the local velocity approaches asymptotically the velocity of the bulk of the fluid stream.

In Fig.2.6. The dashed line OL is so drawn that the velocity changes are confined between this line and the trace of the wall. Because the velocity lines are asymptotic with respect to distance from the plate, it is assumed, in order to locate the dashed line definitely, that the line passes through all points where the velocity is 99 percent of the bulk fluid velocity $u_\infty$. Line OL represents an imaginary surface that separates the fluid stream into two parts: one in which the fluid velocity is constant and the other in which the velocity varies from zero at the wall to a velocity substantially equal to that of the undisturbed fluid. This imaginary surface separates the fluid that is directly affected by the plate from that in which the local velocity is constant and equal to the initial velocity of the approach fluid. The zone, or layer, between the dashed line and the plate constitutes the boundary layer.

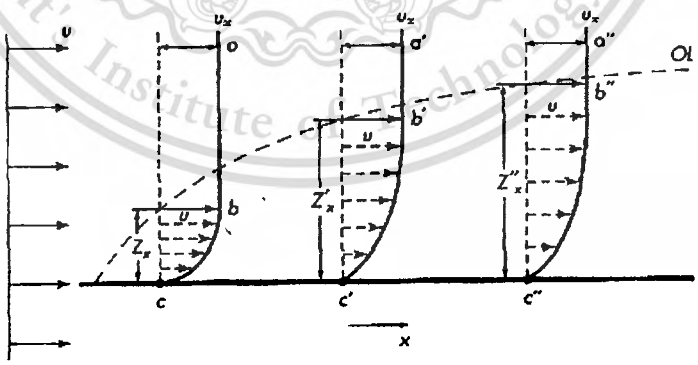



**FIGURE 2.6** Prandtl boundary layer: x, distance from leading edge: $u_\infty$ , velocity of undisturbed stream; $Z_x$ , thickness of boundary layer at distance x ; u, local velocity ; abc, a'b'c', a"b"c", curves of velocity versus distance from wall at points c, c', c" ; OL, outer limit of boundary layer. (The vertical scale is greatly exaggerated.) [6

Laminar and turbulent flow boundary layers

The fluid velocity at the solid-fluid interface is zero, and the velocities close to the solid surface are, of necessity, small. Flow in this part of the boundary layer very near the surface therefore is essentially laminar. Actually it is laminar most of the time, but occasionally eddies from the main portion of the flow or the outer region of the boundary layer move very close to the wall, temporarily disrupting the velocity profile. These eddies may have little effect on the average velocity profile near the wall, but they can have a large effect on the profiles of temperature or concentration when heat or mass is being transferred to or from the wall. This effect is most pronounced for mass transfer in liquids.

Farther away from the surface the fluid velocities, though less than the velocity of the undisturbed fluid, may be fairly large, and flow in this part of the boundary layer may become turbulent. Between the zone of fully developed turbulence and the region of laminar flow is a transition, or buffer, layer of intermediate character. Thus a turbulent boundary layer is considered to consist of three zones: the viscous sublayer, the buffer layer, and the turbulent zone. The existence of a completely viscous sublayer is questioned by some, since mass-transfer studies suggest that some eddies penetrate all the way through the boundary layer and reach the wall.

Near the leading edge of a flat plate immersed in a fluid of uniform velocity, the boundary layer is thin, and the flow in the boundary layer is entirely laminar. As the layer thickens, however, at distances farther from the leading edge, a point is reached where turbulence appears. The onset of turbulence is characterized by a sudden rapid increase in the thickness of the boundary layer, as shown in Fig. 2.7.

When flow in the boundary layer is laminar, the thickness $Z_x$ of the layer increases with $x^{0.5}$ where x is the distance from the leading edge of the plate. For a short time after turbulence appears, $Z_x$ increases with $x^{1.5}$ and then, after turbulence is fully developed, with $x^{0.8}$.

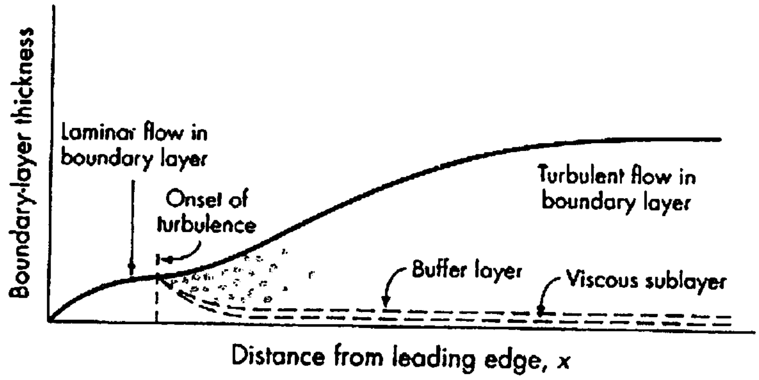



FIGURE 2.7 Development of turbulent boundary layer on a flat plate. [6]

The initial, fully laminar part of the boundary layer may grow to a moderate thickness of perhaps 2 mm with air or water moving at moderate velocities. Once turbulence begins, however, the thickness of the laminar part of the boundary layer diminishes considerably, typically to about 0.2 mm.

Transition from laminar to turbulent flow; Reynolds number. The factors that determine the point at which turbulence appears in a laminar boundary layer are coordinated by the dimensionless Reynolds number defined by the equation

$$\mathrm{Re}_x = \frac{x u_\infty \rho}{\mu} \tag{2.21}$$

where $x$ = distance from leading edge of plate

$u_\infty$ = bulk fluid velocity

$\rho$ = density of fluid

$\mu$ = viscosity of fluid

With parallel flow along a plate, turbulent flow first appears at a critical Reynolds number between about $10^5$ and $3 \times 10^6$. The transition occurs at the lower Reynolds numbers when the plate is rough and the intensity of turbulence in the approaching stream is high, and at the higher values when the plate is smooth and the intensity of turbulence in the approaching stream is low.

**Boundary layer formation in straight tubes**

Consider a straight, thin-walled tube with fluid entering it at a uniform velocity. As shown in Fig. 2.8, a boundary layer begins to from at the entrance to the tube, and as the fluid moves through the first part of the channel, the layer thickens. During this stage the boundary layer occupies only part of the cross section of the tube, and the total stream consists of a core of fluid flowing in rodlike manner at constant velocity and an annular boundary layer between the wall and the core. In the boundary layer the velocity increases from zero at the wall to the constant velocity existing in the core. As the stream moves farther down the tube, the boundary layer occupies an increasing portion of the cross section. Finally, at a point well downstream from the entrance, the boundary layer reaches the center of the tube, the rodlike core disappears, and the boundary layer occupies the cross section of the stream. At this point the velocity distribution in the tube reaches its final form, as shown by the last curve at the right of Fig. 2.8, and remains unchanged during the remaining length of the tube. Such flow with an unchanging velocity distribution is called fully developed flow.

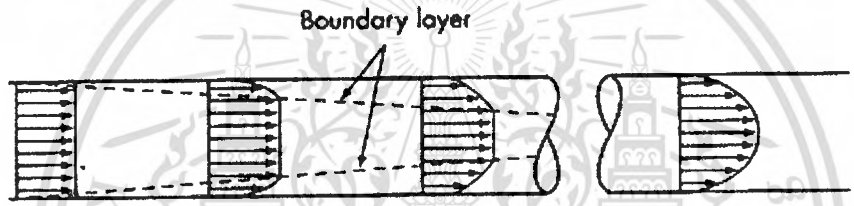



FIGURE 2.8 Development of boundary layer flow in pipe. [6]

Transition length for laminar and turbulent flow. The length of the entrance region of the tube necessary for the boundary layer to reach the center of the tube and for fully developed flow to be established is called the transition length. Since the velocity varies not only with length of tube but also with radial distance from the center of the tube, flow in the entrance region is two-dimensional.

The approximate length of straight pipe necessary for completion of the final velocity distribution is, for laminar flow.

$$Re_x = \frac{x u_\infty \rho}{\mu}$$

Where $x_t$ = transition length

D = diameter of pipe

Equation (2.21), originally proposed by Nikuradse, was verified experimentally by Rothfus and Prehgle. Equation (2.21) shows that for a 50-mm-(2-in.-) ID pipe and a Reynolds number of 1,500, the transition length is 3.75 m (12.3 ft). If the fluid entering the pipe is turbulent and the velocity in the tube is above the critical, the transition length is nearly independent of the Reynolds number and is about 40 to 50 pipe diameters, with little difference between the distribution at 25 diameters and that at greater distances from the entrance. For a 50-mm-ID pipe, a longer transition length, as large as 100 pipe diameters, is needed.

Boundary layer separation and formation

In the preceding paragraphs the growth of boundary layers has been discussed. Now consider what happens at the far side of a submerged object, where the fluid leaves the solid surface.

At the trailing edge of a flat plate that is parallel to the direction of flow, the boundary layers on the two sides of the plate have grown to a maximum thickness. For a time after the fluid leaves the plate, the layers and velocity gradients persist. Soon, however, the gradients fade out, the boundary layers intermingle and disappear, and the fluid once more moves with a uniform velocity. This is shown in Fig. 2.9a.

Suppose, now, the plate is turned at right angles to the direction of flow, as in Fig. 2.9b. A boundary layer forms as before in the fluid flowing over the upstream face. When the fluid reaches the edge of the plate, however, its momentum prevents it from making the sharp turn around the edge, and it separates from the plate and proceeds outward into the bulk of the fluid. Behind the plate is a backwater zone of strongly decelerated fluid, in which large eddies, called vortices, are formed. This zone is known as the wake. The eddies in the wake are kept in motion by the shear stresses between the wake and the separated current. They consume considerable mechanical energy and may lead to a large pressure loss in the fluid.

Boundary layer separation occurs whenever the change in velocity of the fluid, in either magnitude or direction, is too large for the fluid to adhere to the solid surface. It is most frequently encountered when there is an abrupt change in the flow channel such as a sudden expansion or contraction, a sharp bend, or an obstruction around which the fluid must flow. Separation may also occur from the velocity decrease in a smoothly diverging channel. Because of the large energy losses resulting from the formation of a wake, it is often desirable to minimize or prevent boundary layer separation. In some cases this can be done by suction, i. e., by drawing part of the fluid into the solid surface at the area of potential separation. Most often, however, separation is minimized by avoiding sharp changes in the cross-sectional area of the flow channel and by streamlining any objects over which the fluid must flow. For some purposes, such as the promotion of heat transfer or mixing in a fluid, boundary layer separation may be desirable.

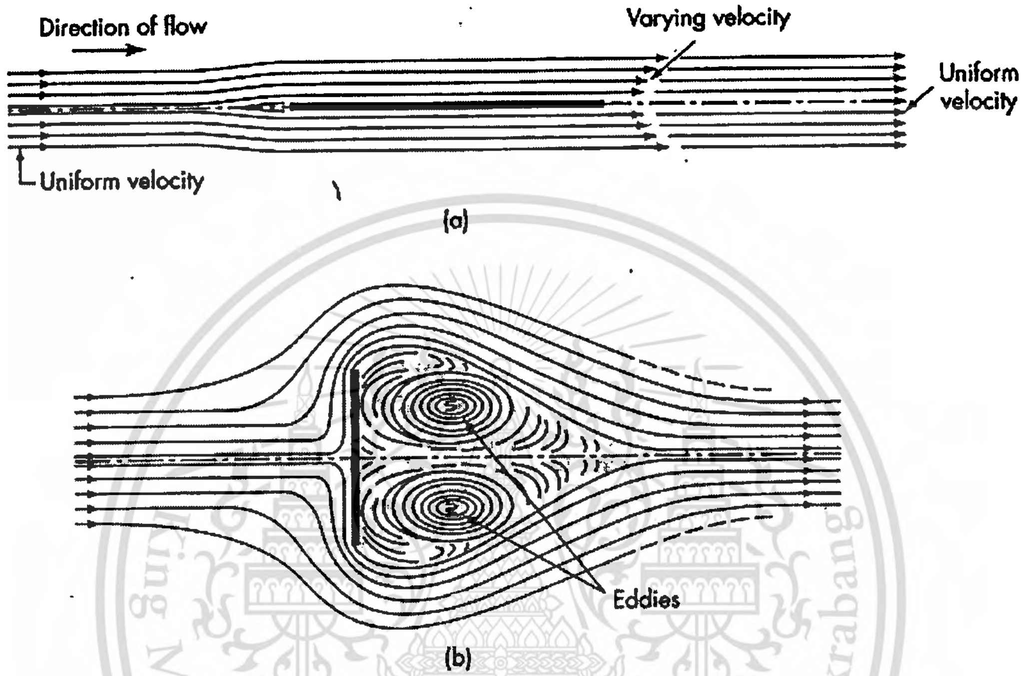



FIGURE 2.9 Flow past flat: (a) flow parallel with plate; (b) flow perpendicular to plate. [6]

## 2.3 HEAT EXCHANGER TYPES

In industrial processes heat energy is transferred by a variety of methods, including conduction in electric-resistance heaters; conduction-convection in exchangers, boilers, and condensers; radiation in furnaces and radiant heat dryers; and by special methods such as dielectric heating. Often the equipment operates under steady-state conditions, but in many processes it operates cyclically, as in regenerative furnaces and agitated process vessels.

The equipment types that are of greatest interest to a process engineer are tubular and plate exchangers, extended-surface equipment, mechanically aided heat-transfer devices, condensers and vaporizers, and packed-bed reactors or regenerators.

General design of heat-exchanger

The design and testing of practical heat-exchanger are based on the general principles. From material and energy balances, the required heat-transfer rate is calculated. Then, using the overall coefficient and the average $\Delta T$, the required heat-transfer area is determined, and in cyclic equipment, considerable accuracy, but in complex processing units the evaluation may be difficult and subject to considerable uncertainty. The final design is nearly always a compromise, based on engineering judgment, to give the best overall performance in light of the service requirements.

Sometimes the design is governed by considerations that have little to do with heat transfer, such as the space available for the equipment or the pressure drop that can be tolerated in the fluid streams. Tubular exchangers are, in general, designed in accordance with various standards and codes, such as the Standards of the Tubular Exchanger Manufacturers Association (TEMA) and the ASME-API Unfired Pressure Vessel Code.

In designing an exchanger many decisions---some of them arbitrary---must be made to specify the materials of construction, tube diameter, tube-length, baffle spacing, number of passes, and so forth. Compromises must also be made. For example, a high fluid velocity inside small tubes leads to improved heat-transfer coefficients and a small required area, but increases the friction losses and pumping costs. The design of an individual exchanger may be optimized by a formal procedure to balance the heat-transfer area, and hence the price of the equipment and the fixed costs, against the cost of energy to pump the fluids. In processing plants, however, the exchangers are components of a complex network of heat-transfer equipment, and it is the network, not the individual units, that is optimized to give minimum investment and operating costs.

### 2.3.1 SHELL-AND-TUBE HEAT EXCHANGERS

Tubular heat exchangers are so important and so widely used in the process industries tries that their design has been highly developed devised and accepted by **TEMA** are available covering in detail the materials, methods of construction, technique of design, and dimensions for exchangers. The following sections describe the more important types of exchanger and cover the fundamentals of their engineering, design, and operation.

#### a.) Single-pass 1-1 exchanger

The simple double-pipe exchanger is inadequate for flow rates that cannot readily be handled in a few tubes. If several double pipes are used in parallel, the weight of metal required for the outer tubes becomes so large that the shell-and-tube construction, such as that shown in Fig. 2.10, where one shell serves for many tubes, is more economical. This exchanger, because it has one shell-side pass and one tube-side pass, is a 1-1 exchanger.

In an exchanger the shell-side and tube-side heat-transfer coefficients are of comparable importance, and both must be large if a satisfactory overall coefficient is to be attained. The velocity and turbulence of the shell-side liquid are as important as those of the tube-side fluid. To promote crossflow and raise the average velocity of the shell-side fluid, baffles are installed in the shell. In the construction shown in Fig. 2.10, baffles A consist of circular disks of sheet metal with one side cut away. Common practice is to cut away a segment having a height equal to one-fourth the inside diameter of the shell. Such baffles are called 25 percent baffles. The baffles are perforated to receive the tubes. Velocity of the shell-side fluid, baffles are installed in the shell. In the construction shown in Fig. 2.10, baffles A consist of circular disks of sheet metal with one side cut away. Common practice is to cut away a segment having a height equal to one-fourth the inside diameter of the shell. Such baffles are called 25 percent baffles.

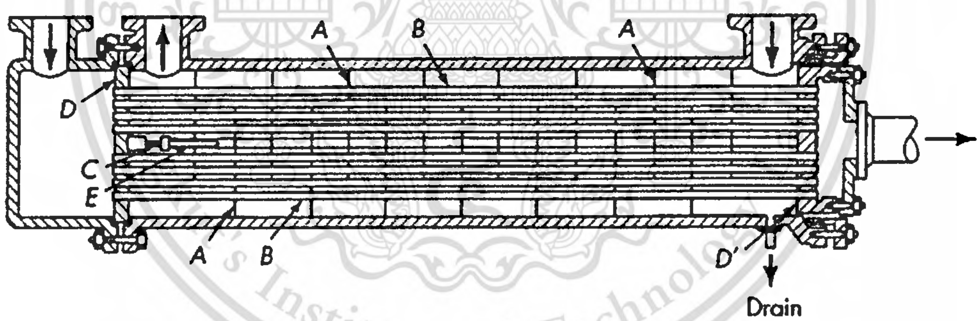



**FIGURE 2.10** Single pass 1-1 counterflow heat exchanger (A), baffle (B), tubes (C), guide rods (D and D'), tube sheets (E), spacer tubes. [6]

The baffles are perforated to receive the tubes.To minimize leakage, the clearances between baffles and shell and tubes should be small. The baffles are supported by one or more guide rods C, which are fastened between the tube sheets D and D' by setscrews. To fix the baffles in place, short sections of tube E are slipped over rod C between the baffles. In assembling such an exchanger, it is necessary to do the tube sheets, support rods, spacers, and baffles first and then to install the tubes.

The stuffing box shown at the right-hand end of Fig. 2.10 provides for expansion. This construction is practicable only for small shells.

Tubes and tube sheets

Tubes are drawn to definite wall thickness in terms of BWG and true outside diameter (OD), and they are available in all common metals. Tables of dimensions of standard tubes are given in App. 4. Standard lengths of tubes for heat-exchanger construction are 8, 12,16, and 20 ft. Tubes are arranged in a triangular or square layout, known as triangular pitch or square pitch (pitch is the distance between centers of adjacent tubes). Triangular pitch is used unless the shell side tends to foul badly, because more heat-transfer area can be packed into a shell of given diameter than with square pitch. If the center-to-center distance between tubes is too small, tubes in triangular pitch cannot be cleaned by running a brush between rows, whereas in square pitch are readily cleaned. Also, square pitch gives a lower shell-side pressure drop than triangular pitch.

TEMA standards specify a minimum pitch of 1.25 times the outside diameter of the tubes for triangular pitch and a minimum cleaning lane of ¼ in. for square pitch.

Shell and baffles

Shell diameters are standardized. For shells up to and including 23 in. the diameters are fixed in accordance with American Society for Testing and Materials (ASTM) pipe standard inside diameters are 8, 10, 12, 13¼, 15¼, 17¼, 19¼, 21¼, and 23¼ in. then 25, 27 in., and so on in 2 in. increments. These shells are constructed of rolled plated.

The distance between battles (center to center) is the baffle pitch, or baffle spacing. It should not be less than one-fifth the diameter of the shell or more than the inside diameter of the shell.

Tubes are usually attached to the tube sheets by grooving the holes circumferentially and rolling the tube ends into the holes by means of a rotating tapered mandrel, which stresses the metal of the tube beyond the elastic limit, so the metal flows into the grooves. In high-pressure exchangers, the tubes are welded or brazed to the tube sheet after rolling.

b.) Alternate designs

Shell-and-tube exchangers with segmented plate baffles may have vibration problems caused by the fluid flowing at high velocity across the tubes. In the ROD baffle exchanger developed by Phillips Petroleum Company, metal rods rather than sheet-metal baffles are used to support the tubes, and flow in the shell is mainly parallel to the tube axis.

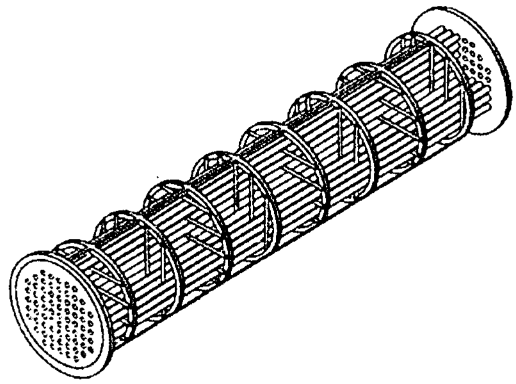


**FIGURE 2.11** ROD baffle exchanger. [6]

The tubes are arranged in square pitch, and rods with a diameter equal to the clearance between tube rows are attached to ring supports and placed between alternate tube in both horizontal and vertical directions. The normal rod diameter is in., and each tube is supported on all four sides at several points along the exchanger, as shown in Fig. 2.11.

Correlations for the outside film coefficient have been developed using the hydraulic diameter for the Reynolds and Nusselt numbers and allowing for the effects of baffle spacing and leakage around the tube bundle. Flow across the rods leads to vortex formation, and the coefficients for turbulent flow are about 1.5 times those predicted for the same Reynolds number using the Dittus-Boelter equation. The coefficients are not as high as those for a segmentally baffled exchanger with close baffle spacing, but the lower pressure drop and reduced vibration failure make the ROD baffle exchanger preferred for many applications.

Another design that requires no baffles uses tubes that are twisted into a helical shape with an oval cross section, so that each tube is supported over its entire length by multiple contact points with adjacent tubes. The twisted tubes give improved heat-transfer coefficients inside and outside because of greater turbulence, and the decrease in required surface area may more than offset the higher cost per square foot. This design eliminates tube vibration and may also reduce the rate of fouling.

With both the twisted tube and ROD baffle exchangers, flow distribution on the shell side is a problem in large-diameter units. With a single inlet pipe, tubes near the inlet would get more than the average flow, and those opposite the inlet would get little flow for an appreciable distance down the exchanger. Flow distribution is improved by enlarging the shell at the ends of the exchanger to make annular zones where fluid enters or leaves radially at lower velocity.

c.) Multipass exchangers

The 1-1 exchanger has limitations, because when the tube-side flow is divided evenly among all the tubes, the velocity may be quite low, giving a low heat-transfer coefficient. If the number of tubes is reduced and the length increased so that the velocity is sufficiently high, the tube length may be impractical.

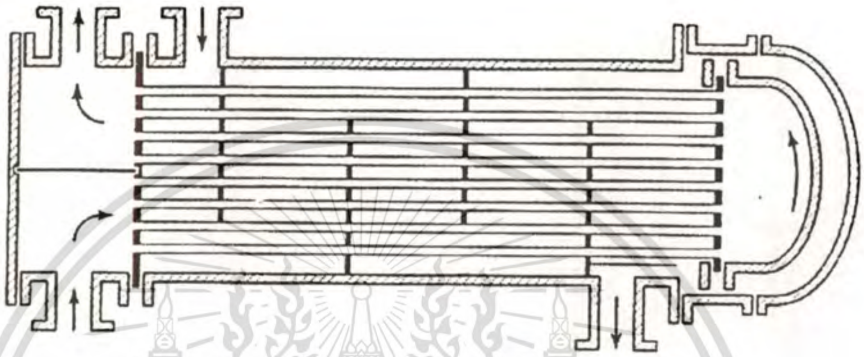


**FIGURE 2.12** A 1-2 parallel counterflow exchanger. [6]

Using multipass construction with two, four, or more tube passes permits the use of standard tube lengths while ensuring a high velocity and a high tube-side coefficient. The disadvantages are that (1) the construction of the exchanger is slightly more complicated, (2) some sections in the exchanger have parallel flow, which limits the temperature approach, and (3) the friction loss is greatly increased. For example, the average velocity in the tubes of a four-pass exchanger is 4 times that in a single-pass exchanger haying the same number and size of tubes and operated at the same liquid flow rate. The tube-side coefficient of the four-pass exchanger is approximately 4 = 3.03 times that for the single-pass exchanger, or even more if the velocity in the single-pass unit is sufficiently low to give laminar flow. The pressure drop per unit length is 4 times greater, and the length is increased by 4 times; consequently the total friction loss is 4 = 48.5 times that in the single-pass unit, not including the additional expansion and contraction losses. The most economic design calls for such a velocity in the tubes that the increased cost of power for pumping is offset by the decreased cost of the apparatus.

An even number of tube-side passes are used in multipass exchangers. The shell side may be either single-pass or multipass. A common construction is the 1-2 parallel-counterflow exchanger, in which the shell-side liquid flows in one pass and the tube-side liquid in two passes. Such an exchanger is shown in Fig. 15.3. In multipass exchangers, floating heads are frequently used, and the bulge in the shell of the condenser in Fig. 11.1 and the stuffing box shown in Fig 15.1 are unnecessary. The tube-side liquid enters and leaves through the same head, which is divided by a baffle to separate the entering and leaving tube-side streams.

The 1-2 exchanger is normally arranged so that the cold fluid and the hot fluid enter at same end of the exchanger, giving parallel flow in the first tube pass and counterflow in the second. This permits a close temperature approach, at least at the exit end of the exchanger.

2-4 Exchanger

The 1-2 exchanger has an important limitation. Because of the parallel-flow pass, the exchanger is unable to bring the exit temperature of one fluid very near to the entrance temperature of the other. Another way of stating the same limitation is that heat recovery of a 1-2 exchanger is inherently poor.

Figure 2.13 A 2-4 exchanger. [6]

A better recovery can be obtained by adding a longitudinal baffle to give two shell passes. A 2-2 exchanger of this kind closely approximates the performance of a double-pipe exchanger, but even with two tube-side passes the total tube length may be insufficient for good heat transfer. More common is the 2-4 exchanger, which has two shell-side and four tube-side passes. This type of exchanger also gives higher velocities and a larger overall heat-transfer coefficient than a 1-2 exchanger having two tube-aide passes and operating with the same flow rates. An example of a 2-4 exchanger is shown in Fig.2.13.

<u>Temperature patterns in multipass exchangers</u>

Temperature-length curves for a 1-2 exchanger are shown in Fig.2.14a using the following temperature designations:

Inlet temperature of hot fluid $T_{ha}$

Inlet temnerature of cold fluid $T_{ca}$

Outlet temperature of hot fluid $T_{hb}$

Outlet temperature of cold fluid $T_{cb}$

Intermediate temperature of cold fluid $T_{ci}$

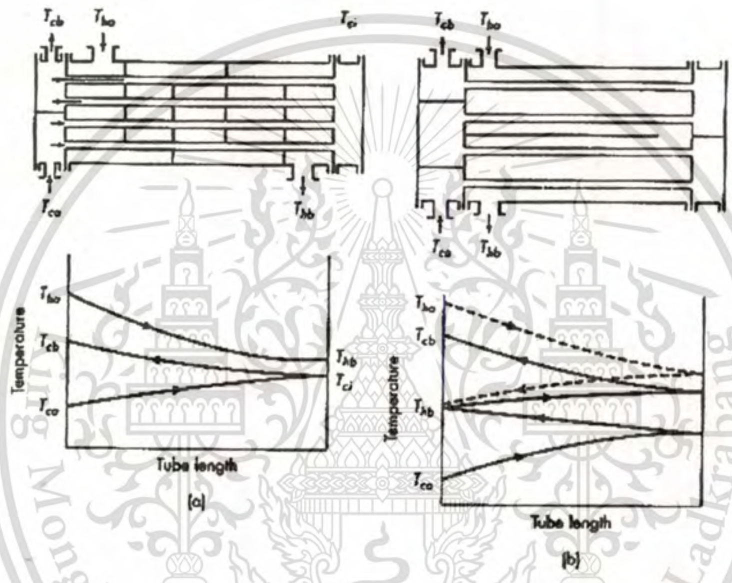



**FIGURE 2.14** Temperature length curves: (a) 1-2 exchanger, (b) 2-4 exchanger. [6]

Curve $T_{ha} - T_{hb}$ applies to the shell-side fluid, which is assumed to be the hot fluid. Curve $T_{ca} - T_{ci}$ applies to the first pass of the tube-side liquid, and curve $T_{ci} - T_{cb}$ to the second pass of the tube-side liquid. In Fig. 2.14a curves $T_{ha} - T_{hb}$ and $T_{ca} - T_{ci}$ taken together are those of a parallel-flow exchanger, and curves $T_{ha} - T_{hb}$ and $T_{ci} - T_{cb}$ taken together correspond to a countercurrent exchanger. The curves for a 2-4 exchanger are given in Fig. 2.14. The dotted lines refer to the shell-side fluid and the solid lines to the tube-side fluid. Again it is assumed that the hotter fluid is in the Shell. The hotter pass of the shell-side fluid is in thermal contact with the two hottest tube-side passes and the cooler shell-side pass with the two coolest tube-side passes. The exchanger as a whole approximates a true countercurrent unit more closely than is possible with a 1-2 exchanger.

<u>Correction of LMTD in multipass exchangers</u>

In multipasses exchangers which have more tube passes than shell passes, the flow is countercurrent in some sections and parallel in others. Figure 2.15 and b shows factor $F_G$ for

1-2 and 2-4 exchangers, respectively, derived on the assumptions that the overall elements of a given fluid stream have the same thermal history in passing through the exchanger. Each curved line in the figure corresponds to a constant value of the dimenstionless ratio Z, defined as

$$Z = \frac{T_{ha} - T_{hb}}{T_{cb} - T_{ca}} \qquad (2.22)$$

and the abscissas are values of the dimenstionless ration $\eta_H$ , defined as

$$\eta_H = \frac{T_{cb} - T_{ca}}{T_{ha} - T_{ca}} \qquad (2.23)$$

The factor Z is the ratio the fall in temperature of the hot fluid to the rise in temperature of the cold fluid. The factor $\eta_H$ is the *heating effectiveness*, or the ratio of the actual temperature rise of the cold fluid to the maximum possible temperature rise obtainable if the warm end approach were zero (based on countercurrent flow). From the numerical valves of $\eta_H$ and Z, factor $F_g$ is read from Fig.2.15, interpolating between lines of constant Z where necessary, and multiplied by the LMTD for counterflow to give the true mean temperature drop.

Factor $F_G$ is always less than unity. The mean temperature drop, and therefore the capacity of the exchanger, is less than that of a countercurrent exchanger having the same LMTD. When $F_G$ is less than about 0.8, the exchanger should be redesigned with more passes or larger temperature differences; otherwise the heat transfer surface is inefficiently used, and there is danger that small changes in conditions may cause the exchanger to become inoperable. When $F_G$ is less than 0.75, it falls rapidly as $\eta_H$ increases, so that operation is sensitive to small changes. In this region, also, any deviations from the basic assumptions on which the charts are based become important especially that of a uniform thermal history for all elements of fluid. Leakage through and around the baffles may partially invalidate this assumption.

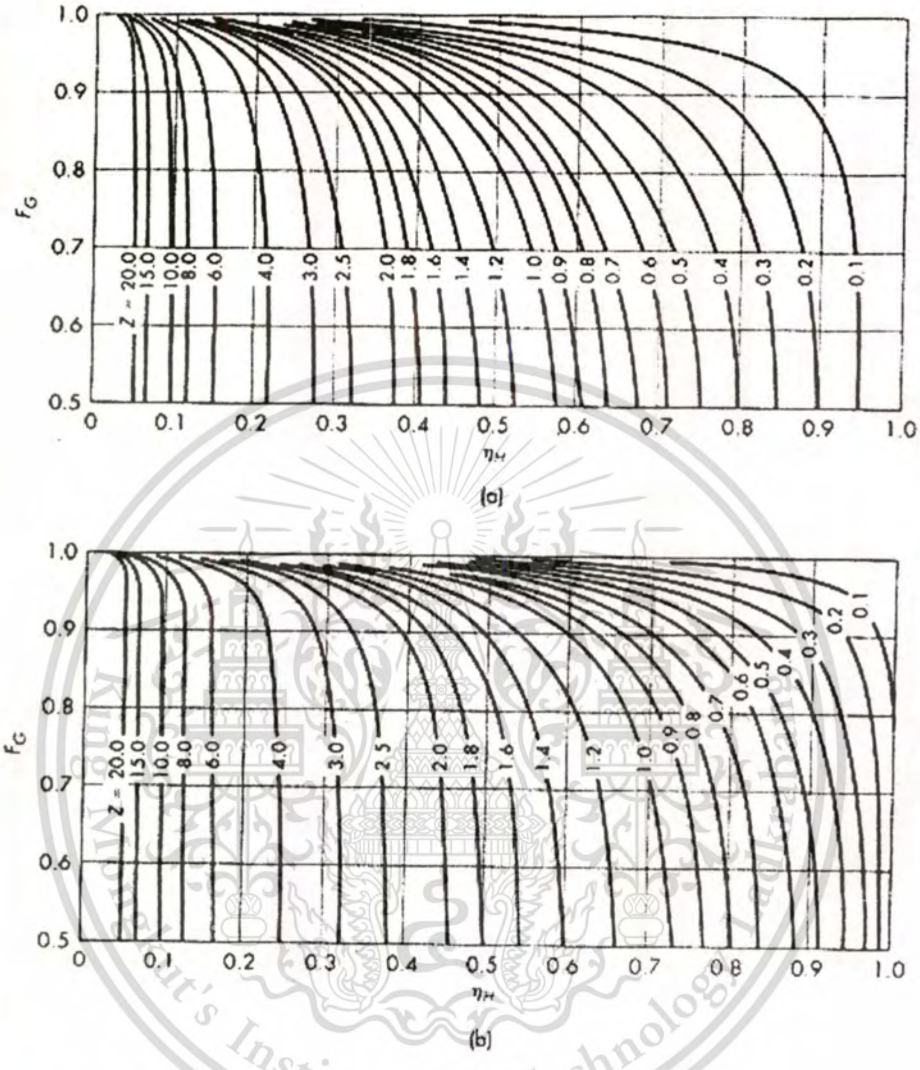



**FIGURE 2.15** Correction of LMTD: (a) 1-2 exchangers, (b) 2-4 exchangers. [6]

Other combinations of shell side passes and tube side passes are used, but the 1-2 and 2-4 are the most common. As the number of shell passes increases, with an even number of tube passes (usually 2 times the number of shell passes). $F_G$ increase and exchanger performance approaches that for true counterflow.

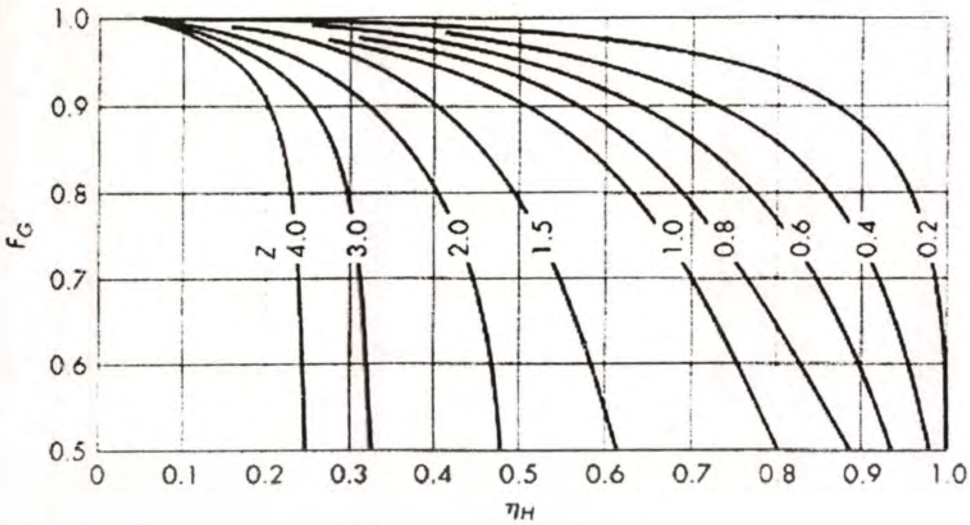



**FIGURE 2.16** Correction of LMTD for crossflow. [6]

Heat transfer coefficients in shell and tube exchangers

The coefficient for the shell side $h_o$ cannot be calculated because the direction of flow is partly parallel to the tubes and partly across them, and because the cross sectional area of the stream and the mass velocity of the stream vary as the fluid crosses the tube bundle back and forth across the shell. Also, leakage between baffles and shell and between baffles and tubes short circuit some of the shell side liquid and reduces the effectiveness of the exchanger. An approximate but generally useful equation for predicting shell side coefficients is the *Donohue equation*, which is based on a weighted average mass velocity $G_e$ of the fluid flowing parallel with the tubes and that flowing across the tubes. The mass velocity $G_b$ parallel with the tubes is the mass flow rate divided by the free area for flow in the baffle window $S_b$. (The baffle window is the portion of the shell cross section not occupied by the baffle). This area is the total area of the baffle window less the area occupied by the tubes, or

$$S_b = f_b \frac{\pi D_s^2}{4} - N_b \frac{\pi D_o^2}{4} \tag{2.24}$$

Where

$f_b$ = fraction of cross sectional area of shell occupied by baffle window (commonly 0.1995)

$D_s$ = inside diameter of shell

$N_b$ = number of tubes in baffle window

$D_o$ = outside diameter of tubes

In crossflow the mass velocity passes through a local maximum each time the fluid passes a row of tubes. For correlating purposes the mass velocity $G_c$ for crossflow is based on the area $S_c$ for transverse flow between the tubes in the row at or closest to the centerline of the exchanger. In a large exchanger $S_c$ can be estimated form the equation

$$S_c = P D_s\left(1 - \frac{D_o}{p}\right) \tag{2.25}$$

Where

$p$ = center to center distance between tubes

$\mathbf{P}$ = baffle pitch

The mass velocities are then

$$G_b = \frac{\dot{m}}{S_b} \quad \text{and} \quad G_c = \frac{\dot{m}}{S_c} \tag{2.26}$$

The Donohue equation is

$$\frac{h_o D_o}{k} = 0.2\left(\frac{D_o G_e}{\mu}\right)^{0.6}\left(\frac{c_p \mu}{k}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{2.27}$$

Where $G_e = (G_b G_c)^{1/2}$.

This is equation tends to give conservatively low values of $h_o$ especially at low Renolds numbers. More elaborate methods of estimating shell side coefficient are available for the specialist. In j - factor form Eq. (2.27) becomes

$$\frac{h_o}{c_p G_e}\left(\frac{c_p \mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = j_H = 0.2\left(\frac{D_o G_e}{\mu}\right)^{-0.4} \tag{2.28}$$

After the individual coefficients are known, the total area required is found in the usual way from the overall coefficient. The LMTD must often be corrected for the departure from true counterflow.

### 2.3.2 PLATE TYPE EXCHANGERS

For many applications at moderate temperature and pressure, an alternative to the shell and tube exchanger is the gasketed plate exchanger, which consists of many corrugated stainless sheets separated by polymer gaskets and clamped in a steel frame. Inlet portals and slots in the gaskets direct the hot and cold fluid to alternate spaces between the plates.

The corrugation induce turbulence for improved heat transfer, and each plate is supported by multiple contacts with adjoining plates, which have a different pattern or angle of corrugation. The space between plates is equal to the depth of the corrugations and is usually 2 to 5 mm. A typical plate design is shown in Fig. 2.17

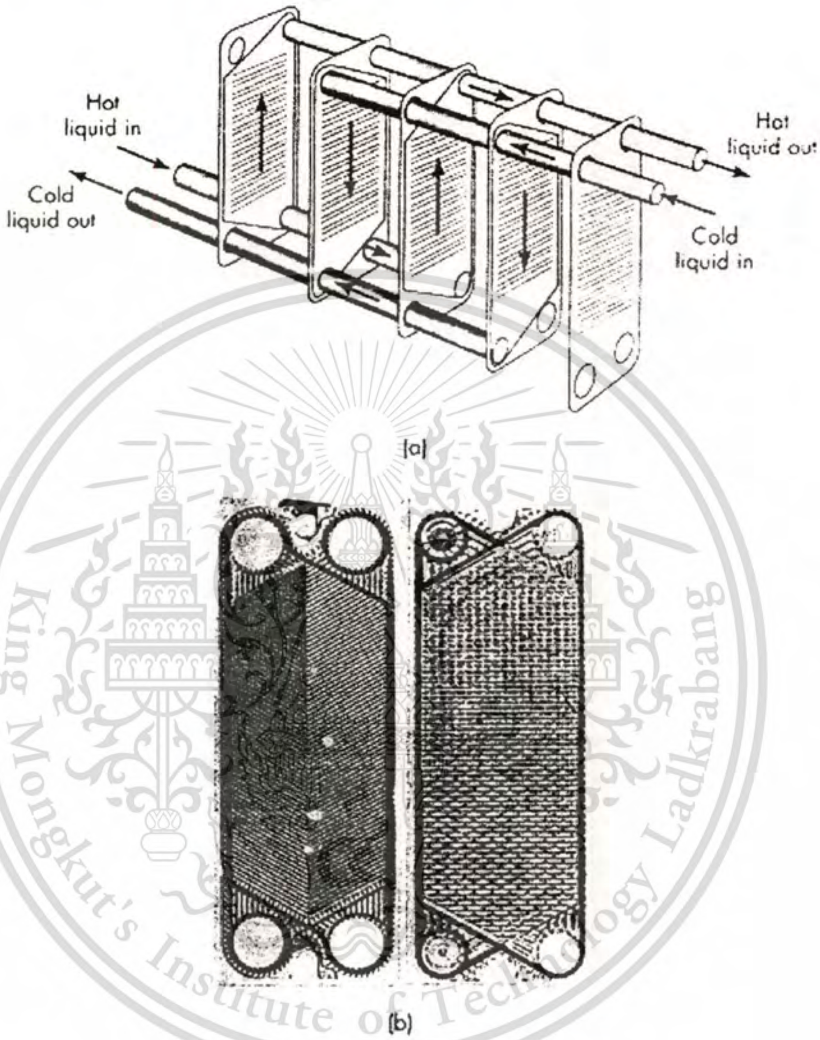



**FIGURE 2.17** Plate heat exchager: (a) general, (b) detail of plate design. [6]

For a liquid-liquid exchanger, the usual fluid velocity is 0.2 to 1.0 m/s, and because of the small spacing, the Reynolds number is often less than 2,100. However, the corrugation give the flow turbulent characteristics at Reynolds numbers of 100 to 400, depending on the plate design. Evidence for turbulent flow is flow is that the heat transfer coefficient varies with the 0.6 to 0.8 power of the flow rate, and the pressure drop depends on the 1.7 to 2.0 power of the flow rate. The heat transfer correlation for a common plate design is

$$\mathrm{Nu} = \frac{h D_e}{k} = 0.37\ \mathrm{Re}^{0.67}\ \mathrm{Pr}^{0.33} \tag{2.29}$$

The pressure drop is given by the Fanning equation with the following friction factor:

$$f = 2.5\ \mathrm{Re}^{-0.3} \tag{2.30}$$

In Eq. (2.29), h is based on the nominal area of the corrugated plate. (Other correlations may be based on the corrugated area). The equivalent diameter is 4 times the hydraulic radius, which for most exchangers is twice the plate spacing. If the hot and cold flows are equal and are introduced at opposite ends of the plates, performance close to counterflow can be obtained in the end section, fluid is heated or from only one side, which slightly reduces the efficiency, and the zigzag flow path also deviates from the ideal pattern. The correction factor decreases as the number of heat transfer units increases and is about 0.95 where $N_H$ = 3.0. When the flows are not balanced two or more passes can be used for the stream with the lower flow rate; but then the exchanger has some parallel and some counterflow sections, and the LMTD correction factor $F_G$ may be 0.7 to 0.9.

With water or aqueous solutions on both sides, the overall coefficient for a clean plate type exchanger may be 3,000 to 6,000 $W/m^2.K$ (500 to1000 $Btu/h.ft^2.°F$), several times the normal value for a shell and tube exchanger. Because of high shear rates, the fouling factor experienced are much lower than those for shell and tube exchangers, and the designer may just add 10% to the calculated area to allow for fouling. The units can easily be taken apart for thorough cleaning.

Plate exchanges were originally used mainly in the dairy and food processing industries, but they have now found many other applications. New designs and better gaskets permit operation at up 200°C and 25 atm. Exchangers with plate areas of 2 $m^2$ and a total area of 1,500 $m^2$ are available.

### 2.3.3 DOUBLE-PIPE HEAT EXCHAGERS

A double-pipe heat exchanger is illustrated in Figure 2.18 As indicated , it consists of the rube within another. One fluid flows through the inner tube. Another fluid flows through the annulus. Usually common copper water tubing is used to construct the exchanger, although pipe and specialty tubing can also be used. Our objective in analyzing a double pipe heat exchanger will be to predict its performance-the amount of heat transferred, the outlet temperature of one (or both) fluid stream (s), or the required area, depending on what is known. For an existing exchanger the fluid-flow rates, inlet and outlet temperatures, and physical data (diameters, etc.) are known or can be measured.

$$\boxed{\begin{aligned} \sum R = {} & \text{Film resistances for the inside tube-surface area} \\ & + \text{Tube} - \text{wall conduction resistance} \\ & + \text{Film resistance for the outside tube} - \text{surface area} \end{aligned}} \qquad (2.31)$$


Figure 2.18 A double-pipe heat exchanger. [6]

Figure 2.18 is a sketch of temperature versus length for two systems: counterflow and parallel flow. As shown, the fluids flow in opposite directions for counterflow, and in the same direction for parallel flow. (Note that the notation for temperature is upper case for the warmer fluid and lower case for the colder fluid.) In both cases the annulus fluid is being heated, and the pipe fluid is being cooled. The "1" subscript refers to an inlet temperature, while "2" refers to an outlet temperature.

The two fluid-stream temperatures vary with distance z within the exchanger. It is usually assumed that all heat lost by the warmer fluid is transferred to the cooler fluid. The heat being transferred thus encounters a tube-film resistance, a tube-wall resistance, and an annulus-film resistance:

Equation 2.31 must contain appropriate areas to evaluate the resistances. The film coefficient that applies to the inside of the inner tube $\hbar_i$ is based on the inside tube-surface area, while the film coefficient $\hbar_0$ is based on the outside surface area of the inner tube, and rewrite the sum of the resistances in terms of an overall heat-transfer coefficient $U_o$ :

$$\frac{1}{U_o} = \frac{1}{\hbar_i\,(A_i/A_0)} + \frac{D_0}{2k}\ln\frac{D_0}{D_i} + \frac{1}{\hbar_o} \qquad (2.32)$$

Where $\mathbf{D_0}$ is the outer diameter (**OD**) of the inner tube, $\mathbf{D_i}$ is the inside diameter of the inner tube, $\mathbf{A_i} = ¶\, D_i\, L$, and $\mathbf{A_o} = ¶\, D_o L$.

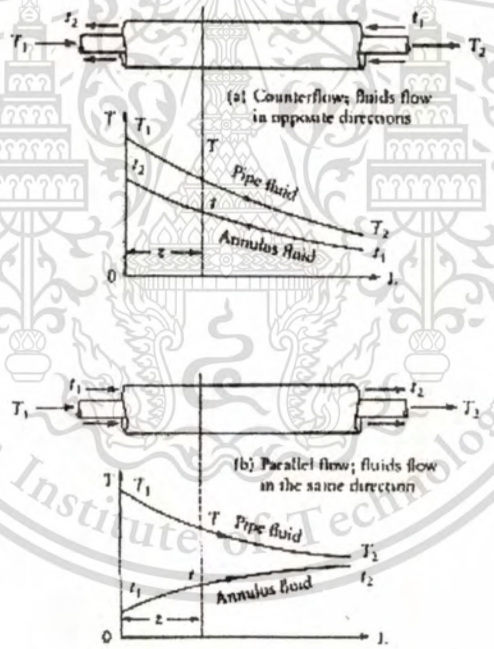



Figure 2.19 Temperature-length relationships for double-pipe heat exchangers. [6]

The pipe itself is made of metal and thus offers a comparatively negligible resistance to heat transfer except in the case of condensation or vaporization. (For the phase-change fluid, the convection coefficient is very high, and the corresponding resistance has a magnitude that is comparable to that of the tube-wall) Neglecting the tube wall resistance, Equation 2.32 can be written as

$$\frac{1}{U_o} = \frac{D_o}{\hbar_i D_i} + \frac{1}{\hbar_o} \tag{2.33}$$

where the overbar on the convection coefficients has been dropped, although it should be remembered that the overall coefficients based on the tube length appear in the equation. We introduce the coefficient $\hbar_{io}$, which is defined as being equal to $\hbar_i D_i / D_o$. This new coefficient in effect is the inside-wall-area convection coefficient referred to the outside-wall-area. The overall coefficient can now de written as

$$\frac{1}{U_o} = \frac{1}{\hbar_{i0}} + \frac{1}{\hbar_o} \tag{2.34}$$

The heat transferred for the entire exchanger then is

$$q = U_0 A_0 \Delta t \tag{2.35}$$

where $\Delta t$ is an as-yet-undetermined temperature difference, which should reflect the fact that we have two inlet and two outlet temperatures. The overall heat-transfer coefficient can be calculated if the film coefficients are known, using empirical correlations.

At any $z$ location within the exchanger (Figure 2.19) the temperature difference is $T - t$. This difference varies throughout the exchanger. It is desirable to determine the overall heat transfer, however, rather than a local value. Further, it is convenient to use the inlet and outlet temperatures again rather than local values because the inlet and outlet temperatures can be easily measured. Thus it is convenient to express the driving potential $\Delta t$ of Equation 2.35 in terms of the inlet and outlet temperatures of both streams rather than a local temperature difference.

Although there will be heat transferred in either parallel or counter flow configurations, the temperature difference is not identical (numerically) for both cases. Beginning with a counterflow arrangement, we will derive the appropriate temperature difference $\Delta t$ under the following assumptions:

1. The overall heat-transfer coefficient $U_o$ (based on the outside surface area of the inner tube $A_0$) is a constant over the length of the exchanger.
2. Steady flow exists.
3. Fluid properties are constant.
4. There are no phase changes in the system.
5. There are no heat losses; that is, all heat transferred from the warmer fluid goes to the cooler fluid.

# CHAPTER 3

# EXPERIMENTAL DETAILS

## 3.1 CHEMICAL

Water

The properties of water were shown in Table 3.1

**TABLE 3.1** Properties of liquid water [7]

| Temperature (K) | Heat Capacity (J/kg.K) | Thermal Conductivity (W/m.K) | Density ($kg/m^3$) |
|---|---|---|---|
| 298 (25°C) | 4177.9 | 0.606 | 996.5 |
| 323 (50°C) | 4183.5 | 0.640 | 988.1 |
| 343 (70°C) | 4213.2 | 0.661 | 978.0 |
| 353 (80°C) | 4236.8 | 0.669 | 60.58 |

## 3.2 APPARATUS

- Fiber glass insulator from Fiber Glass Industrial Limited.
- Heat Exchangers model Ref. BME/3000 by PIGNAT s.a.

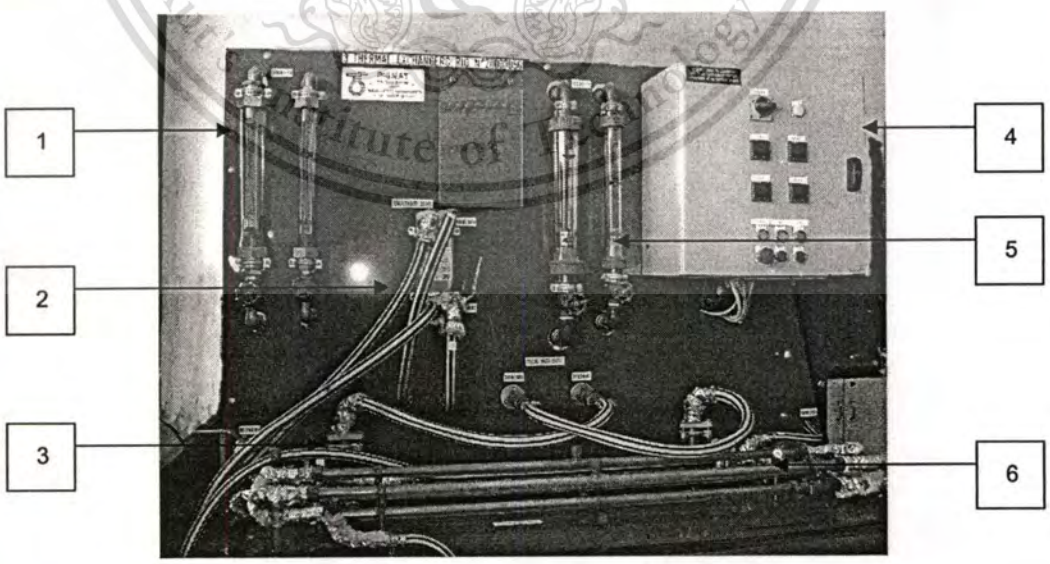



**Figure 3.1** Heat Exchangers model

The component of Heat Exchangers model Ref. BME/3000 by PIGNAT s.a. is consists of: (as shown in Figure 3.1)

1. Flow meter for warm water circuit
2. Plate heat exchanger
3. Shell and tube heat exchanger
4. Controller
5. Flow meter for cold water circuit
6. Monotube heat exchanger

## 3.3 EXPERIMENTAL

### 3.3.1 EXPERIMENTAL PARAMETERS

There were many parameters to study. They were

- Heat exchanger types
- The direction of flow : co-current and countercurrent
- Warm water and cold water flow rate : co-current and counter-current
- Temperature of warm and cold water
- Shielded heat exchanger with fiber glass insulator

### 3.3.2 OPERATION MODE

For investigation of each parameters in heat exchanger types, we did experiment in the same way as follows :

I. Select a heat exchanger (Monotube, Shell and tube, or Plate heat exchanger)
II. Choose the flow direction (co-current or countercurrent) by change direction of inlet cold water line.
III. Switch ON machine on controller, (see Fig 3.2).
IV. Adjust inlet temperature of warm water on controller, (see Fig 3.3).
V. Switch ON pump and heater on controller.
VI. Adjust flow rate of warm water (laminar or turbulent flow) by using valve under flow meter, (see Fig 3.4)
VII. Adjust flow rate of cold water (laminar or turbulent flow) by using valve under flow meter.
VIII. Wait for temperatures stability.

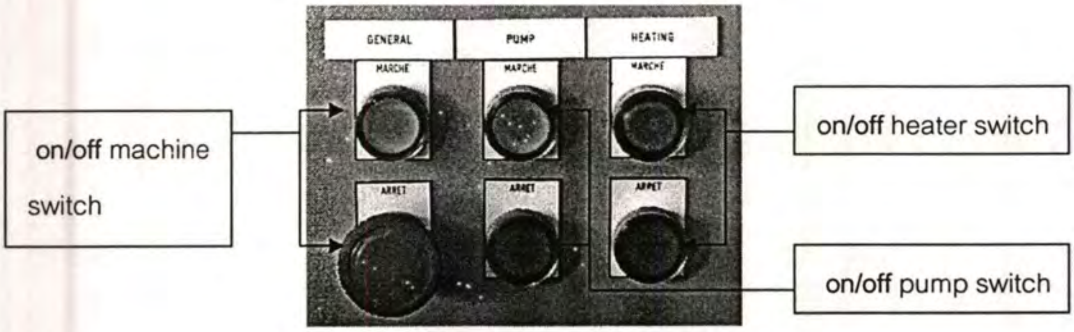



Figure 3.2 An important switch on controller.

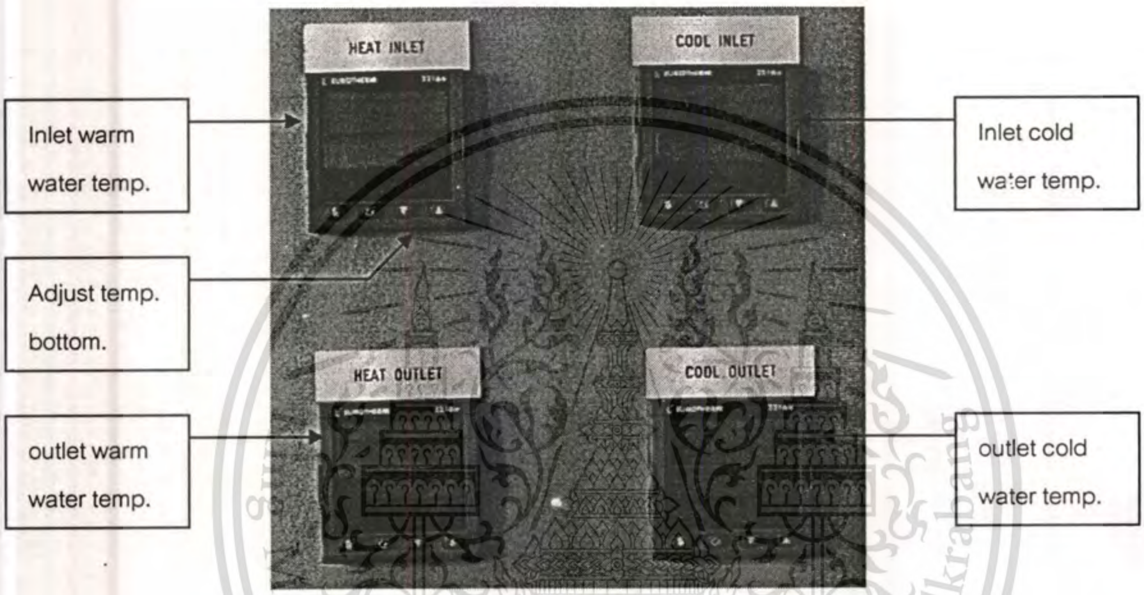
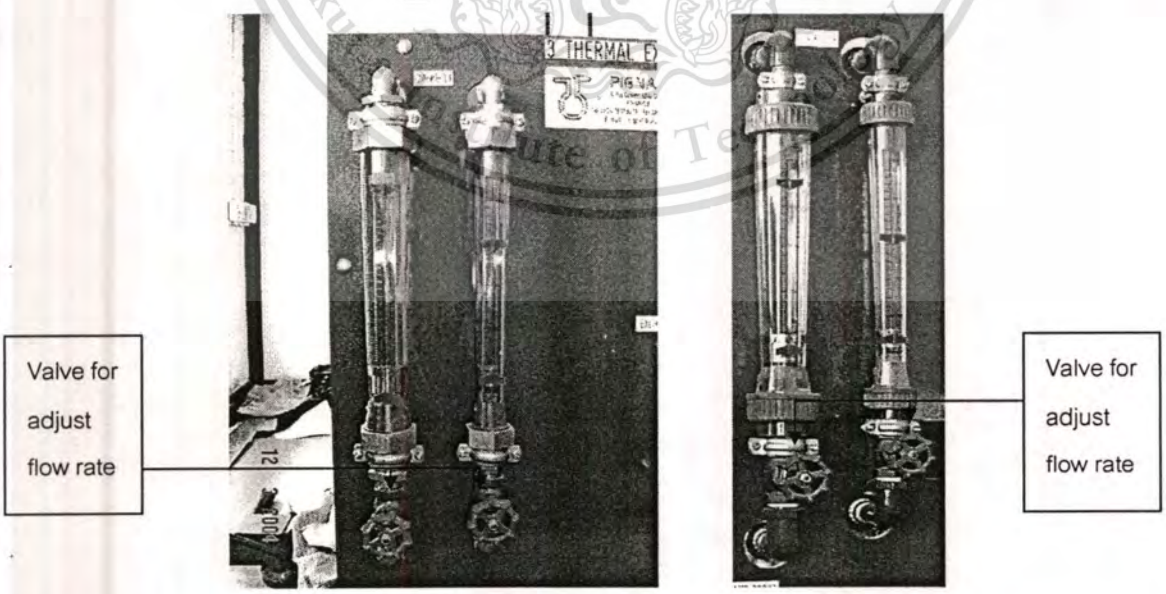



Figure 3.3 Important displays on controller.




(a) (b)

Figure 3.4 a) A flow meter for warm water circuit, b) A flow meter for cold water circuit.

### 3.3.3 EXPERIMENTAL STATEMENT

As soon as temperature was stable:

- Recorded the values of warm water and cold water flow rates from flow meter.
- Recorded temperatures on warm water circuit and cold water circuit from controller. (see Fig 3.3)

Changed flow rate of warm (and/or cold) water, also changed inlet temperature of warm water and waited until stable conditions before further did new experiment.

### 3.3.4 ADDITIONAL EXPERIMENT

Use an insulator (fiber glass type) shielded all valves in warm and cold water circuit, and also heating lost area between system and air could be occurred. Selected the conditions that given the highest heat transfer efficiency from each types of heat exchanger and did an experiment once. When temperature were stable, recorded the values of warm water and cold water flow rates and also recorded temperatures of warm water and cold water circuit. (see Fig 3.5 and Fig 3.6)

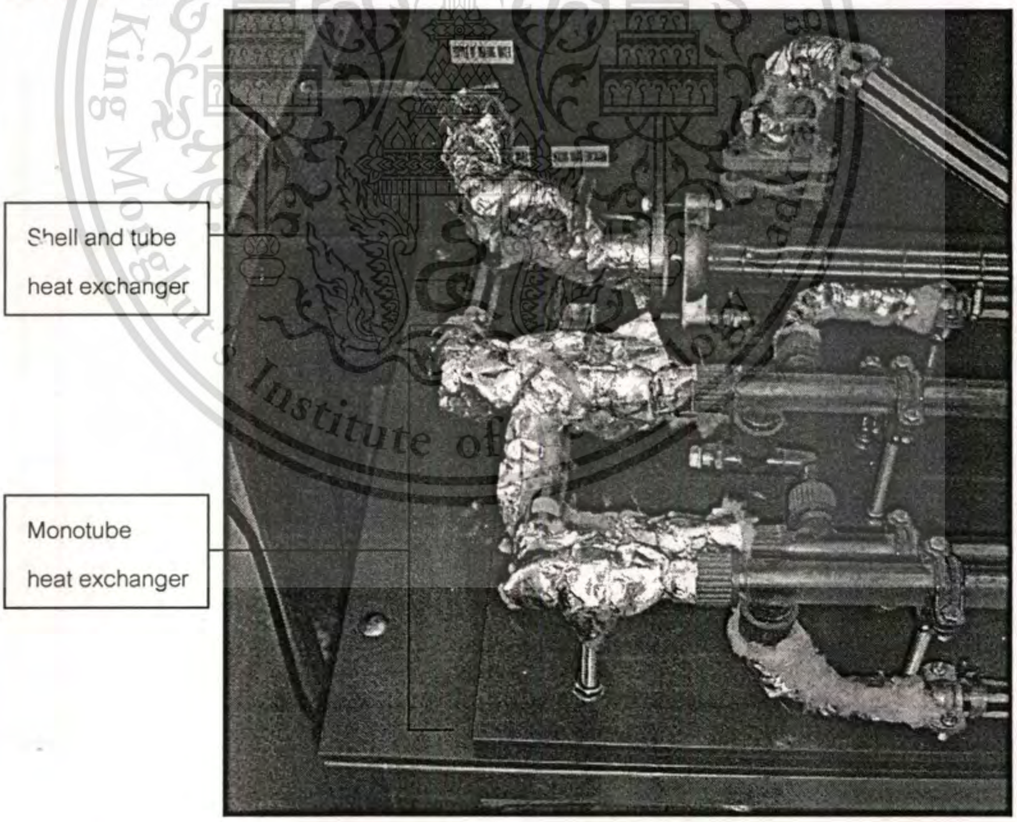



**Figure 3.5** Monotube and shell and tube heat exchanger shielded with fiber glass insulator.

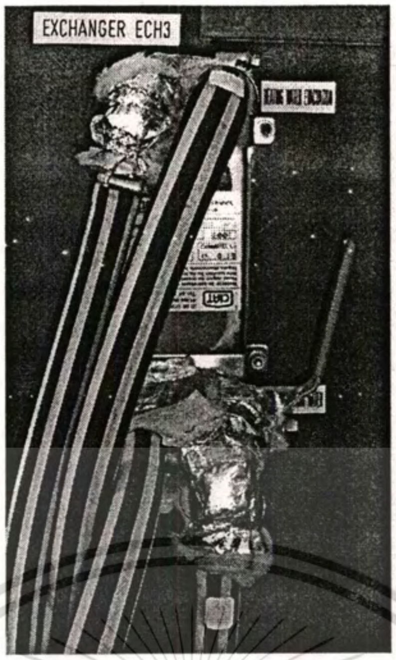



Figure 3.6 Plate heat exchanger shielded with fiber glass insulator.

# CHAPTER 4

# RESULTS AND DISCUSSION

After the appropriated inlet and outlet temperature of warm water and the inlet and outlet temperature of cold water were chosen from the experiment, then we calculated heat exchanger efficiencies of heat transfer ($\eta$) for investigation of parameter for each experiment.

## 4.1 EFFECT OF HEAT EXCHANGER TYPES ON HEAT TRANSFER RATE

Types of heat exchanger were studied and were compared to each other. The results were shown in Figures 4.1. The flow rate temperature and direction of flow of heat exchanger were indicated as follow.

| | |
|---|---|
| Monotube and shell tubes | : cold water = 450 L/h (turbulent) |
| | : warm water = 40.7 L/h (laminar) |
| Plate type | : cold water = 450 L/h (turbulent) |
| | : warm water = 100 L/h (laminar) |
| Temperature | : inlet temperature of warm water = 70°C |
| Direction of flow | : counter current |

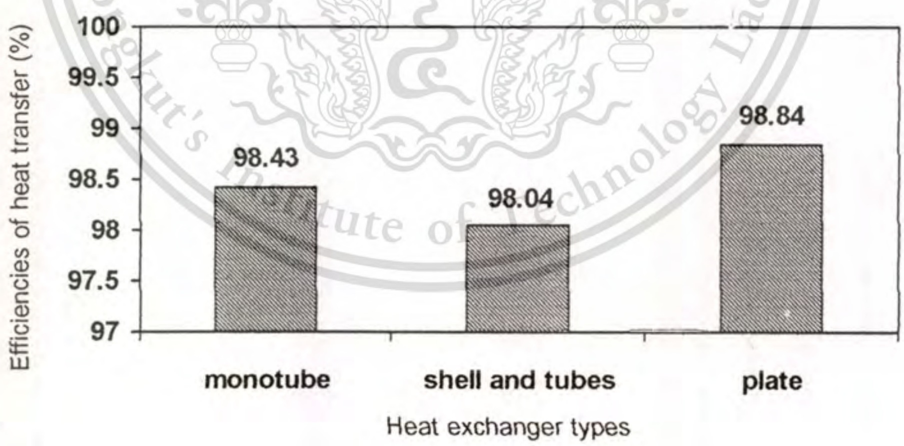



**Figure 4.1** Relationship between efficiencies heat transfer rate and heat exchanger types.

From figure 4.1, it pointed out that efficiency heat transfer rate for each type of heat exchanger was not quite different. This was meant that surface of exchanging area for each type was not so significant.

## 4.2 EFFECT OF DIRECTON OF FLOW AND INLET TEMPEATURE OF WARM WATER ON HEAT TRANSFER RATE

The direction of flow was studied to indicate the flow rate condition and temperature of water for each type of heat exchanger was studied to show the effect on heat transfer rate for each type of heat exchanger. The results were shown in Figure 4.2 to 4.4.

### 4.2.1 MONOTUBE HEAT EXCHANGER

Cold water flow rate was turbulent (450 L/h) and warm water flow rate was laminar (40.7 L/h). The results were shown in Figure 4.2

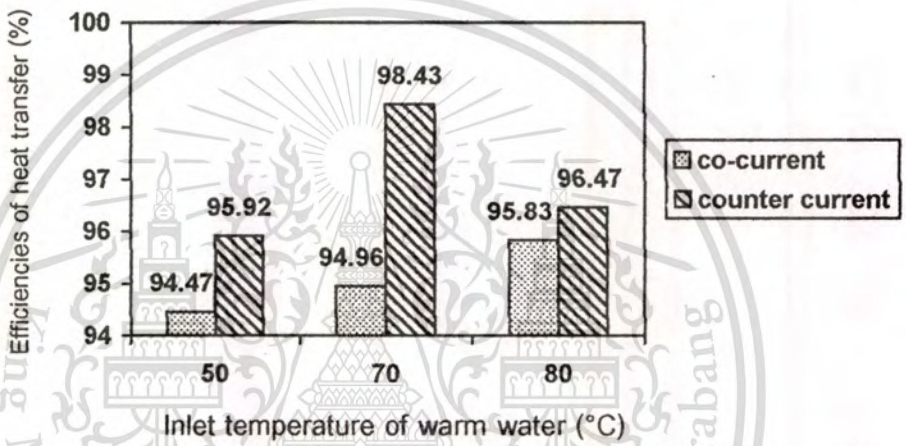



**Figure** 4.2 Relationship between efficiencies heat transfer rate and inlet temperature of warm water of monotube heat exchanger.

From Figure 4.2, the efficiency of heat transfer rate of warm water was so significant if the direction of flow was counter-current.

4.2.2 SHELL AND TUBES HEAT EXCHANGER

Cold water flow rate was turbulent (450 L/h) and warm water flow rate was laminar (40.7 L/h). The results were shown in Figure 4.3

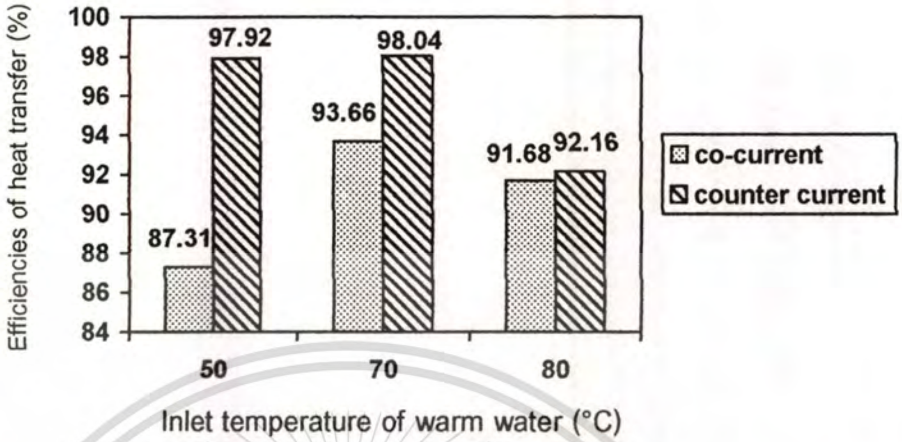



**Figure 4.3** Relationship between efficiencies of heat transfer rate and inlet temperature of warm water of shell and tubes heat exchanger.

From Figure 4.3 the efficiencies of heat transfer of warm water were still dominant if the direction of flow was counter-current.

4.2.3 PLATE HEAT EXCHANGER

Cold water flow rate was turbulent (850 L/h) and warm water flow rate was laminar (100 L/h). The results were shown in Figure 4.4.

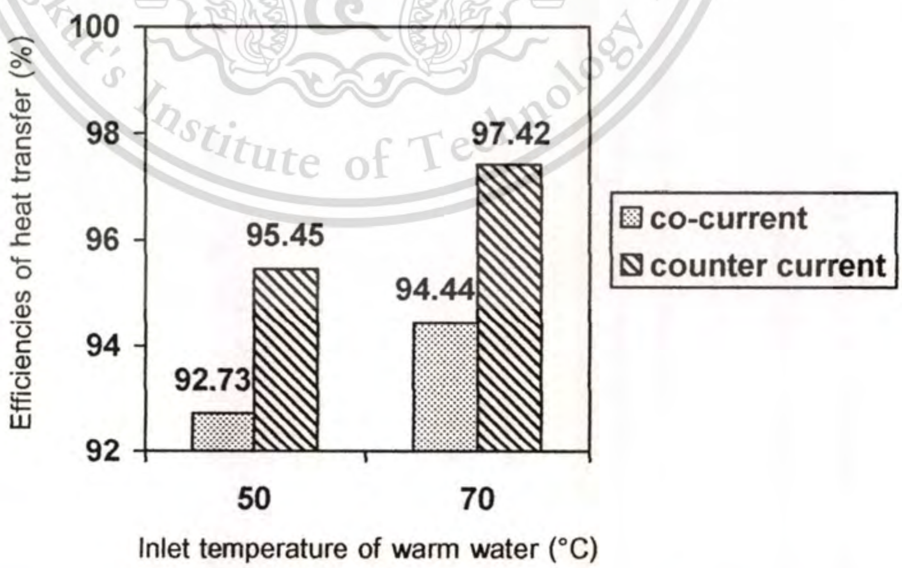



**Figure 4.4** Relationship between efficiencies of heat transfer rate and inlet temperature of warm water of plate heat exchanger.

From Figure 4.4 the efficiency of heat transfer of warm water was so important if the direction of flow was counter-current. This could be the same as monotube and shell and tubes heat exchanger characteristics.

Figure 4.2 to 4.4 were shown the effect of flow direction at any temperatures. It indicated that counter-current had higher efficiencies of heat transfer than co-current flow. This due to the distance between entrance and exit temperature of heat exchanger dominated in counter-current direction. In monotube and shell and tubes heat exchanger, the efficiencies of heat transfer rate were decreased at the inlet temperature of 80°C. It was meant that the temperature of warm water was increased as the volume of cold water per hour was constant. Therefore, heat of warm water could not transfer to cold water whereas inlet temperature was low.

4.3 EFFECT OF FLOW RATE ON HEAT TRANSFER RATE

The flow rate is the other parameter that is so important to heat transfer efficiency. The results were shown in Figure 4.5 to 4.7.

4.3.1 MONOTUBE HEAT EXCHANGER

The conditions for monotube heat exchanger operations were as follows,

L/L was warm water with laminar flow (40.7 L/h) and cold water with laminar flow (127.2 L/h).

L/T was warm water with laminar flow (40.7 L/h) and cold water with turbulent flow (450 L/h).

T/L was warm water with turbulent flow (300 L/h) and cold water with laminar flow (170 L/h).

T/T was warm water with turbulent flow (300L/h) and cold water with turbulent flow (450L/h).

The direction of flow was counter-current and the inlet temperature of warm water was 70°C.

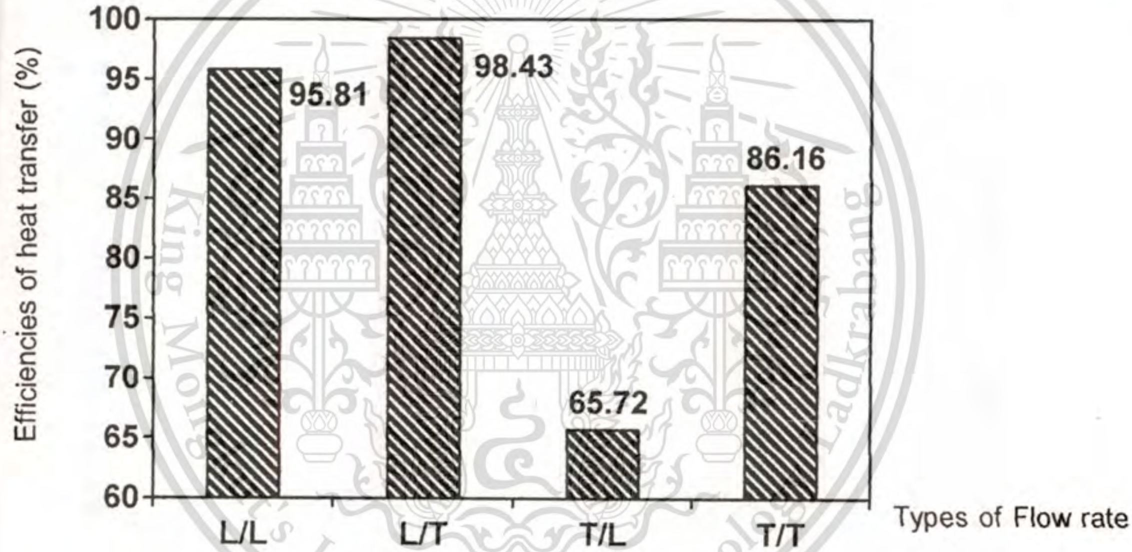



Figure 4.5 Relationship between efficiencies of heat transfer rate and types flow rate of warm water of monotube heat exchanger.

From Figure 4.5, the L/T showed the most significant heat transfer's efficiency as the T/L indicated the lowest one. It pointed out that the turbulent of cold water was dominated more than laminar of cold water. Because the turbulent of cold water could be exchanged of heat transfer to laminar flow of warm water.

### 4.3.2 SHELL AND TUBE HEAT EXCHANGER

The conditions for shell and tube heat exchanger operations were as follows :

L/L was warm water with laminar flow (40.7L/h) and cold water with laminar flow (170L/h).

L/T was warm water with laminar flow (40.7L/h) and cold water with turbulent flow (450L/h).

T/L was warm water with turbulent flow (500L/h) and cold water with laminar flow (170L/h).

T/T was warm water with turbulent flow (500L/h) and cold water with turbulent flow (550L/h).

The direction of flow was counter-current and the inlet temperature of warm water was 70 °C.

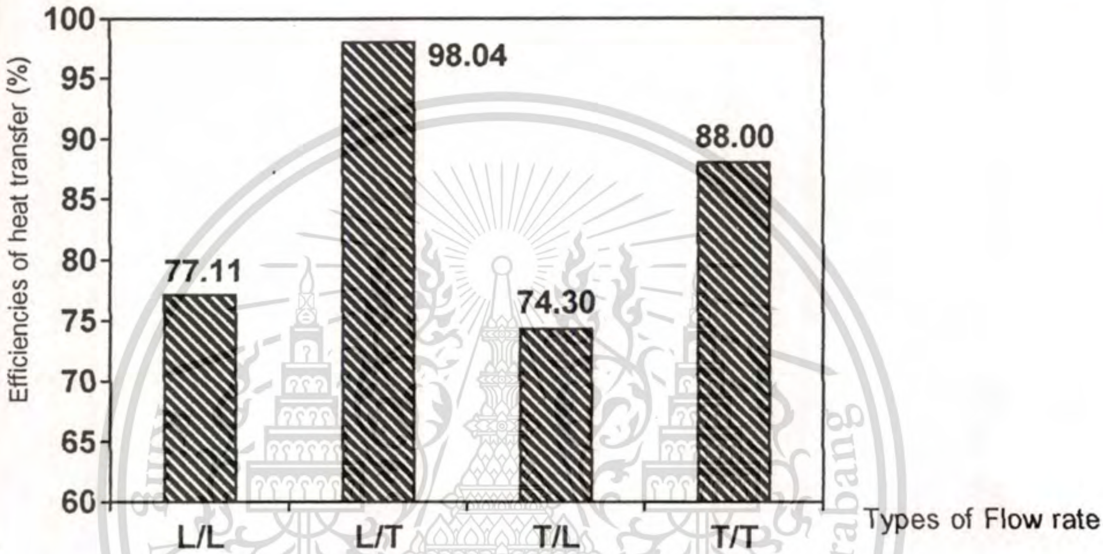



Figure 4.6 Relationship between efficiencies of heat transfer rate and flow rate of warm water of shell and tubes heat exchanger.

From Figure 4.6 the L/T was also shown the most significant heat transfer efficiency. The heat transfer efficiency was quite similar to monotube heat exchanger. Because the interchange between surface areas of monotube (0.113 $m^2$) was close to shell and tube (0.110 $m^2$) heat exchanger.

4.3.3 PLATE HEAT EXCHANGER

The conditions for plate heat exchanger operations were as follows :

L/L was warm water with laminar flow (100 L/h) and cold water with laminar flow (300 L/h).

L/T was warm water with laminar flow (100 L/h) and cold water with turbulent flow (900 L/h).

T/L was warm water with turbulent flow (900 L/h) and cold water with laminar flow (300 L/h).

T/T was warm water with turbulent flow (900L/h) and cold water with turbulent flow (850L/h).

The direction of flow was counter-current and the inlet temperature of warm water was 70 °C.

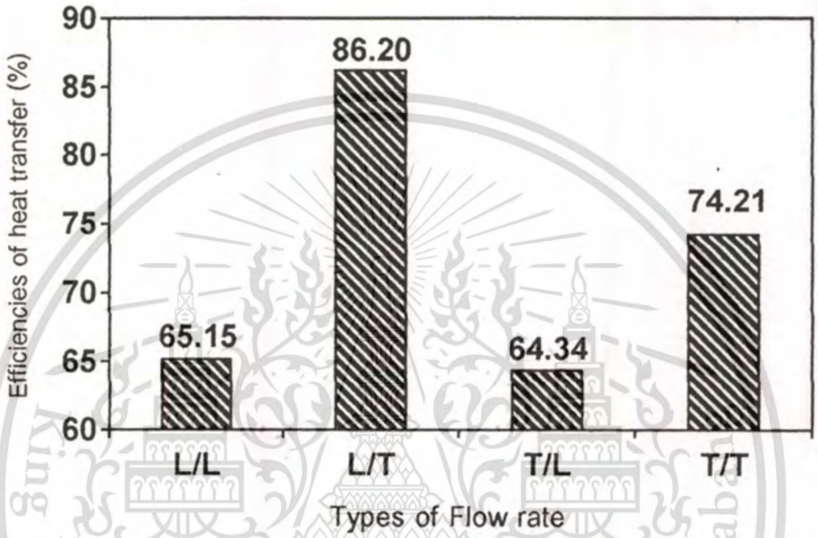



Figure 4.7 Relationship between efficiencies of heat transfer rate and flow rate of warm water of plate heat exchanger.

From Figure 4.7 the L/T was still shown the most significant heat exchanger. The heat transfer efficiency was quit less than monotube and shell and tube heat exchanger. This effect was due to higher flow rate of warm water and cold water even though the plate heat exchanger had the highest surface area (0.140 $m^2$).

## 4.4 EFFECT OF FIBER GLASS INSULATOR ON HEAT TRANSFER RATE

Fiber glass is a popular material for insulation. It is widely used in heat exchanger. In this research, it was used to shield heat exchanger for protection heat loss. The results were shown in Figure 4.8 to 4.10.

### 4.4.1 MONOTUBE HEAT EXCHANGER

Co-current flow had inlet temperature of warm water 50 °C and the flow rate of warm water laminar 33.9 L/h and the flow rate of cold water turbulent 500 L/h. As the counter-current had inlet temperature of warm water 70 °C and the flow rate of warm water laminar 40.7 L/h and the flow rate of cold water turbulent 450 L/h.

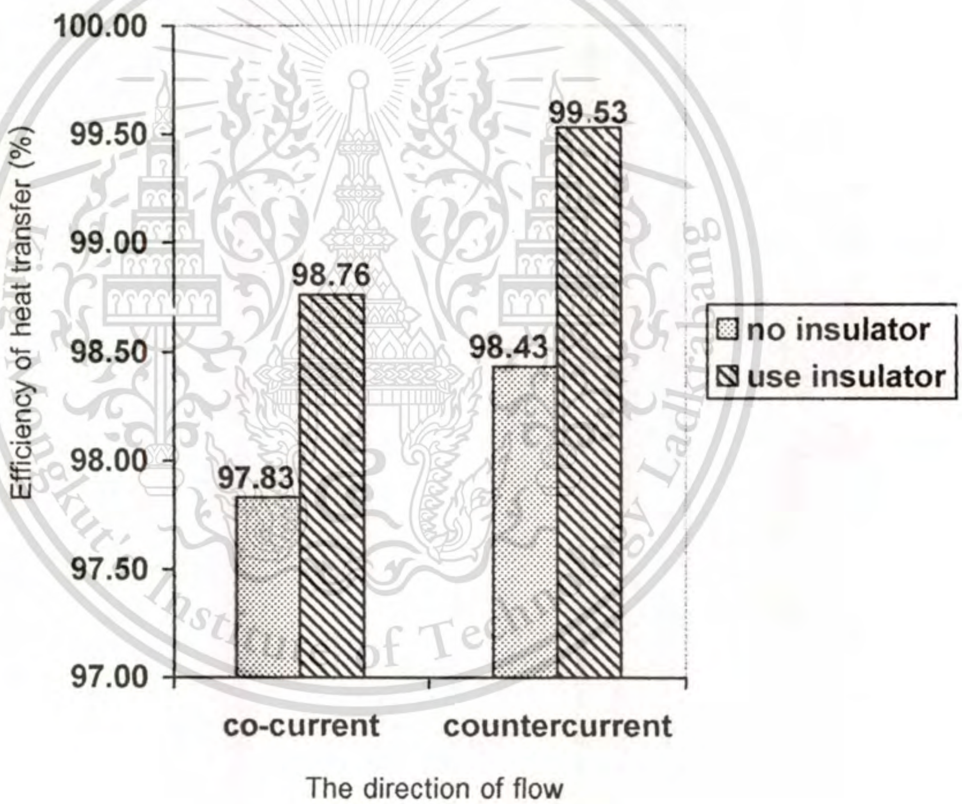



**Figure 4.8** Relationship between efficiencies of heat transfer rate and the direction of flow of monotube heat exchanger when shielded with fiber glass insulator.

From Figure 4.8 the fiber glass was so less significant to monotube heat exchanger. The efficiency of heat transfer was only increased to 0.95% for co-current and 1.11% for counter-current flow.

4.4.2 SHELL AND TUBES HEAT EXCHANGER

Co-current flow had inlet temperature of warm water 80 °C and the flow rate of warm water laminar 33.9 L/h and the flow rate of cold water turbulent 450 L/h. As the counter-current had inlet temperature of warm water 80 °C and the flow rate of warm water laminar 33.9 L/h and the flow rate of cold water turbulent 750 L/h.

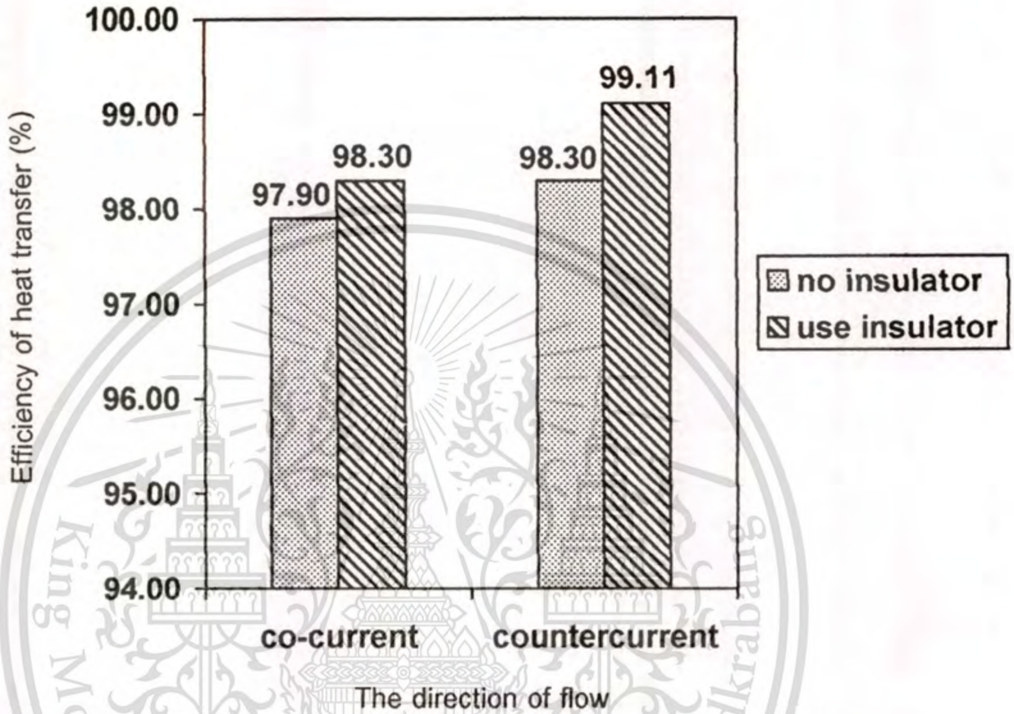



Figure 4.9 Relationship between efficiencies of heat transfer rate and the direction of flow of shell and tubes heat exchanger when shielded with fiber glass insulator.

From Figure 4.9 the fiber glass was so less significant to shell and tubes heat exchanger. The efficiency of heat transfer was only increased to 0.40% for co-current and 0.80% for counter-current flow.

### 4.4.3 PLATE HEAT EXCHANGER

Co-current flow had inlet temperature of warm water 70°C and the flow rate of warm water laminar 100 L/h and the flow rate of cold water turbulent 850 L/h. As the counter-current had inlet temperature of warm water 70°C and the flow rate of warm water laminar 100 L/h and the flow rate of cold water turbulent 850 L/h.

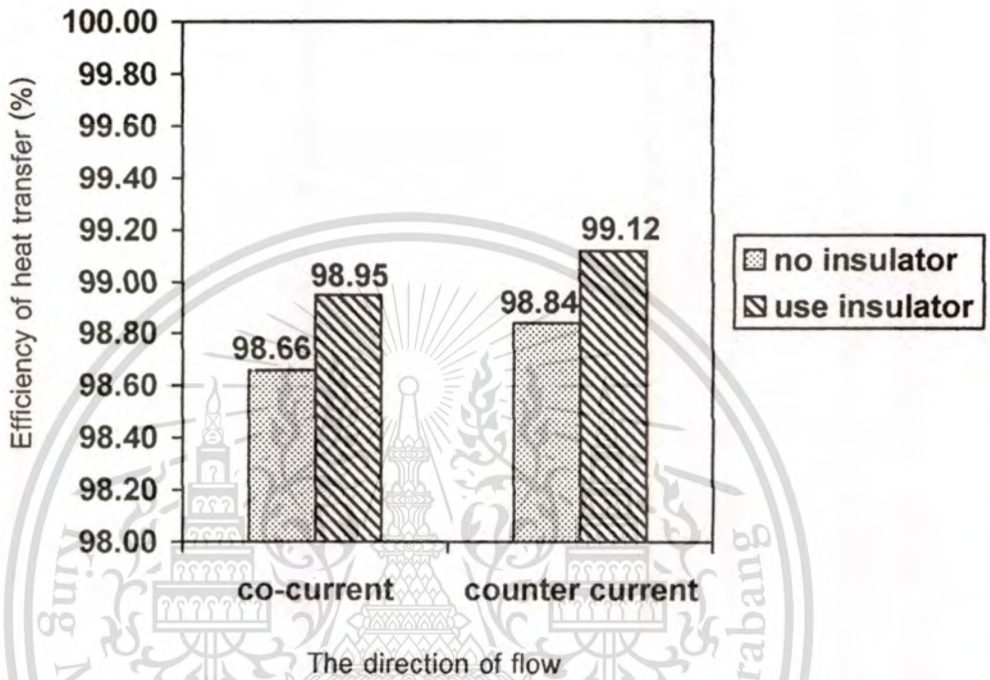



Figure 4.10 Relationship between efficiencies of heat transfer rate of plate heat exchanger when shielded with fiber glass insulator.

From Figure 4.10 the fiber glass was less significant to plate heat exchanger. The efficiency of heat exchanger was only increased to 0.30% for co-current and 0.28% for counter-current flow.

From Figure 4.8 to Figure 4.10, the fiber glass had no effect to any types of heat exchanger. Because the shielded areas with fiber glass were so small. Therefore, there was less heat transfer in particular area.

## 4.5 SUMMARIES OF THE NOTES OF CALCULATION

The effect of heat exchange types, direction of flow and inlet temperature, flow rate and fiber glass as an insulator were calculated for heat transfer as shown in Table 4.1 to 4.6 and appendix A and B.

### 4.5.1 MONOTUBE HEAT EXCHANGER

**TABLE 4.1** Calculation of counter-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | 0.113 $m^2$ | 0.113 $m^2$ |
| Flow of fluids | counter-current | counter-current |
| Flow rate of warm water ($Q_{vc}$) | 40.7 L/h | 40.7 L/h |
| Flow rate of cold water ($Q_{vf}$) | 450 L/h | 450 L/h |
| Efficiencies of heat exchanger ($\eta$) | 98.43% | 99.53% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 977.93 | 1068.47 |
| Inlet warm water temperature | 70 °C | 70 °C |

From Table 4.1, it pointed out that the fiber glass insulator was so less significant to monotube heat exchanger for counter-current flow. The efficiency of heat transfer was increased only 1.1%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to 90.54$kW.m^{-2}.K^{-1}$. Because the fiber glass insulator was protected heating loss between warm water and air so the ($U_{experiment}$) of heat transfer was increased, but it was insignificant according to small surface area for shielding protection.

TABLE 4.2 Calculation of co-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | 0.113 $m^2$ | 0.113 $m^2$ |
| Flow of fluids | co-current | co-current |
| Flow rate of warm water ($Q_{vc}$) | 33.9 L/h | 33.9 L/h |
| Flow rate of cold water ($Q_{vf}$) | 500 L/h | 500 L/h |
| Efficiencies of heat exchanger ($\eta$) | 97.83% | 98.76% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 781.34 | 810.74 |
| Inlet warm water temperature | 50°C | 50°C |

From Table 4.2, the fiber glass was also less significant to monotube heat exchanger for co-current flow. The efficiency of heat transfer was increased only 0.93%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to 29.4 $kW.m^{-2}.K^{-1}$. Because the fiber glass insulator was protected heating loss between warm water and air so the ($U_{experiment}$) of heat transfer was increased, but it was insignificant according to small surface area for shielding protection.

### 4.5.2 SHELL AND TUBE HEAT EXCHANGER

TABLE 4.3 Calculation of counter-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | 0.110 $m^2$ | 0.110 $m^2$ |
| Flow of fluids | counter-current | Counter-current |
| Flow rate of warm water ($Q_{vo}$) | 33.9 L/h | 333.9 L/h |
| Flow rate of cold water ($Q_{vf}$) | 750 L/h | 750 L/h |
| Efficiencies of heat exchanger ($\eta$) | 98.30% | 99.11% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 530.68 | 586.97 |
| Inlet warm water temperature | 80 °C | 80 °C |

From Table 4.3, the fiber glass was so less significant to shell and tube heat exchanger counter-current flow. The efficiency of heat transfer was increased only 0.81%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to 56.29 $kW.m^{-2}.K^{-1}$. Because the fiber glass insulator was protected heating loss between warm water and air so the ($U_{experiment}$) of heat transfer was increased, but it was insignificant according to small surface area for shielding protection.

TABLE 4.4 Calculation of co-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | 0.110 $m^2$ | 0.110 $m^2$ |
| Flow of fluids | co-current | co-current |
| Flow rate of warm water ($Q_{vc}$) | 33.9 L/h | 33.9 L/h |
| Flow rate of cold water ($Q_{vf}$) | 450 L/h | 450 L/h |
| Efficiencies of heat exchanger ($\eta$) | 97.90% | 98.30% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 437.61 | 470.07 |
| Inlet warm water temperature | 80 °C | 80 °C |

From Table 4.4, the fiber glass was still less significant to shell and tube heat exchanger co-current flow. The efficiency of heat transfer was increased only 0.40%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to 32.46 $kW.m^{-2}.K^{-1}$. Because the fiber glass insulator was protected heating loss between warm water and air so the ($U_{experiment}$) of heat transfer was increased, but it was insignificant according to small surface area for shielding protection.

### 4.5.3 PLATE HEAT EXCHANGER

TABLE 4.5 Calculation of counter-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | $0.140\ m^2$ | $0.140\ m^2$ |
| Flow of fluids | counter-current | counter-current |
| Flow rate of warm water ($Q_{vc}$) | 100 L/h | 100 L/h |
| Flow rate of cold water ($Q_{vf}$) | 850 L/h | 850 L/h |
| Efficiencies of heat exchanger ($\eta$) | 98.84% | 99.12% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 1145.61 | 1474.40 |
| Inlet warm water temperature | $70^{\circ}C$ | $70^{\circ}C$ |

From Table 4.5, the fiber glass was less significant to plate heat exchanger for flow was counter-current flow. The efficiency of heat transfer was increased only 0.28%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to 328.79 $kW.m^{-2}.K^{-1}$. Because the fiber glass insulator was protected heating loss between warm water and air so the ($U_{experiment}$) of heat transfer was increased, but it was insignificant according to small surface area for shielding protection.

TABLE 4.6 Calculation of co-current

| | NO INSULATOR | USED INSULATOR |
|---|---|---|
| Surface of exchange | 0.140 $m^2$ | 0.140 $m^2$ |
| Flow of fluids | co-current | co-current |
| Flow rate of warm water ($Q_{vc}$) | 100 L/h | 100 L/h |
| Flow rate of cold water ($Q_{vf}$) | 850 L/h | 850 L/h |
| Efficiencies of heat exchanger ($\eta$) | 98.66% | 98.95% |
| $U_{experiment}$ ($kW.m^{-2}.K^{-1}$) | 1336.37 | 1351.84 |
| Inlet warm water temperature | 70°C | 70°C |

From Table 4.6, the fiber glass was less significant to plate heat exchanger for co-current flow. The efficiency of heat transfer was increased only 0.29%. Similarly, the overall heat transfer coefficient of experimental evaluation ($U_{experiment}$) was increased to15.47 $kW.m^{-2}.K^{-1}$

From Table 4.1 to 4.6 we got the same results of ($U_{experiment}$) for counter-current and co-current but the counter-current was more significant than co-current because $U_{experiment}$ was depended on temperature difference between the warm fluid and cold fluid.

In counter-current mode, this difference was completely heat exchanger constant. It implied that overall thermal exchange was completely constant.

In co-current mode, temperature difference was decreased. It implied that thermal exchange was less transfer. It evolution of the gradient of temperature allowed to show counter-current, it would be more effective than the co-current flow according to any point of heat exchanger, the gradient temperature was constant. Therefore, $U_{experiment}$ of counter-current was more significant than co-current.

# CHAPTER 5
# CONCLUSION AND SUGGESTION

## 5.1 CONCLUSION

From the experiment we could be concluded that the efficiencies of heat transfer were depend on the types of heat exchanger, the direction of water flow, flow rate of water and inlet temperature of warm water. Effect of heat exchanger types on heat transfer rate was depending on interchange between surface area. This led to enhancement of heat transfer rate from warm liquid to cold liquid.

The direction of flow (co-current and counter current) of two liquids had an important role in thermal exchanges. Indeed, the vector of the exchanger was the distance from temperature between warm liquid to cold liquid.

According to the profiles of temperatures, the distance from temperature was more important to counter-current flow than co-current flow.

The flow rate (laminar and turbulent flow) of two liquids also had effect in thermal exchanges. The flow rate of warm water should be laminar flow (low flow rate) and cold water should be turbulent (high flow rate). Because these conditions were given appropriate times for heat transfer from warm water to cold water. If the flow rate of warm water was high and cold water was low, the efficiencies of heat transfer were decreased because warm water had less time to transfer heat to cold water.

The last effect on heat transfer rate was the inlet temperature of warm water. When the inlet temperature of warm water was increased but the flow rate of cold water was constant, the efficiencies of heat transfer were decreased because the volumes of cold water per unit time were constant. Therefore heat from warm water could not transfer to cold water as well as the low inlet temperature.

For improving the efficiencies of heat exchanger, the fiber glass insulator was used to shield heating loss areas between system and air around the machines. According to fiber glass insulator was protected heating loss between warm water and air so the efficiencies of heat transfer were increased, but it was insignificant. Because it had less surface area for shielding protection.

## 5.2 RECOMMENDATION

i. Should be using electronic flow meter which can control via computer for improving the accurate measurement of flow rate.

ii. Should be using the electronic valves which can control via computer for improving an adjustment of water flow rate.

iii. Should be using water tank for storage and circulation of cold water system for water saving.

iv. Should be using water pump in cold water circuit for increasing water flow rate.

v. Should be studying an effect of extended surfaces of heat exchanger on heat transfer rate.

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# APPENDIX A

# EXPERIMENTAL RESULTS

<u>Monotube</u> (Co-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 50.1 | 34.3 | 28.8 | 32.2 | 80.72 |
| 33.9 | 152.6 | 50.1 | 33.9 | 28.8 | 31.9 | 86.11 |
| 33.9 | 170.0 | 50.0 | 33.6 | 28.8 | 31.7 | 88.64 |
| 40.7 | 127.2 | 50.1 | 35.2 | 28.9 | 33.5 | 96.51 |
| 40.7 | 152.6 | 50.2 | 34.8 | 28.9 | 32.8 | 94.97 |
| 40.7 | 170.0 | 50.1 | 34.1 | 28.9 | 32.1 | 83.55 |
| 45.2 | 127.2 | 50.2 | 37.3 | 30.7 | 34.9 | 91.62 |
| 45.2 | 152.6 | 50.1 | 36.1 | 30.7 | 34.2 | 84.40 |
| 45.2 | 170.0 | 50.2 | 35.5 | 30.7 | 33.7 | 76.75 |

<u>Note</u>

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 49.9 | 41.4 | 26.4 | 36.4 | 74.82 |
| 200 | 152.6 | 49.8 | 40.3 | 26.4 | 35.9 | 76.30 |
| 200 | 170.0 | 50.2 | 39.2 | 26.4 | 34.9 | 65.68 |
| 300 | 127.2 | 49.9 | 43.5 | 26.4 | 38.4 | 79.50 |
| 300 | 152.6 | 50.1 | 42.4 | 26.4 | 37.2 | 71.34 |
| 300 | 170.0 | 50.1 | 34.1 | 28.9 | 32.1 | 71.87 |
| 400 | 127.2 | 50.1 | 44.5 | 26.4 | 39.0 | 71.55 |
| 400 | 152.6 | 49.9 | 43.8 | 26.4 | 38.2 | 73.79 |
| 400 | 170.0 | 50.2 | 43.0 | 26.4 | 37.6 | 66.11 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Laminar flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 70.2 | 40.1 | 29.0 | 36.5 | 93.46 |
| 33.9 | 152.6 | 70.2 | 38.9 | 29.0 | 35.7 | 96.32 |
| 33.9 | 170.0 | 70.1 | 38.3 | 28.9 | 35.0 | 96.16 |
| 40.7 | 127.2 | 70.2 | 42.0 | 28.9 | 37.5 | 95.33 |
| 40.7 | 152.6 | 70.1 | 41.3 | 28.9 | 36.3 | 96.36 |
| 40.7 | 170.0 | 70.3 | 39.7 | 28.9 | 35.8 | 94.21 |
| 45.2 | 127.2 | 70.1 | 42.4 | 30.6 | 39.8 | 93.46 |
| 45.2 | 152.6 | 70.2 | 41.9 | 30.6 | 38.4 | 93.05 |
| 45.2 | 170.0 | 70.1 | 40.3 | 30.6 | 37.4 | 85.82 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 70.1 | 54.0 | 26.6 | 46.8 | 79.79 |
| 200 | 152.6 | 69.8 | 52.6 | 26.6 | 45.2 | 82.51 |
| 200 | 170.0 | 70.5 | 50.3 | 26.5 | 43.4 | 71.11 |
| 300 | 127.2 | 69.7 | 57.8 | 26.6 | 49.6 | 81.94 |
| 300 | 152.6 | 69.9 | 55.9 | 26.6 | 47.5 | 75.93 |
| 300 | 170.0 | 70.1 | 53.6 | 26.6 | 45.3 | 64.22 |
| 400 | 127.2 | 70.1 | 60.2 | 26.6 | 51.4 | 79.66 |
| 400 | 152.6 | 70.2 | 58.8 | 26.7 | 49.6 | 76.63 |
| 400 | 170.0 | 70.1 | 57.3 | 26.7 | 48.3 | 71.72 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Laminar flow of warm water ($80^{\circ}C$) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 80.1 | 42.4 | 29.0 | 38.3 | 92.53 |
| 33.9 | 152.6 | 80.2 | 41.6 | 29.0 | 37.0 | 93.26 |
| 33.9 | 170.0 | 80.1 | 40.6 | 28.9 | 36.4 | 95.18 |
| 40.7 | 127.2 | 80.1 | 42.8 | 29.0 | 39.2 | 85.48 |
| 40.7 | 152.6 | 80.0 | 42.3 | 29.0 | 38.3 | 32.51 |
| 40.7 | 170.0 | 80.1 | 41.6 | 29.0 | 37.5 | 92.24 |
| 45.2 | 127.2 | 80.1 | 43.2 | 29.0 | 39.7 | 81.60 |
| 45.2 | 152.6 | 80.0 | 42.6 | 29.0 | 38.9 | 89.36 |
| 45.2 | 170.0 | 80.1 | 41.6 | 29.0 | 37.7 | 84.99 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 80.3 | 60.0 | 26.4 | 51.3 | 78.01 |
| 200 | 152.6 | 80.4 | 58.1 | 26.4 | 50.2 | 81.43 |
| 200 | 170.0 | 80.3 | 56.3 | 26.4 | 48.5 | 78.27 |
| 300 | 127.2 | 80.2 | 63.7 | 26.5 | 53.8 | 70.15 |
| 300 | 152.6 | 80.4 | 60.1 | 26.5 | 52.7 | 65.65 |
| 300 | 170.0 | 80.3 | 58.4 | 26.5 | 50.1 | 61.06 |
| 400 | 127.2 | 80.2 | 66.4 | 26.5 | 55.4 | 66.59 |
| 400 | 152.6 | 80.3 | 62.4 | 26.5 | 53.6 | 57.75 |
| 400 | 170.0 | 80.3 | 60.4 | 26.5 | 52.5 | 55.52 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Laminar flow of warm water ($50^{\circ}C$) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 50.6 | 29.9 | 26.2 | 27.7 | 96.16 |
| 33.9 | 475 | 50.6 | 29.7 | 26.2 | 27.6 | 93.83 |
| 33.9 | 500 | 50.8 | 29.7 | 26.1 | 27.5 | 97.83 |
| 40.7 | 450 | 50.1 | 30.2 | 26.2 | 27.9 | 94.47 |
| 40.7 | 475 | 50.3 | 30.0 | 26.2 | 27.9 | 92.61 |
| 40.7 | 500 | 50.2 | 29.8 | 26.3 | 27.9 | 96.37 |
| 45.2 | 450 | 50.5 | 30.4 | 26.2 | 28.1 | 94.10 |
| 45.2 | 475 | 50.5 | 30.2 | 26.2 | 28.0 | 93.18 |
| 45.2 | 500 | 50.6 | 30.0 | 26.2 | 27.9 | 91.28 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 49.9 | 35.9 | 27.0 | 31.9 | 78.75 |
| 200 | 475 | 50.1 | 35.3 | 27.1 | 31.6 | 72.21 |
| 200 | 500 | 50.1 | 35.2 | 27.1 | 31.6 | 75.50 |
| 300 | 450 | 49.9 | 38.4 | 27.2 | 33.6 | 83.47 |
| 300 | 475 | 50.1 | 38.5 | 27.2 | 33.5 | 85.99 |
| 300 | 500 | 50.0 | 38.2 | 27.2 | 33.2 | 84.74 |
| 400 | 450 | 49.9 | 40.1 | 27.2 | 34.3 | 81.50 |
| 400 | 475 | 49.9 | 39.9 | 27.2 | 34.1 | 81.93 |
| 400 | 500 | 49.9 | 39.7 | 27.2 | 34.0 | 83.33 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 70.5 | 33.2 | 26.7 | 29.4 | 96.05 |
| 33.9 | 475 | 69.9 | 33.0 | 26.9 | 29.4 | 94.90 |
| 33.9 | 500 | 70.1 | 32.8 | 26.8 | 29.2 | 94.87 |
| 40.7 | 450 | 70.1 | 34.0 | 26.7 | 29.8 | 94.96 |
| 40.7 | 475 | 70.2 | 33.9 | 26.9 | 29.9 | 91.39 |
| 40.7 | 500 | 70.1 | 33.5 | 26.8 | 29.6 | 94.00 |
| 45.2 | 450 | 70.2 | 34.5 | 26.6 | 30.1 | 97.60 |
| 45.2 | 475 | 70.1 | 34.0 | 26.8 | 30.1 | 96.06 |
| 45.2 | 500 | 70.5 | 33.8 | 26.9 | 30.1 | 96.45 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water ($70^{\circ}C$) and Turbulent of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 69.9 | 44.2 | 27.2 | 37.2 | 87.54 |
| 200 | 475 | 70.0 | 44.0 | 27.2 | 36.8 | 87.69 |
| 200 | 500 | 70.1 | 43.3 | 27.2 | 36.6 | 87.68 |
| 300 | 450 | 70.2 | 48.7 | 27.2 | 39.5 | 85.81 |
| 300 | 475 | 69.9 | 48.2 | 27.2 | 39.5 | 89.74 |
| 300 | 500 | 69.7 | 47.9 | 27.2 | 39.1 | 90.97 |
| 400 | 450 | 70.2 | 51.1 | 26.0 | 40.7 | 86.58 |
| 400 | 475 | 70.1 | 50.7 | 26.0 | 40.3 | 87.53 |
| 400 | 500 | 70.1 | 50.4 | 26.0 | 40.1 | 89.46 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 80.2 | 34.7 | 26.9 | 30.2 | 96.24 |
| 33.9 | 475 | 80.4 | 34.5 | 27.0 | 30.1 | 94.60 |
| 33.9 | 500 | 80.3 | 34.3 | 26.9 | 29.9 | 96.16 |
| 40.7 | 450 | 79.9 | 36.0 | 27.1 | 30.9 | 95.83 |
| 40.7 | 475 | 80.2 | 35.7 | 27.1 | 30.8 | 92.05 |
| 40.7 | 500 | 80.3 | 35.4 | 27.1 | 30.6 | 95.78 |
| 45.2 | 450 | 80.1 | 36.4 | 27.0 | 31.2 | 95.79 |
| 45.2 | 475 | 80.0 | 36.0 | 26.9 | 30.9 | 95.64 |
| 45.2 | 500 | 80.3 | 35.7 | 26.9 | 30.8 | 96.83 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Co-current)

At Turbulent flow of warm water ($80^{\circ}C$) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 80.1 | 48.3 | 26.0 | 39.7 | 96.93 |
| 200 | 470 | 80.1 | 47.9 | 26.0 | 39.1 | 96.62 |
| 200 | 500 | 80.1 | 47.6 | 26.1 | 38.8 | 97.69 |
| 300 | 450 | 79.9 | 52.5 | 26.1 | 41.2 | 82.66 |
| 300 | 475 | 79.9 | 51.7 | 26.1 | 41.2 | 84.78 |
| 300 | 500 | 80.0 | 51.2 | 26.1 | 40.8 | 85.07 |
| 400 | 450 | 79.9 | 55.6 | 26.1 | 43.6 | 81.01 |
| 400 | 475 | 79.9 | 54.6 | 26.1 | 42.7 | 77.91 |
| 400 | 500 | 79.9 | 53.7 | 26.1 | 42.3 | 77.29 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 50.1 | 33.4 | 30.0 | 33.7 | 83.11 |
| 33.9 | 152.6 | 50.1 | 33.0 | 30.0 | 33.2 | 84.21 |
| 33.9 | 170.0 | 50.1 | 32.5 | 30.0 | 32.7 | 76 .91 |
| 40.7 | 127.2 | 50.1 | 34.6 | 30.0 | 34.8 | 96.80 |
| 40.7 | 152.6 | 50.2 | 33.9 | 30.0 | 34.0 | 92.03 |
| 40.7 | 170.0 | 50.2 | 33.2 | 30.0 | 33.3 | 81.10 |
| 45.2 | 127.2 | 50.1 | 35.1 | 30.0 | 35.2 | 97.55 |
| 45.2 | 152.6 | 50.0 | 34.2 | 30.0 | 34.5 | 96.15 |
| 45.2 | 170.0 | 49.9 | 33.6 | 30.0 | 33.7 | 85.37 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 50.3 | 40.2 | 30.0 | 39.1 | 57.30 |
| 200 | 152.6 | 50.3 | 39.9 | 30.0 | 38.5 | 62.36 |
| 200 | 170.0 | 50.2 | 39.9 | 30.2 | 38.4 | 67.66 |
| 300 | 127.2 | 50.1 | 43.2 | 30.2 | 40.9 | 65.75 |
| 300 | 152.6 | 50.1 | 42.4 | 30.0 | 40.0 | 66.06 |
| 300 | 170.0 | 50.0 | 42.3 | 30.1 | 39.9 | 72.12 |
| 400 | 127.2 | 50.1 | 44.8 | 30.1 | 41.5 | 68.40 |
| 400 | 152.6 | 50.1 | 43.8 | 30.1 | 40.7 | 64.18 |
| 400 | 170.0 | 50.0 | 43.7 | 30.0 | 40.5 | 70.83 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 70.2 | 37.4 | 30.0 | 38.5 | 97.20 |
| 33.9 | 152.6 | 70.4 | 36.8 | 30.0 | 37.3 | 97.77 |
| 33.9 | 170.0 | 70.3 | 36.0 | 30.0 | 36.6 | 96.46 |
| 40.7 | 127.2 | 70.3 | 38.0 | 30.0 | 39.9 | 95.81 |
| 40.7 | 152.6 | 70.3 | 37.3 | 30.0 | 38.5 | 96.59 |
| 40.7 | 170.0 | 70.3 | 36.4 | 30.0 | 37.8 | 96.12 |
| 45.2 | 127.2 | 70.3 | 39.2 | 30.0 | 40.7 | 96.82 |
| 45.2 | 152.6 | 70.2 | 38.6 | 30.0 | 39.1 | 97.22 |
| 45.2 | 170.0 | 70.2 | 37.8 | 30.0 | 38.3 | 96.34 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 69.9 | 51.8 | 30.0 | 49.9 | 69.92 |
| 200 | 152.6 | 70.1 | 50.6 | 30.1 | 48.8 | 73.17 |
| 200 | 170.0 | 70.1 | 49.4 | 30.1 | 47.3 | 70.62 |
| 300 | 127.2 | 70.1 | 56.4 | 30.1 | 52.9 | 70.56 |
| 300 | 152.6 | 70.0 | 54.0 | 30.1 | 50.4 | 64.53 |
| 300 | 170.0 | 70.1 | 53.2 | 30.1 | 49.7 | 65.72 |
| 400 | 127.2 | 70.1 | 59.5 | 30.1 | 56.2 | 78.30 |
| 400 | 152.6 | 70.2 | 58.7 | 30.1 | 55.4 | 89.93 |
| 400 | 170.0 | 70.1 | 56.9 | 30.1 | 54.4 | 78.23 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127.2 | 80.0 | 39.2 | 30.0 | 40.0 | 91.93 |
| 33.9 | 152.6 | 80.4 | 38.6 | 30.0 | 38.8 | 94.73 |
| 33.9 | 170.0 | 80.3 | 37.7 | 30.0 | 38.1 | 95.32 |
| 40.7 | 127.2 | 80.4 | 40.5 | 30.0 | 42.5 | 97.93 |
| 40.7 | 152.6 | 80.1 | 39.2 | 30.0 | 40.3 | 94.44 |
| 40.7 | 170.0 | 80.4 | 38.4 | 30.0 | 39.6 | 95.49 |
| 45.2 | 127.2 | 80.5 | 41.2 | 30.0 | 43.4 | 95.95 |
| 45.2 | 152.6 | 80.4 | 39.6 | 30.0 | 41.7 | 96.81 |
| 45.2 | 170.0 | 80.5 | 38.8 | 30.0 | 40.8 | 97.40 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 127.2 | 80.2 | 57.2 | 26.7 | 55.8 | 80.46 |
| 200 | 152.6 | 80.0 | 55.0 | 26.7 | 53.4 | 81.48 |
| 200 | 170.0 | 80.4 | 53.3 | 26.7 | 51.3 | 77.15 |
| 300 | 127.2 | 80.0 | 63.0 | 26.7 | 59.5 | 81.80 |
| 300 | 152.6 | 79.9 | 60.8 | 26.7 | 57.0 | 80.90 |
| 300 | 170.0 | 80.8 | 59.5 | 26.7 | 55.2 | 75.82 |
| 400 | 127.2 | 80.5 | 66.2 | 26.7 | 61.3 | 76.94 |
| 400 | 152.6 | 80.2 | 64.7 | 26.7 | 59.6 | 81.23 |
| 400 | 170.0 | 80.0 | 63.3 | 26.7 | 57.3 | 77.87 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 50.5 | 30.6 | 27.4 | 28.8 | 93.59 |
| 33.9 | 475 | 50.5 | 30.5 | 27.4 | 28.8 | 98.30 |
| 33.9 | 500 | 50.6 | 30.4 | 27.4 | 28.7 | 95.12 |
| 40.7 | 450 | 50.4 | 30.8 | 37.4 | 29.1 | 95.92 |
| 40.7 | 475 | 50.4 | 30.9 | 27.5 | 29.1 | 90.74 |
| 40.7 | 500 | 50.5 | 30.7 | 27.4 | 28.9 | 93.09 |
| 45.2 | 450 | 50.6 | 31.1 | 27.4 | 29.3 | 97.25 |
| 45.2 | 475 | 50.6 | 31.1 | 27.5 | 29.3 | 97.25 |
| 45.2 | 500 | 50.6 | 30.9 | 27.4 | 29.1 | 95.70 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 49.9 | 34.7 | 27.4 | 33.0 | 83.16 |
| 200 | 475 | 50.2 | 34.5 | 27.4 | 32.5 | 77.14 |
| 200 | 500 | 50.2 | 34.3 | 27.3 | 32.2 | 77.28 |
| 300 | 450 | 50.1 | 37.5 | 27.4 | 34.3 | 82.14 |
| 300 | 475 | 50.0 | 37.3 | 27.5 | 34.2 | 83.53 |
| 300 | 500 | 50.1 | 37.0 | 27.4 | 33.9 | 83.01 |
| 400 | 450 | 50.1 | 39.4 | 27.4 | 35.4 | 84.50 |
| 400 | 475 | 50.0 | 39.2 | 27.4 | 35.2 | 85.76 |
| 400 | 500 | 50.0 | 38.8 | 27.3 | 34.8 | 83.70 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 70.2 | 31.6 | 27.5 | 30.3 | 96.38 |
| 33.9 | 475 | 69.9 | 31.5 | 27.6 | 30.2 | 94.83 |
| 33.9 | 500 | 70.1 | 31.3 | 27.4 | 29.9 | 95.12 |
| 40.7 | 450 | 70.2 | 32.0 | 27.4 | 30.8 | 98.43 |
| 40.7 | 475 | 70.1 | 31.5 | 27.5 | 30.7 | 96.90 |
| 40.7 | 500 | 70.0 | 31.8 | 27.5 | 30.5 | 96.62 |
| 45.2 | 450 | 69.8 | 32.4 | 27.4 | 31.0 | 95.95 |
| 45.2 | 475 | 70.0 | 32.0 | 27.3 | 30.9 | 96.91 |
| 45.2 | 500 | 70.0 | 32.0 | 27.4 | 30.7 | 96.19 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (70°C) and Turbulent of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 70.1 | 41.6 | 27.6 | 38.9 | 89.27 |
| 200 | 475 | 70.1 | 41.3 | 27.6 | 38.5 | 90.04 |
| 200 | 500 | 70.1 | 41.0 | 27.5 | 37.9 | 89.34 |
| 300 | 450 | 70.0 | 46.1 | 27.5 | 41.2 | 86.16 |
| 300 | 475 | 70.2 | 45.4 | 27.5 | 40.6 | 83.63 |
| 300 | 500 | 70.1 | 45.2 | 27.6 | 40.4 | 85.84 |
| 400 | 450 | 69.9 | 50.2 | 27.5 | 43.4 | 90.79 |
| 400 | 475 | 69.8 | 49.6 | 27.5 | 42.8 | 89.94 |
| 400 | 500 | 69.8 | 49.3 | 27.5 | 42.4 | 90.85 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 80.0 | 32.3 | 27.4 | 30.9 | 97.37 |
| 33.9 | 475 | 79.9 | 32.1 | 27.4 | 30.7 | 96.60 |
| 33.9 | 500 | 80.0 | 32.0 | 27.4 | 30.6 | 98.40 |
| 40.7 | 450 | 80.2 | 33.2 | 27.5 | 31.6 | 96.47 |
| 40.7 | 475 | 80.1 | 32.9 | 27.4 | 31.2 | 89.13 |
| 40.7 | 500 | 80.0 | 32.7 | 27.4 | 31.1 | 96.22 |
| 45.2 | 450 | 80.2 | 33.5 | 27.5 | 32.0 | 95.93 |
| 45.2 | 475 | 80.1 | 33.2 | 27.4 | 31.7 | 96.37 |
| 45.2 | 500 | 80.1 | 33.0 | 27.4 | 31.5 | 93.37 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Monotube (Counter-current)

At Turbulent flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 200 | 450 | 79.6 | 45.7 | 27.5 | 42.0 | 96.38 |
| 200 | 470 | 79.6 | 44.8 | 27.5 | 41.5 | 95.68 |
| 200 | 500 | 79.6 | 44.7 | 27.5 | 40.8 | 95.27 |
| 300 | 450 | 79.6 | 50.3 | 27.6 | 44.5 | 86.66 |
| 300 | 475 | 80.4 | 49.8 | 27.6 | 44.0 | 84.99 |
| 300 | 500 | 79.8 | 49.6 | 27.5 | 43.6 | 88.85 |
| 400 | 450 | 79.9 | 53.7 | 27.5 | 46.7 | 82.44 |
| 400 | 475 | 79.8 | 53.5 | 27.6 | 46.5 | 85.33 |
| 400 | 500 | 79.9 | 53.3 | 27.5 | 46.3 | 88.34 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 50.0 | 34.6 | 26.8 | 30.1 | 80.25 |
| 33.9 | 152 | 50.1 | 34.4 | 26.8 | 29.8 | 85.65 |
| 33.9 | 170 | 50.0 | 34.0 | 26.8 | 29.3 | 78.33 |
| 40.7 | 127 | 50.1 | 36.2 | 26.9 | 30.7 | 85.33 |
| 40.7 | 152 | 50.1 | 35.9 | 26.9 | 30.1 | 84.18 |
| 40.7 | 170 | 50.1 | 35.6 | 26.9 | 29.8 | 82.99 |
| 45.2 | 127 | 50.5 | 36.9 | 26.7 | 30.8 | 83.33 |
| 45.2 | 152 | 50.1 | 36.1 | 26.7 | 30.2 | 79.27 |
| 45.2 | 170 | 50.2 | 35.4 | 26.9 | 29.7 | 71.16 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Turbulent flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 50.1 | 45.4 | 27.6 | 39.6 | 75.41 |
| 430 | 152 | 50.1 | 44.4 | 27.5 | 37.8 | 63.88 |
| 430 | 170 | 50.1 | 44.1 | 27.6 | 37.2 | 62.93 |
| 500 | 127 | 50.1 | 46.2 | 27.5 | 40.4 | 82.63 |
| 500 | 152 | 50.0 | 45.4 | 27.5 | 38.7 | 74.02 |
| 500 | 170 | 50.1 | 45.0 | 27.4 | 38.0 | 70.67 |
| 600 | 127 | 50.1 | 46.2 | 27.4 | 40.2 | 69.20 |
| 600 | 152 | 50.1 | 45.4 | 27.4 | 39.1 | 63.06 |
| 600 | 170 | 50.2 | 45.0 | 27.4 | 38.2 | 58.85 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 70.3 | 42.6 | 26.8 | 33.8 | 94.64 |
| 33.9 | 152 | 70.2 | 41.7 | 26.9 | 32.7 | 91.22 |
| 33.9 | 170 | 70.2 | 41.0 | 26.9 | 31.8 | 84.13 |
| 40.7 | 127 | 70.1 | 45.0 | 26.8 | 39.3 | 93.26 |
| 40.7 | 152 | 70.2 | 44.5 | 26.8 | 33.3 | 94.48 |
| 40.7 | 170 | 70.2 | 44.2 | 26.8 | 32.3 | 88.38 |
| 45.2 | 127 | 70.1 | 48.0 | 26.8 | 34.1 | 92.81 |
| 45.2 | 152 | 70.0 | 47.2 | 26.8 | 33.4 | 97.34 |
| 45.2 | 170 | 70.1 | 46.1 | 26.8 | 32.8 | 94.03 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

## Shell and Tubes (Co-current)

At Turbulent flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | T1 (°C) | T2 (°C) | t1 (°C) | t2 (°C) | η (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 70.2 | 61.7 | 27.7 | 52.2 | 85.13 |
| 430 | 152 | 70.1 | 60.3 | 27.6 | 49.4 | 78.63 |
| 430 | 170 | 70.1 | 59.8 | 27.6 | 47.9 | 77.92 |
| 500 | 127 | 70.2 | 62.5 | 27.8 | 53.4 | 84.45 |
| 500 | 152 | 70.1 | 61.0 | 27.2 | 50.1 | 74.50 |
| 500 | 170 | 70.1 | 60.4 | 27.6 | 48.6 | 73.61 |
| 600 | 127 | 70.2 | 63.0 | 27.6 | 54.1 | 77.90 |
| 600 | 152 | 70.3 | 62.4 | 27.6 | 51.6 | 76.96 |
| 600 | 170 | 70.2 | 61.5 | 27.5 | 49.6 | 71.97 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 80.4 | 47.8 | 26.7 | 35.2 | 97.65 |
| 33.9 | 152 | 80.3 | 46.7 | 26.7 | 33.9 | 96.05 |
| 33.9 | 170 | 80.2 | 46.1 | 26.7 | 33.0 | 92.48 |
| 40.7 | 127 | 80.3 | 49.8 | 26.8 | 36.2 | 96.19 |
| 40.7 | 152 | 80.4 | 48.1 | 26.8 | 34.4 | 87.90 |
| 40.7 | 170 | 80.3 | 47.4 | 26.8 | 33.5 | 85.08 |
| 45.2 | 127 | 80.3 | 50.9 | 26.6 | 36.8 | 97.48 |
| 45.2 | 152 | 80.2 | 41.0 | 26.6 | 35.3 | 93.77 |
| 45.2 | 170 | 80.4 | 48.5 | 26.3 | 30.1 | 91.96 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

## Shell and Tubes (Co-current)

At Turbulent flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 80.4 | 70.0 | 27.5 | 57.3 | 84.63 |
| 430 | 152 | 80.3 | 68.7 | 27.5 | 54.6 | 82.58 |
| 430 | 170 | 80.3 | 67.3 | 27.5 | 53.7 | 79.68 |
| 500 | 127 | 80.3 | 70.6 | 27.6 | 59.5 | 83.53 |
| 500 | 152 | 80.3 | 68.9 | 27.5 | 56.5 | 77.33 |
| 500 | 170 | 80.2 | 68.3 | 27.5 | 54.5 | 77.14 |
| 600 | 127 | 80.3 | 71.3 | 27.4 | 59.8 | 76.20 |
| 600 | 152 | 80.3 | 70.4 | 27.4 | 57.2 | 75.49 |
| 600 | 170 | 80.3 | 69.9 | 27.4 | 56.3 | 78.73 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 50.2 | 30.5 | 27.0 | 28.3 | 87.57 |
| 33.9 | 600 | 50.1 | 30.2 | 27.0 | 28.0 | 88.91 |
| 33.9 | 750 | 50.1 | 29.9 | 27.8 | 28.5 | 87.59 |
| 40.7 | 450 | 50.0 | 31.0 | 27.0 | 28.5 | 87.31 |
| 40.7 | 600 | 50.1 | 30.6 | 27.0 | 28.2 | 90.74 |
| 40.7 | 750 | 50.1 | 30.1 | 27.0 | 27.9 | 82.94 |
| 45.2 | 450 | 50.0 | 31.5 | 27.2 | 28.9 | 91.49 |
| 45.2 | 600 | 50.1 | 31.3 | 27.2 | 28.5 | 91.79 |
| 45.2 | 750 | 50.0 | 29.9 | 27.2 | 28.2 | 82.55 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 50.0 | 42.2 | 27.4 | 32.6 | 85.27 |
| 430 | 600 | 50.0 | 41.8 | 27.4 | 32.2 | 81.68 |
| 430 | 750 | 50.1 | 41.5 | 27.4 | 31.6 | 84.17 |
| 500 | 550 | 50.1 | 42.8 | 27.4 | 32.9 | 84.38 |
| 500 | 600 | 50.0 | 42.6 | 27.3 | 32.6 | 85.95 |
| 500 | 750 | 50.1 | 42.2 | 27.2 | 31.8 | 87.34 |
| 600 | 550 | 50.1 | 43.5 | 27.4 | 33.5 | 84.72 |
| 600 | 600 | 50.1 | 42.7 | 27.4 | 32.8 | 72.97 |
| 600 | 750 | 50.0 | 42.4 | 27.4 | 32.1 | 78.95 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 70.1 | 38.6 | 27.4 | 29.5 | 88.47 |
| 33.9 | 600 | 70.1 | 38.4 | 27.4 | 29.1 | 94.89 |
| 33.9 | 750 | 70.1 | 38.6 | 27.4 | 28.8 | 98.30 |
| 40.7 | 450 | 70.0 | 39.7 | 27.5 | 30.1 | 93.66 |
| 40.7 | 600 | 70.3 | 39.8 | 27.4 | 29.2 | 87.02 |
| 40.7 | 750 | 70.1 | 39.5 | 27.4 | 28.9 | 90.21 |
| 45.2 | 450 | 70.1 | 42.4 | 27.4 | 30.0 | 93.28 |
| 45.2 | 600 | 70.1 | 42.1 | 27.4 | 29.4 | 94.82 |
| 45.2 | 750 | 70.1 | 41.8 | 27.4 | 29.0 | 93.81 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Turbulent flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 70.2 | 55.0 | 27.3 | 38.0 | 90.04 |
| 430 | 600 | 70.1 | 54.5 | 27.3 | 37.2 | 88.55 |
| 430 | 750 | 70.2 | 53.9 | 27.3 | 35.9 | 92.02 |
| 500 | 550 | 70.1 | 56.2 | 27.3 | 38.9 | 85.35 |
| 500 | 600 | 70.0 | 55.8 | 27.3 | 38.1 | 90.88 |
| 500 | 750 | 70.2 | 55.4 | 27.3 | 36.8 | 95.96 |
| 600 | 550 | 70.2 | 57.1 | 27.3 | 39.4 | 84.35 |
| 600 | 600 | 70.1 | 56.6 | 27.3 | 38.7 | 84.44 |
| 600 | 750 | 70.1 | 55.7 | 27.2 | 36.9 | 84.20 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 80.1 | 43.5 | 26.6 | 29.3 | 97.90 |
| 33.9 | 600 | 80.4 | 43.1 | 26.6 | 28.6 | 94.87 |
| 33.9 | 750 | 80.3 | 42.4 | 26.6 | 28.2 | 93.37 |
| 40.7 | 450 | 80.2 | 41.6 | 26.5 | 29.7 | 91.68 |
| 40.7 | 600 | 80.3 | 42.5 | 26.6 | 28.9 | 92.04 |
| 40.7 | 750 | 80.3 | 42.8 | 26.6 | 28.4 | 88.36 |
| 45.2 | 450 | 80.2 | 45.6 | 26.6 | 29.9 | 94.95 |
| 45.2 | 600 | 80.2 | 45.3 | 26.6 | 28.1 | 95.09 |
| 45.2 | 750 | 80.3 | 44.8 | 26.6 | 28.6 | 93.48 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Co-current)

At Turbulent flow of warm water (80 °C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 80.3 | 60.5 | 27.3 | 40.4 | 84.62 |
| 430 | 600 | 80.2 | 60.2 | 27.3 | 39.8 | 87.21 |
| 430 | 750 | 80.2 | 59.3 | 27.3 | 38.1 | 89.71 |
| 500 | 550 | 80.2 | 62.7 | 27.3 | 41.7 | 90.51 |
| 500 | 600 | 80.3 | 61.5 | 27.2 | 40.4 | 84.26 |
| 500 | 750 | 80.2 | 59.7 | 27.2 | 38.5 | 82.52 |
| 600 | 550 | 80.2 | 63.3 | 27.3 | 42.4 | 81.63 |
| 600 | 600 | 80.3 | 62.5 | 27.3 | 41.3 | 78.65 |
| 600 | 750 | 80.2 | 60.5 | 27.3 | 39.2 | 75.51 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 50.0 | 33.7 | 26.4 | 30.3 | 88.46 |
| 33.9 | 152 | 50.1 | 33.5 | 26.4 | 29.5 | 83.71 |
| 33.9 | 170 | 50.0 | 33.2 | 26.3 | 29.2 | 83.55 |
| 40.7 | 127 | 50.1 | 35.0 | 26.4 | 30.5 | 83.71 |
| 40.7 | 152 | 50.1 | 34.8 | 26.4 | 29.8 | 83.01 |
| 40.7 | 170 | 50.1 | 34.7 | 26.9 | 29.4 | 80.03 |
| 45.2 | 127 | 50.1 | 36.6 | 26.4 | 31.0 | 95.74 |
| 45.2 | 152 | 50.0 | 36.2 | 26.4 | 29.8 | 82.85 |
| 45.2 | 170 | 50.0 | 35.8 | 26.4 | 29.6 | 84.76 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Turbulent flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 50.0 | 44.6 | 27.1 | 39.4 | 66.39 |
| 430 | 152 | 50.1 | 44.1 | 27.1 | 28.5 | 67.16 |
| 430 | 170 | 50.1 | 43.7 | 27.1 | 37.3 | 63.01 |
| 500 | 127 | 50.1 | 44.7 | 27.0 | 40.6 | 78.51 |
| 500 | 152 | 50.0 | 45.0 | 27.1 | 39.3 | 73.44 |
| 500 | 170 | 50.1 | 44.3 | 27.0 | 38.2 | 65.37 |
| 600 | 127 | 50.0 | 46.0 | 27.0 | 41.1 | 74.61 |
| 600 | 152 | 50.1 | 45.6 | 27.0 | 39.7 | 71.50 |
| 600 | 170 | 50.1 | 45.0 | 27.0 | 38.5 | 63.89 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Laminar flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 70.1 | 41.7 | 27.4 | 34.6 | 94.12 |
| 33.9 | 152 | 70.1 | 40.8 | 27.4 | 33.6 | 94.85 |
| 33.9 | 170 | 70.0 | 40.0 | 27.4 | 32.6 | 86.90 |
| 40.7 | 127 | 70.0 | 42.0 | 27.4 | 35.0 | 84.72 |
| 40.7 | 152 | 70.0 | 41.0 | 27.4 | 33.9 | 83.73 |
| 40.7 | 170 | 70.1 | 40.3 | 27.4 | 32.9 | 77.11 |
| 45.2 | 127 | 70.1 | 43.1 | 27.4 | 36.0 | 89.49 |
| 45.2 | 152 | 70.2 | 42.0 | 27.4 | 34.5 | 84.67 |
| 45.2 | 170 | 70.1 | 40.8 | 27.4 | 33.3 | 75.73 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

<u>Shell and Tubes</u> (Counter-current)

At Turbulent flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 70.0 | 60.7 | 27.3 | 53.0 | 81.02 |
| 430 | 152 | 70.1 | 59.6 | 27.3 | 50.6 | 78.44 |
| 430 | 170 | 70.1 | 58.9 | 27.3 | 48.8 | 75.89 |
| 500 | 127 | 70.2 | 61.7 | 27.7 | 54.6 | 81.88 |
| 500 | 152 | 70.1 | 60.7 | 27.2 | 51.4 | 77.85 |
| 500 | 170 | 70.1 | 59.9 | 27.2 | 49.6 | 74.30 |
| 600 | 127 | 70.1 | 62.4 | 27.2 | 54.7 | 75.11 |
| 600 | 152 | 70.0 | 61.2 | 27.2 | 52.1 | 71.68 |
| 600 | 170 | 70.2 | 60.6 | 27.2 | 50.5 | 69.49 |

<u>Note</u>

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Laminar flow of warm water ($80^{\circ}C$) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 127 | 80.4 | 45.1 | 27.4 | 36.6 | 97.47 |
| 33.9 | 152 | 80.3 | 44.5 | 27.3 | 34.8 | 93.78 |
| 33.9 | 170 | 80.3 | 43.7 | 27.3 | 33.8 | 88.91 |
| 40.7 | 127 | 80.2 | 48.7 | 27.3 | 37.1 | 97.10 |
| 40.7 | 152 | 80.3 | 47.5 | 27.3 | 35.6 | 94.53 |
| 40.7 | 170 | 80.3 | 46.6 | 27.2 | 34.3 | 87.89 |
| 45.2 | 80.2 | 80.3 | 50.2 | 27.1 | 37.6 | 97.85 |
| 45.2 | 152 | 80.2 | 48.8 | 27.0 | 35.9 | 95.16 |
| 45.2 | 170 | 80.3 | 48.2 | 27.1 | 34.7 | 89.05 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Turbulent flow of warm water (80$^{\circ}$C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 127 | 80.4 | 68.7 | 27.3 | 60.4 | 83.56 |
| 430 | 152 | 80.3 | 67.1 | 27.4 | 57.2 | 75.58 |
| 430 | 170 | 80.3 | 65.9 | 27.3 | 54.4 | 74.14 |
| 500 | 127 | 80.3 | 70.4 | 27.2 | 61.3 | 87.49 |
| 500 | 152 | 80.3 | 68.8 | 27.1 | 57.9 | 81.07 |
| 500 | 170 | 80.4 | 67.7 | 27.1 | 56.0 | 77.04 |
| 600 | 127 | 80.5 | 72.1 | 27.1 | 62.2 | 88.47 |
| 600 | 152 | 80.5 | 69.3 | 27.1 | 59.0 | 72.15 |
| 600 | 170 | 80.4 | 68.6 | 27.1 | 56.6 | 70.53 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Laminar flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 50.0 | 28.9 | 26.7 | 28.2 | 94.34 |
| 33.9 | 600 | 50.1 | 28.1 | 26.7 | 27.9 | 96.51 |
| 33.9 | 750 | 50.1 | 27.9 | 26.7 | 27.6 | 89.67 |
| 40.7 | 450 | 50.1 | 30.9 | 26.6 | 28.3 | 97.92 |
| 40.7 | 600 | 50.1 | 29.1 | 26.7 | 28.0 | 91.28 |
| 40.7 | 750 | 50.1 | 28.6 | 26.6 | 27.5 | 77.16 |
| 45.2 | 450 | 50.0 | 30.5 | 26.6 | 28.5 | 97.00 |
| 45.2 | 600 | 50.1 | 30.0 | 26.7 | 28.1 | 92.46 |
| 45.2 | 750 | 50.1 | 29.5 | 26.6 | 27.7 | 88.60 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

<u>Shell and Tubes</u> (Counter-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 50.0 | 43.0 | 26.9 | 31.8 | 89.53 |
| 430 | 600 | 50.1 | 42.8 | 26.9 | 31.5 | 87.96 |
| 430 | 750 | 50.1 | 42.3 | 26.9 | 30.7 | 84.97 |
| 500 | 550 | 50.1 | 43.5 | 26.9 | 32.3 | 90.00 |
| 500 | 600 | 50.0 | 43.2 | 26.9 | 31.7 | 84.71 |
| 500 | 750 | 50.1 | 42.8 | 26.9 | 31.0 | 84.25 |
| 600 | 550 | 50.1 | 43.9 | 26.9 | 32.7 | 85.75 |
| 600 | 600 | 50.1 | 43.4 | 26.9 | 32.4 | 82.09 |
| 600 | 750 | 50.0 | 42.9 | 26.9 | 31.5 | 80.98 |

<u>Note</u>

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 70.1 | 39.8 | 26.7 | 28.9 | 96.35 |
| 33.9 | 600 | 70.2 | 38.8 | 26.7 | 28.4 | 95.80 |
| 33.9 | 750 | 70.1 | 38.4 | 26.6 | 27.9 | 90.70 |
| 40.7 | 450 | 70.1 | 41.9 | 26.6 | 29.1 | 98.04 |
| 40.7 | 600 | 70.2 | 40.7 | 26.6 | 28.5 | 94.97 |
| 40.7 | 750 | 70.1 | 40.2 | 26.5 | 27.7 | 73.98 |
| 45.2 | 450 | 70.1 | 41.2 | 26.6 | 29.3 | 96.17 |
| 45.2 | 600 | 70.1 | 41.7 | 26.7 | 28.7 | 93.48 |
| 45.2 | 750 | 70.1 | 40.7 | 26.6 | 28.2 | 90.30 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Turbulent flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 70.0 | 58.5 | 27.0 | 34.7 | 85.64 |
| 430 | 600 | 70.1 | 58.1 | 27.0 | 34.3 | 84.88 |
| 430 | 750 | 70.1 | 57.4 | 27.0 | 27.5 | 78.28 |
| 500 | 550 | 70.1 | 59.1 | 27.0 | 35.8 | 88.00 |
| 500 | 600 | 70.1 | 58.2 | 27.0 | 35.5 | 85.71 |
| 500 | 750 | 70.0 | 57.0 | 27.0 | 34.1 | 81.92 |
| 600 | 550 | 70.0 | 59.8 | 27.1 | 36.6 | 85.38 |
| 600 | 600 | 70.0 | 59.1 | 27.0 | 36.2 | 84.40 |
| 600 | 750 | 70.0 | 58.2 | 27.0 | 34.9 | 83.69 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

## Shell and Tubes (Counter-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 80.3 | 40.5 | 27.6 | 30.3 | 90.03 |
| 33.9 | 600 | 80.4 | 40.2 | 27.6 | 29.8 | 96.83 |
| 33.9 | 750 | 80.3 | 39.8 | 27.6 | 29.7 | 98.30 |
| 40.7 | 450 | 80:3 | 43.1 | 27.6 | 30.7 | 92.16 |
| 40.7 | 600 | 80.2 | 42.8 | 27.6 | 30.0 | 94.50 |
| 40.7 | 750 | 80.3 | 42:5 | 27.6 | 29.6 | 97.52 |
| 45.2 | 450 | 80.3 | 44.2 | 27.6 | 30.8 | 88.25 |
| 45.2 | 600 | 80.3 | 44.0 | 27.6 | 30.2 | 95.08 |
| 45.2 | 750 | 80.3 | 43.5 | 27.6 | 29.7 | 94.69 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Shell and Tubes (Counter-current)

At Turbulent flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 430 | 550 | 80.2 | 60.3 | 27.0 | 40.2 | 84.84 |
| 430 | 600 | 80.2 | 59.8 | 27.0 | 39.3 | 84.13 |
| 430 | 750 | 80.3 | 58.5 | 27.0 | 38.0 | 88.01 |
| 500 | 550 | 80.3 | 60.8 | 27.0 | 41.3 | 80.67 |
| 500 | 600 | 80.2 | 60.2 | 27.0 | 39.8 | 76.20 |
| 500 | 750 | 80.2 | 59.5 | 27.0 | 38.0 | 79.71 |
| 600 | 450 | 80.1 | 61.4 | 27.0 | 41.9 | 73.04 |
| 600 | 600 | 80.2 | 61.1 | 27.1 | 41.2 | 73.82 |
| 600 | 750 | 80.2 | 60.3 | 27.0 | 39.8 | 80.40 |

Note

$T_1$ is inlet temperature of warm water

$T_2$ is outlet temperature of warm water

$t_1$ is inlet temperature of cold water

$t_2$ is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Platetube (Co-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 100 | 49.9 | 35.0 | 26.8 | 39.7 | 86.57 |
| 100 | 200 | 50.1 | 32.1 | 26.8 | 34.3 | 83.33 |
| 100 | 300 | 50.2 | 31.1 | 26.8 | 32.5 | 89.52 |
| 200 | 100 | 49.9 | 39.8 | 26.9 | 43.0 | 79.70 |
| 200 | 200 | 50.1 | 35.8 | 26.8 | 37.7 | 76.22 |
| 200 | 300 | 49.9 | 34.4 | 26.9 | 35.6 | 84.19 |
| 300 | 100 | 50.0 | 42.1 | 26.8 | 45.2 | 77.63 |
| 300 | 200 | 49.9 | 38.8 | 26.9 | 40.5 | 81.68 |
| 300 | 300 | 49.9 | 37.1 | 26.9 | 37.7 | 84.37 |

At Laminar flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 100 | 70.2 | 53.4 | 27.6 | 37.2 | 57.14 |
| 100 | 200 | 70.3 | 53.0 | 27.6 | 33.2 | 64.74 |
| 100 | 300 | 70.2 | 50.5 | 27.6 | 31.4 | 57.89 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Co-current)

At Turbulent flow of warm water ($50^{\circ}C$) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 100 | 49.9 | 46.7 | 27.0 | 47.6 | 71.52 |
| 900 | 200 | 49.9 | 44.7 | 26.9 | 44.4 | 74.78 |
| 900 | 300 | 50.0 | 43.0 | 26.9 | 41.7 | 70.47 |

At Turbulent flow of warm water ($70^{\circ}C$) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 100 | 70.2 | 65.3 | 27.0 | 58.1 | 70.52 |
| 900 | 200 | 70.2 | 61.5 | 27.1 | 533.3 | 66.92 |
| 900 | 300 | 70.1 | 58.7 | 27.1 | 51.6 | 71.64 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Co-current)

At Laminar flow of warm water (50 $^{\circ}$C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 50.4 | 33.9 | 26.6 | 28.4 | 92.73 |
| 100 | 875 | 50.4 | 32.1 | 26.6 | 28.5 | 90.84 |
| 100 | 900 | 50.3 | 28.4 | 26.6 | 28.6 | 82.19 |

At Laminar flow of warm water (70 $^{\circ}$C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 70.3 | 36.7 | 27.4 | 31.3 | 98.66 |
| 100 | 875 | 70.2 | 35.8 | 27.5 | 31.0 | 89.03 |
| 100 | 900 | 70.2 | 35.1 | 27.5 | 30.1 | 82.05 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Co-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 850 | 50.1 | 42.0 | 27.2 | 33.3 | 71.12 |
| 900 | 875 | 50.1 | 41.8 | 27.2 | 33.0 | 67.94 |
| 900 | 900 | 50.2 | 41.6 | 27.2 | 32.8 | 65.12 |

At Turbulent flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 850 | 70.2 | 48.2 | 27.2 | 43.7 | 70.83 |
| 900 | 875 | 70.3 | 48.0 | 27.2 | 42.5 | 66.77 |
| 900 | 900 | 70.2 | 47.7 | 27.2 | 42.1 | 66.22 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Laminar flow of warm water (50°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 100 | 50.2 | 38.1 | 27.4 | 37.3 | 85.12 |
| 100 | 200 | 50.1 | 35.4 | 27.4 | 33.5 | 82.99 |
| 100 | 300 | 50.1 | 33.6 | 27.4 | 31.7 | 78.18 |
| 200 | 100 | 50.1 | 42.1 | 26.4 | 39.6 | 82.50 |
| 200 | 200 | 50.0 | 39.5 | 26.4 | 34.9 | 80.95 |
| 200 | 300 | 50.1 | 38.5 | 26.4 | 32.5 | 78.88 |
| 300 | 100 | 50.1 | 44.6 | 26.5 | 40.8 | 86.67 |
| 300 | 200 | 50.1 | 42.3 | 26.5 | 35.7 | 78.88 |
| 300 | 300 | 50.1 | 41.3 | 26.5 | 33.1 | 75.00 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Turbulent flow of warm water ($50^{\circ}C$) and Laminar flow of cold water

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 100 | 50.0 | 47.0 | 27.4 | 45.4 | 66.66 |
| 900 | 200 | 50.1 | 45.0 | 27.4 | 42.2 | 64.49 |
| 900 | 300 | 50.1 | 44.1 | 27.4 | 39.0 | 64.44 |
| 1000 | 100 | 50.1 | 47.9 | 27.4 | 45.7 | 83.18 |
| 1000 | 200 | 50.0 | 46.0 | 27.4 | 42.3 | 74.50 |
| 1000 | 300 | 50.1 | 45.1 | 27.4 | 39.3 | 71.40 |
| 1100 | 100 | 50.0 | 47.0 | 27.4 | 45.4 | 54.54 |
| 1100 | 200 | 50.1 | 45.0 | 27.4 | 42.2 | 52.76 |
| 1100 | 300 | 50.1 | 44.1 | 27.4 | 39.0 | 52.73 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Laminar flow of warm water (70°C) and Laminar flow of cold wate

| Warm water (L/h) | Cold water (L/h) | T1 (°C) | T2 (°C) | t1 (°C) | t2 (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 100 | 70.1 | 52.9 | 27.4 | 37.7 | 59.88 |
| 100 | 200 | 70.1 | 52.0 | 27.4 | 33.5 | 67.40 |
| 100 | 300 | 70.2 | 50.4 | 27.4 | 31.7 | 65.15 |
| 200 | 100 | 70.1 | 59.6 | 26.4 | 39.6 | 63.10 |
| 200 | 200 | 70.0 | 54.6 | 26.4 | 34.9 | 55.19 |
| 200 | 300 | 70.1 | 52.6 | 26.4 | 32.5 | 52.28 |
| 300 | 100 | 70.1 | 61.4 | 26.5 | 40.8 | 54.48 |
| 300 | 200 | 70.0 | 60.8 | 26.5 | 35.7 | 66.66 |
| 300 | 300 | 70.2 | 59.5 | 26.5 | 33.1 | 61.68 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Turbulent flow of warm water (70°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 100 | 70.2 | 64.6 | 26.6 | 63.0 | 72.22 |
| 900 | 200 | 70.2 | 61.0 | 26.6 | 56.7 | 72.70 |
| 900 | 300 | 70.5 | 57.6 | 26.9 | 51.8 | 64.34 |
| 1000 | 100 | 70.2 | 65.2 | 26.7 | 62.8 | 72.20 |
| 1000 | 200 | 70.2 | 61.2 | 26.6 | 56.7 | 66.89 |
| 1000 | 300 | 70.1 | 58.7 | 26.6 | 52.0 | 66.84 |
| 1100 | 100 | 70.1 | 65.4 | 26.7 | 63.6 | 71.37 |
| 1100 | 200 | 70.2 | 61.9 | 26.7 | 57.5 | 67.47 |
| 1100 | 300 | 70.2 | 59.0 | 26.6 | 52.8 | 63.80 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Laminar flow of warm water (80°C) and Laminar flow of cold water.

| Warm water (L/h) | Cold water (L/h) | T1 (°C) | T2 (°C) | t1 (°C) | t2 (°C) | η (%) |
|---|---|---|---|---|---|---|
| 100 | 100 | 80.2 | 53.6 | 27.3 | 52.3 | 93.98 |
| 100 | 200 | 80.2 | 51.3 | 27.3 | 40.5 | 91.35 |
| 100 | 300 | 80.2 | 43.5 | 27.4 | 38.5 | 90.74 |
| 200 | 100 | 80.2 | 60.3 | 26.4 | 57.1 | 77.14 |
| 200 | 200 | 80.3 | 57.5 | 26.4 | 43.5 | 75.00 |
| 200 | 300 | 80.2 | 47.6 | 26.4 | 41.5 | 69.48 |
| 300 | 100 | 80.2 | 64.5 | 26.5 | 59.6 | 70.28 |
| 300 | 200 | 80.2 | 60.5 | 26.5 | 45.6 | 64.64 |
| 300 | 300 | 80.3 | 53.6 | 26.5 | 42.3 | 59.18 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Laminar flow of warm water ($50^{\circ}C$) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 50.2 | 31.5 | 26.5 | 28.6 | 95.45 |
| 100 | 875 | 50.1 | 30.9 | 26.5 | 28.5 | 91.15 |
| 100 | 900 | 50.1 | 30.7 | 28.3 | 70.2 | 83.50 |
| 200 | 850 | 50.1 | 33.5 | 26.4 | 30.1 | 94.73 |
| 200 | 875 | 50.0 | 33.3 | 26.4 | 29.9 | 91.69 |
| 200 | 900 | 50.1 | 32.9 | 26.4 | 29.7 | 86.33 |
| 300 | 850 | 50.1 | 35.0 | 26.5 | 31.0 | 84.44 |
| 300 | 875 | 50.1 | 34.9 | 26.5 | 30.9 | 84.43 |
| 300 | 900 | 50.1 | 34.7 | 26.5 | 30.7 | 81.82 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platétube (Counter-current)

At Turbulent flow of warm water (50°C) and Turbulent flow of cold water

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 850 | 50.0 | 41.6 | 27.3 | 35.2 | 88.82 |
| 900 | 875 | 50.1 | 41.3 | 27.3 | 35.0 | 85.07 |
| 900 | 900 | 50.1 | 41.0 | 27.4 | 34.8 | 81.32 |
| 1000 | 850 | 50.1 | 42.1 | 26.4 | 35.4 | 95.63 |
| 1000 | 875 | 50.0 | 41.9 | 26.4 | 34.9 | 91.82 |
| 1000 | 900 | 50.1 | 41.4 | 26.4 | 34.2 | 80.69 |
| 1100 | 850 | 50.0 | 42.3 | 27.3 | 36.0 | 87.31 |
| 1100 | 875 | 50.1 | 42.1 | 27.3 | 35.7 | 83.52 |
| 1100 | 900 | 50.1 | 42.0 | 27.4 | 35.5 | 81.82 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 70.2 | 35.8 | 27.6 | 31.6 | 98.84 |
| 100 | 875 | 70.2 | 35.0 | 27.6 | 31.3 | 91.97 |
| 100 | 900 | 70.2 | 34.7 | 27.6 | 31.0 | 86.20 |
| 200 | 850 | 70.2 | 35.5 | 27.6 | 34.7 | 86.96 |
| 200 | 875 | 70.1 | 35.2 | 27.8 | 33.8 | 75.21 |
| 200 | 900 | 70.1 | 35.0 | 27.6 | 33.4 | 41.57 |
| 300 | 850 | 70.2 | 35.9 | 27.6 | 37.5 | 81.78 |
| 300 | 875 | 70.2 | 35.6 | 27.6 | 34.9 | 61.54 |
| 300 | 900 | 70.2 | 34.7 | 27.5 | 33.8 | 53.24 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Platetube (Counter-current)

At Turbulent flow of warm water (70$^{\circ}$C) and Turbulent flow of cold wat

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 900 | 850 | 70.2 | 47.8 | 27.3 | 44.9 | 74.21 |
| 900 | 875 | 70.1 | 47.5 | 27.3 | 44.5 | 73.99 |
| 900 | 900 | 70.1 | 47.2 | 27.4 | 44.1 | 72.93 |
| 1000 | 850 | 70.1 | 48.2 | 26.4 | 45.3 | 73.36 |
| 1000 | 875 | 70.2 | 47.8 | 26.4 | 44.9 | 72.26 |
| 1000 | 900 | 70.1 | 47.5 | 26.4 | 44.2 | 70.88 |
| 1100 | 850 | 70.1 | 49.1 | 26.4 | 45.5 | 70.28 |
| 1100 | 875 | 70.2 | 48.2 | 26.4 | 45.2 | 67.97 |
| 1100 | 900 | 70.1 | 47.3 | 26.4 | 45.0 | 66.75 |

Note

$T_1$ is temperature of warm water inlet

$T_2$ is temperature of warm water outlet

$t_1$ is temperature of cold water inlet

$t_2$ is temperature of cold water outlet

$\eta$ is efficiencies of heat transfer

Monotube (cover with insulator) (Co-current)

At Laminar flow of warm water (50°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 500 | 50.5 | 29.6 | 26.2 | 27.6 | 98.76 |

Monotube (cover with insulator) (Counter-current)

At Laminar flow of warm water (70°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 40.7 | 450 | 70.1 | 30.1 | 26.2 | 29.8 | 99.53 |

Shelltube (cover with insulator) (Co-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 450 | 80.1 | 42.3 | 26.6 | 29.4 | 98.3 |

Shelltube (cover with insulator) (Counter-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 33.9 | 750 | 82 | 39.9 | 27.7 | 29.5 | 99.82 |

Note

T1 is inlet temperature of warm water

T2 is outlet temperature of warm water

t1 is inlet temperature of cold water

t2is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

Platetube (cover with insulator) (Co-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 70.1 | 36.6 | 27.5 | 31.4 | 98.95 |

Platetube (cover with insulator) (Counter-current)

At Laminar flow of warm water (80°C) and Turbulent flow of cold water.

| Warm water (L/h) | Cold water (L/h) | $T_1$ (°C) | $T_2$ (°C) | $t_1$ (°C) | $t_2$ (°C) | $\eta$ (%) |
|---|---|---|---|---|---|---|
| 100 | 850 | 80.6 | 38.2 | 27.6 | 29.5 | 99.12 |

Note

T1 is inlet temperature of warm water

T2 is outlet temperature of warm water

t1 is inlet temperature of cold water

t2is outlet temperature of cold water

$\eta$ is efficiencies of heat transfer

# APPENDIX B

# CALCULATION

# APPENDIX B

# CALCULATION

## B.1 DETERMINATION OF THE REGIMES OF FLOW

The conditions of functioning are defined with the number of Reynolds (**Re**). This criteria allows to define the flow rate of flow of fluids for a flow in laminar regime and a flow in turbulent regime.

According to the value of this number **Re**, the regimes of flow are different.

TABLE B.1 Reynolds number

| Regime of flow | Reynolds number (**Re**) |
|---|---|
| Laminar regime | **Re** < on 2000 |
| Transitory regime | On 2000 < **Re** < 5000 |
| Turbulent regime | **Re** > 5000 |

This number is defined by the relation:

$$\boxed{Re = \frac{\rho\, Um\, D}{\mu}} \tag{B.1}$$

Where $\rho$ is specific gravity of the fluid ($kg/m^3$)

$\mu$ is viscosity of the fluid (Pa·s)

$D$ is diameter of the channeling (m)

$Um$ is average speed of the fluid (m/s)

**Um** is defined by the relation:

$$\boxed{Um = \frac{Qv}{\Omega}} \tag{B.2}$$

Where $Qv$ is volumic flow rate of the fluid ($m^3/s$)

$\Omega$ is section of the passage of the fluid ($m^2$)

$$\Omega = \frac{\pi D^2}{4} \quad \text{(B.3)}$$

The number of Reynolds becomes:

$$Re = \frac{\rho Qv D}{\Omega \mu} = \frac{4 \rho Qv D}{\pi \mu D^2} = \frac{4 \rho Qv}{\pi \mu D} \quad \text{(B.4)}$$

This expression allows us to define flow rate for the various regimes of functioning.

## B.2 THERMAL BALANCES

### B.2.1 THERMAL BALANCE ON THE COLD WATER

The thermal balance is given by the relation:

$$\Phi_f = Qv_f \cdot Cp_f \cdot \rho_f \cdot (t_2 - t_1) \quad \text{(B.5)}$$

where **Cp** is heat capacity ($J \cdot kg^{-1} \cdot °C^{-1}$)

$t_1$ is inlet temperature of cold water (°C)

$t_2$ is outlet temperature of cold water (°C)

### B.2.2 THERMAL BALANCE ON THE WARM WATER

The thermal balance is given by the relation:

$$\Phi_C = Qv_C \cdot Cp_C \cdot \rho_C \cdot (T_2 - T_1) \quad \text{(B.6)}$$

where **Cp** is heat capacity ($J \cdot kg^{-1} \cdot °C^{-1}$)

$T_1$ is inlet temperature of warm water (°C)

$T_2$ is outlet temperature of warm water (°C)

### B.2.3 YIELD ON THE HEAT EXCHANGER

The yield on the heat exchanger can be expressed by the relation:

$$\eta = \frac{Qv_f \cdot Cp_f \cdot \rho_f \cdot (t_2 - t_1)}{Qv_C \cdot Cp_C \cdot \rho_C \cdot (T_2 - T_1)} \times 100 \qquad \text{(B.7)}$$

## B.3 OVERALL HEAT TRANSFER COEFFICIENT OF EXPERIMENTAL EVALUATION OF U :($U_{EXP}$)

Relation allows to defined the coefficient of global thermal exchanger $U_{EXP}$

$$\Phi = U \cdot S \cdot \Delta Tmlog \qquad \text{(B.8)}$$

where **U** is the overall heat transfer coefficient of global exchange is through the surface (S)

**S** is the surface of exchange ($m^2$)

### B.3.1 AVERAGE DIFFERENCE OF TEMPERATURE

$\Delta$Tmlog is the logarithmic average of the temperatures of fluids on surfaces. It is defined by the relation:

$$\Delta Tmlog = \frac{\Delta T_1 - \Delta T_2}{Ln\,(\Delta T_1 / \Delta T_2)} \qquad \text{(B.9)}$$

where $\Delta T_1$ is difference of the temperature between fluids chill in the extremity where it is the strongest.

$\Delta T_2$ is difference of the temperature between fluids chill in the extremity where it is the weakest.

### B.3.2 AVERAGE THERMAL STREAM

For the calculation of the overall heat transfer coefficient of thermal exchange $U_{EXP}$, we take the average thermal stream. It is calculated by making average arithmetic enters the thermal stream $\Phi_C$ warm and the thermal stream $\Phi_f$ of the cold fluid.

$$\boxed{\Phi_{AVR} = \frac{\Phi_C + \Phi_f}{2}} \qquad \text{(B.10)}$$

### B.3.3 OVERALL HEAT TRANSFER COEFFICIENT OF THERMAL EXCHANGE : ($U_{EXP}$)

Relation allows to defined the coefficient of exchange:

$$\Phi = U \cdot S \cdot \Delta Tmlog$$

So

$$\boxed{U_{EXP} = \frac{\Phi_{AVR}}{\Delta Tmlog \cdot S}} \qquad \text{(B.11)}$$

## B.4 OVERALL HEAT TRANSFER COEFFICIENT OF THEORETICAL EVALUATION OF U : ($U_{THEO}$)

The thermal stream which exchanges in the wall of the heat exchanger is due to three terms

- Convection forced by the warm fluid
- Conduction through the wall of the heat exchanger
- Convection forced by the cold fluid

The following relation allows to define the global thermal coefficient of exchangers:

$$\boxed{U = \frac{1}{\frac{1}{hc} + \frac{e}{\lambda} + \frac{1}{hf}}} \quad (B.12)$$

### B.4.1 TERM OF THE CONDUCTION

Coefficient due to the conduction is

$$\boxed{Kcd = \frac{\lambda}{ec}} \quad (B.13)$$

where **ec** is thickness of the wall warm tube (nm).

$\lambda$ is thermal specific conductivity of stainless steel ($W \cdot m^{-1} \cdot °C^{-1}$)

### B.4.2 TERM DUE TO THE CONVECTION

#### B.4.2.1 CALCULATION OF THE NUMBER OF PRANDTL : (Pr)

Relation allows to define this number of Prandtl is:

$$\boxed{Pr = \frac{\mu Cp}{\lambda}} \quad (B.14)$$

#### B.4.2.2 CALCULATION OF THE NUMBER OF NUSSELT : (Nu)

The empirical relation which allows us to define the number of Nusselt is:

$$\boxed{Nu = 0.023 Re^{0.8} Pr^{\frac{1}{3}}} \quad (B.15)$$

### B.4.2.3 <u>CALCULATION OF THE COEFFICIENT OF CONVECTION ON WARM FLUID SIDE (hc) AND COLD FLUID SIDE (hf)</u>

Relation defines the number of Nusselt by:

$$\boxed{Nu = \frac{h\,L}{\lambda}} \qquad (B.16)$$

where **L** is characteristic which corresponds for a smooth tube to the internal diameter (D).

So

$$\boxed{hc\ (\text{or } hf) = \frac{\lambda \times Nu}{D}}$$

### B.4.2.4 <u>OVERALL HEAT TRANSFER COFFICIENT OF GLOBAL THERMAL EXCHANGE</u> : ($U_{THEO}$)

Equation defining the global coefficient of thermal exchange for a tube is :

$$\boxed{\frac{1}{U} = \frac{R_2}{R_1\,hc} + \frac{R_2\,Ln\,\frac{R_2}{R_1}}{\lambda} + \frac{1}{hf}} \qquad (B.17)$$

where $R_1$ is corresponds to the internal warm beam (shelf)

$R_2$ is corresponds to the outside warm beam (shelf)

## B.5 EXAMPLES OF CALCULATION

### CALCULATION : MONOTUBE HEAT EXCHANGER

### GEOMETRIC CHARACTERISTICS

Internal tube : Flow of the warm water

Shell : Flow of the cold water

| Name | Abbreviation | Value | Unity |
|---|---|---|---|
| Warm outside diameter | Dec | $10\times10^{-3}$ | m |
| Thickness of the warm tube | ec | $1\times10^{-3}$ | m |
| Length warmly | L | 4 | m |
| Cold outside diameter | Def | $25\times10^{-3}$ | m |
| Thickness of the cold Shell | ef | $2.5\times10^{-3}$ | m |

Material (subject) internal tube : stainless steel 316 L

: thermal specific conductivity of the stainless steel : 23.2 $W.m^{-1}$ $°C^{-1}$

Averaged Surfaceof exchange Sm

Sm : Surface of exchange : It corresponds to the average surface of the warm tube.

We use Dmc (average diameter of the warm tube) :

$Dmc = Dec - ec = 10 \times 10^{-3} - 1 \times 10^{-3} = 9\times10^{-3}$ m

$Dmc = 9 \times10^{-3}$ m

$$Sm = ¶ \times Dmc \times L = ¶ \times 9 \times 10^{-3} \times 4 = 1.13 \times 10^{-1}\ m^2$$

Section of passage of the warm fluid $\Omega C$

The section of passage of the warm fluid is :

$$\Omega C = \frac{¶ \times Dic^2}{4} = \frac{¶ \times (8 \times 10^{-3})^2}{4} = 5.03 \times 10^{-5}\ m^2$$

Section of passage of the cold fluid

The section of passage of the cold fluid corresponds to a ring.

The section corresponds to the internal section of the Shell decreased in the outside section of the warm tube.

$$\Omega F = \frac{¶ \times Dif^2}{4} - \frac{¶ \times Dec^2}{4} = ¶ \times (20 \times 10^{-3})^2 = 2.36 \times 10^{-4}\ m^2$$

Cold hydraulic diameter Dhf

Pm : perimeter wet by the cold fluid

$$Pm = ¶ (Dec + Dif) = ¶ (10 \times 10^{-3} + 20 \times 10^{-3}) = 9.42 \times 10^{-2} m$$

Dhf : cold hydraulic diameter

$$Dhf = \frac{4\Omega f}{Pm} = \frac{4 \times 2.36 \times 10^{-4}}{1 \times 10^{-2}} = 1 \times 10^{-2} m$$

For the warm fluid

Laminar regime : Re < on 2000

$$Qvc_{laminar} = \frac{Re\ ¶\ Dic\ \mu}{4\ \rho} = \frac{2000 \times ¶ \times 8 \times 10^{-3} \times 1 \times 10^{-3}}{4*1000} = 1.26 \times 10^{-5}\ m^3.s^{-1}$$

i.e. $Qvc_{laminar}$ lower than 45 l/h

Turbulent Regime : Re > 5000

$$Qvc_{turbulent} = \frac{Re\ ¶\ Dic\ \mu}{4\ \rho} = \frac{5000 \times ¶ \times 8 \times 10^{-3} \times 1 \times 10^{-3}}{4 \times 1000} = 3.14 \times 10^{-5}\ m^3.s^{-1}$$

i.e. $Qvc_{turbulent}$ superior to 113 l/h

For the cold fluid

Remark : cold fluid circulates in an annular canal as a ring.

The diameter of the canal is defined with the cold hydraulic diameter Dhf.

$$Dhf = \frac{4\ \Omega f}{Pm} = \frac{4¶ (Dif^2 - Dec^2)}{4 \times ¶ (Dif + Dec)} = (Dif - Dec) = 1*10^{-2}\ m$$

$$Re = \frac{\rho\ Umf\ Dhf}{\mu}$$

Let us look for Um's valve

$$Um = \frac{Qvf}{\Omega f} = \frac{4\ Qvf}{¶ (Dif^2 - Dec^2)}$$

Re becomes :

$$Re = \frac{\rho\ Umf\ Dhf}{\mu} = \frac{\rho \times 4 Qvf \times (Dif - Dec)}{¶ (Dif^2 - Dec^2)\ \mu} = \frac{4\ \rho \times Qvf}{¶\ \mu\ (Dif + Dec)}$$

Where from one deducts from it relation defining Qvf :

$$Qvf = \frac{Re \times ¶ (Dif + Dec)\ \mu}{4\ \rho}$$

$\mu$ : viscosity of the water : $1 \times 10^{-3}$ Pa.s

$\rho$ : specific gravity of the fluid : 1000 kg.m$^{-3}$

Dif + Dec = $(20 \times 10^{-3} + 10 \times 10^{-3}) = 3 \times 10^{-2}$ m

Laminar regime : Re < on 2000

$$Qvf_{laminar} = \frac{Re \times ¶\,(Dif + Dec)\,\mu}{4\rho} = \frac{2000 \times ¶ \times 3 \times 10^{-2} \times 1 \times 10^{-3}}{4 \times 1000} = 4.71 \times 10^{-5}\,m^3.s^{-1}$$

i.e.Qvc $_{laminar}$ lower than 170 l/h

Turbulent Regime : Re > 5000

$$Qvf_{turbulent} = \frac{Re \times ¶\,(Dif + Dec)\,\mu}{4\rho} = \frac{5000 \times ¶ \times 3 \times 10^{-2} \times 1 \times 10^{-3}}{4 \times 1000} = 1.18 \times 10^{-4}\,m^3.s^{-1}$$

i.e.Qvc $_{turbulent}$ superior to 424 l/h

Conclusion

| Regime of flow | Cold flow rate Qvf L.h$^{-1}$ | Warm flow rate Qvc L.h$^{-1}$ |
|---|---|---|
| laminar | < 170 | < 45 |
| turbulent | > 424 | > 113 |

NO SHIELD WITH FIBER GLASS INSULATOR

THERMAL BALANCE

Experimental results were chosen as follows

Flow with countercurrent 70 °C Cold water 450 L/h (TF) and warm water 40.7L/h (LF)

| On the cold fluid | On the warm fluid |
|---|---|
| $t_1$ = 27.4 °C | $T_1$ = 70.2 °C |
| $t_2$ = 30.8 °C | $T_2$ = 32 °C |
| Qvf = 450 L/h soit $450 \times 10^{-3} m^3 \times h^{-1}$ | Qvc = 40.7 L/h soit $450 \times 10^{-3} m^3 \times h^{-1}$ |

Physico – chemical characteristics of water :

$\mu$ : viscosite' de l'eau : $1.10^{-3}$ Pa.s

$\rho$ : specific gravity of the fluid : 1000 kg.m$^{-3}$

$C_p$ : chaleur massique de l'eau : 4181 J.kg$^{-1}$. °C$^{-1}$

$\lambda_L$ : conductivite' de l'eau : 0.598 W.m$^{-1}$.K$^{-1}$

Thermal balance on the cold water

The thermal balance is given by the relation:

$$\Phi_f = Qvf \times C\rho f \times \rho f \times (t_2 - t_1)$$

$$\Phi_f = \frac{450 \times 10^{-3}}{3600} \times 1000 \times 4181 \times (30.8 - 27.4) = 1776.92 \text{ W}$$

Thermal balance on the warm water

The thermal balance is given by the relation :

$$\Phi_c = Qvc \times C\rho c \times \rho c \times (T_1 - T_2)$$

From experimental results :

$$\Phi_c = \frac{40.69 \times 10^{-3}}{3600} \times 1000 \times 4181 \times (70.2 - 32) = 1805.21 \text{ W}$$

Yield on the heat exchanger

$$\eta = \frac{Qvf \times C\rho f \times \rho f \times (t_2 - t_1)}{Qvc \times C\rho c \times \rho c \times (T_1 - T_2)} \times 100$$

$$\eta = \frac{1776.92}{1805.2} \times 100 = 98.3\ \%$$

OVERALL HEAT TRANSFER COEFFICIENT OF EXPERIMENTAL EVALUATION OF $U_{EXP}$

Relation allows to define the coefficient of global thermal exchange $U_{EXP}$

$$\Phi = U \times S \times \Delta Tmlog$$

K the coefficient of global exchange is through the surface S.

Average different of temperature

$\Delta$Tmlog is the logarithmic average of the temperatures of fluids on surfaces. It is defined by the relation :

$$\Delta Tmlog = \frac{\Delta T_1 - \Delta T_2}{Ln\,(\Delta T_1 / \Delta T_2)}$$

$\Delta T_1$ = Difference of temperature between fluids chill in the extremity where it is the strongest.

$\Delta T_2$ = Difference of temperature between fluids chill in the extremity where it is the weakest .

$$\Delta T_1 = T_1 - t_2 = 70.2 - 30.8 = 39.4\ °C$$

$$\Delta T_2 = T_2 - t_1 = 32 - 27.4 = 4.6\ °C$$

$$\Delta T\ mlog = \frac{39.4 - 4.6}{Ln \frac{39.4}{4.6}} = 16.208\ °C$$

Average thermal stream

For the calculation of the coefficient of thermal exchange $U_{EXP}$, we take the average thermal stream. It is calculated by making average arithmetic enters the thermal stream $\Phi_C$ warm and the thermal stream $\Phi_f$ of the cold fluid.

$$\Phi_{AVR} = \frac{\Phi_C + \Phi_f}{2}$$

Relation allows to define the coefficient of exchange

$$\Phi_{AVR} = \frac{1805.21+1829.19}{2} = 1817.2\ W.$$

The surface of exchange Sm was calculated previously and is equal in :

$$Sm = ¶ \times Dmc \times L = ¶ \times 9 \times 10^{-3}\ 4 = 1.13 \times 10^{-1} m^{-2}$$

$$U_{exp} = \frac{\Phi_{AVR}}{\Delta Tmlog \cdot S}$$

$$= \frac{1817.2\ W}{(16.208\ K)\ (1.13 \times 10^{-1} m^{-3})} = 992.19\ Watt.m^{-2}.k^{-1}$$

OVERALL HEAT TRANSFER COEFFICIENT OF THEORETICAL EVALUATION OF U : $U_{THEO}$

The thermal stream which exchanges in the wall of the heat exchanger is due to three terms :

Convection forced by the warm fluid

Conduction through the wall of the heat exchanger

Convection forced by the cold fluid

The following relation allows to define the global thermal coefficient of exchanges:

$$U = \frac{1}{\frac{1}{hc} + \frac{e}{\lambda} + \frac{1}{hf}}$$

Term of the conduction

Coefficient due to the conduction is : Ucd = $\frac{\lambda}{ec}$

ec : thickness of the wall warm tube : 1mm

$\lambda$ : thermal specific conductivity ofstainless steel : 23.2 $W.m^{-1}.\ °C^{-1}$

$$Kcd = \frac{\lambda}{ec} = \frac{23.2}{1 \times 10^{-3}} = 23200\ W.m^{-2}.\ °C^{-1}$$

Term due to the convoction of the warm fluid hc

<u>Calculation of the number of Reynolds Re:</u>

Relation defining the number of Reynolds is :

$$Re = \frac{\rho\ Um\ D}{\mu}$$

$\rho$ : specific gravity of the fluid $kg/m^3$

$\mu$ : viscosity of the fluid Pa.s

D : diameter of the channeling m

Um : average speed of the fluid m/s

$$Um = \frac{Qv}{\Omega}$$

Um is defined by the relation : Um = <u>Qv</u>

Qv : volumic flow rate of the fluid $m^3 / s$

$\Omega$ : section of passage of the fluid m2

$$\Omega = \frac{¶ \times D^2}{4}$$

The number of Reynold becomes :

$$Re = \frac{\rho\ Qv\ D}{\Omega\ \mu} = \frac{4\ \rho\ Qv\ D}{¶\ \mu\ D^2} = \frac{4\ \rho\ Qv}{¶\ \mu\ D}$$

Qvc : 40.69 L/h

Dic : $8 \times 10^{-3}$

$\mu$ : $1 \times 10^{-3}$ Pa.s

$\rho$ : 1000 $kg.m^{-3}$ AND Re = 1800

The number of Reynold is widely superior to 5,000 so we are in Turbulent regime.

Calculation of the number of Prandtl Pr

Relation allows to define this number of Prandtl

$$Pr = \frac{\mu Cp}{\lambda}$$

Cp : 4181 $J.kg^{-1}{}^{o}C^{-1}$

$\mu$ : $1 \times 10^{-3}$ Pa.s

$\lambda_L$ : 0.598 $W.m^{-1}\ {}^{o}C^{-1}$

$$= \frac{1 \times 10^{-3} \times 4181}{0.598} = 6.99$$

Calculation of the number of Nusselt Nu

The condition of flow are :

Turbulent regime.

Smooth tube.

The emperical relation which allows us to define the number of Nusselt is:

$$Nu = 0.023\ Re^{0.8}\ Pr^{1/3}$$

Re : $4.42 \times 10^{3}$

Pr : 6.99

$$Nu = 0.023\ Re^{0.8}\ Pr^{1/3} = 0.023 \times (1800)^{0.8} \times 6.99^{1/3} = 17.05$$

Calculation of the coefficient of convection on warm fluid side hc

Relation defines the number of Nusselt by :

$$Nu = \frac{hL}{\lambda}$$

$\lambda_L$ : 0.598 $W.m^{-1}{}^{o}C^{-1}$

L : Characteristic dimension which corresponds for a smooth tube to the internal diameter Dic.

$$hc = \frac{\lambda_L \times Nu}{Dic} = \frac{(0.598\ W.m^{-1}.\ {}^{o}C^{-1})\ (17.55)}{(8 \times 10^{-3})}$$

$$hc = 1311.8625\ W.m^{-2}.\ {}^{o}C^{-1}$$

Term of in the convection of the cold fluid hf

Calculation of the number of Reynolds Re :

Relation defining the number of Reynolds is

$$Re = \frac{\rho \; Umf \; Dhf}{\mu}$$

In the case of the flow of the cold water, the section of passage is acrown.

The diameter is defined with the hydraulic diameter Dhf.

Cold hydraulic diameter Dhf

$$Dhf = \frac{4\,\Omega f}{Pm} = \frac{4 \times 2.36 \times 10^{-4}}{9.42 \times 10^{-2}} = 1 \times 10^{-2}$$

Calculation of the number of Prandtl Pr

Relation allows to define this number of Prandtl :

$$Pr = \frac{\mu Cp}{\lambda_L}$$

Cp : 4181 $J.kg^{-1}{}^{o}C^{-1}$

$\mu$ : $1 \times 10^{-3}$ Pa.s

$\lambda_L$ : 0.598 $W.m^{-1}\,{}^{o}C^{-1}$

$$Pr = \frac{\mu Cp}{\lambda_L} = \frac{1 \times 10^{-3} \times 4181}{0.598} = 6.99$$

Calculation of the number of Nusselt Nu

The condition of flow are :

Turbulent regime.

Smooth tube.

The empirical relation which allows us to define the number of Nusselt is :

$$Nu = 0.023\, Re^{0.8}\, Pr^{1/3}$$

Re : $4.42 \times 10^{4}$

Pr : 6.99

$$Nu = 0.023\, Re^{0.8}\, Pr^{1/3} = 0.023 \times (5296.61)^{0.8} \times 6.99^{1/3}$$

$$= 41.64$$

Calculation of the coefficient of convection on cold fluid side hf

Relation defines the number of Nusselt by :

$$Nu = \frac{hL}{\lambda}$$

$\lambda L$ : 0.598 $W.m^{-1}°C^{-1}$

L : Characteristic dimension which corresponds to the hydraulic diameter Dhf.

$$h_f = \frac{\lambda L \times Nu}{Dhf}$$

$$= \frac{0.598 \times 41.64}{1 \times 10^{-2}}$$

$$= 2490.072 \ W.m^{-2}.°C^{-1}$$

Overall heat transfer coefficient of global thermal exchange $U_{theo}$

Equation defining the global coefficient of thermal exchange foe a tube is :

$$\frac{1}{U} = \frac{R_2}{R_1 hc} + \frac{R_2 \, Ln \frac{R_2}{R_1}}{\lambda} + \frac{1}{hf}$$

R1 correspond to the internal warm beam (shelf) : $4 \times 10^{-3}$ m

R2 correspond to the outside warm beam (shelf) : $5 \times 10^{-3}$ m

hc : 1311.8625 $W.m^{-2}.°C^{-1}$

hf : 2490.072 $W.m^{-2}.°C^{-1}$

$\lambda$ : 23.2 $W.m^{-1}.°C^{-1}$

$$\frac{1}{U} = \frac{5 \times 10^{-3}}{(4 \times 10^{-3})(1311.8625)} + \frac{5 \times 10^{-3} \times Ln\,(5 \times 10^{-3}/4 \times 10^{-3})}{23.2} + \frac{1}{2490.072}$$

$U_{theo}$ = 713.012 $Watt.m^{-2}.k^{-1}$

THE COMPARISON OF TWO COEFFICIENTS :

$U_{exp}$ = 992.19 $Watt.m^{-2}.k^{-1}$

$U_{theo}$ = 713.012 $Watt.m^{-2}.k^{-1}$

WHEN SHIELDED WITH FIBER GLASS INSULATOR

THERMAL BALANCE

Experimental results chosen as these calculations are the following ones

Flow with countercurrent 70 °C Cold water 450 L/h (TF) and warm water 40.7L/h (LF)

On the cold fluid

$t_1 = 26.2$ °C

$t_2 = 29.8$ °C

On the warm fluid

$T_1 = 70.1$ °C

$T_2 = 30.1$ °C Qvf = 45 soit $450 \times 10^{-3} m^3 .h^{-1}$

Qvc = 40.7 L/h soit $450 \times 10^{-3} m^3 \times h^{-1}$

Physico – chemical characteristics of water :

$\mu$ : viscosite' de l'eau : $1 \times 10^{-3}$ Pa.s

$\rho$ : specific gravity of the fluid : 1000 $kg.m^{-3}$

$C_p$ : chaleur massique de l'eau : 4181 $J.kg^{-1}.°C^{-1}$

$\lambda_L$ : conductivite' de l'eau : 0.598 $W.m^{-1}.K^{-1}$

Thermal balance on the cold water

The thermal balance is given by the relation :

$$\Phi_f = Qvf \times C\rho f \times \rho f \times (t_2 - t_1)$$

$$\Phi_f = \frac{450 \times 10^{-3}}{3600} \times 1000 \times 4181 \times (30.1 - 26.2) = 1881.45 \text{ W}$$

Thermal balance on the warm water

The thermal balance is given by the relation :

$$\Phi_c = Qvc \times C\rho c \times \rho c \times (T_1 - T_2)$$

From experimental results :

$$\Phi_c = \frac{40.69 \times 10^{-3}}{3600} \times 1000 \times 4181 \times (70.1 - 29.8) = 1890.277$$

Yield on the heat exchanger

$$\eta = \frac{Qvf \times C\rho f \times \rho f \times (t_2 - t_1) \times 100}{Qvc \times C\rho c \times \rho c \times (T_1 - T_2)}$$

$$\eta = \frac{1881.45}{1890.28} \times 100 = 99.53\ \%$$

OVERALL HEAT TRANSFER COEFFICIENT OF EXPERIMENTAL EVALUATION OF EXPERIMENT $U_{EXP}$

Relation allows to define the coefficient of global thermal exchange $K_{EXP}$

$$\Phi = U \times S \times \Delta Tmlog$$

K the coefficient of global exchange is through the surface S.

**Average different of temperature**

$\Delta$Tmlog is the logarithmic average of the temperatures of fluids on surfaces. It is defined by the relation :

$$\Delta Tmlog = \frac{\Delta T_1 - \Delta T_2}{Ln\,(\Delta T_1 / \Delta T_2)}$$

$\Delta T_1$ = Difference of temperature between fluids chill in the extremity where it is the strongest.

$\Delta T_2$ = Difference of temperature between fluids chill in the extremity where it is the weakest .

$$\Delta T_1 = T_1 - t_2 = 70.1 - 29.8 = 40.3\ ^{\circ}C$$

$$\Delta T_2 = T_2 - t_1 = 30.1 - 26.2 = 3.9\ ^{\circ}C$$

$$\Delta T\ mlog = \frac{40.3 - 3.9}{Ln\,\frac{40.3}{3.9}} = 15.62\ ^{\circ}C$$

3.3.4.1 Average thermal stream

For the calculation of the coefficient of thermal exchange $U_{EXP}$, we take the average thermal stream. It is calculated by making average arithmetic enters the thermal stream $\Phi_C$ warm and the thermal stream $\Phi_f$ of the cold fluid.

$$\Phi_{AVR} = \frac{\Phi_C + \Phi_f}{2}$$

Relation allows to define the coefficient of exchange

$$\Phi_{AVR} = \frac{1890.277 + 1881.45}{2} = 1885.86\ W.$$

The surface of exchange Sm was calculated previously and is equal in :

$$Sm = ¶ \times Dmc \times L = ¶ \times 9 \times 10^{-3}\ 4 = 1.13 \times 10^{-1} m^{-2}$$

$$U_{exp} = \frac{\Phi_{AVR}}{\Delta Tmlog \cdot S}$$

$$= \frac{1885.86\ W}{(15.62\ K)\ (1.13 \times 10^{-1} m^{-3})}$$

$$= 1068.47\ Watt.m^{-2}.k^{-1}$$