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Design and Analysis of Heat Exchanger

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BONAFIDE CERTIFICATE

This is to certify that the project report on **"Design and Analysis of Heat Exchanger"** is the bona fide work of **Akash Behl and Yash Rane,** students from Indian Institute of Technology Roorkee, who carried out the project work under my supervision during the summer internship 2012.

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PREFACE

Heat Exchangers play a very crucial role when it comes to transference of heat energy between fluids. Its versatility and diversified applications makes it popular in process industries, conventional and nuclear power stations, steam generators, alternative energy applications including ocean, thermal and geothermal. The classification of heat exchangers is mainly done on the type of fluids that are involved in transference of energy.

This project report is a detailed compendium of the factors that has to be considered during designing of a shell and tube heat exchanger. Starting with the basics of heat exchanger, this report covers – Selection of Tube; its material; its surface area; its length, fin configuration, baffle configuration and their orientation, fan laws and discharge characteristic etc.

All the above factors are covered separately in different section. Each section details a systematic approach that must be followed depending on the boundary conditions.

ACKNOWLEDGEMENT

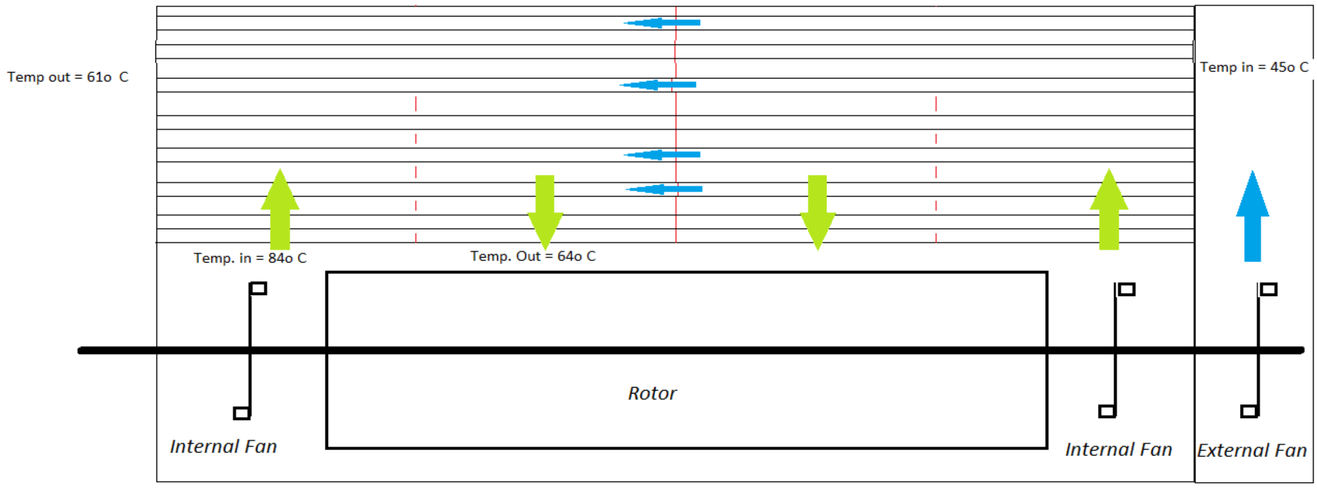
With sheer delight and immense satisfaction, I would like to take this opportunity to convey my gratitude towards all the people who were responsible in making this project a memorable experience.

My foremost thanks to **Mr. Parag Bhandarkar**, Senior Manager – R&D Division, for his support and guidance without whom this project could never have taken off.

We would also like to thank **Mr. Prashant Sharma**, Executive Design Engineer, for his innovative thinking and unconventional strategies that helped me during the entire project.

PROJECT OUTLINE

The ultimate aim of this report is to design an Air Cooled Heat Exchanger, preferably of shell and tube type, that can be used to transfer the heat energy generated in the windings of 3-phase Induction motor working at 800 rpm (Assumed). Following figure details the boundary conditions for the heat exchanger.



Specifications:

- Air Flow Rate from External Fan (Cold Fluid) = 30 m^3/s
- Inlet temperature of cold fluid (external fan) = 45° C
- External temperature of cold fluid (external fan) = 61° C
- Inlet Temperature of Hot Fluid (Internal Fan) = 84° C
- Outlet temperature of Hot Fluid (Internal Fan) = 64° C
- Heat Exchange Capacity = 500 KW
- Dimensions of Heat Exchanger = 2400 x 2250 x 3900 (in mm)

Selection of factors

- Material of Tubes and its cost consideration
- Pattern of Tubes
- Air Flow rate from Internal Fan
- Fin Selection on surface
- Flow of Fluid (cross, parallel and counter)
- Factor of Fouling and Corrosiveness

Assumptions:

- Specific Heat Capacity air is assumed to be constant for temperature difference of 20° C
- Velocity of air does not change in the different region as mentioned in diagram.

Operational Flowchart

Step 1
- Selection of standard HE tube dimension

Step 2
- Calculation of Reynolds no. for 6 regions .

Step 3
- Calculation of Nusslet No. in each region

Step 4
- Calculate Convection Heat Transfer Coefficient for each region

Step 5
- Calculate Total Resistance, heat capacity rate and hence total no. of transfer units (NTU)

Step 6
- Evaluate Effectiveness, Q(maximum) and hence Q(actual)

Step 7
- Depict outlet temperatures of hot and cold fluid

Step 8
- Selection of standard Fin specifications

Step 9
- Calculate Fin surface area and ξ

Step 10
- Using Proper graph for fin profile calcuate efficieny

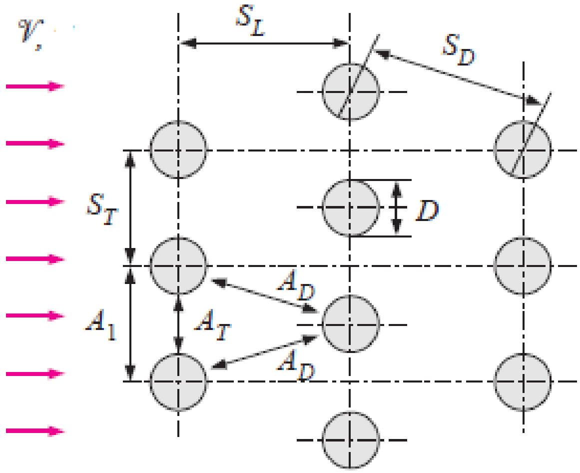
Step 11
- Calculate total area of fin and hence calculate total heat dissipated

Chapter 1
Designing and Analysis of Project

Step 1: Slection of Tube Dimensions, Material, Configuration

In heat exchanger we use standardized tube dimension that is clearly depicted in Appendix 1. This table is a description of tubes based on their BWG parameter and gives Inner Diameter, External surface area, internal surface area, flow area etc.

For the first iteration we have selected the outer diameter as 1.5 inch and thickness as .065 inch (BWG 16). Using this data, we calculate inner diameter, internal and external surface area per unit length, inner and outer cross section area. As we get this data it is easy to calculate transverse and longitudinal pitch. Optimally transverse pitch (S_t) to outer diameter (OD) ratio is 1.25 – 1.5. **We assumed it to be 1.5 for first iteration.** As mentioned in the Chapter 2 of this report, for maximum efficiency, staggered configuration is preferred (60^o). So clearly, $S_t = S_d$ and S_l= 1.732*S_t/2



We have considered 2.4x2.05x3.7 (in m) as the total region for piping because the complete volume cannot be used for piping. So total no pipes can be calculated be determining no. of columns (N_c) and no. of tubes in each column(N_{tc}). Hence total no. of pipes are N_c X $N_{tc.}$

For deciding the baffle cut, it can range from 20% - 49%. **We assumed it to be 25% for first iteration**. Once baffle cut is determined, we can calculate no. of tubes in cross flow and parallel/counter flow region by assuming the piping density to be constant.
So total no. of pipes in Cross Flow = 0.75* Total no. of pipes (N_t)
And total no. of pipes in Parallel/ Counter Flow = 0.25 * Total No. of Pipes (N_t)

Step 2: Calculation of Reynolds no. for 6 regions

To calculate Reynolds no. inside and outside the tubes we need to calculate mass flow V maximum on both the sides. *For inside the tube*:

1. Total volume flow rate is given. We divide it by total no. of tubes to determine the volume flow rate in each tube.

2. Using Q = A x V. We know Q in each pipe and Flow area of air inside the pipe and hence we calculate V max.
3. For inside the tube, Critical length is inner diameter and kinematic viscosity is know. So by using the following relation, we can calculate Reynolds No.
   Reynolds No. = Critical Length * V (max.) / Kinematic Viscosity

For calculation of Reynolds No. outside the tube:

1. We know the volume flow rate of the internal fan. So again using the Q = A x V. We can calculate the V max. Here the area is assumed to be consist of 1/6rd length of total length of Heat Exchanger i.e. 0.4m. So total area is 0.4m * 2.25m.
2. Once we have the V max. We can again use the aforementioned relation of Reynolds no., but the critical length will be outside diameter of the tube.

Reynolds No. will be same for inside the tube flow but it will change in case of the shell flow because the flow area in each region Is different that will change the Vmax. and hence the Reynolds No. and Nusselt No.

Step 3: Calculation of Nusselt No.

Nusselt no. calculations are totally different for the air flow inside and outside the HE tubes. It also depends on the laminar or turbulent flow of the air. Let's discuss each of the air flow separately:

Inside the tube:

1. Laminar Flow: For the fully developed laminar flow inside the tube, the Nusselt no. is constant. For isothermal surface, it is 3.66 and for constant heat flux condition, it is 4.36.
2. Turbulent Flow: The flow in pipes is fully turbulent for Re 10,000. Turbulent flow is commonly utilized in practice because of the higher heat transfer coefficients associated with it. To calculate the Nusselt no. is case of turbulent flow, we use Gnielinski (1976) equation. F is Darcy's friction factor given as,

$$\mathrm{Nu} = \frac{(f/8)\,\mathrm{Re}\,\mathrm{Pr}}{1.07 + 12.7(f/8)^{0.5}\,(\mathrm{Pr}^{2/3} - 1)} \qquad \begin{pmatrix} 0.5 \le \mathrm{Pr} \le 2000 \\ 10^4 < \mathrm{Re} < 5 \times 10^6 \end{pmatrix}$$

$$f = (0.790 \ln \mathrm{Re} - 1.64)^{-2}$$

Now, while calculating Nusselt no. outside the tube i.e. in the shell, we have to consider three different flow viz. Cross Flow, Parallel Flow and Counter Flow. Let's study their relation one-by-one:

Outside the tube:

- Cross Flow: When the hot air will flow from the tube bank, Nusselt no. mainly depends on the transverse and longitudinal pitch, Reynolds. No. and Prandtl no. Following table is the Nusselt no. expression for cross flow over tube banks for N>16 and 0.7<Pr<500

Staggered	0–500	$Nu_D = 1.04\, Re_D^{0.4} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	500–1000	$Nu_D = 0.71\, Re_D^{0.5} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$1000–2 \times 10^5$	$Nu_D = 0.35 (S_T/S_L)^{0.2}\, Re_D^{0.6} Pr^{0.36} (Pr/Pr_s)^{0.25}$
	$2 \times 10^5–2 \times 10^6$	$Nu_D = 0.031 (S_T/S_L)^{0.2}\, Re_D^{0.8} Pr^{0.36} (Pr/Pr_s)^{0.25}$

*All properties except Pr_s are to be evaluated at the arithmetic mean of the inlet and outlet temperatures of the fluid (Pr_s is to be evaluated at T_s).

- Parallel Flow / Counter Flow: For the parallel flow, we have to consider the fact, whether the flow is laminar or turbulent. Nusselt no. is case of parallel as well as counter flow is given by:

Nu = 0.2 * (Re) ^0.6 * (Pr)^0.33

Step 4: Calculation of Convection Heat transfer coefficient (h)

After calculating Nusselt no., h can be very easily determined using **Nu= h*l /k** Where l = critical length and k = Conduction heat transfer coefficient of fluid. For flow inside and outside the tube, value of critical length is inner and outside diameter of the tube respectively

Step 5: Calculate Total Resistance, heat capacity rate and hence total no. of transfer units (NTU)

Total resistance in case of heat exchanger is calculated by:

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$

Value of UAs obtained by the expression is then multiplies with total no. of tubes in each region, so as to consider the total heat exchange in each region.

Heat capacity rate Is determined to calculate the capacity ratio. Using the data we calculate no of transfer units (NTU)., UAs/Cmin.

Step 6: Evaluate Effectiveness, Q(maximum) and hence Q(actual)

Next step is to calculate the effectiveness for the cross, parallel and counter flow, which is shown in the figure below.

Now, **Qmax= Cmin($T_h$in - $T_c$in)**

And we know the effectiveness from the above expression, so Q = ε * Qmax.

Effectiveness relations for heat exchangers: $NTU = UA_s/C_{min}$ and $c = C_{min}/C_{max} = (\dot{m}C_p)_{min}/(\dot{m}C_p)_{max}$ (Kays and London, Ref. 5.)

Heat exchanger type	Effectiveness relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counter-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 - c)]}{1 - c\exp[-NTU(1 - c)]}$
2 *Shell and tube:*	
One-shell pass 2, 4, . . . tube passes	$\varepsilon = 2\left\{1 + c + \sqrt{1 + c^2}\,\dfrac{1 + \exp[-NTU\sqrt{1 + c^2}]}{1 - \exp[-NTU\sqrt{1 + c^2}]}\right\}^{-1}$
3 *Cross-flow (single-pass)*	
Both fluids unmixed	$\varepsilon = 1 - \exp\left\{\dfrac{NTU^{0.22}}{c}[\exp(-c\ NTU^{0.78}) - 1]\right\}$
C_{max} mixed, C_{min} unmixed	$\varepsilon = \dfrac{1}{c}(1 - \exp\{1 - c[1 - \exp(-NTU)]\})$
C_{min} mixed, C_{max} unmixed	$\varepsilon = 1 - \exp\left\{-\dfrac{1}{c}[1 - \exp(-c\ NTU)]\right\}$
4 *All heat exchangers with $c = 0$*	$\varepsilon = 1 - \exp(-NTU)$

Step 7: Depict outlet temperature of hot and cold fluid

Once we have the heat exchanged in the region, the outlet temperatures can be calculated by basic heat expression i.e.

$Q = C\ hot\ (T_h in - T_h out) = C\ cold\ (T_c out - T_c in)$

The outlet temperatures so obtained are used as inlet temperature for successive regions.

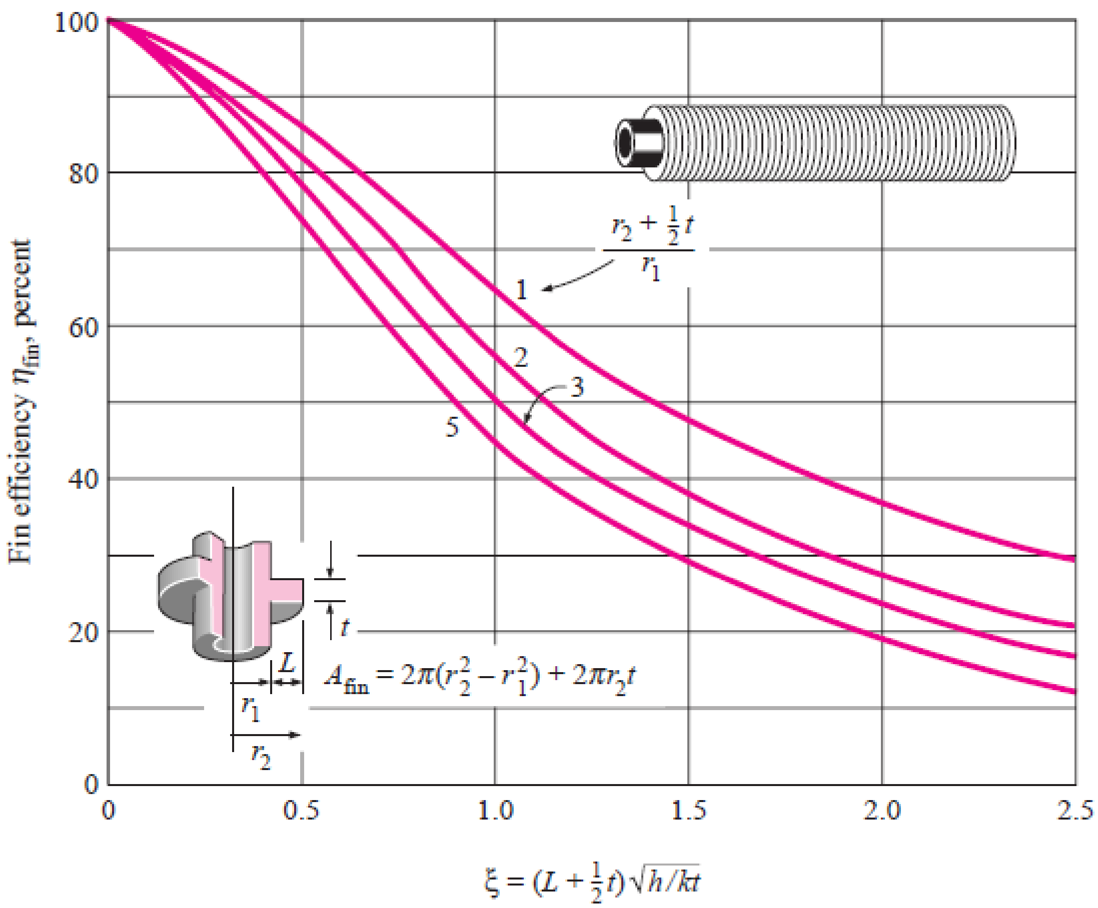
Step 8: Selection of standard Fin specifications

Detailed theory of fin specification, effectiveness, efficiency and heat transfer is detailed in Chapter 2. For the first iteration, the specifications considered are depicted in the figure given below:

Fin Height (in inch)	0.035
Fin thickness (in inch)	0.01
Fin thickness (in metre)	0.000254
Fin density (inches)	3
Fin Material's thermal conductivity (Aluminium)	270

Step 9 and 10: Calculate Fin surface area and ξ

Fin surface area is different for each type of profile. It's the total area of fin through which heat exchange is taking place. We have considered annular fins whose A_{fin} is given the diagram below. We also need to calculate ξ. Once we get the value of ξ, we use this chart to determine the corresponding value of efficiency of the fins.



Step 11: Calculate total area of fin and hence calculate total heat dissipated

Now to calculate the total heat dissipation, we need to determine the total surface area of the fin and pipe combined, which is given by the expression

$$\dot{Q} = [\eta_{fin} A_{fin} + (A - A_{fin})]h\theta_0$$

Where, Q = total heat generated from one pipe
H = efficiency of fine calculated in Step 9 & 10.
A_{fin} = Area of one fin * fin density * length of tube
A = total surface area of tube
h= convection heat transfer coefficient in different region.
Θ = Temperature difference of mean cold and hot fluid
To calculate the total heat generated we calculate Q in each region, multiply it with total no. of pipes in that region and then add all the heat generated in region.

RESULTS

Following is the complete data sheet of the specifications and results obtained:

	Volume	Effective Volume for Piping
Length (in m)	2.4	2.4
Breath (in m)	2.25	2.05
Height (in m)	3.9	3.7

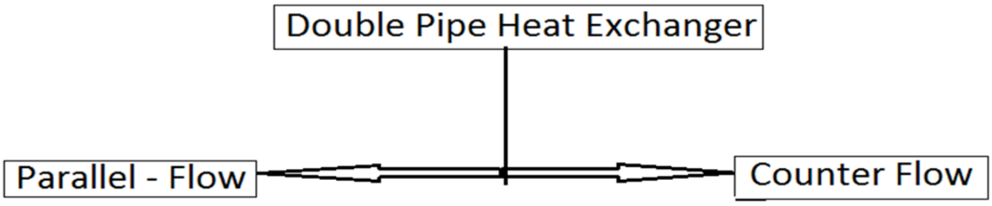
Material Properties of Pipe	
Total Length of Pipe in One Circuit of HE	1.2
Baffle position from External Fan	0.333
Length of Pipe in region 1 and 4 (in m)	0.3996
Length of Pipe in region 3 and 6 (in m)	0.8004
Length of Pipe in region 2 and 5 (in m)	1.2
Outer Diameter(in inch)	1.5
Outer Diameter(in m)	0.0381
Thickness (in inch)	0.065
Thickness (in m)	*0.001651*
Inner Diameter (in m)	0.034798
Inner Cross Section Flow Area of Pipe	0.000950557
External Surface Area per unit meter(in m^2)	0.119634
External Surface Area(in m^2) (region 1 and 4)	0.047805746
Internal Surface Area per unit meter (in m^2)	0.10926572
Internal Surface Area (in m^2) (region 1 and 4)	0.043662582
Outer Cross Section Area of Pipe	0.001139514
Volume flow rate from External Fan(in m^3/sec)	30
Volume flow rate in each pipe (in m^3/sec)	0.011432927
Mass flow rate in each pipe (Kg/sec)	0.012884909
Maximum Velocity inside the pipe(m/sec)	12.02760618
Diagonal Pitch	0.05715
Longitudinal Pitch (in m)	0.0494919
Transverse Pitch (in m)	0.05715
No. of Longitudinal Pipes in each column	64
No. of columns	41
Total No of Pipes	2624
Baffle Cut	0.25
Pipes in Cross Flow	1968
Pipes for Parallel/Couter Flow	656
Fouling Factor of Air, Rf (m2 · °C/W)	0.0004
Thermal Conductivity of Mild Steel	40

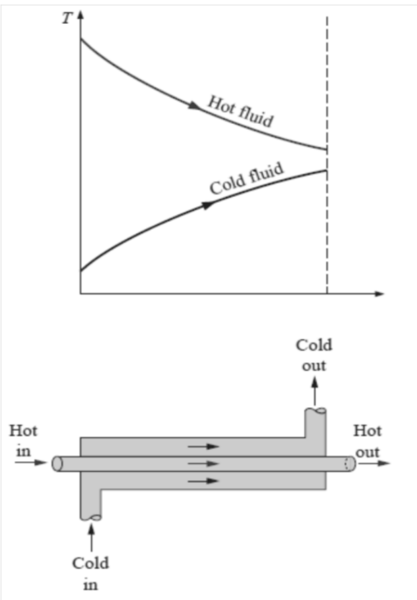
Fin Specifications	
Fin Height (in inch)	0.035
Fin Height (in metre)	0.000889
Fin thickness (in inch)	0.01
Fin thickness (in metre)	0.000254
Fin density (inches)	3
Fin Material's thermal conductivity (Alluminium)	270
Fin surface area	0.000249478
Fin diameter	0.039878
External surface area in region 1 and 4	0.047805746
External surface area in region 2 and 5	0.1435608
External surface area in region 3 and 6	0.095755054
Fin density (metres)	118.1102362
Number of fins in region 1 and 4	47.19685039
Number of fins in region 2 and 5	141.7322835
Number of fins in region 3 and 6	94.53543307
Area no fin (for 1 fin)	3.0387E-05
Temp Difference in region 1	28.06182162
Temp Difference in region 2	15.51166808
Temp Difference in region 3	7.229359105
Temp Difference in region 4	15.63122529
Temp Difference in region 5	20.56968641
Temp Difference in region 6	10.71557757

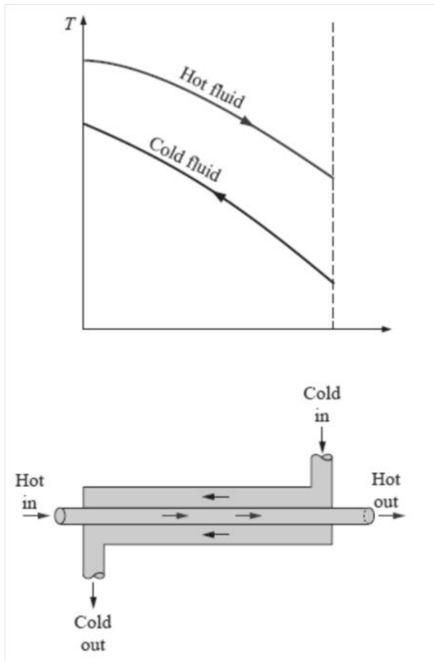
Calculation Of Effectiveness	
Region 1: Cross Flow	
effectiveness	0.029852612
efficiency	0.9
Qtotal	186273.8077
Region 2 : Parallel Flow	
effectiveness	0.017554463
efficiency	0.9
Qtotal	35640.13719
Region 3: Cross Flow	
effectiveness	0.024484571
efficiency	0.9
Qtotal	64660.29375
Region 4: Cross Flow	
effectiveness	0.030264766
efficiency	0.9
Qtotal	106644.6186
Region 5: Counter Flow	
effectiveness	0.017403842
efficiency	0.9
Qtotal	46454.0629
Region 6: Cross Flow	
effectiveness	0.025365108
efficiency	0.9
Qtotal	102858.9004
Q Net (in KW)	542.5318205

Chapter 2
Study of Heat Exchangers

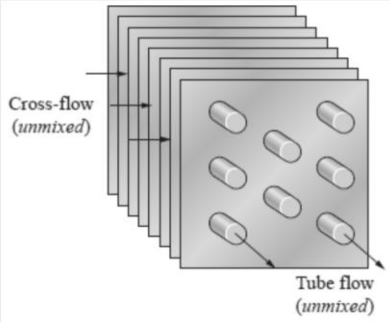
Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment. Simplest type of heat exchanger consists of two concentric pipes of different diameters called the **double-pipe** heat exchanger. One fluid in a double-pipe heat exchanger flows through the smaller pipe while the other fluid flows through the annular space between the two pipes. This are further divided as



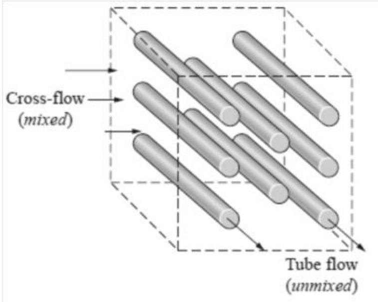




There is also a third category known as compact heat exchangers, in which the two fluids usually move *perpendicular* to each other, and such flow configuration is called **cross-flow.** The cross-flow is further classified as *unmixed* and *mixed flow,* depending on the flow configuration, as shown in Figure below. In (*a*) the cross-flow is said to be *unmixed* since the plate fins force the fluid to flow through a particular inter-fin spacing and prevent it from moving in the transverse direction (i.e., parallel to the tubes). The cross-flow in (*b*) is said to be *mixed* since the fluid now is free to move in the transverse direction. Both fluids are unmixed in a car radiator. The presence of mixing in the fluid can have a significant effect on the heat transfer characteristics of the heat exchanger.

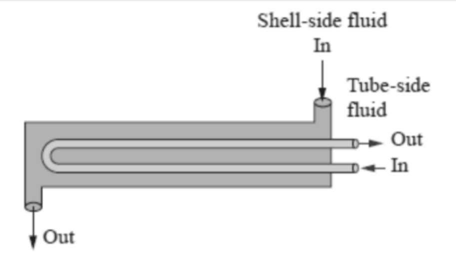


(a) Both fluids unmixed

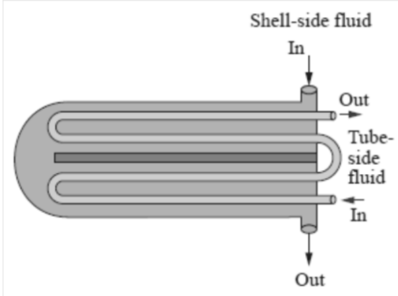


(b) One fluid mixed, one fluid unmixed

Shell-and-tube heat exchangers are further classified according to the number of shell and tube passes involved. Heat exchangers in which all the tubes make one U-turn in the shell, for example, are called *one-shell-pass and two tube- passes* heat exchangers. Likewise, a heat exchanger that involves two passes in the shell and four passes in the tubes is called a *two-shell-passes and four-tube-passes* heat exchanger.



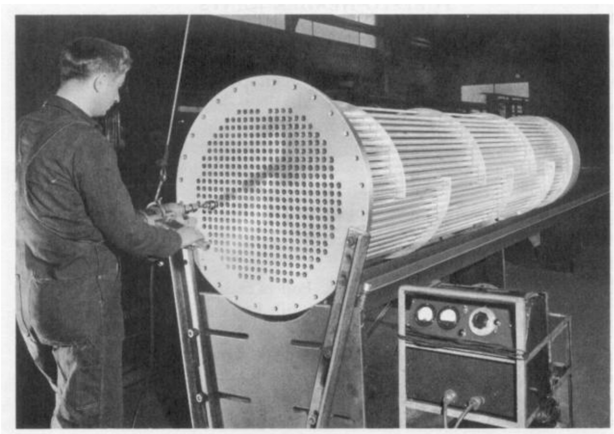
(a) One-shell pass and two-tube passes

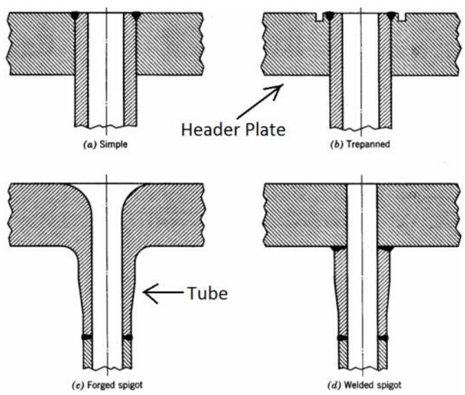


(b) Two-shell passes and four-tube passes

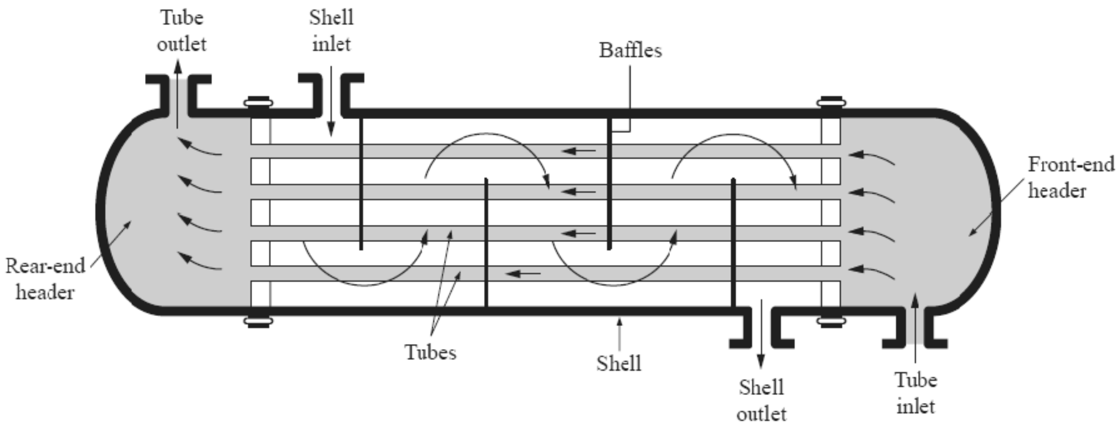
HEADER PLATE SELECTION

Tubes are arranged in a **bundle** and held in place by **header plate** (tube sheet). The number of tubes that can be placed within a shell depends on Tube layout, tube outside diameter, pitch, number of passes and the shell diameter. When the tubes are too close to each other, the header plate becomes to weak. The methods of attaching tubes to the header plate are shown in figure below.





Basics of Heat Exchanger



THE OVERALL HEAT TRANSFER COEFFICIENT

A heat exchanger typically involves two flowing fluids separated by a solid wall. Heat is first transferred from the hot fluid to the wall by *convection,* through the wall by *conduction,* and from the wall to the cold fluid again by *convection.* Any radiation effects are usually included in the convection heat transfer coefficients. The thermal resistance network associated with this heat transfer process involves two convection and one conduction resistances. T*otal thermal resistance*

$$R = R_{\text{total}} = R_i + R_{\text{wall}} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o}$$

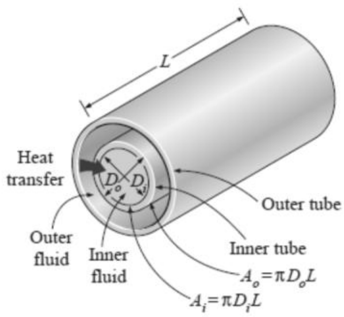
becomes

On considering factor of fouling,

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$

The total rate of heat transfer between the two fluids can be expressed as

$$\dot{Q} = \frac{\Delta T}{R} = UA\,\Delta T = U_i A_i\,\Delta T = U_o A_o\,\Delta T$$

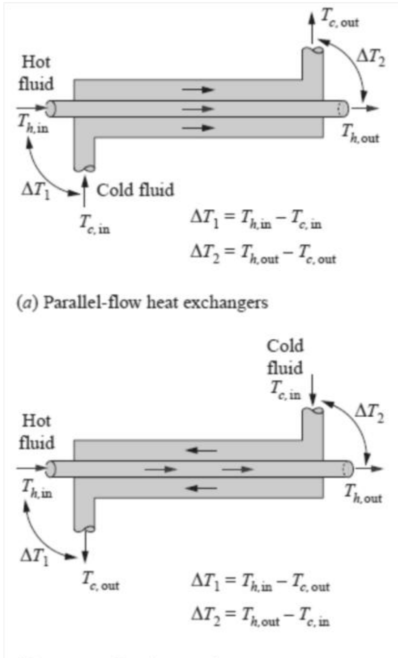


ANALYSIS OF HEAT EXCHANGERS

Heat exchangers are commonly used in practice, and an engineer often finds himself in a position to select a heat exchanger that will achieve a specified temperature change in a fluid stream of known mass flow rate, or to predict the outlet temperatures of the hot and cold fluid streams in a specified Heat exchanger. The two methods used in the analysis of heat exchangers are the log mean temperature difference (or LMTD) method which is best suited for the first task and the effectiveness–NTU method for the second task. But first we present some general considerations. As such, the mass flow rate of each fluid remains constant, and the fluid properties such as temperature and velocity at any inlet or outlet remain the same. Also, the fluid streams experience little or no change in their velocities and elevations, and thus the *kinetic* and *potential energy changes* are negligible. The *specific heat* of a fluid, in general, changes with temperature but for a small temperature change of 20^{o} – *50^{0}C, we assume it to be constant.*

Log Mean Temperature Difference Method

The log mean temperature difference (LMTD) method is easy to use in heat exchanger analysis when the inlet and the outlet temperatures of the hot and cold fluids are known or can be determined from an energy balance. This method is mainly used to determine the Net Surface Area for the heat transfer, for the given heat rating. Therefore, the LMTD method is very suitable for determining the *size* of a heat exchanger to realize prescribed outlet temperatures when the mass Flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified.



(*a*) Parallel-flow heat exchangers

(*b*) Counter-flow heat exchangers

$$\dot{Q} = UA_s\,\Delta T_{\mathrm{lm}}$$

Where,

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

is the **log mean temperature difference,**

ΔT_1 and ΔT_2 is shown in figure

A_s = Outside tube surface area

Q = Heat duty – heat exchange between tube and shell side

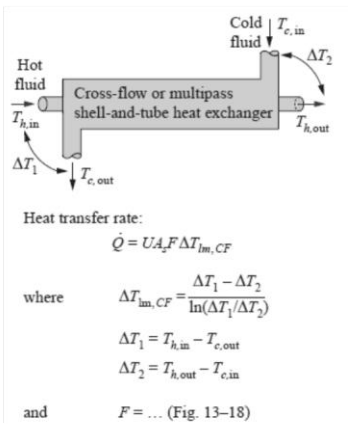
$$\Delta T_{\mathrm{lm}} = F\,\Delta T_{\mathrm{lm},\,CF}$$

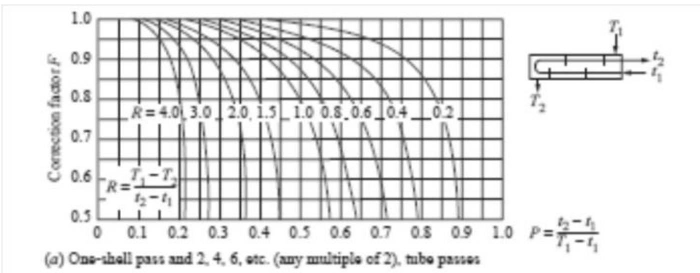
where F is the **correction factor,** which depends on the geometry of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams.

The correction factor is less than unity for a cross-flow and multi-pass shell and-tube heat exchanger. That is, F < 1. The limiting value of F=1 corresponds, to the counter-flow heat exchanger. Thus, the correction factor F for a heat exchanger is a measure of deviation of the from the corresponding values for the counter-flow case.

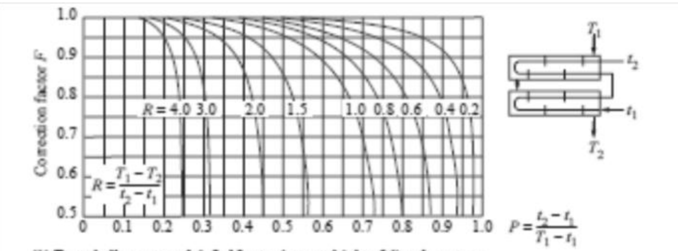
The correction factor F for common cross-flow and shell-and-tube heat exchanger Configurations is given in Figure 13–18 versus two temperature ratios P and R defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}C_p)_{\text{tube side}}}{(\dot{m}C_p)_{\text{shell side}}}$$

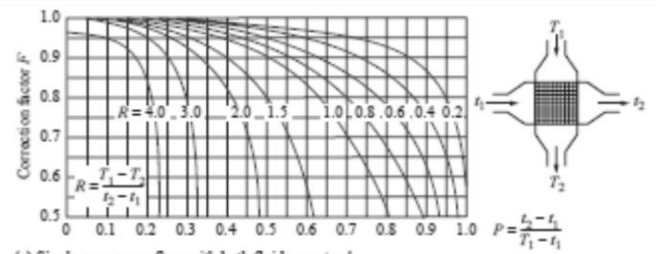




(*a*) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(c) Single-pass cross-flow with both fluids unmixed

With the LMTD method, the task is to *select* a heat exchanger that will meet the prescribed heat transfer requirements. The procedure to be followed by the selection process is:

1. Select the type of heat exchanger suitable for the application.
2. Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.
3. Calculate the log mean temperature difference ΔTlm and the correction factor *F*, if necessary.
4. Obtain (select or calculate) the value of the overall heat transfer coefficient *U.*
5. Calculate the heat transfer surface area *As.*

The Effective NTU Method

A second kind of problem encountered in heat exchanger analysis is the determination of the *heat transfer rate* and the *outlet temperatures* of the hot and cold fluids for prescribed fluid mass flow rates and inlet temperatures when the *type* and *size* of the heat exchanger are specified. The heat transfer surface area *A* of the heat exchanger in this case is known, but the *outlet temperatures* are not. Here the task is to determine the heat transfer performance of a specified heat exchanger or to determine if a heat exchanger available in storage will do the job. Kays and London came up with a method in 1955 called the **effectiveness–NTU method,** which greatly simplified heat exchanger analysis. This method is based on a dimensionless parameter called the **heat transfer effectiveness**, defined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

The actual heat transfer rate in a heat exchanger can be determined from an energy balance on the hot or cold fluids and can be expressed as **Q = Cc(Tc, out _ Tc, in) = Ch(Th, in _ Th, out)**
Where, C_c = mc $_.$Cp$_c$ and C_h= m$_c$ $_.$Cp$_h$ are the heat capacity rates of the cold and the hot fluids, respectively. To determine the maximum possible heat transfer rate in a heat exchanger, we first recognize that the maximum temperature difference in a heat exchanger is the difference between the *inlet* temperatures of the hot and cold fluids. That is,

$$\Delta T_{max} = T_{h,\,in} - T_{c,\,in}$$

The heat transfer in a heat exchanger will reach its maximum value when (1) the cold fluid is heated to the inlet temperature of the hot fluid or (2) the hot fluid is cooled to the inlet temperature of the cold fluid. These two limiting conditions will not be reached simultaneously unless the heat capacity rates of the hot and cold fluids are identical (i.e., Cc = Ch). When Cc = Ch, which is usually the case, the fluid with the smaller heat capacity rate will experience a larger temperature change, and thus it will be the first to experience the maximum temperature, at which point the heat transfer will come to a halt. Therefore, the maximum possible heat transfer rate in a heat exchanger is

$$\dot{Q}_{max} = C_{min}(T_{h,\,in} - T_{c,\,in})$$

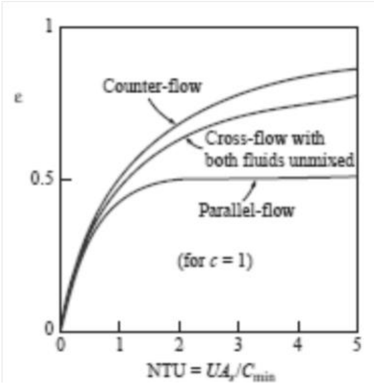
Effectiveness relations of the heat exchangers typically involve the dimensionless group UAs /Cmin. This quantity is called the **number of transfer units NTU** and is expressed as

$$\text{NTU} = \frac{UA_s}{C_{min}} = \frac{UA_s}{(\dot{m}C_p)_{min}}$$

where U is the overall heat transfer coefficient and As is the heat transfer surface area of the heat exchanger. Note that, larger the NTU, the larger the heat exchanger.

We make these following observations from the effectiveness relations:

- The value of the effectiveness ranges from 0 to 1. It increases rapidly with NTU for small values (up to about NTU=1.5) but rather slowly for larger values. Therefore, the use of a heat exchanger with a large NTU (usually larger than 3) and thus a large size cannot be justified economically.
- For a given NTU and capacity ratio *c = Cmin /Cmax*, the *counter-flow* heat exchanger has the *highest* effectiveness, followed closely by the cross-flow heat exchangers with both fluids unmixed. As you might expect, the lowest effectiveness values are encountered in parallel-flow heat exchangers.
- The value of the capacity ratio *c* ranges between 0 and 1. For a given NTU, the effectiveness becomes a *maximum* for *c*=0 and a *minimum* for c=1. The case *c = Cmin /Cmax* $\longrightarrow$ 0 corresponds to *Cmax* $\longrightarrow \infty$ which is realized during a phase-change process in a *condenser* or *boiler.* All effectiveness relations in this case reduce to

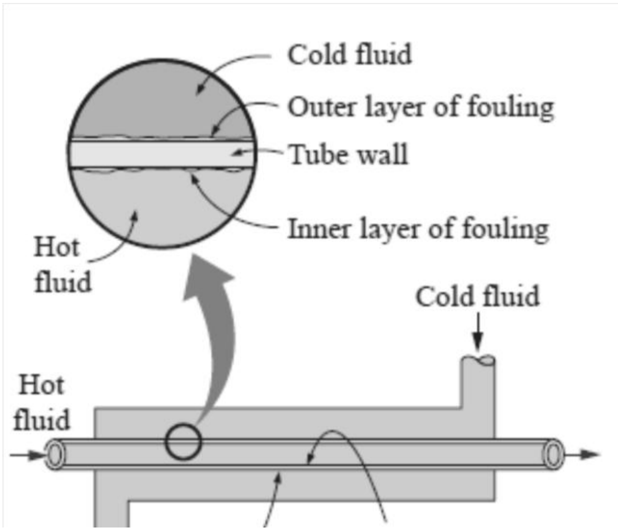


$$\varepsilon = \varepsilon_{max} = 1 - \exp(-NTU)$$

Fouling Factor

The performance of heat exchangers usually deteriorates with time as a result of accumulation of *deposits* on heat transfer surfaces. The layer of deposits represents *additional resistance* to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease. The net effect of these accumulations on heat transfer is represented by a **fouling factor** *Rf* , which is a measure of the *thermal resistance* introduced by fouling. Some of the fouling factors (representing thermal resistance due to fouling for a unit surface area) :

- Alcohol vapors : $R_d = 0.00009\ (m^2K/W)$
- Boiler feed water, treated above 325 K : $R_d = 0.0002\ (m^2K/W)$
- Fuel oil : $R_d = 0.0009\ (m^2K/W)$
- **Industrial air :** $\boldsymbol{R_d = 0.0004\ (m^2K/W)}$
- Quenching oil : $R_d = 0.0007\ (m^2K/W)$
- Refrigerating liquid : $R_d = 0.0002\ (m^2K/W)$
- Seawater below 325 K : $R_d = 0.00009\ (m^2K/W)$
- Seawater above 325 K : $R_d = 0.0002\ (m^2K/W)$
- Steam : $R_d = 0.00009\ (m^2K/W)$



Baffle selection

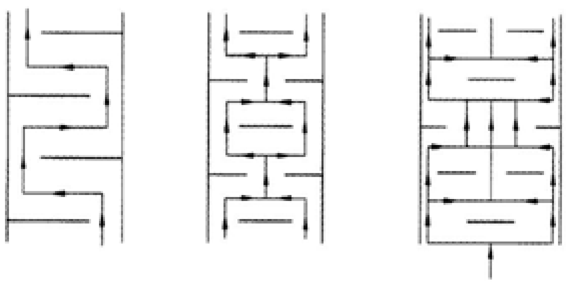
Baffle is the part of shell and tube heat exchanger and is used to support tubes, maintain spacing between tubes as well as to direct fluid at shell side across or along tube bundle in regular pattern. We can install baffles in different ways in heat exchanger in such a way that it gives required flow pattern that we want. Baffle enhance heat transfer rate at the expense of increased pressure drop. Tubular exchanger manufacturing association (TEMA) provides complete guideline of segmental baffles which is available in the annual TEMA journal at a price of $400.

Baffles may have holes through which steel tie-rods are passed to provide structural support to the pipes Baffle spacing, cut and orientation are the key features of the TEMA baffles design. In heat exchanger, baffles are installed normal or parallel to tubes. Similarly baffles are classified into two types which are following

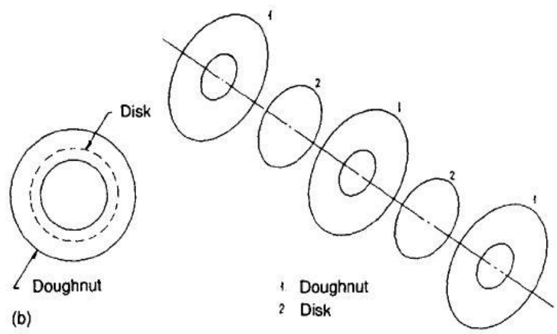
Transverse baffles

This baffle is used to give path to the fluid at shell side into tube bundle perpendicular to tubes. In this way, fluid becomes turbulent and will get more heat transfer. Transverse baffles are actually plate baffles which is divided into three types which are following

1. ***Segmental baffles:*** Segmental baffles are actually a circular disk inside shell having a removed segment. This removed segment is denoted as **BAFFLE CUT** and we express it as shell percentage inside diameter. It is also referred as single segmental baffle. To reduce baffle spacing, we need to use multi-segmental baffles. By using this, we can reduce fluid cross-flow at shell side. This will lead to less pressure drop as compare to segmental baffles. if we use single segmental baffle, fluid will pass through tube bank. But in case of double segmental baffle, fluid will divide into two streams inside shell and in three in case of triple segmental baffle. Because of this property of fluid stream distribution, we can handle more fluid flow as compare to single segmental baffle.
2. ***Disk and doughnut baffle:*** This baffle is made up of disks and doughnuts placed alternatively. This type of baffle has importance in nuclear heat exchangers because it gives us lower pressure drops as compare to single segmental baffle for same unsupported tube span.
3. ***Orifice baffle:*** We rarely use this design because of its properties. In it we have large clearance between tube and baffle. Due to this large clearance, it acts as an orifice. This type of baffle doesn't support tubes and fouling behaviour from fluid can be easily seen. So they cannot be cleaned but easily changed.



(a) Segmental Baffles



(b) Disc and Doughnut Baffle

Longitudinal baffles

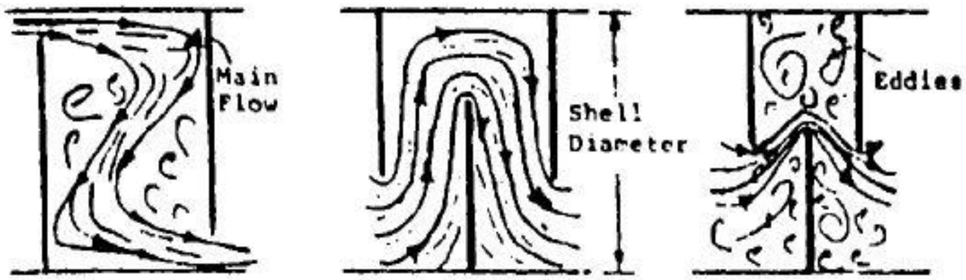
We use longitudinal baffle to control the direction of fluid flow in shell side rather than giving path as we did in transverse baffle. Longitudinal baffles have the ability to divide shell into further sections which will lead to provide multi-pass in shell. To use it, we are facing one restriction that baffle must be welded with shell and tube sheet. If we don't weld baffles than bypass will takes place and it will affect heat transfer coefficient and create problems. So to avoid such malfunctions, we should welded baffle with shell and tube sheet accurately. The types of longitudinal baffles are following

- Rod baffles
- NEST baffles and egg-crate tube support
- Grimmas baffle

Baffle Cut

Due to the presence of baffle cut, heat transfer and pressure drop of crossflow bundles are greatly affected. The percentage of baffle cut is 20 to 49% in which 20 to 25% is commonly used. If we use less than 20%, it will result into higher pressure drop and create stagnant regions or areas having low flow velocities.

Baffle spacing



Practically segmental baffle spacing varies from 1/5 to 1 inch shell diameter. Generally we use baffles between nozzles. The spacing between inlet and outlet baffles is more as compare to the baffle we use in centre. Similarly it is according to the process requirement that we should not use nozzles near tube sheet.

Baffle thickness

Baffle thickness is also one of the most important factors to be discussed. About the thickness of baffles for different types of materials we need to see the tables of TEMA because they have all knowledge regarding to the thickness of baffle for every type of material.

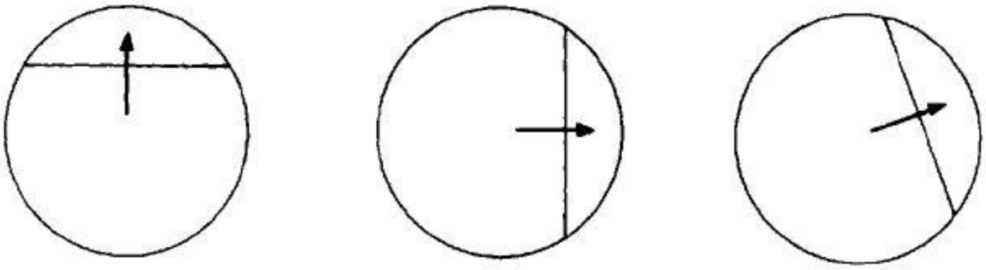
Shell side flow distribution

During designing if we don't use correct baffle cut ratio then segmental baffle will do poor flow distribution. If baffle cut ratio is low or high, it will give us inefficient heat transfer as well as favour fouling tendency. If we are designing shell and tube heat exchanger for low pressure drop, we need to choose the type of baffle which gives us more uniform flow which are rod baffles, multi-segmental and disc and doughnuts baffles.

Orientation of baffle

After arranging a baffle inside shell, we arrange other segmental baffle at 180˚ because it will lead the fluid to approach cross-flow while passing through bundle and axial flow while passing through baffle window zone. Mostly we use segmental baffle which has horizontal baffle cut. A baffle which has horizontal baffle cut should be used for single phase application because it will reduce the chances of fluid to be deposit at the bottom of shell.

Similarly segmental baffle with vertical direction is used for condensation, for boiling fluid, for solid entrained in liquid.



(a) (b) (c)

Tube Selection

Tube Material

Materials selection and compatibility between construction materials and working fluids are important issues, in particular with regard to **corrosion** and/or operation at **elevated temperatures**.
Requirement for low cost, light weight, high conductivity, and good joining characteristics often leads to the selection of Aluminium for the heat transfer surface. But aluminium can cost up to Rs 130 per kg. On the other side, stainless steel is used for food processing or fluids that require corrosion resistance.
Although copper can also be used because of its high thermal conductivity but it will lead to expensive heat exchanger. Moreover, the frame of the heat exchanger and header plate must be of the material with least thermal conductivity and at the same time it should be light weight. Mild steel is greatly preferred because of it low cost of nearly Rs 50 per kg and moderately low thermal conductivity of 45-64 W/m/K.

Material	Thermal Conductivity (W/m.k)
Silver	420
Copper	401
Gold	317
Aluminium	237
Tungsten	174
Brass	150
Zinc	116
Platinium	107

Metal or Alloy	Density (kg/m^3)
Aluminum	2712
Titanium	4500
Vanadium	5494
Zirconium	6570
Antimony	6690
Zinc	7135
Chromium	7190
Tin	7280
Wrought Iron	7750
Iron	7850
Steel	7850
Incoloy	8027
copper	8900

Tube Wall thickness and Diameter

The wall thickness of heat exchanger tubes is standardized in terms of Birmingham Wire Gage BWG of the tube. Small tube diameters (8 to 15mm) are preferred for greater area to volume density but are limited for the purposes of cleaning. Large tube diameters are often required for condensers and boilers. From the heat transfer viewpoint, smaller-diameter tubes yield higher heat transfer coefficients and result in a more compact exchanger. However, larger-diameter tubes are easier to clean and more rugged. The foregoing common sizes represent a compromise. For chemical cleaning, smaller sizes can be used provided that the tubes never plug completely.

The **Stubs Iron Wire Gauge** system (also known as the **Birmingham Wire Gauge**) is used to specify thickness or diameter of metal wire, strip, and tube products. Wall thickness of a pipe is normally given in decimal part of an inch rather than as fraction or gage number. When gauge numbers is given for a pipe without reference to a system, Birmingham Wire Gauge - BWG - is implied.

The table shows the standardized data for commercial tubing obtained from BWG.

The gauge starts at the lowest gauge number of 5Ø or 00000, corresponding to the largest size of 0.500" (12.7mm) to the highest gauge number of 36, corresponding to the smallest size of 0.004" (0.102mm) Birmingham Wire Gauge is also known as Stubs' Wire Gauge, used for drill rod and tool steel wire.

The table on the next page shows some commercial tubing data with thickness, outer/inner diameter, and internal flow area, internal and external surface are per Ft. of length and weight of per Ft. of pipe of mild steel. (Detailed Chart: Appendix 1)

Gauge	Birmingham Wire Gage B.W.G. *(inches)*
00000 (5/0)	0.5
0000 (4/0)	0.454
000 (3/0)	0.425
00 (2/0)	0.38
0	0.34
1	0.3
2	0.284
3	0.259
4	0.238
5	0.22
6	0.203
7	0.18
8	0.165
9	0.148
10	0.134
11	0.12
12	0.109
13	0.095
14	0.083
15	0.072
16	0.065
17	0.058
18	0.049
19	0.042
20	0.035
21	0.032
22	0.028
23	0.025
24	0.022
25	0.02
26	0.018
27	0.016
28	0.014
29	0.013
30	0.012
31	0.01
32	0.009
33	0.008
34	0.007
35	0.005
36	0.004

Tube Length

Thermal expansion of tubes needs to be taken into account for heat exchangers operating at elevated temperatures. But at the temperature difference of 20^{o} C – 50^{o} C, deflection can be neglected. Tube elongation due to thermal expansion causes: Header plate deformation,
Shell wall deformation near the header plate and Fatigue strength of the tube. Tube length also affects the cost and operation of heat exchangers. Longer the tube length (for any given surface area), Fewer tubes are needed, Shell diameter decreases resulting in lower cost. Shell-diameter-to-tube-length ratio should be within limits of 1/5 to 1/15.

TABLE 8.1 (CONTINUED)

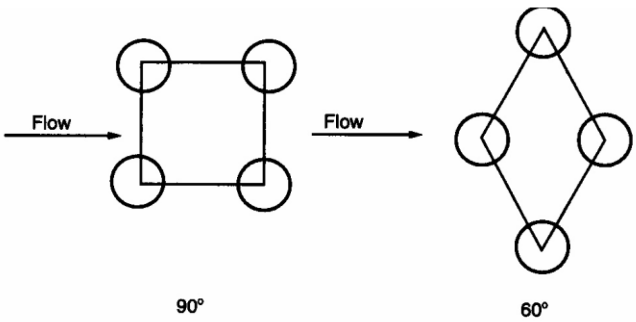
Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1	18	0.049	0.6390	0.2618	0.2361	0.496	0.902	1.109
1	20	0.035	0.6793	0.2618	0.2435	0.360	0.930	1.075
1-1/4	7	0.180	0.6221	0.3272	0.2330	2.057	0.890	1.404
1-1/4	8	0.165	0.6648	0.3272	0.2409	1.921	0.920	1.359
1-1/4	10	0.134	0.7574	0.3272	0.2571	1.598	0.982	1.273
1-1/4	11	0.120	0.8012	0.3272	0.2644	1.448	1.010	1.238
1-1/4	12	0.109	0.8365	0.3272	0.2702	1.329	1.032	1.211
1-1/4	12	0.095	0.8825	0.3272	0.2773	1.173	1.060	1.179
1-1/4	14	0.083	0.9229	0.3272	0.2838	1.033	1.084	1.153
1-1/4	16	0.065	0.9852	0.3272	0.2932	0.823	1.120	1.116
1-1/4	18	0.049	1.042	0.3272	0.3016	0.629	1.152	1.085
1-1/4	20	0.035	1.094	0.3272	0.3089	0.456	1.180	1.059
1-1/2	10	0.134	1.192	0.3927	0.3225	1.955	1.232	1.218
1-1/2	12	0.109	1.291	0.3927	0.3356	1.618	1.282	1.170
1-1/2	14	0.083	1.398	0.3927	0.3492	1.258	1.334	1.124
1-1/2	16	0.065	1.474	0.3927	0.3587	0.996	1.370	1.095
2	11	0.120	2.433	0.5236	0.4608	2.410	1.760	1.136
2	13	0.095	2.573	0.5236	0.4739	1.934	1.810	1.105
2-1/2	9	0.148	3.815	0.6540	0.5770	3.719	2.204	1.134

Source: Courtesy of the Tubular Exchanger Manufacturers Association.

Tube Layout

Tube layout is characterized by the included angle between tubes. Two standard types of tube layouts are the square/ in-line and the staggered/ equilateral triangle. Triangular pitch (30^{o} layout) is better for heat transfer and surface area per unit length (greatest tube density.) Square pitch (45 & 90 layouts) is needed for mechanical cleaning. Note that the 30°,45° and 60° are staggered, and 90° is in line. For the identical tube pitch and flow rates, the tube layouts in decreasing order of shell-side heat transfer coefficient and pressure drop are: 30°,45°,60°, 90°.



The 90° layout will have the lowest heat transfer coefficient and the lowest pressure drop. The square pitch is generally not used in the fixed header sheet design because cleaning is not feasible.

The triangular pitch provides a more compact arrangement, usually resulting in smaller shell, and the strongest header sheet for a specified shell-side flow area. It is preferred when the operating pressure difference between the two fluids is large.

Tube Pitch

The selection of tube pitch is a compromise between a **Close pitch** (small values of Pt/do) for increased shell-side heat transfer and surface compactness, and an o**pen pitch** (large values of Pt/ do) for decreased shell-side plugging and ease in shell-side cleaning. Tube pitch (PT) is chosen so that the pitch ratio is $1.25 < PT/do < 1.5$. When the tubes are too close to each other (Pt/do less than 1.25) , the header plate (tube sheet) becomes too weak for proper rolling of the tubes and cause leaky joints.

Fin Selection

USE of FINS

For a given LMTD the only way of increasing heat transfer rates in a heat-exchanger is to increase the surface area. One way of achieving this is through the use of extended or finned surfaces.
With Liquid /Gas HX's very often the heat transfer coefficient on the liquid side is much greater than that on the gas side. Fins would then be used on the gas side so that the resistance to heat transfer was approximately the same on both sides. Fin comes in many shapes and sizes. They can be broadly classified into :

- Fins of constant cross-section : a)rectangular b) pin (spike) fins
- Fins of varying cross-section.: tapered fins.

Fin geometry and density

Fins improve heat transfer in two ways. One way is by creating turbulent flow through fin geometry, which reduces the thermal resistance (the inverse of the heat transfer coefficient) through the nearly stagnant film that forms when a fluid flows parallel to a solid surface. A second way is by increasing the fin density, which increases the heat transfer area that comes in contact with the fluid.
Fin geometries and densities that create turbulent flow and improve performance also increase pressure drop, which is a critical requirement in most high performance applications. The optimum fin geometry and fin density combination is then a compromise of performance, pressure drop, weight, and size. Aside from fin geometry, parameters such as thickness, height, pitch, and spacing can also be altered to improve performance. Typically, fin thicknesses vary from 0.004 in (0.1 mm) to 0.012 in (0.3 mm), heights vary from 0.035 in (0.89 mm) to 0.6 in (15.24 mm), and densities vary from 8 to 30 FPI (Fins Per Inch).

In most high performance applications, fins are made of copper or aluminium. Aluminium fins are preferred in due to their lighter weight. There are many different fin geometries used in heat transfer applications. Some of the most commonly used are louvered, lanced offset, straight, and wavy fins.

If we want to maximize heat transfer we must minimize thermal resistance. To accomplish this, we must increase the corresponding heat transfer areas, the film coefficients, or both. Increasing the heat transfer area is relatively easy in concept, though sometimes constrained by application requirements such as weight, size, and pressure drop. An effective way to increase the heat transfer area is to increase the fin density (fins per unit length). Increasing the film coefficient is more complicated, however, because the film coefficient is dependent upon the properties of the fluid being considered, the fluid velocity, and the fin geometry.

Fin Performance

It is possible using methods we have already used to analyse the heat transfer characteristics (convection/ conduction) of fins. For long constant cross-section fins (where heat transfer through the tip is ignored (insulated), and assuming a constant heat transfer coefficient along its length, the temperature difference between the fin and the fluid is given by :-

$$\frac{\theta}{\theta_0} = \frac{\cosh m(L-x)}{\cosh mL}$$

Where;
Θ_0 = Temperature difference at the root of the fin (the wall end);
m = $(hp/\lambda A)^{1/2}$
p = perimeter of the fin cross-section;
A = fin cross-sectional area;
L = fin length
Λ = thermal conductivity

The heat transfer rate from the fin may be obtained by
Determining the conduction at the root.

$$\dot{Q}_0 = m\lambda A\theta_0 \tanh mL$$

For a comparatively short constant cross-section fin we cannot ignore the heat transfer at its tip, and when this is included we obtain:-

$$\frac{\theta}{\theta_0} = \frac{\cosh m(L-x) + \frac{h}{\lambda m}\sinh m(L-x)}{\cosh mL + \frac{h}{\lambda m}\sinh mL}$$

$$\dot{Q}_0 = m\lambda A\theta_0 \frac{\tanh mL + \frac{h}{\lambda m}}{1 + \frac{h}{\lambda m}\tanh mL}$$

Note: if $h/\lambda m = 1$, $\dot{Q}_0 = h\,A\,\theta_0$
ie the same as with no fin.

if $h/\lambda m < 1$ a fin is worthwhile

if $h/\lambda m > 1$ it would reduce heat transfer!
(ie act as small insulators)

Fin Efficiency / Effectiveness

If the whole fin were at the wall (or root) temperature, the increase in heat transfer rate would be in direct proportion to the increase in surface area. Owing to the temperature gradient along a fin this is not achievable.
We therefore define fin **efficiency** as:-

$$\eta_{fin} = \frac{\text{Actual fin heat transfer rate}}{\text{Heat transfer rate if the whole fin were at the root temperature}}$$

For long constant cross-section fins this is:

$$\eta_{fin} = \frac{m\,\lambda\,A\,\theta_0 \tanh mL}{h\,p\,L\,\theta_0} = \frac{\tanh mL}{mL}$$

For short constant cross-section fins this is:

$$\eta_{fin} = \frac{\tanh mL + h/\lambda m}{(1 + h/\lambda m \tanh mL)(h/\lambda m + mL)}$$

Finned Surfaces

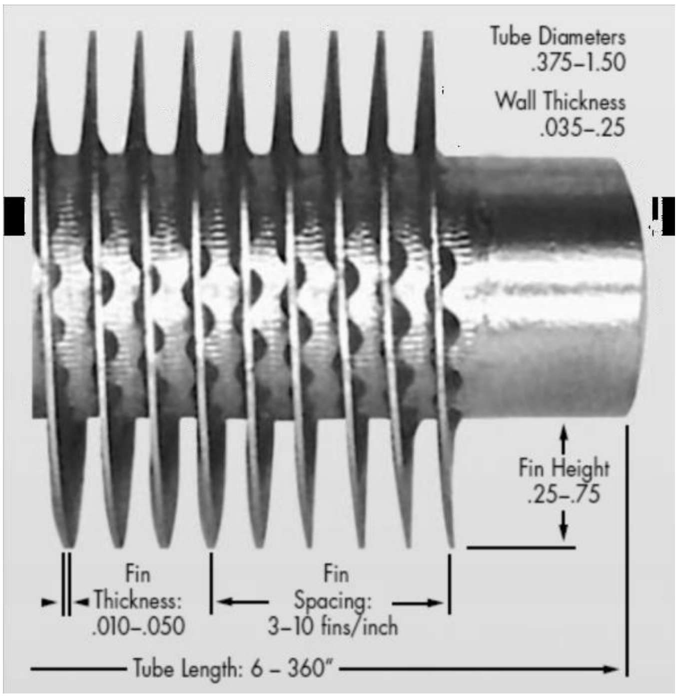
If A = **total** surface area (including fin area), and A_{fin} = fin surface area, let b = A_{fin} /A.
The total heat transfer from a finned surface is then given by:-

$$\begin{aligned}\dot{Q} &= [\eta_{fin} A_{fin} + (A - A_{fin})] h\theta_0 \\ &= \frac{[\eta_{fin} A_{fin} + (A - A_{fin})] A h\theta_0}{A} \\ &= [\eta_{fin}\beta + (1-\beta)] A h\theta_0 \\ &= \eta' h A\theta_0\end{aligned}$$

η' = 1 - b (1 - η_{fin}) is the area weighted fin efficiency η ' can be incorporated into the calculation for the overall heat transfer coefficient (U) for heat exchangers. In this case the fins are on the 'h1' side.

$$\frac{1}{UA} = \frac{1}{\eta' h_1 A_1} + \frac{1}{h_2 A_2}$$

Although analysis of varying cross-section fins is more complex than for constant cross-section fins, it can be done (!), and graphs are available which give fin efficiency as a function of their geometry and dimensions. These can be used to estimate the heat transfer enhancement achieved by using these types of fins.



Fan Discharge

The air flow and the air speed between ribs of frame must meet minimum the figures given below as to shaft height. Figures correspond 50 Hz ne t supply, with 60 Hz 20 % must be added.

Air speed and Air flow :

Shaft height	Pole number	Air speed m/s	Air flow m^3/s
56	2	1.5	0.12
	4	0.75	0.04
	6	NA	NA
	8	NA	NA
63	2	2	0.16
	4	1	0.07
	8	0.5	0.03
71	2	2.5	0.21
	4	1.5	0.10
	6	1.0	0.07
	8	0.75	0.06
80	2	3.5	0.31
	4	2.5	0.19
	6	1.5	0.12
	8	1.2	0.09
90	2	4.5	0.36
	4	3.0	0.28
	6	2.0	0.17
	8	1.6	0.14
100	2	7.5	0.69
	4	4.5	0.42
	6	3	0.25
	8	2.5	0.19
112	2	11	0.15
	4	7	0.10
	6	7	0.10
	8	7	0.10
132	2	12	0.25
	4	9	0.20
	6	8	0.15
	8	8	0.15

Air speed and Air flow :

Shaft height	Pole number	Air speed m/s	Air flow m^3/s
160	2	11	0.35
	4	8	0.25
	6	6	0.20
	8	3	0.10
180	2	11	0.45
	4	8	0.30
	6	6	0.25
	8	4	0.15
200	2	10	0.45
	4	8	0.35
	6	5	0.25
	8	5	0.25
225	2	10	0.50
	4	10	0.55
	6	9	0.45
	8	7	0.35
250	2	10	0.55
	4	12	0.65
	6	9	0.45
	8	6	0.30
280	2	9.6	0.46
	4	8.5	0.39
	6	6.5	0.32
	8	7.6	0.36
315	2	8.3	0.46
	4	9.4	0.56
	6	7.5	0.40
	8	7.6	0.43
355	2	10	0.82
	4	13	1.1
	6	11.5	1.0
	8	8.5	0.7
400	2	15	1.4
	4	15	1.5
	6	11	1.1
	8	8	0.8
450	2	15	2.0
	4	15	2.0
	6	13	1.7
	8	10	1.25

Appendix 1: Tube Specifications

Tubing OD, in	BWG gauge	Wall thickness, in	Inside cross-sectional area, in^2	Ft^2 external surface per ft length	Ft^2 internal surface per ft length	Weight per ft length steel, lb †	Tubing ID, in	Moment of inertia, in^4	Section modulus, in^3	Radius of gyration, in	Costant C ‡	OD / ID	Transverse metal area, in^2
1/4	22	0.028	0.0295	0.0655	0.0508	.0066	0.194	0.00012	0.00098	0.0792	46	1.289	0.0195
1/4	24	.022	.0333	.0655	.0539	.054	.206	.00011	.00083	.0810	52	1.214	.0159
1/4	26	.018	.0360	.0655	.0560	.045	.214	.00009	.00071	.0824	56	1.168	.0131
1/4	27	.016	.0373	.0655	.0570	.040	.218	.00008	.00064	.0829	58	1.146	.0117
3/8	18	.049	.0603	.0982	.0725	.171	.277	.00068	.0036	.1164	94	1.354	.0502
3/8	20	.035	.0731	.0982	.0798	.127	.305	.00055	.0029	.1213	114	1.233	.0374
3/8	22	.028	.0799	.0982	.0835	.104	.319	.00046	.0025	.1227	125	1.176	.0305
3/8	24	.022	.0860	.0982	.0867	.083	.331	.00038	.0020	.1248	134	1.133	.0244
1/2	16	.065	.1075	.1309	.0969	.302	.370	.0022	.0086	.1556	168	1.351	.0888
1/2	18	.049	.1269	.1309	.1052	.236	.402	.0018	.0072	.1606	198	1.244	.0694
1/2	20	.035	.1452	.1309	.1126	.174	.430	.0014	.0056	.1649	227	1.163	.0511
1/2	22	.028	.1548	.1309	.1162	.141	.444	.0012	.0046	.1671	241	1.126	.0415
5/8	12	.109	.1301	.1636	.1066	.602	.407	.0061	.0197	.1864	203	1.536	.177
5/8	13	.095	.1486	.1636	.1139	.537	.435	.0057	.0183	.1903	232	1.437	.158
5/8	14	.083	.1655	.1636	.1202	.479	.459	.0053	.0170	.1938	258	1.362	.141
5/8	15	.072	.1817	.1636	.1259	.425	.481	.0049	.0156	.1971	283	1.299	.125
5/8	16	.065	.1924	.1636	.1296	.388	.495	.0045	.0145	.1993	300	1.263	.114
5/8	17	.058	.2035	.1636	.1333	.350	.509	.0042	.0134	.2016	317	1.228	.103
5/8	18	.049	.2181	.1636	.1380	.303	.527	.0037	.0118	.2043	340	1.186	.089
5/8	19	.042	.2298	.1636	.1416	.262	.541	.0033	.0105	.2068	358	1.155	.077
5/8	20	.035	.2419	.1636	.1453	.221	.555	.0028	.0091	.2089	377	1.126	.065
3/4	10	.134	.1865	.1963	.1262	.844	.482	.0129	.0344	.2229	285	1.556	.260
3/4	11	.120	.2043	.1963	.1335	.809	.510	.0122	.0326	.2267	319	1.471	.238
3/4	12	.109	.2223	.1963	.1393	.748	.532	.0116	.0309	.2299	347	1.410	.220
3/4	13	.095	.2463	.1963	.1466	.666	.560	.0107	.0285	.2340	384	1.339	.196
3/4	14	.083	.2679	.1963	.1529	.592	.584	.0098	.0262	.2376	418	1.284	.174
3/4	15	.072	.2884	.1963	.1587	.520	.606	.0089	.0238	.2410	450	1.238	.153
3/4	16	.065	.3019	.1963	.1623	.476	.620	.083	.0221	.2433	471	1.210	.140
3/4	17	.058	.3157	.1963	.1660	.428	.634	.0076	.0203	.2455	492	1.183	.126
3/4	18	.049	.3339	.1963	.1707	.367	.652	.0067	.0178	.2484	521	1.150	.108
3/4	20	.035	.3632	.1963	.1780	.269	.680	.0050	.0134	.2532	567	1.103	.079
7/8	10	.134	.2892	.2291	.1589	1.061	.607	.0221	.0505	.2662	451	1.441	.312
7/8	12	.109	.3390	.2291	.1720	.891	.657	.0196	.0449	.2736	529	1.332	.262
7/8	13	.095	.3685	.2291	.1793	.792	.685	.0180	.0411	.2778	575	1.277	.233
7/8	14	.083	.3948	.2291	.1856	.704	.709	.0164	.0374	.2815	616	1.234	.207
7/8	16	.065	.4359	.2291	.1950	.561	.745	.0137	.0312	.2873	680	1.174	.165
7/8	18	.049	.4742	.2291	.2034	.432	.777	.0109	.0249	.2925	740	1.126	.127
7/8	20	.035	.5090	.2291	.2107	.313	.805	.0082	.0187	.2972	749	1.087	.092
1	8	.165	.3526	.2618	.1754	1.462	.670	.0392	.0784	.3009	550	1.493	.430
1	10	.134	.4208	.2618	.1916	1.237	.732	.0350	.0700	.3098	656	1.366	.364
1	11	.120	.4536	.2618	.1990	1.129	.760	.0327	.0654	.3140	708	1.316	.332
1	12	.109	.4803	.2618	.2047	1.037	.782	.0307	.0615	.3174	749	1.279	.305
1	13	.095	.5153	.2618	.2121	.918	.810	.0280	.0559	.3217	804	1.235	.270
1	14	.083	.5463	.2618	.2183	.813	.834	.0253	.0507	.3255	852	1.199	.239
1	15	.072	.5755	.2618	.2241	.714	.856	.0227	.0455	.3291	898	1.167	.210
1	16	.065	.5945	.2618	.2278	.649	.870	.0210	.0419	.3314	927	1.149	.191
1	18	.049	.6390	.2618	.2361	.496	.902	.0166	.0332	.3366	997	1.109	.146
1	20	.035	.6793	.2618	.2435	.360	.930	.0124	.0247	.3414	1060	1.075	.106
1 1/4	7	.180	.6221	.3272	.2330	2.057	.890	.0890	.1425	.3836	970	1.404	.605

1 1/4	8	.165	.6648	.3272	.2409	1.921	.920	.0847	.1355	.3880	1037	1.359	.565
1 1/4	10	.134	.7574	.3272	.2571	1.598	.982	.0741	.1186	.3974	1182	1.273	.470
1 1/4	11	.120	.8012	.3272	.2644	1.448	1.010	.0688	.1100	.4018	1250	1.238	.426
1 1/4	12	.109	.8365	.3272	.2702	1.329	1.032	.0642	.1027	.4052	1305	1.211	.391
1 1/4	13	.095	.8825	.3272	.2775	1.173	1.060	.0579	.0926	.4097	1377	1.179	.345
1 1/4	14	.083	.9229	.3272	.2838	1.033	1.084	.0521	.0833	.4136	1440	1.153	.304
1 1/4	16	.065	.9852	.3272	.2932	.823	1.120	.0426	.0682	.4196	1537	1.116	.242
1 1/4	18	.049	1.042	.3272	.3016	.629	1.152	.0334	.0534	.4250	1626	1.085	.185
1 1/4	20	.035	1.094	.3272	.3089	.456	1.180	.0247	.0395	.4297	1707	1.059	.134
1 1/2	10	.134	1.192	.3927	.3225	1.955	1.232	.1354	.1806	.4853	1860	1.218	.575
1 1/2	12	.109	1.291	.3927	.3356	1.618	1.282	.1159	.1546	.4933	2014	1.170	.476
1 1/2	14	.083	1.398	.3927	.3492	1.258	1.334	.931	.1241	.5018	2181	1.124	.370
1 1/2	16	.065	1.474	.3927	.3587	.996	1.370	.0756	.1008	.5079	2299	1.095	.293
2	11	.120	2.433	.5236	.4608	2.410	1.760	.3144	.3144	.6660	3795	1.136	.709
2	14	.038	9	.5236	.4801	1.699	1.834	.2300	.2300	.6784	4121	1.090	.500
2 1/2	9	.148	3.815	.6540	.5770	3.719	2.204	.7592	.6074	.8332	5951	1.134	1.094

† **Weights are based on low-carbon steel with a density of 0.2833 lb/in^3. For other metals multiply by the following factors:**

Aluminum..0.35	Aluminum brass.....................1.06
Titanium..0.58	Nickel-chrome-iron..............1.07
AISI 400 series stainless steels...0.99	Admiralty.................................1.09
AISI 300 series stainless steels...1.92	Nickel and nickel-copper.....1.13
Aluminum bronze.................................1.04	Copper and cupronickels....1.14

‡ Liquid velocity = (lb per tube per h / C x sp gr of liquid) in ft / s [sp gr of water at 16°C (60°F) = 1.0]

Appendix 2: Air Properties

Temperature	Density	Specific heat capacity	Thermal conductivity	Kinematic viscosity	Expansion coefficient	Prandtl's number
- *t* -	- -	- c_p -	- *l* -	- *v* -	- *b* -	- P_r -
(^oC)	(kg/m^3)	(kJ/kg.K)	(W/m.K)	$\times 10^{-6}$ (m^2/s)	$\times 10^{-3}$ (1/K)	
-150	2.793	1.026	0.0116	3.08	8.21	0.76
-100	1.98	1.009	0.016	5.95	5.82	0.74
-50	1.534	1.005	0.0204	9.55	4.51	0.725
0	1.293	1.005	0.0243	13.3	3.67	0.715
20	1.205	1.005	0.0257	15.11	3.43	0.713
40	1.127	1.005	0.0271	16.97	3.2	0.711
60	1.067	1.009	0.0285	18.9	3	0.709
80	1	1.009	0.0299	20.94	2.83	0.708
100	0.946	1.009	0.0314	23.06	2.68	0.703
120	0.898	1.013	0.0328	25.23	2.55	0.7
140	0.854	1.013	0.0343	27.55	2.43	0.695
160	0.815	1.017	0.0358	29.85	2.32	0.69
180	0.779	1.022	0.0372	32.29	2.21	0.69
200	0.746	1.026	0.0386	34.63	2.11	0.685
250	0.675	1.034	0.0421	41.17	1.91	0.68
300	0.616	1.047	0.0454	47.85	1.75	0.68
350	0.566	1.055	0.0485	55.05	1.61	0.68
400	0.524	1.068	0.0515	62.53	1.49	0.68