FUNDAMENTALS OF HEAT EXCHANGERS

Associate Professor **Dr. Osama Mohammed Elmardi Suleiman Khayal**
Department of Mechanical Engineering, Faculty of Engineering and Technology, Nile Valley University, Atbara, Sudan

Abstract— Heat exchangers are used to transfer heat between two sources. The term exchanger applies to all equipment used to transfer heat between two streams. However, the term is commonly used to equipment in which two process streams exchange heat with each other. This research paper explains the basics of heat exchangers, covering such topics as: classification; design methods; pressure drop; analysis of extended surfaces; typical designs of double pipe, shell and tube compact, plate and frame, boilers, condensers and evaporators; and fouling of heat exchangers.

Keywords— **Heat exchangers, classifications, losses, extended surfaces, designs, fouling.**

I.   INTRODUCTION

Heat exchangers are devices used to transfer heat between two or more fluid streams at different temperatures. Heat exchangers find widespread use in power generation, chemical processing, electronics cooling, air-conditioning, refrigeration, and automotive applications. In this research paper we will examine the basic theory of heat exchangers and consider many applications. In addition, we will examine various aspects of heat exchanger design and analysis.

II.   HEAT EXCHANGER CLASSIFICATION

Due to the large number of heat exchanger configurations, a classification system was devised based upon the basic operation, construction, heat transfer, and flow arrangements. The following classification as outlined by Kakac and Liu (1998) will be discussed:
- Recuperators and regenerators.
- Transfer processes: direct contact or indirect contact.
- Geometry of construction: tubes, plates, and extended surfaces.
- Heat transfer mechanisms: single phase or two-phase flow.
- Flow Arrangement: parallel flow, counter flow, or cross flow.

III.   HEAT EXCHANGER DESIGN METHODS

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing. We will limit ourselves to the design of recuperators only. That is, the design of a two-fluid heat exchanger used for the purposes of recovering waste heat.

We will begin first, by discussing the basic principles of heat transfer for a heat exchanger. We may write the enthalpy balance on either fluid stream to give:

$$Q_c = \dot{m}_c \left(h_{c2} - h_{c1} \right) \qquad (1)$$

and

$$Q_h = \dot{m}_h \left(h_{h1} - h_{h2} \right) \qquad (2)$$

For constant specific heats with no change of phase, we may also write

$$Q_c = (\dot{m}c_p)_c (T_{c2} - T_{c1})$$ (3)

and

$$Q_h = (\dot{m}c_p)_h (T_{h2} - T_{h1})$$ (4)

Where $(\dot{m}c_p)_c$ and $(\dot{m}c_p)_h$ are the thermal capacities of the cold and hot fluid respectively.

Now from energy conservation we know that $Q_c = Q_h = Q$, and that we may relate the heat transfer rate Q and the overall heat transfer coefficient U, to some mean temperature difference ΔT_m by means of

$$Q = UA\Delta T_m$$ (5)

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

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

It is now clear that the problem of heat exchanger design comes down to obtaining an expression for the mean temperature difference. Expressions for many flow configurations, i.e. parallel flow, counter flow, and cross flow, have been obtained in the heat transfer field. We will examine these basic expressions later. Two approaches to heat exchanger design that will be discussed are the LMTD method and the effectiveness - NTU method. Each of these methods has particular advantages depending upon the nature of the problem specification.

A. Overall Heat Transfer Coefficient

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

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_i} + \frac{1}{Sk_w} + \frac{1}{(\eta_o hA)_o}$$ (7)

where η_0 is the surface efficiency of inner and outer surfaces, h is the heat transfer coefficients for the inner and outer surfaces, and S is a shape factor for the wall separating the two fluids.

The surface efficiency accounts for the effects of any extended surface which is present on either side of the parting wall. It is related to the fin efficiency of an extended surface in the following manner:

$$\eta_o = \left(1 - (1 - \eta_f)\frac{A_f}{A}\right)$$ (8)

The thermal resistances include: the inner and outer film resistances, inner and outer extended surface efficiencies, and conduction through a dividing wall which keeps the two fluid streams from mixing. The shape factors for a number of useful wall configurations are given below in Table 1 . Additional results will be presented for some complex doubly connected regions.

Equation (7) is for clean or unfouled heat exchanger surfaces. The effects of fouling on heat exchanger performance is discussed in a later section. Finally, we should note that

$$UA = U_o A_o = U_i A_i$$ (9)

however,

$$U_o \neq U_i$$ (10)

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

Table –1 Shape Factors

Geometry	S
Plane Wall	$\dfrac{A}{t}$
Cylindrical Wall	$\dfrac{2\pi L}{\ln\left(\dfrac{r_o}{r_i}\right)}$

| Spherical Wall | $\dfrac{4\pi r_i r_o}{r_o - r_i}$ |

Table –2 Order of Magnitude of h Kakac (1991)

Fluid	$h[W/m^2K]$
Gasses (natural convection)	5 – 25
Gasses (forced convection)	10 – 250
Liquids (non-metal)	100 – 10,000
Liquids (metal)	5000 – 250,000
Boiling	1000 – 250,000
Condensation	1000 – 250,000

B. LMTD Method

The logarithmic mean temperature difference (LMTD) is derived in all basic heat transfer texts. It may be written for a parallel flow or counter flow arrangement. The LMTD has the form:

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln \dfrac{\Delta T_2}{\Delta T_1}} \qquad (11)$$

where ΔT_1 and ΔT_2 represent the temperature difference at each end of the heat exchanger, whether parallel flow or counter flow. The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required.

The LMTD method is also applicable to crossflow arrangements when used with the crossflow correction factor. The heat transfer rate for a crossflow heat exchanger may be written as:

$$Q = FUA\Delta T_{LMTD} \qquad (12)$$

where the factor F is a correction factor, and the logarithmic mean temperature difference is based upon the counter flow heat exchanger arrangement.

The LMTD method assumes that both inlet and outlet temperatures are known. When this is not the case, the solution to a heat exchanger problem becomes somewhat tedious. An alternate method based upon heat exchanger effectiveness is more appropriate for this type of analysis. If $\Delta T_1 = \Delta T_2 = \Delta T$, then the expression for the LMTD reduces simply to ΔT.

C. $\in - NTU$ Method

The effectiveness / number of transfer units (NTU) method was developed to simplify a number of heat exchanger design problems. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate if there were infinite surface area. The heat exchanger effectiveness depends upon whether the hot fluid or cold fluid is a minimum fluid. That is the fluid which has the smaller capacity coefficient $C = \dot{m}C_p$. If the cold fluid is the minimum fluid then the effectiveness is defined as:

$$\in = \frac{C_{\max}\left(T_{H,in} - T_{H,out}\right)}{C_{\min}\left(T_{H,in} - T_{C,in}\right)} \qquad (13)$$

otherwise, if the hot fluid is the minimum fluid, then the effectiveness is defined as:

$$\in = \frac{C_{\max}\left(T_{C,out} - T_{C,in}\right)}{C_{\min}\left(T_{H,in} - T_{C,in}\right)} \qquad (14)$$

We may now define the heat transfer rate as:

$$Q = \in C_{\min}\left(T_{H,in} - T_{C,in}\right) \qquad (15)$$

It is now possible to develop expressions which relate the heat exchanger effectiveness to another parameter referred to as the number of transfer units (NTU). The value of NTU is defined as:

$$NTU = \frac{UA}{C_{\min}} \qquad (16)$$

It is now a simple matter to solve a heat exchanger problem when

$$\in = f(NTU, C_r) \qquad (17)$$

where $C_r = C_{\min}/C_{\max}$.

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The paper summarizes a number of these solutions and the special cases which may be derived from them.

For convenience the $\in$−NTU relationships are given for a simple double pipe heat exchanger for parallel flow and counter flow:

Parallel Flow

$$\in = \frac{1 - \exp[-NTU(1 + C_r)]}{1 + C_r} \qquad (18)$$

or

$$NTU = \frac{-\ln[1 - \in(1 + C_r)]}{1 + C_r} \qquad (19)$$

Counter Flow

$$\in = \frac{1 - \exp[-NTU(1 - C_r)]}{1 + C_r \exp[-NTU(1 - C_r)]}, when\, C_r < 1 \quad (20)$$

and

$$\in = \frac{NTU}{1 + NTU}, when\, C_r = 1 \qquad (21)$$

or

$$NTU = \frac{1}{C_r - 1}\ln\left(\frac{\in -1}{\in C_r -1}\right),\, when\, C_r < 1 \quad (22)$$

and

$$NTU = \frac{\in}{1 - \in}, when\, C_r = 1 \qquad (23)$$

For other configurations, the student is referred to Heat Transfer textbooks, or the handout of Elmardi August (2018) and September (2018). Often manufacturer's choose to present heat exchanger performance in terms of the inlet temperature difference ITD = $(T_{h,i} - T_{c,i})$. This is usually achieved by plotting the normalized parameter Q/ITD = Q/ $(T_{h,i} - T_{c,i})$. This is a direct consequence of the $\in - NTU$ method.

IV. HEAT EXCHANGER PRESSURE DROP

Pressure drop in heat exchangers is an important consideration during the design stage. Since fluid circulation requires some form of pump or fan, additional costs are incurred as a result of poor design. Pressure drop calculations are required for both fluid streams, and in most cases flow consists of either two internal streams or an internal and external stream. Pressure drop is affected by a number of factors, namely the type of flow (laminar or turbulent) and the passage geometry.

First, a fluid experiences an entrance loss as it enters the heat exchanger core due to a sudden reduction in flow area, then the core itself contributes a loss due to friction and other internal losses, and finally as the fluid exits the core it experiences a loss due to a sudden expansion. In addition, if the density changes through the core as a result of heating or cooling an acceleration or deceleration in flow is experienced. This also contributes to the overall pressure drop (or gain). All of these effects are discussed below.

Entrance Loss

The entrance loss for an abrupt contraction may be obtained by considering Bernoulli's equation with a loss coefficient combined with mass conservation to obtain:

$$\Delta p_i = \left(1 - \sigma_i^2 + K_c\right)\frac{1}{2}\frac{G^2}{\rho_i} \qquad (24)$$

where σ is the passage contraction ratio and G = $\dot{m}$/A, the mass flux of fluid. In general,

$$\sigma = \frac{\text{minimum flow area}}{\text{frontal area}} \qquad (25)$$

Core Loss

In the core, we may write the pressure drop in terms of the Fanning friction factor:

$$\Delta p_c = \frac{4fL}{D_h}\frac{1}{2}\frac{G^2}{\rho_m} \qquad (26)$$

Since the fluid density may change appreciably in gas flows, acceleration or deceleration may occur. We consider a momentum balance across the core

$$\Delta p_a A_c = \dot{m}(V_e - V_i) \qquad (27)$$

which may be written as

$$\Delta p_a = G^2\left(\frac{1}{\rho_e} - \frac{1}{\rho_i}\right) \qquad (28)$$

after writing $V = G/\rho$, since $G_i = G_e$.

Exit Loss

Finally, as the flow exits the core, the fluid may pass through a sudden expansion. Application of Bernoulli's equation with mass conservation results in

$$\Delta p_e = -\left(1 - \sigma_e^2 - K_e\right)\frac{1}{2}\frac{G^2}{\rho_e} \qquad (29)$$

where we have assumed that pressure drop (rise) is from left to right. Once again σ is the area contraction ratio and G is the mass flux of fluid.

Total Pressure Drop

The total pressure drop across the heat exchanger core is obtained by taking the sum of all of these contributions. That is

$$\Delta p = \Delta p_i + \Delta p_c + \Delta p_a + \Delta p_e \qquad (30)$$

Combining all of the effects and rearranging, yields the following general expression for predicting pressure drop in a heat exchanger core:

$$\Delta p = \frac{G^2}{2\rho_i}\Big[(1 - \sigma_i^2 + K_c) + f\frac{4L}{D_h}\langle\frac{\rho_i}{\rho_m}\rangle$$
$$+ 2\langle\frac{\rho_i}{\rho_e} - 1\rangle - (1 - \sigma_e^2 + K_e)\langle\frac{\rho_i}{\rho_e}\rangle\Big] \qquad (31)$$

Now the fluid pumping power is related to the overall pressure drop through application of conservation of energy

$$\dot{W}_p = \frac{1}{\eta_p}\frac{\dot{m}}{\rho}\Delta P \qquad (32)$$

where η_p is the pump efficiency. The efficiency accounts for the irreversibility in the pump, i.e. friction losses. It is clear that a Reynolds number dependency exists for the expansion and contraction loss coefficients. However, this dependency is small. For design purposes we may approximate the behavior of these losses by merely considering the Re = ∞ curves. These curves have the following approximate equations:

$$K_e = (1 - \sigma)^2 \qquad (33)$$

and

$$K_c \approx 0.42(1 - \sigma^2)^2 \qquad (34)$$

.

V. ANALYSIS OF EXTENDED SURFACES

Extended surfaces also known as fins, are widely used as a means of decreasing the thermal resistance of a system. The addition of fins as a means of increasing the overall heat transfer rate is widely employed in compact heat exchanger and heat sink design. The aim of this paper is to develop and present the theory of extended surfaces. A large number of analytic solutions for various types of fins will be presented in detail in addition to the development of numerical methods for complex fin geometries whose solution are not possible by analytic means.

A. One Dimensional Conduction with Convection

We begin first, by deriving the equation for one dimensional conduction with convection from first principles. The governing equation will be derived in general terms such that the results may be applied to both axial flow

systems such as longitudinal and pin fins, as well as radial flow systems such as the family of circular annular fins.

The governing equation for one dimensional flow with convection may be derived in general terms for longitudinal, radial, or pin fin configurations. Beginning with an arbitrary control volume, the conduction into the left face is given by the Fourier rate equation

$$Q_{cond,u} = -kA\frac{dT}{du} \qquad (35)$$

where k is the thermal conductivity, A = A(u) is the cross-sectional area, T = T(u) is the temperature, and u is the inward directed normal with respect to a particular coordinate system, i.e., for longitudinal or pin fins u = x and for radial fins u = r.

The conduction leaving the right face is

$$Q_{cond,u+du} = -kA\frac{dT}{du}\frac{d}{du}\left(-kA\frac{dT}{du}\right)du \qquad (36)$$

where du is the width of the control volume in the direction of conduction.

The convection loss at the surface of the control volume is obtained from Newton's Law of cooling

$$Q_{conv} = hPds\,(T - T_f) \qquad (37)$$

where h is the convection heat transfer coefficient, P = P(u) is the perimeter, ds is the arc length of the lateral surface, and T_f is the ambient fluid temperature.

Taking an energy balance over the control volume requires that

$$Q_{cond,u} - Q_{cond,u+du} - Q_{conv} = 0 \qquad (38)$$

which results in the following differential equation

$$\frac{d}{du}\left(kA\frac{dT}{du}\right) - hP\frac{ds}{du}\left(T - T_f\right) = 0 \qquad (39)$$

Now, introducing the temperature excess $\theta = T(u) - T_f$

$$\frac{d}{du}\left(A\frac{d\theta}{du}\right) - \frac{hP}{k}\frac{ds}{du}\theta = 0 \qquad (40)$$

gives the governing equation for one dimensional heat conduction with convection. Expansion of the differential yields

$$A\frac{d^2\theta}{du^2} + \frac{dA}{du}\frac{d\theta}{du} - \frac{hP}{k}\frac{ds}{du}\theta = 0 \qquad (41)$$

The governing equation is valid for both axial and radial systems having varying cross-sectional area and profile. The term ds/du is the ratio of lateral surface area to the projected area. It is related to the profile function y(u) through

$$\frac{ds}{du} = \sqrt{1 + \left(\frac{dy}{du}\right)^2} \qquad (42)$$

The above equation may be taken as unity, i.e. $(dy/du)^2 \approx 0$ for slender fin profiles, without incurring large errors. Thus, for slender fins having varying cross-sectional area and profile the governing equation becomes

$$A\frac{d^2\theta}{du^2} + \frac{dA}{du}\frac{d\theta}{du} - \frac{hP}{k}\theta = 0 \qquad (43)$$

The governing equation for one dimensional conduction with convection is applicable to systems in which the lateral conduction resistance is small relative to the convection resistance. Under these conditions the temperature profile is one dimensional. The conditions for which Equation (37) is valid are determined from the following criterion:

$$Bi = \frac{hb}{k} < 0.1 \qquad (44)$$

where Bi is the Biot number based upon the maximum half thickness of the fin profile. The fin Biot number is simply the ratio of the lateral conduction to lateral convection resistance

$$Bi = \frac{R_{conduction}}{R_{convection}} = \frac{\dfrac{b}{kA}}{\dfrac{1}{hA}} \qquad (45)$$

B. Boundary Conditions

The general fin equation is subject to the following boundary conditions at the fin tip $(u = u_e)$

$$\frac{d\theta(u_e)}{du} + \frac{h_e}{k}\,\theta(u_e) = 0 \qquad (46)$$

for truncated fins where h_e is the convection heat transfer coefficient for the edge surfaces, or

$$\frac{d\theta(u_e)}{du} = 0 \qquad (47)$$

the adiabatic tip conditions. At the fin base $(u = u_o)$

$$\theta(u_o) = \theta_o \qquad (48)$$

is generally prescribed.

For axial fins it will be convenient to take $u_e = 0$ and $u_o = L$, while for radial fins $u_e = r_o$ and $u_o = r_i$. In subsequent sections, analytic results will be obtained for each class of fin for various profile shapes. Once the solution for the temperature excess for a particular case has been found, the solution for the heat flow at the fin base may be obtained from the Fourier rate equation

$$Q_b = -kA\frac{d\theta}{du} \qquad (49)$$

applied to the base of the fin.

C. Fin Performance

Fin performance has traditionally been measured by means of the fin efficiency or fin effectiveness. Fin efficiency may be defined as

$$\eta_f = \frac{Q_b}{Q_{max}} = \frac{Q_b}{hA_s\theta_b} \qquad (50)$$

where Q_{max} is the maximum heat transfer rate if the temperature at every point within the fin were at the base temperature θ_b. The fin effectiveness may be defined as

$$\in = \frac{Q_{b,fin}}{Q_{b,bare}} = \frac{Q_b}{hA_b\theta_b} \qquad (51)$$

where $Q_{b,bare}$ is the heat transfer from the base of the fin when the fin is not present, i.e. $L \rightarrow 0$.

In many heat sink design applications, it is often more convenient to consider the fin resistance defined as

$$R_{fin} = \frac{\theta_b}{Q_b} \qquad (52)$$

The use of the fin resistance is more appropriate for modeling heat sink systems, since additional resistive paths may be considered.

D. Analytical Solutions

In this section we examine the many analytical solutions which have been obtained in various fin configurations. In most cases, the solutions for the temperature distribution involve special functions such as the modified Bessel functions. A thorough review of analytical methods pertaining to extended surfaces may be found in the classic text by Kern and Kraus (1972). While more brief reviews are found in most advanced texts on heat conduction such as those by Arpaci (1966) and Schnieder (1955), along with the more general advanced heat transfer texts such as those by Jakob (1949) and Eckert and Drake (1972). Analytical methods have been successfully applied to a number of applications of extended surfaces such as longitudinal fins, pin fins, and circular annular fins.

VI.   TYPICAL HEAT EXCHANGER DESIGNS

We will now examine several common heat exchanger designs and highlight the differences in each of the configurations. We shall consider: Double Pipe Heat Exchangers, Shell and Tube Heat Exchangers, Compact Heat Exchangers, Plate and Frame Heat Exchangers, and Boilers, Condensers, and Evaporators.

A. Double Pipe Exchangers

The double pipe heat exchanger is probably one of the simplest configurations found in applications. It consists of two concentric circular tubes with one fluid flowing inside the inner tube and the other fluid flowing inside the annular space between the tubes. Its primary uses are in cooling process fluids where small heat transfer areas are required. It may be designed in a number of arrangements such as parallel flow and counter flow, and combined in series or parallel arrangements with other heat exchangers to form a system.

For this configuration the overall heat transfer coefficient is given by:

$$\frac{1}{UA} = \frac{1}{h_i(2\pi r_i L)} + \frac{\ln(r_o/r_i)}{2\pi k_w L} + \frac{1}{h_o(2\pi r_o L)} \quad (53)$$

where r_i and r_o denote the radii of the inner pipe. The heat transfer coefficient h_i is computed for a pipe while the heat transfer coefficient h_o is computed for the annulus. If both fluids are in turbulent flow, the heat transfer coefficients may be computed using the same correlation with $D = D_h$, otherwise, special attention must be given to the annular region.
The pressure drop for each fluid may be determined from:

$$\Delta p \left[\sum \frac{4fL}{D_h} + \sum K \right] \frac{1}{2}\rho V^2 \quad (54)$$

However, care must be taken to understand the nature of the flow, i.e. series, parallel, or series-parallel.

B. Shell and Tube Exchangers

Shell and tube heat exchangers are widely used as power condensers, oil coolers, preheaters, and steam generators. They consist of many tubes mounted parallel to each other in a cylindrical shell. Flow may be parallel, counter, or cross flow and in some cases combinations of these flow arrangements as a result of baffling. Shell and tube designs are relatively simple and most often designed according to the Tubular Exchanger Manufacturer's Association (TEMA) standards.
For this configuration the overall heat transfer coefficient is given by (ignoring fouling):

$$\frac{1}{UA} = \frac{1}{h_i A_i} + R_w + \frac{1}{h_o A_o} \quad (55)$$

where A_i and A_o denote the inner and outer areas of the tubes. The heat transfer coefficient h_i is computed for a tube while the heat transfer coefficient h_o is computed for tube bundles in either parallel or cross flow depending on whether baffling is used. Special attention must be given to the internal tube arrangement, i.e. baffled, single pass, multi-pass, tube pitch and arrangement, etc., to properly predict the heat transfer coefficient. Often, unless drastic changes occur in the tube count, the shell side heat transfer coefficient will not vary much from an initial prediction. Often a value of $h_o = 5000$ W/m^{2}K is used for preliminary sizing.
The heat transfer surface area is calculated from:

$$A_o = \pi d_o N_t L \quad (56)$$

where d_o is the outer diameter of the tubes, N_t is the number of tubes, and L is the length of the tubes.
The number of tubes that can fit in a cylindrical shell is calculated from:

$$N_t = CTP \frac{\pi D_s^2}{4 CL P_t^2} \quad (57)$$

The factor CTP is a constant that accounts for the incomplete coverage of circular tubes in a cylindrical shell, i.e. one tube pass CTP = 0.93, two tube passes CTP = 0.9, and three tube passes CTP = 0.85. The factor CL is the tube layout constant given by CL = 1 for 45 and 90-degree layouts, and CL = 0.87 for 30 and 60-degree layouts. Finally, P_t is the tube pitch and D_s is the shell diameter.
The shell diameter may be solved for using the above two equations, to give:

$$D_s = 0.637 \sqrt{\frac{CL}{CTP}} \left[\frac{A_o P_t^2}{d_o L} \right]^{1/2} \quad (58)$$

The shell side heat transfer coefficient is most often computed from the following experimental correlation:

$$Nu_{D_e} = 0.36 \, Re_{D_s}^{0.55} \, Pr^{1/3} \quad (59)$$

for

$$2 \times 10^3 < Re_{D_e} < 1 \times 10^6$$

. The effective diameter D_e is obtained from

$$D_e = \frac{4\left(P_t^2 - \pi d_o^2/4\right)}{\pi d_o} \quad (60)$$

for a square tube arrangement, and

$$D_e = \frac{8\left(\sqrt{3}P_t^2/4 - \pi d_o^2/8\right)}{\pi d_o} \quad (61)$$

for a triangular tube arrangement. The tube side heat transfer coefficient is computed from an appropriate tube model depending on the type of flow, i.e. laminar or turbulent.

The pressure drop for the tube side is often predicted using the following formula (Kakac and Liu, 1998):

$$\Delta p_t = \left[\frac{4fLN_p}{d_i} + 4N_p \right] \frac{1}{2} \rho V^2 \qquad (62)$$

where N_p is the number of tube passes.

This accounts for tube friction and internal losses due to the return bends. The friction factor f is computed from an appropriate tube model depending on the type of flow, i.e. laminar or turbulent. The shell side pressure drop may be predicted from

$$\Delta p_s = \frac{fG_s^2(N_b + 1)D_s}{2\rho D_e} \qquad (63)$$

and

$$G_s = \frac{\dot{m}P_t}{D_s CB} \qquad (64)$$

where N_b is the number of baffles, C is the clearance between adjacent tubes, B is the baffle spacing, while f is determined from

$$f = \exp\left[0.576 - 0.19\ln(G_sD_e/\mu)\right] \qquad (65)$$

which is valid for $400 < G_sD_e/\mu < 1 \times 10^6$. These correlations have been tested against many shell and tube designs and have been found to provide very good results. However, care must be taken to understand the nature of the flow, i.e. parallel, cross flow, or cross flow with baffles. In general, when designing shell and tube heat exchangers, the TEMA standards should be followed.

C. Compact Heat Exchangers

Compact heat exchangers offer a high surface area to volume ratio typically greater than 700 m^2/m^3 for gas-gas applications, and greater than 400 m^2/m^3 for liquid-gas applications. They are often used in applications where space is usually a premium such as in aircraft and automotive applications. They rely heavily on the use of extended surfaces to increase the overall surface area while keeping size to a minimum. As a result, pressure drops can be high. Typical applications include gas-to-gas and gas-to-liquid heat exchangers. They are widely used as oil coolers, automotive radiators, intercoolers, cryogenics, and electronics cooling applications.

For this configuration the overall heat transfer coefficient is given by:

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_i} + \frac{t}{k_w A_w} + \frac{1}{(\eta_o hA)_o} \qquad (66)$$

where η_o is the overall surface efficiency. In most compact heat exchanger design problems, the heat transfer and friction coefficients are determined from experimental performance charts or models for enhanced heat transfer surfaces. The pressure drop is also computed using the general method discussed section 4.

D. Plate and Frame Exchangers

Plate heat exchangers consist of a series of thin corrugated formed metal plates. Each pair of plates forms a complex passage in which the fluid flows. Each pair of plates are then stacked together to form a sandwich type construction in which the second fluid flows in the spaces formed between successive pairs of plates. These types of heat exchangers provide for a compact and lightweight heat transfer surface. As a result of the small plate spacing and corrugated design, high heat transfer coefficients result along with strong eddy formation which helps minimize fouling. Because of the simple construction, they are easily cleaned and find wide use in food processing applications.

E. Boilers, Condensers, and Evaporators

A condenser and evaporator are heat exchangers in which a change of phase results. In a condenser, a vapor is converted into a liquid, while in an evaporator (and a boiler) liquid is converted into a vapor. Due to the two-phase nature of these devices, design is not as straight forward. Two phase fluid flows are much more complex than their single-phase counterparts. Additional understanding of the phase make up and distribution is required to perform the necessary design calculations. In addition, design correlations for two phase flows can be somewhat complicated.

VII. TYPICAL HEAT EXCHANGER DESIGNS

Fouling in heat exchangers represents a major source of performance degradation. Fouling not only contributes to a decrease in thermal efficiency, but also hydraulic efficiency. The buildup of scale or other deposit increases the overall thermal resistance of the heat exchanger core which directly reduces the overall thermal efficiency. If buildup of a fouling deposit is significant, it can also increase pressure drop due to the reduced flow area in the heat exchanger core. The two effects combined can lead to serious performance degradation. In some cases, the degradation in hydraulic performance is greater than the degradation in thermal performance which necessitates cleaning of the heat exchanger on a regular basis. Fouling of heat exchangers results in a number of ways. The two most common are corrosion and scale build up. However, depending upon the nature of the fluid other factors may contribute to fouling.

Table –3 TEMA Design Fouling Resistances Rf for a Number of Industrial Fluids

Fluid	$R_f'' = R_f A$ [m^2K/kW]
Engine Oil	0.176
Fuel Oil no. 2	0.352
Fuel Oil no. 6	0.881
Quench Oil	0.705
Refrigerants	0.176
Hydraulic Fluids	0.176
Ethylene Glycol Solutions	0.352
Exhaust gases	1.761
Natural Gas Flue Gases	0.881
Coal Flue Gases	1.761

Fouling in heat exchangers is traditionally treated using the concept of a fouling resistance. This resistance is added in series to either side of the wall resistance in the definition of the overall heat transfer coefficient.

$$\frac{1}{UA} = \frac{1}{\eta_i h_i A_i} + R_{f,i} + \frac{1}{Sk_w} + R_{f,o} + \frac{1}{\eta_o h_o A_o} \qquad (67)$$

The fouling resistance may be computed from

$$R_f = \frac{t_f}{k_f A_w} \qquad (68)$$

for a plane wall, and

$$R_f = \frac{\ln(d_f / d_c)}{2\pi k_f L} \qquad (69)$$

for a tube.

Unfortunately, fouling in heat exchangers has not been modeled adequately for predictive purposes. Some typical values of fouling resistances are given in Table 3 for a number of fluids.

Another method of designing for fouling, is through the specification of percentage over surface, that is increasing the surface area to initially provide for a heat exchanger which exceeds the design heat transfer rate. This is often done for heat exchangers which cannot be cleaned easily such as shell and tube heat exchangers. Often as a rule 25 percent over surface is prescribed. The percentage over surface is defined as

$$\%OS = \left(\frac{A_f}{A_c} - 1\right) \times 100 \qquad (70)$$

The effect that fouling has on a heat exchanger's performance can be seen by examining the change in pressure drop:

$$\frac{\Delta p_f}{\Delta p_c} = \frac{f_f}{f_c} \frac{d_c}{d_f} \left(\frac{V_f}{V_c}\right)^2 \qquad (71)$$

If the mass flow rate is constant, then $\dot{m} = \rho V_c A_c = \rho V_f A_f$, and we obtain

$$\frac{\Delta p_f}{\Delta p_c} = \frac{f_f}{f_c} \left(\frac{d_c}{d_f}\right)^5 \qquad (72)$$

If the pressure drop is maintained constant, then we obtain, after substituting for mass flow rate:

$$\frac{\dot{m}_f}{\dot{m}_c} = \frac{f_c}{f_f}\left(\frac{d_f}{d_c}\right)^3 \tag{73}$$

For any other condition, such as pump driven flow, we may solve for the new operating point. Fouling will effectively increase the system curve, which shifts the operating point to the left, i.e. lowering the actual flow. If the flow rate decreases, then the heat transfer coefficient also decreases. This decrease combined with the increased resistance due to the fouling layer, leads to an overall decrease in thermal performance, i.e. a lowering of the $Q/(T_{h,i} - T_{c,i})$ curve.

VIII. CONCLUSIONS

This research paper explains the basics of heat exchangers, their design methods and how to measure their performance. Therefore, it is useful for undergraduate and graduate students.

ACKNOWLEDGEMENT

IX.   REFERENCES

[1] Bejan, A., Heat Transfer, 1993, Wiley, New York, NY.
[2] Kakac, S. (ed.), Boilers, Evaporators, and Condensers, 1991, Wiley, New York, NY.
[3] Kakac, S. and Liu, H., Heat Exchangers: Selection, Rating, and Thermal Performance, 1998, CRC Press, Boca Raton, FL.
[4] Kays, W.M. and London, A.L., Compact Heat Exchangers, 1984, McGraw-Hill, New York, NY.
[5] Kern, D.Q. and Kraus, A.D., Extended Surface Heat Transfer, 1972, McGraw-Hill, New York, NY.
[6] McQuiston, F.C. and Parker, J.D., Heating, Ventilation, and Air Conditioning: Analysis and Design, 1988, Wiley, New York, NY.
[7] Elmardi, Osama Mohammed, Solution of Problems in Heat Exchanger in Arabic, August (2018), www.ektab.com, Jordan.
[8] Elmardi, Osama Mohammed, Heat Transfer: Solved Examples and Additional Problems in Arabic, September (2018), www.ektab.com, Jordan.
[9] Rohsenow, W.M., Hartnett, J.P, Cho, Y.I., Handbook of Heat Transfer, 1998, McGraw-Hill, New York, NY.
[10] Shah, R.K. and Sekulic, D., Fundamentals of Heat Exchanger Design, 2003, Wiley, New York, NY.
[11] Smith, E.M., Thermal Design of Heat Exchangers, 1995, Wiley, New York, NY.

Author

Osama Mohammed Elmardi Suleiman was born in Atbara, Sudan in 1966. He received his diploma degree in mechanical engineering from Mechanical Engineering College, Atbara, Sudan in 1990. He also received a bachelor degree in mechanical engineering from Sudan University of science and technology – Faculty of engineering in 1998, and a master degree in solid mechanics from Nile valley university (Atbara, Sudan) in 2003, and a PhD in structural engineering in 2017. He contributed in teaching some subjects in other universities such as Red Sea University (Port Sudan, Sudan), Kordofan University (Obayed, Sudan), Sudan University of Science and Technology (Khartoum, Sudan) and Blue Nile university (Damazin, Sudan). In addition, he supervised more than hundred and fifty under graduate studies in diploma and B.Sc. levels and about fifteen master theses. The author wrote about twenty-two engineering books written in Arabic language, and twelve books written in English language and fifty research papers in fluid mechanics, thermodynamics, internal combustion engines and analysis of composite structures. He is currently an assistant professor in department of mechanical engineering, Faculty of Engineering and Technology, Nile Valley University. His research interest and favorite subjects include structural mechanics, applied mechanics, control engineering and instrumentation, computer aided design, design of mechanical elements, fluid mechanics and dynamics, heat and mass transfer and hydraulic machinery. The author is also works as a technical manager and superintendent of Al – Kamali mechanical and production workshops group which specializes in small, medium and large automotive overhaul maintenance and which situated in Atbara town in the north part of Sudan, River Nile State.