**Unit 8 Energy Performance assessment of Heat Exchangers**

**Objectives:** At the end of this chapter the reader would be able to describe the various terms associated with the performance assessment of heat exchangers and explain how the performance assessment of a heat exchanger is carried out.

**Pre-requisites:** Understanding of basic thermodynamic and heat transfer principles, basic understanding of fluid mechanics.

**8.1 Performance terms and Methodology of performance assessment**

A *heat exchanger* is a device that is used to transfer thermal energy between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact[[1]](#footnote-1). The various types of heat exchangers have been discussed in Unit 6. Now let us discuss how the performance of heat exchangers can be assessed.

Determination of heat transfer and pressure drop performance of either an existing exchanger or an already sized exchanger is also known as the rating problem. Inputs to the rating problem are the heat exchanger construction, flow arrangement and overall dimensions, complete details on the materials and surface geometries, fluid flow rates, inlet temperatures, and fouling factors. The fluid outlet temperatures, total heat transfer rate, and pressure drops on each side of the exchanger are then determined in the rating problem. The rating problem is also sometimes referred to as the performance or simulation problem.

In general the performance of a heat exchanger is influenced by –

1. Temperature Difference (ΔT) between the two fluids - This is the driving force in heat exchange principles. The greater the ΔT, the greater the heat transfer rate.
2. Fluid flow rate - Increasing flow rate will increase heat transfer rate.
3. The nature of the heat conducting materials - Some materials have a high conductivity while others don't. This factor is 'built-in' in the design of the exchanger and choice of materials.
4. Surface area - The larger the surface area of the conducting interfaces, the greater the heat transfer rate.

Heat transfer characteristics of an exchanger surface are presented in terms of the Nusselt number, Prandtl number, Stanton number, Colburn factor, Reynolds number, etc. Flow friction characteristics are presented in terms of the Fanning friction factor vs. Re. Some of the important dimensionless groups used in heat exchanger design and study are summarized in the table below with their physical meanings.

**Table 8.1: Some dimensionless groups used in heat transfer study[[2]](#footnote-2)**

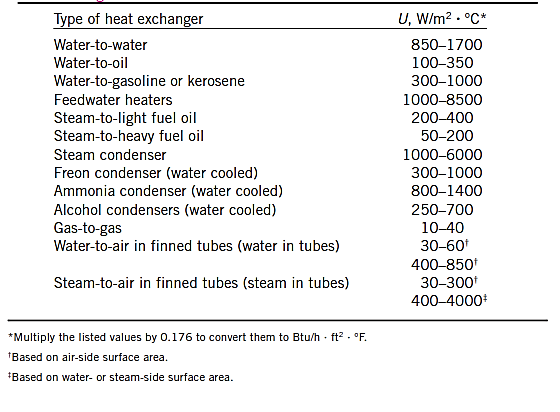
|  |  |
| --- | --- |
| Dimensionless Group | Physical Meaning and Comment |
| Reynolds Number (Re) | Flow modulus, proportional to the ratio of flow momentum rate ("inertia force") to viscous force |
| Fanning friction factor (*f)* | Ratio of wall shear (skin frictional) stress to the flow kinetic energy per unit volume |
| Nusselt number (Nu) | Ratio of the convective conductance h to pure molecular thermal conductance k=Dh over the hydraulic diameter |
| Prandtl number (Pr) | Fluid property modulus representing the ratio of momentum diffusivity to thermal diffusivity of the fluid |
| Stanton number (St) | Ratio of convection heat transfer (per unit duct surface area) to the enthalpy rate change of the fluid reaching the wall temperature; St does not depend on any geometric characteristic dimension |
| Colburn factor (*j*) | Modified Stanton number to take into account the moderate variations in the Prandtl number for 0.5 ≤ Pr ≤ 10 in turbulent flow |

The influence of various parameters on the heat transfer rate can be studied by developing a theoretical model based on different equations. Models are theoretical representations that simulate the behavior or activity of systems, processes, or phenomena. They include the use of mathematical equations, computers, and other electronic equipment. Heat transfer modeling helps us to numerically define the heat transfer characteristics of a system. This characterization helps in predicting the performance of the thermal system with variation in different parameters.

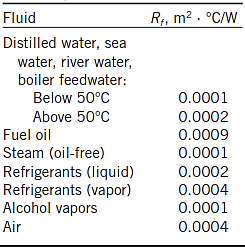
In the analysis of heat exchangers, it is convenient to combine all the thermal resistances in the path of heat flow from the hot fluid to the cold one into a single resistance *R*, and to express the rate of heat transfer between the two fluids as

--------- (8.1)

where *U* is the **overall heat transfer coefficient,** whose unit is W/m2 °C, which is identical to the unit of the ordinary convection coefficient *h. A* is the heat transfer surface area and the subscripts *i* and *o* denote the inside and outside surface respectively. Table 8.2 shows the representative values of the overall heat transfer coefficients in some types of heat exchangers.

**Table 8.2: Overall Heat Transfer Co-efficient of some Heat Exchangers[[3]](#footnote-3)** 

Over the period of time the performance of a heat exchanger deteriorates due to deposition and accumulation on the heat transfer surfaces. These layers of deposits act as additional resistance to the transfer of heat between the fluids. The **fouling factor** *Rf*, gives a measure of the *thermal resistance* introduced by fouling. Table 8.3 shows the representative values of fouling factor.

**Table 8.2: Fouling Factor values of some commercially used fluids[[4]](#footnote-4)** 

In heat exchanger analysis, the product of the *mass flow rate* and the *specific heat* of a fluid is combined into a single quantity called the **heat capacity rate** and is defined for the hot and cold fluid streams as

----------(8.2)

The heat capacity rate of a fluid stream represents the rate of heat transfer needed to change the temperature of the fluid stream by 1°C as it flows through a heat exchanger.

The temperature difference between the hot and cold fluids varies along the heat exchanger, and it is convenient to have a *mean temperature difference* Δ*Tm* for use in the relation

--------(8.3)

If we try to develop a relation for the equivalent mean temperature difference between the two fluids (hot and cold) for a parallel flow heat exchanger we arrive at the relation

----------(8.4)

where

----------(8.5)

*ΔTlm* is known as the log mean temperature difference which is the equivalent mean temperature difference between the two fluids. Here *ΔT1* and *ΔT2* represent the temperature difference between the two fluids at the two ends (inlet and outlet).

It is observed that the same relations hold good for the analysis of counter flow heat exchangers also. However, for cross flow and multi-pass shell and tube heat exchangers a correction factor *F* has to be applied. The log mean temperature difference in these cases become *ΔTlm* = *F. ΔTlm,CF* where *ΔTlm,CF* is the log mean temperature difference for the counter flow arrangement. The correction factor *F* depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The correction factor *F* for different heat exchanger configurations and fluid flow patterns are shown in figures 8.1 (a) to (d)[[5]](#footnote-5).

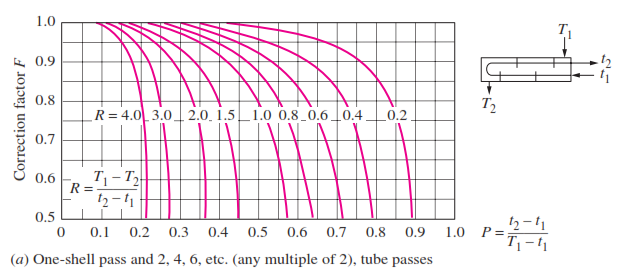


Figure 8.1 (a)

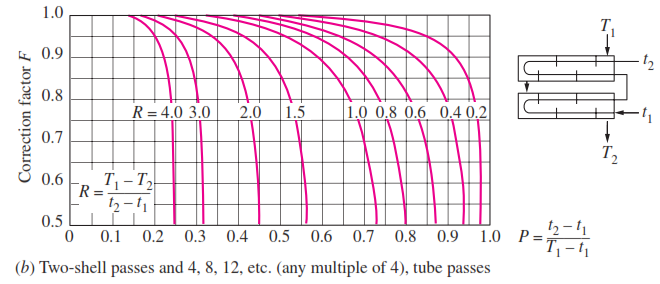


Figure 8.1 (b)

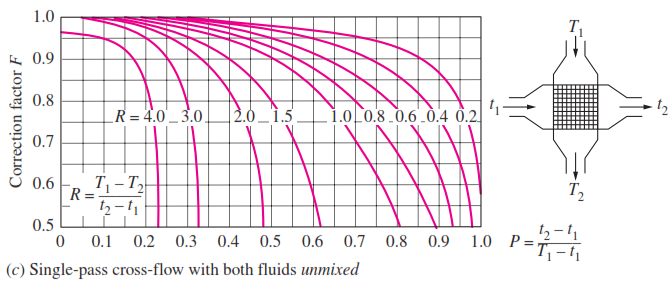


Figure 8.1 (c)

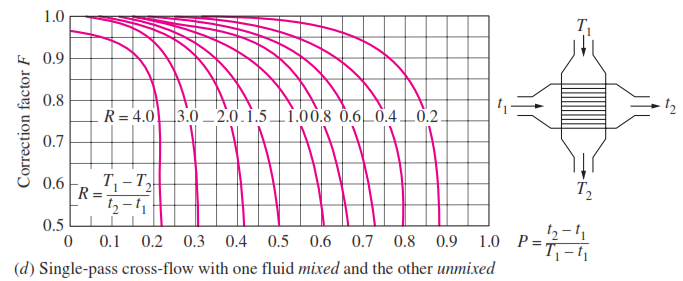


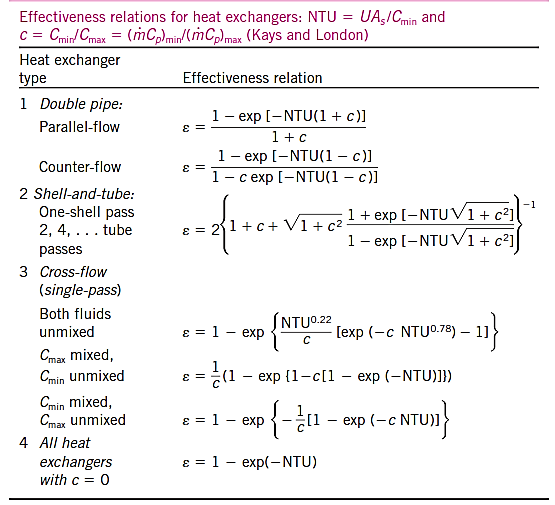
Figure 8.1 (d)

The LMTD method is used to determine the size of the heat exchanger when the mass flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified. A second kind of problem encountered in the analysis of heat exchangers is the determination of the heat transfer rates and the outlet temperatures of the hot and cold fluids when the mass flow rate is prescribed. Although LMTD method can be used for this but the process would involve a large number of iterations. So a different approach known as the effectiveness-NTU method is used. This method is based on the dimensionless parameter called the effectiveness, ɛ and defined as

----------(8.6)

The effectiveness of a heat exchanger depends on the *geometry* of the heat exchanger as well as the *flow arrangement.* Therefore, different types of heat exchangers have different effectiveness relations. Effectiveness relations for some types of heat exchangers are shown in Table 8.5.

Table 8.5[[6]](#footnote-6)



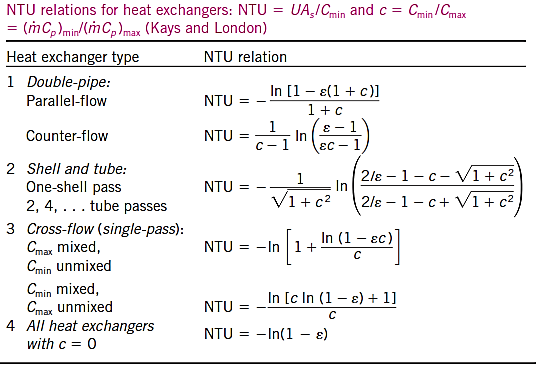
Effectiveness relations of the heat exchangers typically involve the *dimensionless* group *UAs/Cmin* called the **number of transfer units NTU** where U is the overall heat transfer coefficient, As is the heat transfer surface area and Cmin is the minimum of the heat capacities (Cc or Ch).

Another dimensionless quantity that comes into picture is the heat capacity ratio *c* and is given by

*c = Cmin/Cmax* ----------(8.7)

Knowing the inlet and outlet conditions of the hot and the cold sides the ɛ-NTU method can also be used to determine the size of the heat exchanger. The NTU value gives a measure of the size of the heat exchanger. Table 8.6 shows the NTU relations for different heat exchangers.

Table 8.6[[7]](#footnote-7)



Having discussed the various terms related with the study of a heat exchanger let us now discuss a proven technique through which the performance of a heat exchanger can be evaluated in the field. A step by step method to evaluate the performance of a heat exchanger is discussed below[[8]](#footnote-8).

**Step – A**

**The steady parameters are monitored and recorded in the format shown below:**

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameters** | **Units** | **Inlet** | **Outlet** |
| Hot fluid flow,Wh | kg/h |  |  |
| Cold fluid flow,Wc | kg/h |  |  |
| Hot fluid Temp, Th | OC |  |  |
| Cold fluid Temp,Tc | OC |  |  |
| Hot fluid Pressure,Ph | bar |  |  |
| Cold fluid Pressure, Pc | bar |  |  |

**Step – B**

**With the monitored data, the parameters required for the evaluation of the thermal data are calculated as shown in the table below**

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameters** | **Units** | **Inlet** | **Outlet** |
| Hot fluid density, ρh | kg/m3 |  |  |
| Cold fluid density, ρc | kg/m3 |  |  |
| Hot fluid Viscosity, µh | MpaS\* |  |  |
| Cold fluid Viscosity, µc | MPaS |  |  |
| Hot fluid Thermal Conductivity, kh | kW/(m. K) |  |  |
| Cold fluid Thermal Conductivity, kc | kW/(m. K) |  |  |
| Hot fluid specific heat Capacity, Cph | kJ/(kg. K) |  |  |
| Cold fluid specific heat Capacity, Cpc | kJ/(kg. K) |  |  |

\* MpaS – Mega Pascal Second

Density and viscosity can be determined by analysis of the samples taken from the flow stream at the recorded temperature in the plant laboratory. Thermal conductivity and specific heat capacity if not determined from the samples can be collected from handbooks.

**Step – C**

**Thermal parameters of heat exchanger are calculated and compared with the design data**

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameters** | **Units** | **Test Data** | **Design Data** |
| Heat Duty, Q | kW |  |  |
| Hot fluid side pressure drop, ∆Ph | bar | \* |  |
| Cold fluid side pressure drop, ∆Pc | bar | \* |  |
| Temperature Range hot fluid , ∆T | °C |  |  |
| Temperature Range cold fluid , ∆t | °C |  |  |
| Capacity ratio, R | ----- |  |  |
| Effectiveness, S | ----- |  |  |
| Corrected LMTD, MTD | °C |  |  |
| Heat Transfer Coefficient, U | kW/(m2. °C) |  |  |

\* - The pressure drop for the design flow can be rated with the relation Pressure drop is proportional to (Flow)1.75

**Step – D**

**The following formulae are used for calculating the thermal parameters:**

1. Heat Duty, *Q = qs + ql* ----------(8.8)

Where qs is the sensible heat and ql is the latent heat

**For Senisble heat**

*qs = Wx Cph x(Ti- To)/1000/3600* *in kW (or)* ----------(8.9)

*qs = w x Cpc x (To-Ti)/1000/3600 in kW* ----------(8.10)

**For Latent heat**

*ql= W x λh ,* ----------(8.11)

*λh* – Latent heat of Condensation of a hot condensing vapour

(or)

*ql = w x λc* , where *λc* - Latent heat of Vaporization ----------(8.12)

2. Hot Fluid Pressure Drop, *∆Ph = Phi – Pho* -----------(8.13)

3. Cold fluid pressure drop, *∆Pc = Pco – Pci* -----------(8.14)

4. Temperature range hot fluid, *∆Th = Thi- Tho* -----------(8.15)

5. Temperature range cold fluid, *∆Tc = Tco- Tci* -----------(8.16)

6. Capacity ratio, c using equation 8.7

7. Effectiveness, ɛ using table 8.5

8. LMTD using equation 8.5 for parallel flow heat exchangers

9. Corrected LMTD using figure 8.1

10. Overall Heat Transfer Co-efficient using equation 8.1

**8.2 Case study[[9]](#footnote-9)**

A finned tube exchanger of following configuration is considered being used for heating air with steam in the tube side. The monitored parameters are as below:

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameters** | **Units** | **Inlet** | **Outlet** |
| Hot fluid flow, W | kg/h | 3000 | 3000 |
| Cold fluid flow, w | kg/h | 92300 | 92300 |
| Hot fluid Temp, T | OC | 150 | 150 |
| Cold fluid Temp, t | OC | 30 | 95 |
| Hot fluid Pressure, P | Bar g |  |  |
| Cold fluid Pressure, p | Bar g | 200 mbar | 180 mbar |

**Calculation of Thermal data:**

Bare tube Area = 42.8 m2; Fined tube area = 856 m2

1.Duty:

Hot fluid, Q = 1748 kW Cold Fluid, Q = 1726 kW

2. Hot Fluid Pressure Drop

Pressure Drop = Negligible

3. Cold Fluid Pressure Drop

Pressure Drop = 20 mbar.

4. Temperature range hot fluid

Temperature Range ∆Th = Not required.

5. Temperature Range Cold Fluid

Temperature Range ∆Tc = 65 oC.

6. Capacity Ratio

Capacity ratio, c = Not significant in evaluation here.

7. Effectiveness

Effectiveness, ɛ = (to – ti) / (Ti – ti) = Not significant in evaluation here.

8. LMTD

Calculated considering condensing part only

a) LMTD, Counter Flow =((150 – 30)-(150-95)/ ln ((150-30)/(150-95)) = 83.3 oC.

b) Correction Factor to account for cross flow F = 0.95

9. Corrected LMTD

MTD = F x LMTD = 0.95 x 83.3 = 79 oC.

10. Overall Heat Transfer Co-efficient (HTC)

U = Q/ A ∆T = 1748/ (856 x 79) = 0.026 kW/m2. K

**Comparison of Calculated data with Design Data**

|  |  |  |  |
| --- | --- | --- | --- |
| **Parameters** | **Units** | **Test Data** | **Design Data** |
| Duty, Q | kW | 1748 | 1800 |
| Hot fluid side pressure drop, ∆Ph | Bar | Neg | Neg |
| Cold fluid side pressure drop, ∆Pc | Bar | 20 | 15 |
| Temperature Range hot fluid, ∆T | °C |  |  |
| Temperature Range cold fluid, ∆t | °C | 65 | 65 |
| Capacity ratio, R | ----- |  |  |
| Effectiveness, S | ----- |  |  |
| Corrected LMTD, MTD | °C | 79 | 79 |
| Heat Transfer Coefficient, U | kW/(m2. K) | 0.026 | 0.03 |

**Inferences:**

Heat Duty: The difference inferred from the duty as the exchanger is under performing than required

Pressure drop: The airside pressure drop has increased in spite of condensation at the steam side. Indication of choking and dirt blocking at the airside.

Temperature range: No deviations

Heat Transfer coefficient: Decreased because of decreased fin efficiency due to choking on air side.

Trouble shooting: Operations may be checked to perform pulsejet cleaning with steam / blow air jet on air side if the facility is available. Mechanical cleaning may have to be planned during any down time in the immediate future.

**Self-Assessment Exercise**

**Q1.** Explain the various terms associated with the performance evaluation of a heat exchanger.

**Q2.** Draw the temperature profiles for a cross flow and counter flow heat exchanger and derive their LMTD.

**Q3.** Explain what is meant by NTU.

**Q4.** Explain the various factors that influence the performance of a heat exchanger.

1. . Shah R. K. and Sekulic D. P. (2003) *Fundamentals of Heat Exchanger Design*, John Wiley & Sons, New Jersey [↑](#footnote-ref-1)
2. . Incropera F. P. and DeWitt D. P. (2005) *Introduction to Heat Transfer*, John Wiley and Sons, New York [↑](#footnote-ref-2)
3. . Cengel, Y. A., *Heat Transfer: A Practical Approach*, Tata McGraw Hill [↑](#footnote-ref-3)
4. . Cengel, Y. A., *Heat Transfer: A Practical Approach*, Tata McGraw Hill [↑](#footnote-ref-4)
5. . Cengel, Y. A., *Heat Transfer: A Practical Approach*, Tata McGraw Hill [↑](#footnote-ref-5)
6. . Cengel, Y. A., *Heat Transfer: A Practical Approach*, Tata McGraw Hill [↑](#footnote-ref-6)
7. . Cengel, Y. A., Heat Transfer: A Practical Approach, Tata McGraw Hill [↑](#footnote-ref-7)
8. . This section has been adapted from *Bureau of Energy Efficiency Guide Books*, Chapter: Energy Performance assessment of Heat Exchangers [↑](#footnote-ref-8)
9. . This section has been adapted from *Bureau of Energy Efficiency Guide Books*, Chapter: Energy Performance assessment of Heat Exchangers [↑](#footnote-ref-9)