Chapter – 3

Double Pipe Heat Exchanger

NON-PRINT ITEMS

Abstract

The chapter details design of double pipe heat exchanger with and without fins. The types of double pipe and their construction features along with the advantages and limitations of this class of exchanger are discussed. A data sheet format is provided along with applicable codes and standards. Typical exchanger configurations using standard Schedule 40 pipes with and without fins using applicable equations for thermal design and hydraulic design are presented. Analytical expressions of F_T for series-parallel configuration are also provided. Steps for designing a double pipe exchanger have been formalised and a design example is included to help the designer.

Key Words

Double pipe exchanger; Multitube double pipe exchanger; Finned tubes; Double pipe exchanger data sheet; Thermal design of double pipe exchanger; Hydraulic design of double pipe exchanger; Series-parallel configuration of hairpins; Design steps; Design example

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3.1 INTRODUCTION

Double pipe heat exchanger is probably the simplest to construct and is used for low heat load applications. As shown in Fig. 3.1, these may comprise of single tube or multiple tubes inside a shell. Commercially available single tube double-pipe sections range from 50 mm through 100 mm (2” through 4”) pipe size shells with inner tubes varying from 19 to 65 mm (¾” to 2½”) pipe size.

Multi-tube double pipe exchanger

The exchanger with a bundle of U tubes inside a pipe of 150 mm diameter and above uses segmental baffles (discussed in Chapter 4) and is referred to as hairpin or jacketed U tube exchanger. Multi tubular double pipe sections may contain 7 to 64 tubes within the outer tube. Nevertheless, sections containing more than 7 tubes per section are rarely used since they have limited, if any, economic advantage for most services. If the particular service requires fractional portions or short tube lengths of a multi-tube section, single tube sections are more economical. One end of the tube element is free-floating for thermal expansion. The two fluids usually flow in counter current mode for the highest thermal performance for a given surface area. However, for an almost constant wall temperature, the flow can be cocurrent. Double pipe sections have been designed for up to 165 bar (g) (2400 psig) on the shell side and up to 1033 bar (g) (15000 psig) on the tube side. Metal-to metal ground joints, ring joints or confined ‘O’-rings are used in the front end closures at lower pressures.

Finned tube

In general, the tubes used are plain but some applications use low-fin tubes for the inner pipes that provide about 2.5 times the external surface area. Finned tube in double pipe exchangers are economical if the heat transfer coefficient for the fluid flowing in the annular area is less than 75% of the tube side coefficient. The fins are longitudinally attached to the inner tube either by welding, brazing or mechanical bonding. Usually single tubes have longitudinal fins while multi-tubes have radial fins. The fins 16 to 48 per tube are 12.5 to 25mm (½ to 1 inch) high and 0.9 to 1.3 mm (35 to 50 mils) thick, and the fin height is dictated by the clearance between the inner and the outer pipe. Shorter fins have higher fin efficiency. The minimum thickness is rarely below 0.8 mm. Fin efficiency increases with decreasing annular coefficient and increasing fin thermal conductivity. Low-fin tubes are costlier by 50 to 70% compared to plain tubes.

Multitube exchangers with fins, typically use 12 to 20 fins per tube that are nominally 6 mm (1/4”) high and 0.9 mm thick. Normally, only bare tubes are used in sections containing more than 19 tubes.

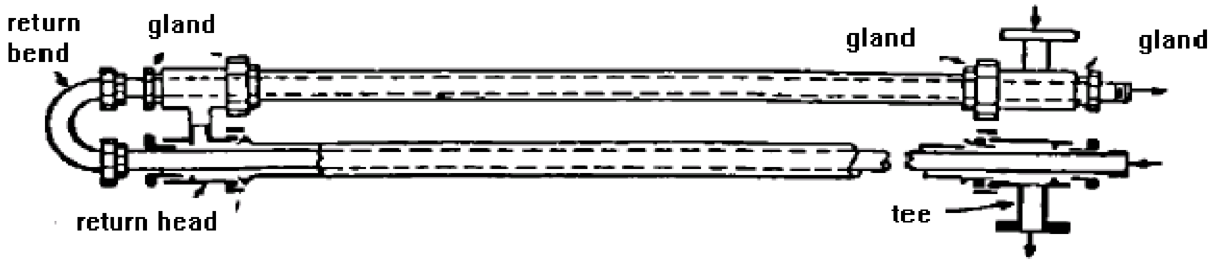


Figure 3.1a: Double pipe heat exchanger

Figure 3.1b: Double pipe heat exchanger stack

Figure 3.1c: Multi-tubular double pipe heat exchanger

In a double pipe exchanger sealing between the outer and the inner tube is by a gland seal. Gland packings are common wherever there is a shaft protruding through a body and the leakage of a fluid from the body through the junction of the shaft is to be prevented. Examples this are common – every valve has its stem passing through such a gland seal. The seal is provided by a packing between the inner pipe/ shaft and the outer pipe. The packing, uniformly compressed against a restrictor by a ring, provides a leak proof seal. The pressure on the ring is varied by tightening a gland which may either be threaded or flanged (in case of high pressure application). The gland not only prevents any leakage of fluid from the annular space but also ensures the concentric configuration of the inner and the outer pipes.

> Sealing arrangement between the outer and inner tube

The detail of the sealing gland sealing arrangement can be seen in Fig. 3.1a.

Double pipe sections can be combined in a variety of series / parallel arrangements to provide the required surface area while maintaining pressure drop limitations. Sections installed in series are normally mounted one on top of the other (Fig. 3.1b) and the sections in parallel are placed side by side. A combination of series-parallel arrangement elaborated in para 3.3 can be achieved by a combination of side-by-side and one-over-the other modules.

Series/Parallel arrangement

Advantages and disadvantages

- Double pipe exchangers are perhaps the simplest heat exchanger.
- Flow distribution is not a problem and disassembly and cleaning is easily.
- As the dimensions are small, these exchangers are suitable when either or both the fluids are at very high pressure.
- Since double-pipe sections permit true counter-current or true co-current flow, they may be of particular advantage when very close temperature approaches are required. Counter current flow results in lower surface area requirements, usually below 28 m^2 (300 sq. ft) for services having a temperature cross.
- In some cases where the thermal resistances of the two fluid films are essentially the same, it is found that for small heat loads, the installation of double pipe units are more economical than shell and tube units which are economic mostly in larger sizes.
- Hairpin exchangers are cheaper than shell and tube exchangers at very small sizes and can be specified for areas from 7 m^2 to 150 m^2
- They are easier to fabricate using standard bought out pipes and pipe fittings. Shortened delivery time result from the use of stock components that can be assembled into standard sections.
- Potential need for expansion joint is eliminated in U-tube construction.
- Double pipe exchangers are modular and are used in applications requiring adding and dismantling the modules or the rearrangement of sections for new services, thus ensuring flexibility.

Nevertheless, multiple hairpin sections are not always economically competitive with a single shell and tube heat exchanger. They are more expensive on a cost per unit area basis and are generally used for small capacity applications where the total heat transfer surface area is less than 50 m^2. Compared to shell and tube exchangers or other more compact heat exchanger types, these require more floor space and also entail a large number of points at which leakage may occur. In addition, proprietary closure design requires special gaskets and longer length is required to bring about the required heat transfer.

3.2 DESIGN

Double pipe heat exchanger design involves estimation of heat transfer area and pressure drop at the tube and the shell side. After determining the required heat exchanger surface area for counter flow or parallel flow, the pipe sizes and number of bends is finalised.

3.2.1 Input Data

The input data is same as that in case of any other type of exchanger. Table 3.1 lists the items of input data pertaining to the inner and outer fluids. In addition, information on the nature of the fluids e.g. flammability, corrosive nature, fouling tendency, solid concentration etc as applicable are also considered.

3.2.2. Deliverables

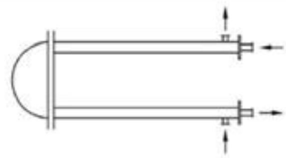
Design output is the details to be filled in the heat exchanger data sheet. A typical data sheet is shown in Table 3.1.

In addition, the design references: Process calculation references (Methods – Kern, HTRI etc.); Mechanical standard class (TEMA, BIS, etc.) are also to be furnished as part of design documentation.

Complete fabrication drawing consisting of the following set are required to be included – General arrangement drawings including stacking plan, if required; shell, nozzles and support details, other connections (vent, drain, instruments, etc.); tube bundle and its component details, if provided; details of head.

Table 3.1: Double Pipe Exchanger data sheet

DOUBLE PIPE HEAT EXCHANGER DATA SHEET						
UNITS OF MEASUREMENT : (SI)						
1	Service Of Unit:		No Of Units:		Item No:	
2	Site:		Manufacturer			
3	Size Type Arrangement Shell	Parallel	Series: Tube		Parallel:	Series
4	Surface/Unit (Eff.)	m^2	Section/Unit		Surface/Section (Eff.)	m^2
5	PERFORMANCE OF ONE UNIT					
6	Fluid Allocation		SHELL SIDE		TUBE SIDE	
7	Fluid Name					
8	Fluid Quantity, Total	Kg/hr				
9	Vapor (In/Out)	Kg/hr				
10	Liquid	Kg/hr				
11	Steam	Kg/hr				
12	Water	Kg/hr				
13	Non-Condensate (Mw)	Kg/hr				
14	Temperature	°C				
15	Density (Vapor/Liquid)	Kg/m^3				
16	Viscosity (Vapor/Liquid)	cP				
17	Molecular Weight	Kg/Kmol				
18	Specific Heat (Vapor/Liquid)	Kj/Kg°C				
19	Thermal Coductivity (Vapor/Liquid)	W/m°C				
20	Surface Tension	Dyn/cm				
21	Boiling Point	°C				
22	Latent Heat	Kj/Kg				
23	Inlet Pressure	barg				
24	Velocity	m/s				
25	Pressure Drop, Allowable/Calculated	bar				
26	Fouling Resistance (Min.)	m^2 °C/W				
27	Heat Exchanged	MW	MTD (Corrected) (Weighted)			°C
28	Transfer Rate	W/m^2 °C				
29	CONSTRUCTION OF ONE SHELL					
30		SHELL SIDE	TUBE SIDE	Sketch		
31	Design Pressure barg					
32	Design Temperature Max/Min °C					
33	Corrosion Allowance mm					
34	Insulation THK. In/Out mm					
35	Connections In					
36	Size & Out					
37	Rating					
38	Tube No. O.D. (mm);	Thk. mm (Ave/Min)	Length mm;	Pitch mm ;	Flow Angle	Deg
39	Tube Type		Material			
40	Fins: No. Height mm;	Thk. mm	Type		Material	
41	Shell O.D. mm;	Thk. mm	Material			
42	Tube Sheet - Stationary		Impingement Protection			
43	Baffles-Cross Type	%Cut	Spacing:c/c	mm;	Inlet	mm
44	Shell Return Bend - Housing Material		Cover Material			
45	Tube Side Closure - Type				Material	
46	External Return Bend: OD mm;	Thk.	mm;		Material	
47	Gasket - Shell Side		Tube Side			
48	Code Requirements		Stamp NO			
49	Double Pipe Type?		Multi Tube Type?			
50	Remarks:					
51						
52						
53						
Rev.	Date	Description	App.1	App.2	App.3	



3.2.3 Codes and Standards

Common standards for double pipe heat exchangers are TEMA and API 660. There is no Indian (BIS) code. Hairpin sections are specially designed units which are normally not built to any industry standard other than ASME Code. However, TEMA tolerances are normally incorporated wherever applicable.

3.2.4 Guidelines to select inner and outer fluid

The guideline for selecting the inner and outer fluid is same for a shell and tube exchanger and a double pipe exchanger. The general guidelines for preliminary selection are presented in Table 3.2. The are general in nature and not rigid rules. Optimal fluid placement depends on several service specific factors as well.

Table 3.2: General guidelines for selecting the shell and tube side fluids

Tube side fluid	**Shell side fluid**
• *High temperature fluid* At higher temperature the allowable stress is lower. Since tubes have much lower diameter as compared to shell, they can withstand higher pressure at the same temperature. This makes the design safer. Further, this ensures lower heat losses from the exchanger to the surroundings and lower cost of exchanger insulation. • *Dirty and Fouling fluids* Tubes are easier to clean. Fouling tendency is lower due to fewer stagnation points. Usually cooling water is in tubes for this reason. Also the tube fluid, mostly flowing at a higher velocity would have lower fouling (less deposit). Mechanical cleaning is easier for tubes, slurry is preferred in the tube side for this reason. • *More hazardous or expensive fluid* The chance of leaking out is less. • *Fluid at higher pressure* Lower diameter of tubes call for a lower wall thickness compared to the shell. • *Corrosive fluid* Only the tubes and not the shell is exposed to the corrosive environment. A corrosive fluid in shell would affect both the shell and the tubes. In addition, it is cheaper to fabricate tubes from expensive corrosion resistant materials. • *Streams with low flow rates* These are placed in tubes to obtain increased velocity and turbulence.	• *More viscous fluid* The critical Reynolds number for turbulent flow is 200 on the shell side. Thus for the same *Re*, when flow in laminar in tubes, the shell flow may be turbulent. However if the flow is still laminar in the shell, it is directed through the tubes as this ensures more accurate prediction of both heat transfer and flow distribution. • *Liquid with lower flow rate* To avoid multipass construction that will have LMTD correction factor below unity. Turbulent flow may also result due to lower critical Reynolds number for the shell side. • *Fluid undergoing phase change e.g. condensing steam/vapor* Shell side offers a lower pressure drop. Vapor-liquid mixtures resulting from vapor condensation is allowable in vertical condensers. • *Fluid for which pressure drop limit is lower* or there is chance of exceeding the same e.g. fluid of high viscosity. • *Fluid that has poorer heat transfer characteristics:* As the critical Reynolds number for turbulent flow is 200 on the shell side. • *Fluid with large ΔT (>40^{o}C)*

3.2.5 Design considerations

Heat exchangers shall be designed to conform to specified shell side or tube side design pressure with respect to the ambient. Designs based on differential pressure of shell side and tube side is not permitted. Minimum design pressure shall be 10% above the maximum operating pressure or maximum operating pressure plus 2 bar (200 kPa), whichever is greater. Double pipe sections have been designed for up to 165 bar (g) (2400 psig) on the shell side and up to 1033 bar (g) (15000 psig) on the tube side~~.~~

Minimum design temperature shall be 10% above maximum operating temperature, or maximum operating temperature plus 28°C whichever is greater.

Tube elements shall be removable without cutting the shell or connecting piping and without disconnecting the shell piping. One end of the tube element shall be free-floating for thermal expansion. No internal screwed connections shall be allowed. Over-all length shall be approximately 10 meters. Minimum outside tube diameter of the tube element shall be 25.4 mm (1”) and minimum thickness shall be equivalent to 12 BWG tubing or Schedule 40 pipe. All pipe and tubing used in construction of the exchangers shall be seamless.

Minimum corrosion allowance on pressurized steel pressure parts shall be 3 mm for hydrocarbon services, except for tubes.

Heat transfer area

The heat transfer area and heat transfer coefficients shall be based on the total effective outside tube and fin surface. The effective tube wall and fin metal resistance shall be considered in calculating the heat transfer coefficient. Finned tubes should not be used where fouling is expected on the shell side; or the fins are likely to be exposed to a corrosive medium. A hairpin exchanger is not permitted if fouling is expected in the tube side.

Cooling water is normally passed through the tube side. Minimum allowed water velocity is 1 m/sec. Fouling factors for circulating cooling water may be taken 0.35 m^2.°C/ kW or 0.00035 m2.°C/W (0.002 ft2.h.°F/Btu).

The suitability of using hairpin exchanger in a given application may be evaluated by computing the product of heat transfer coefficient and area (UA). For preliminary evaluation, (UA) of 80 kW/K may be considered to be the upper economical limit for applying hairpin type units. Above this value the unit may be uneconomical for a hairpin type design. If a hairpin is applied, it may require multiple ND 400 (16") multitube sections. In the range of 53 to 80 kW/K one or more ND 300 (12") to ND 400 (16") multitube sections will normally be required. In the range of 26 to 53 kW/K one or more ND 100 (4") to ND 300 (12") multitube sections will normally be required. Below 26 kW/K, both double pipe and multitube sections should be compared based on economics. Table 3.3 lists typical sizes for hairpin type exchangers.

Table 3.3: Typical hairpin type exchanger sizes

	Double Pipe	**Multitube**
Shell Dia., ND mm (inch)	50 – 150 (2 – 6”)	80 – 4300 (3”- 16”)
Tube Dia., ND mm (inch)	20 – 100 ($^{3}/_{4}$” – 4”)	20 – 25 ($^{3}/_{4}$” – 1”)
No. of longitudinal fins, N_f, when used	20 to 48	0 or 16 or 20
Fin height, h_f mm (inch), when provided	10 – 25 (0.375 – 1”)	0 – 12.7 (0 – ½”)
Surface m^2/6m (ft^2/20 ft)	3 – 12.2 (10 – 40)	23 – 60 (75 – 1500)
Fin thickness, t_f, mm (inch),	0.889 (0.035”) for weldable metals 0.5 (0.197”) for soldered fins below 12.5 mm height and 0.8 (0.0315”) for fins above 12.5 mm height	

3.2.6 Thermal Design

The following outlines the steps of calculation for a tube-in-tube double pipe exchanger utilising the applicable equations outlined in Chapter 2.

For a double pipe exchanger, the heat transfer area A in Eqn. 2.3 is the outer surface area of the inner conduit. The size designation for heat exchanger tubes is different from pipes. The nominal outside diameter of a heat exchanger tube is its actual (outside) diameter and the wall thickness is specified by Birmingham Wire Gage (BWG) instead of Schedule number.

Thus the design equation is - $A = A_o = \pi D_o L = \dfrac{Q}{U_o \Delta T_{LMTD}}$ (3.1)

U in Eqn. 3.1 is obtained from Eqn. 2.13 based on the outer diameter of the inner pipe, viz.

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{D_o \ln\left(D_o / D_i \right)}{2k_w} + \frac{D_o}{h_i D_i} \qquad (3.2)$$

Subscripts o and i denote conditions at the tube outside and inside respectively. Thus $(1/U_o)$ is the overall thermal resistance based on the tube outside area, h_i and h_o are the heat transfer coefficient for the inner and annular fluids and D_i , D_o are the inner and outer diameter of the tube. Incorporating dirt factor R_{Di} and R_{Do} for the inner and outer wall of the tube, the design overall heat transfer coefficient for a finned multitube double pipe exchanger can be expressed as

Overall Heat Transfer Coefficient

$$\frac{1}{U_D} = \frac{1}{h_o E_{f,effective}} + \frac{A_{total} \ln\left(D_o / D_i \right)}{2\pi k_w L} + \frac{A_{total}}{h_i A_i} + \frac{R_{Do}}{E_{f,effective}} + \frac{R_{Di} A_{total}}{A_i} \qquad (3.3)$$

One may note that U_D is defined with respect to the total area A_{total}. Although fins can be attached to both internal and external pipe surface, external fins are most frequently used. Accordingly, $E_{f,effective}$ the weighted fin efficiency for the entire finned surface is associated with the pipe outer wall only in Eqn. 3.3 where $E_{f,effective}$ is given as

$$E_{f,effective} = \frac{A_{prime} + E_f A_f}{A_{total}} \qquad (3.4)$$

The area of the prime surface (A_{prime}), longitudinal fin surface (A_f) and total cross sectional area

(A_{total}) is -

$$A_{prime} = \left(\pi D_o - N_f t_f \right) nL \qquad (3.5a)$$

$$A_f = 2nN_f \left(h_f + \tfrac{t_f}{2} \right) L \qquad (3.5b)$$

$$A_{total} = A_{prime} + A_f \quad (3.5c)$$

N_f is the number of longitudinal fins having height h_f and thickness t_f on each tube and n is the number of finned tubes, each of length L .

Typical dimensional configuration of finned tubes used in industry is shown in Table 3.4. The exact values can also be obtained from finned tube manufacturers.

Table 3.4: Typical double pipe exchanger configurations (a) single inner pipe, (b) multiple pipes

(a) 40 Schedule single inner pipe						
Outer pipe			Inner pipe			
Nominal Diameter, Inch	Wall thickness, mm	OD, mm	Max. no of fins / tube N_f	OD, mm	Wall thickness, mm	Fin height h_f , mm
2	3.91	60.3	20	25.4	2.77	11.1
3	5.49	88.9	20	25.4	2.77	23.8
3.5	5.49	88.9	36	48.3	3.68	12.7
3.5	5.74	101.6	40	60.3	3.91	12.7
4	6.02	114.3	36	48.3	3.68	25.4
4	6.02	114.3	40	60.3	3.91	19.05
4	6.02	114.3	48	73.0	5.16	12.7

(b) 40 Schedule multiple inner pipes							
Outer pipe			Inner pipe				
Nominal Diameter, Inch	Wall thickness, mm	OD, mm	Number of tubes, *n*	Max. no of fins per tube, N_f	OD, mm	Wall thickness, mm	Fin height h_f , mm
4	6.02	114.3	7	16	19.02	2.11	5.33
4	6.02	114.3	7	20	22.2	2.11	5.33
6	7.11	168.3	19	16	19.02	2.11	5.33
6	7.11	168.3	14	16	19.02	2.11	5.33
6	7.11	168.3	7	20	20.04	2.77	12.7
8	8.18	219.1	19	16	19.02	2.11	8.64
8	8.18	219.1	19	20	22.2	2.11	7.11
8	8.18	219.1	19	20	25.4	2.77	5.33
8	8.18	219.1	19	16	19.02	2.11	7.11
8	8.18	219.1	19	20	22.2	2.11	5.33

Since double pipe exchangers employ longitudinal fins, E_f in Eqn. 3.4 is given by Eqn. 2.18 reproduced below -

$$h_{f,eq} = h_f + {}^{t_f}\!/_{2} \quad (2.18)$$

If both fluids are in turbulent flow, the heat transfer coefficients $\left(h_i\right)$ and $\left(h_o\right)$ for plain tubes may be computed from the same correlation using a suitably defined equivalent diameter (D_e), otherwise, special attention must be given to the annular region. Referring to Table 2.6 and using the nomenclatures defined therein -

Turbulent flow, Re>10000

Individual Heat Transfer Coefficients

$$hD_e/k = 0.027 \left(D_eG/\mu\right)^{0.8} \left(C_p\mu/k\right)^{0.33} \left(\mu/\mu_w\right)^{0.14} \qquad (3.6a)$$

[Some prefer replacing 0.027 with 0.023 for double pipe]

Intermediate flow range (10000>Re>2100)

$$hD_e/k = 0.116\left[\left(D_eG/\mu\right)^{2/3} - 125\right]\left(C_p\mu/k\right)^{1/3} \left(\mu/\mu_w\right)^{0.14} \left[1+(D_i/L)^{2/3}\right] \qquad (3.6b)$$

Laminar flow, $\mathrm{Re}\left(=D_eG/\mu\right) < 2100$

$$hD_e/k = 1.86\left[\left(k/D_e\right)\left(C_p\mu/k\right)D_e/L\right]^{1/3} \left(\mu/\mu_w\right)^{0.14} \qquad (3.6c)$$

The heat transfer coefficients (h_o) for finned tube *in the annulus has* been expressed in terms of j_H factor $\left(j_H = \left(\frac{h_0 D_e}{k}\right)\left(\frac{C_p\mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14}\right)$ as a function of Reynolds number ($\mathrm{Re}_o = \frac{\rho_o V_o D_e}{\mu}$) by Kern and Krauss (1972).

$$j_H = \left(0.0263\,\mathrm{Re}_o^{\,0.9145} + 4.9\times10^{-7}\,\mathrm{Re}_o^{\,2.618}\right)^{1/3} \text{ for } N_f = 24 \qquad (3.7a)$$

$$j_H = \left(0.0116\,\mathrm{Re}_o^{\,1.032} + 4.9\times10^{-7}\,\mathrm{Re}_o^{\,2.618}\right)^{1/3} \text{ for } N_f = 36 \qquad (3.7b)$$

Eqn. 3.7a and 3.7b predict nearly same values of j_H for Re_o >1000.

Fluid flow properties usually are functions of the flow temperature and may be evaluated at the *caloric temperature*. If the temperature difference of flow is moderate or the fluids have viscosity less than 1 cP at cold terminal temperature, $\mathrm{T}_{f,avg}$ (arithmetic average temperature) is used instead of caloric temperature.

Individual Heat Transfer Coefficients

μ_w in the Sieder-Tate correction factor $\left(\mu / \mu_w\right)$ of Eqn. 3.6 is estimated at the average wall temperature of the inner pipe given by

$$T_w = \frac{h_i T_{i,avg} + h_o \left(D_o / D_i\right) T_{o,avg}}{h_i + h_o \left(D_o / D_i\right)} \tag{3.8}$$

Wall Temperature

Eqn. 3.8 is obtained by assuming that the entire heat transfer occurs between the fluids at their average temperature through the wall of the inner pipe. For hot fluid flowing through the inner pipe, this gives

$$h_i A_i \left(T_{i,avg} - T_w\right) \quad = \quad h_o A_o \left(T_w - T_{o,avg}\right) \tag{3.9}$$

Where $T_{i,avg}$ and $T_{o,avg}$ are the average temperature for the inner and outer fluids respectively

Use of Eqn. 3.8 involves an iterative procedure since T_w is required to calculate $\left(h_i\right)$ and $\left(h_o\right)$ and vice versa. Initially the values of $\left(h_i\right)$ and $\left(h_o\right)$ are calculated by assuming $\left(\mu / \mu_w\right) = 1$. The calculated values of $\left(h\ \right)$ are used to calculate T_w and obtain μ_w. The viscosity correction factor for both the fluids is then multiplied to the preliminary values of $\left(h_i\right)$ and $\left(h_o\right)$ to obtain the final values of the film coefficients. A single iteration usually suffices.

For finned tubes, the viscosity correction factor for the fluid in the inner pipe $\left(\mu / \mu_w\right)_i$ is calculated at T_{prime}, the temperature of the prime surface and for the outer fluid $\left(\mu / \mu_w\right)_o$ is calculated at T_{wf}, the weighted average temperature of the extended and prime surfaces. The derivation for the two wall temperatures is based on the assumption that all the heat is transferred between the streams at their average temperatures, $T_{i,avg}$ and $T_{o,avg}$ or

$$Q = h_i A_i \left(T_{i,avg} - T_{prime}\right) \quad = \quad h_o E_f A_{Total} \left(T_{prime} - T_{o,avg}\right) \tag{3.10}$$

Where T_{wf} is defined by $Q = h_o A_{Total} \left(T_{wf} - T_{o,avg}\right)$ (3.11)

This gives the expressions of the wall temperatures as -

$$T_{prime} = \frac{h_i T_{i,avg} + h_o E_{f,effective} \left(A_{total} / A_i\right) T_{o,avg}}{h_i + h_o E_{f,effective} \left(A_{total} / A_i\right)} \tag{3.12a}$$

$$T_{wf} = \frac{h_i E_{f,effective} T_{i,avg} + \left[h_i \left(1 - E_{f,effective}\right) + h_o E_{f,effective} \left(A_{total} / A_i\right)\right] T_{o,avg}}{h_i + h_o E_{f,effective} \left(A_{total} / A_i\right)} \tag{3.12b}$$

The equivalent diameter (D_e) is the inside diameter (D_i) for the inner pipe.

Equivalent diameter (D_e) for the annulus is four times the mean hydraulic radius, r_H that is defined as the ratio of flow area and wetted perimeter.

Equivalent diameter, D_e

$$D_e = D_{io} - D_o \qquad (3.13a)$$

where D_{io} is the inner diameter of the outer pipe.

According to Kern (1950) the wetted perimeter for heat transfer calculations is the outer circumference of the inner tube$\left(\pi D_o\right)$. Therefore, the equivalent diameter ($D_e^{'}$) for thermal calculations as defined by Kern (1950) is

$$D_e^{'} = \frac{D_{io}^{\ 2} - D_o^{\ 2}}{D_o} \qquad (3.13b)$$

Where the cross sectional area is $\frac{\pi}{4}\left(D_{io}^{\ 2} - D_o^{\ 2}\right)$. He has used Eqn. 3.13b for evaluation of both Nusselt Number as well as Reynolds number.

However, the Reynolds number estimation for calculation of pressure drop is always based on Eqn. 3.13a. In this book D_e both for thermal as well as pressure drop calculations have been evaluated by Eqn. 3.13a.

For plain multitube hairpin exchangers containing *n* tubes each of OD (D_o) housed within an outer pipe of diameter (D_{io}), the expressions for flow area [$A = (\pi/4)\left(D_{io}^{\ 2} - nD_o^2\right)$] and wetted perimeter [$(\pi)\left(D_{io} + nD_o\right)$] gives the expression for equivalent diameter as –

$$D_e' = \frac{\left(D_{io}^{\ 2} - nD_o^2\right)}{\left(D_{io} + nD_o\right)} \qquad (3.13c)$$

The above expression reduces to Eqn. 3.13a for $n = 1$

In a finned annulus, with fin length being L_f, the equivalent diameter D_{ef} obtained as 4 times the flow area$\left[A = (\pi/4)\left(D_{io}^{\ 2} - nD_o^2\right) - nN_f L_f t_f\right]$divided by the wetted perimeter for heat transfer [$(\pi)\left(D_{io} + nD_o\right) + 2nN_f L_f$] is

$$D_{ef} = \frac{\pi\left(D_{io}^{\ 2} - nD_o^2\right) - 4nN_f L_f t_f}{\pi\left(D_{io} + nD_o\right) + 2nN_f L_f} \qquad (3.13d)$$

3.2.7 Hydraulic Design

The pressure drop for flow through the straight length of annulus is expressed in *liquid* (fluid) head is

$$\Delta H_{fo} = \frac{4 f_o G_o^2 L_o}{2g\rho_o^{\ 2} De} \qquad (3.14a)$$

Pressure Drop in straight length

and for the inner pipe it is -

$$\Delta H_{fi} = \frac{4 f G_i^2 L_i}{2 g \rho_i^2 D_i} \tag{3.14b}$$

G is mass velocity of the fluid, g is the acceleration due to gravity, ρ is fluid density, L is the length of the corresponding section and f is the *Fanning friction factor*. When several double pipe exchangers are connected in series, annulus to annulus and pipe to pipe, the length (L) in Eqn. (3.14) is the total for the entire path. The friction factor (f) in Eqn. 3.14 is expressed as a function of Reynolds number, defined as -

$$\mathrm{Re}_i = \frac{G_i D_i}{\mu_{i,average}} \text{ for the inner fluid} \tag{3.15a}$$

and $$\mathrm{Re}_o = \frac{G_o D_e}{\mu_{o,average}} \quad \text{based on } D_e \text{ for the annulus} \tag{3.15b}$$

Turbulent flow –

Flow in tubes, with ±5% tolerance: $f_i = 0.0014 + 0.125/(\mathrm{Re})^{0.32}$ (3.16a)

Flow in clean iron and steel pipes, with ±10% tolerance: $f_i = 0.0035 + 0.246/(\mathrm{Re})^{0.42}$ (3.16b)

Laminar flow -

Flow in tubes: $f_i = 16/\mathrm{Re}_i$ (3.16c).

and for the outer fluid

$$f_o = \left(16/\mathrm{Re}_o\right)\left[\frac{1-\left(D_o/D_{io}\right)^2}{1+\left(D_o/D_{io}\right)^2+\left(1-\left(D_o/D_{io}\right)^2\right)/\ln\left(D_o/D_{io}\right)}\right] \tag{3.16d}$$

For longitudinal finned tubes, the friction factor for the annular region is

$$f_{of} = \exp\left[0.08172\left(\ln \mathrm{Re}_{of}\right)^2 - 1.7434\left(\ln \mathrm{Re}_{of}\right) - 0.6806\right] \text{ for } \left(\mathrm{Re} > 400\right) \tag{3.16e}$$

and

$$f_{of} = 16/\mathrm{Re}_{of} \quad \text{for } \left(\mathrm{Re} \le 400\right) \tag{3.16f}$$

Since fins tend to destabilise laminar flow, the critical Reynolds number is 400 in the finned annulus.

A minor modification is often made to Eqn. 3.14 by incorporating a viscosity correction factor (ϕ) to account for the effect of variable fluid property on friction factor in non-isothermal flow, viz

$$\phi = \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad \text{for laminar flow} \tag{3.17a}$$

$$\phi = \left(\frac{\mu}{\mu_w}\right)^{0.25} \quad \text{for turbulent flow} \tag{3.17b}$$

This modifies the pressure drop equation (Eqn. 3.14a & b) for the outer and inner fluid as –

$$\Delta H_{f,o} = \frac{4 f_o G_o^2 L_o}{2g\rho_o^{\ 2} De'}\left(\frac{1}{\phi}\right) \tag{3.18a}$$

$$\Delta H_{f,i} = \frac{4 f G_i^2 L_i}{2g\rho_i^{\ 2} D_i}\left(\frac{1}{\phi}\right) \tag{3.18b}$$

Minor pressure losses due to entrance and exit effects and return bends of each hairpin are usually estimated in terms of velocity heads. For inner pipes of double pipe exchangers connected in series, the bend pressure loss is usually negligible but the same may be significant for the annuli.

Bend pressure drop

In an exchanger with N_{HP} number of hairpins connected in series, the total pressure drop due to direction change is

$$\Delta H_{f,o,bend} = \frac{(2N_{HP} - 1)V_o^2}{2g} \tag{3.19}$$

Where V_o is the velocity of the outer fluid.

Inner pipe: With the inlet and exit piping aligned with the inner pipe the entrance and exit losses can be neglected. However in multitube exchangers the losses at the two tubesheets are taken as one tube velocity head per hairpin for turbulent flow.

Entry and exit losses

Annulus:

Nozzle entry and exit losses are accounted as –

Laminar flow: For Re≥100, 3 velocity heads for head loss in the entry and the exit nozzle together. For Re<100, the loss depends on Re.

Turbulent flow: 1 velocity head for the entry and 0.5 velocity head for the exit nozzle.

For exchangers with internal return bends, nozzle head loss is given by –

$$\Delta H_n = \frac{2(N_{HP})V_n^2}{2g} \text{ , for turbulent flow} \tag{3.20a}$$

$$\Delta H_n = \frac{4(N_{HP})V_n^2}{2g} \text{ , for laminar flow and Re>100} \tag{3.20a}$$

Where V_n is nozzle velocity

In case the exchanger has *external return bends*, the pressure drop is double of the value estimated by Eqn. 3.20.

Total pressure Drop

The total pressure drop in the annular section is:

$$\Delta P_o = \left(\Delta H_{f,o} + \Delta H_{f,o,bend} + \Delta H_n\right)\rho_o g \tag{3.21a}$$

and in the inner pipe is: $\Delta P_i = \left(\Delta H_{f,i}\right)\rho_i g$ (3.21b)

Typically the maximum allowable design pressure drops in a double pipe heat exchanger in 0.7 kg/cm^2 for both inner and outer pipes. If the calculated pressure drop exceeds the allowable limit, the designer needs to select a larger pipe diameter, or decide to connect sections in parallel or a combination of series and parallel. The flow with higher volumetric flow rate is usually sent to the side with higher flow cross sectional area.

3.3 SERIES-PARALLEL CONFIGURATION OF HAIRPINS

Pressure drop constraint in double pipe exchanger can often be met by dividing only the specific stream exceeding the pressure drop limit in parallel branches. Fig. 3.3 shows such a configuration with the annuli in series and the inner pipes connected in parallel just for 2 hairpins. Several such hairpins may be configured. Each hairpin has counterflow but the overall flow arrangement is not true countercurrent. The departure from true counterflow operation in a series–parallel arrangement is accounted for by the LMTD correction factor F_T discussed in Chapter 2. For x number of parallel branches -

$$F_T = \frac{P(1-x)}{x(1-P)\ln\left[\dfrac{(1-x)}{(1-P)^{1/x}} + x\right]} \qquad \text{for } R=1 \qquad (3.22a)$$

$$F_T = \left[\frac{R-x}{x(R-1)}\right]\frac{\ln\left[(1-P)/(1-PR)\right]}{\ln\left[\dfrac{(R-x)}{R(1-PR)^{1/x}} + \dfrac{x}{R}\right]} \qquad \text{for } R\neq 1 \qquad (3.22b)$$

where, $P = \dfrac{T_{p,out} - T_{p,in}}{T_{s,in} - T_{s,\text{out}}}$ and $R = \dfrac{T_{p,out} - T_{p,in}}{T_{s,in} - T_{s,\text{out}}}$

$T_{p,out}$, $T_{p,in}$, $T_{s,\text{out}}$ and $T_{s,in}$ are the outlet and the inlet temperatures of the streams (p and s) that flow parallel and in series through the set of hairpins.

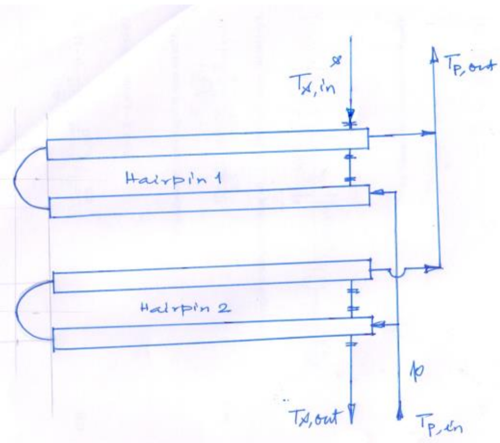


Figure 3.2: Two hairpins with annuli in series and inner pipes in parallel

3.4 DESIGN ILLUSTRATION

3.4.1 Design steps

The design output is the exchanger geometry meeting the heat load target and constraints of pressure drop. This can be met by *several combinations of inner and outer pipe sizes* and corresponding series, or series-parallel configuration of hairpins. Considering the pipe sizes to be the designer's choice the steps to be followed by the designer are the following -

1. Input data: $\dot{m}_h, \dot{m}_c, c_{p,c}, c_{p,h,}$ any 3 of $\{T_{h,\text{in}}, T_{h,out}, T_{c,\text{in}}, T_{h,out},\}$
2. Calculate heat load, Q from enthalpy balance of hot / cold stream.
   Find the unknown variable in the set $\{T_{h,\text{in}}, T_{h,out}, T_{c,\text{in}}, T_{h,out},\}$.
3. Estimate $T_{\text{c,avg}} = (T_{c,\text{in}} + T_{\text{c,out}})/2$, $T_{\text{h,avg}} = (T_{h,\text{in}} + T_{\text{h,out}})/2$,
4. Note $\rho_{c,avg}, k_{c,avg}, \mu_{c,avg}$ at $T_{\text{c,avg}}$ and $\rho_{c,avg}, k_{c,avg}, \mu_{c,avg}$ at $T_{h,\text{avg}}$ for calculations. Also note the variation of viscosity with temperature for both liquids.
5. Decide the maximum limit of $\Delta P_{i,\max}, \Delta P_{o,\max}$ or the same for the two fluids streams.
6. Select inner and outer tube/pipe specifications (typical starting values can be 1.25"and 2" ND 40 Schedule pipes of length 6 m or 6.5 m). Note down values of D_i, $D_{i,o}$, $t_w = (D_i o - D_i)/2$, D_o, L_{std}. Note k_{wall} value.
7. Note values of R_{Do} and R_{Di} to be considered. These are often associated with the two fluids.
8. Select the inner and the annulus fluid and these are henceforth designated by subscript *i* and *o*. The new set of variables with these subscripts is derived from those mentioned in steps 2, 3 and 4. Consider counterflow configuration.
9. $A_i = \pi D_i^2/4$, $A_o = \pi(D_{i,o}{}^2 - D_o{}^2)/4$, $G_i = \dot{m}_i/A_i$, $G_o = \dot{m}_o/A_o$, $D_\text{e} = (D_{i,o} - D_o)$, $\dot{m}_i$ and $\dot{m}_o$ are the mass flow ratesof the inner and the outer fluids
10. $\text{Re}_i = D_i G_i / \mu_{\text{i},avg}$, $\text{Re}_o = D_\text{e} G_o / \mu_{\text{o},avg}$, $\text{Pr}_i = c_{p,\text{i}}\, \mu_{i,avg}/k_{i,avg}$, $\text{Pr}_o = c_{p,0}\, \mu_{o,avg}/k_{o,avg}$
11. Assume $\phi_i = 1$, $\phi_o = 1$ as initial guess.
12. Compute h_i and h_o from Eqn. 3.6 / 3.7
13. IF $\left(\dfrac{h_o D_\text{e}}{h_i D_i}\right) < 0.75$, THEN

    Place fluid with lower *h* in annulus and provide fins on inner pipe. Select N_f, L_f, t_f, h_f for the finned tube(s) from Table 3.3 and Table 3.4 and decide on n. One may start with n=1 and increase later, if required.
    Calculate L_f (Eqn. 2.18), *m* (Eqn. 2.16), E_f (Eqn. 2.15), A_{prime}, *A* and A_{total} (Eqn. 3.5), $E_{f,effective}$ (Eqn. 3.4), D_{ef} (Eqn. 3.13d).
    Calculate h_o (Eqn. 3.7), assuming $\phi_o = 1$
    Calculate (A_{total}/A_{prime}) and use it in Eqn. 3.12 to calculate T_{prime} and T_{wf}. Refer to fluid property data and note $\mu_{\text{o},w}$ value at T_{wf}. Calculate $\phi_{o,new} = (\mu_{\text{o},avg}/\mu_{\text{o},w})^{0.14}$. Calculate T_w (Eqn. 3.8). Refer to fluid property data and note $\mu_{\text{i},w}$ value at T_{wf}. Calculate $\phi_{i,new} = (\mu_{\text{i},avg}/\mu_{\text{i},w})^{0.14}$.

    IF (abs(($\phi_{o,new}$ - ϕ_o)/ $\phi_{o,new}$) <0.02) and (abs(($\phi_{o,new}$ - ϕ_o)/ $\phi_{o,new}$) <0.02) THEN

    GO TO Step 14

ELSE

GO TO Step12

END

ELSE

Calculate T_w from Eqn.3.8 and proceed to calculate h_i from Eqn. 3.6.

GO TO Step 14

END

14. Calculate U_D (Eqn. 3.3). Calculate LMTD using F_T from Eqn. 3.22 if series-parallel configuration is chosen, else LMTD to be calculated directly from $\{T_{h,\text{in}}, T_{h,out}, T_{\text{c,in}}, T_{h,out},\}$.
15. Calculate A_o (Eqn. 3.1). $L_{total} = A_o / (\pi D_{i,o})$, $N_{HP} = L_{total} / (2L_{std})$; Round off L_{total} to next higher value of L_{std} so that there are integral number of hairpins.
16. Calculate f_i corresponding to Re_i (Eqn. 3.16). Calculate D_e' (Eqn. 3.13c). Calculate Re_o (Eqn. 3.15b). Calculate f_o corresponding to Re_o (Eqn. 3.16 d-f).
17. Calculate $\Delta H_{f,o}$ (Eqn. 3.18a) , $\Delta H_{f,o,bend}$ (Eqn. 3.19) and ΔH_n (Eqn. 3.2)

    Calculate ΔP_o (Eqn. 3.21a).

    Calculate $\Delta H_{f,i}$ (Eqn. 3.18b). Calculate ΔP_i (Eqn. 3.21b).
18. IF $\Delta P_i > \Delta P_{i,\max}$ THEN

Switch fluids and check for pressure drop.

IF even after switching fluids, the pressure drop limits are exceeded THEN

connect annuli in parallel and tubes in series. Recalculate F_T using Eqn. 3.22.

Go to step 9.

END

ELSE

Print Design output and fill up the rest of the form shown in Table 3.1.

END

3.4.2 Design example

Problem: Design a double pipe heat exchanger to cool 2000 kg/hr of 5% w/w caustic solution from 80°C to 40° using cooling water available at 33°C. Maximum return temperature for the cooling water stream is 45°C. the dirt factor for caustic and cooling water may be taken as $0.00035\text{m}^2\text{K/W}$ and $0.00018\text{m}^2\text{K/W}$. Maximum pressure for the cooling water and the caustic pump header are 5 and 4 kg/cm^2(g) respectively and the maximum allowable pressure drop is 0.7 kg/cm^2 for both the fluids.

Viscosity variation of cooling water and 5% w/w caustic lye with temperature								
T (°C)	30	40	50	60	70	80	90	100
Water μ_w(Pa.sec)	0.8 $\times 10^{-3}$	0.65 $\times 10^{-3}$	0.55 $\times 10^{-3}$	0.47 $\times 10^{-3}$	0.40 $\times 10^{-3}$	0.35 $\times 10^{-3}$	0.31 $\times 10^{-3}$	0.28 $\times 10^{-3}$
5% w/w Caustic μ_c(Pa.sec)	1.03 $\times 10^{-3}$	0.83 $\times 10^{-3}$	0.69 $\times 10^{-3}$	0.58 $\times 10^{-3}$	0.50 $\times 10^{-3}$	0.43 $\times 10^{-3}$	0.38 $\times 10^{-3}$	0.33 $\times 10^{-3}$

:

Solution –

Caustic **inlet**, 80°C, 2000 kg/hr (0.5556 kg/sec)	-→ *Tc,avg=60°*C *(Following properties are at 60°*C*)* $Cp,c = 3983.2\text{kJ/Kg.K}$ $\mu_c = 5.8\times10^{-4}\,\text{Pa.sec}$ $\rho_c = 1055\text{kg/m}^3$ $k_c = 0.688\text{w/m.K}$ $R_{d,c} = 0.00035\text{m}^2\text{.K/W}$ $P_{r,c} = Cp,c/(\mu_c.k_c) = 3.3579$	Caustic **exit**, 40°C
CW **exit**, 33°C	←- *Tw,avg=39°C* *(Following properties are at 39°C)* $Cp,w = 4185kJ/Kg.K$ $C_{p,w} = 4185\text{kJ/Kg.K}$ $\mu_w = 6.65\times10^{-4}\,\text{Pa.sec}$ $\rho_w = 1000\text{kg/m}^3$ $k_w = 0.6541\text{W/m.K}$ $R_{d,\text{w}} = 0.00018m^2\text{.K/W}$ $P_{r,\text{w}} = C_{p,w}/(\mu_w.k_w) = 4.254$	CW **inlet**. 33°C
$\Delta_1 = 80-45 = 35^O C$	LMTD=17.3974°C	$\Delta_2 = 40-33 = 7^O C$

$Q = m_c.C_{p,c}.(T_{c,in} - T_{c,\text{out}})$ =8.8515e4 W

$m_c = 0.5555; \text{m}_w = Q/(C_{p,\text{w}}(T_{w,out} - T_{w,\text{in}}))$ =1.7630 kg/sec

LMTD=17.3974°C.

We choose 40 sch., 1.5”x 2“ ND double pipe heat exchanger made from steel pipes. Maximum pressure in the system being 6 kg/cm2(g) (~75 psig), we choose all fittings with a50 lbs rating. For this size of piping,

D_i =34.98 mm= 0.03498 m; $D_{i,o}$ =52.48 mm= 0.05248 m, D_o =42.1 mm= 0.0421 m,

$A_i = \pi D_i^2 / 4$ =9.6101e-4 m^2; $A_O = \pi(D_{i,o}^2 - D_o^2)/4$ =7.7106e-4 m^2;

Since $A_i > A_o$, the higher flow of water is considered for the inner pipe

$V_i = m_w / (\rho_w . A_i) = 1.8375\,\text{m/sec}$; $V_o = m_c / (\rho_c . A_o) = 0.6829\,\text{m/sec}$

$G_i = m_w / (A_i) = 1834.5\,\text{kg/(m}^2\text{.sec)}$; $G_o = m_o / (A_o) = 720.5116\,\text{kg/(m}^2\text{.sec)}$

$P_{r,t} = P_{r,w} = 4.254$; $P_{r,o} = P_{r,c} = 3.3579$

$\text{Re}_t = D_i . V_i . \rho_w / \mu_w = 96497$; $\text{Re}_o = (D_{i,o} - D_o) . V_o . \rho_c / \mu_c = 12895$

Assuming ϕ_t =1 and ϕ_o =1, $Nu_t = 0.027.(\text{Re}_t)^{0.8}.(\text{Pr}_t)^{0.33}.\phi_t = 423.1341$;

$Nu_o = 0.027.(\text{Re}_o)^{0.8}.(\text{Pr}_o)^{0.33}.\phi_o = 78.2158$.

$h_i = Nu_t.(k_w / D_i) = 7911.7$ W/(m^2.K)

$h_o = Nu_o.(k_c / (D_o - D_i)) = 5184.2$ W/(m^2.K)

$\frac{h_o D_{io}}{h_i D_o} = 0.7886 > 0.75$, and hence no need of finned tubes.

$$T_w = \frac{h_i T_{w,avg} + h_o \left(D_o / D_i \right) T_{c,avg}}{h_i + h_o \left(D_o / D_i \right)} = 48.2592\,^o\text{C}$$

Estimating ϕ_t and ϕ_o -

$$\phi_{i,new} = \left(\frac{\mu_w @ 39^o C}{\mu_w @ 48.2592^o C} \right)^{0.14} = \left(\frac{6.65 \times 10^{-4}}{5.6741 \times 10^{-4}} \right)^{0.14} = 1.0225$$

$$\phi_{o,new} = \left(\frac{\mu_c @ 40^o C}{\mu_w @ 48.2592^o C} \right)^{0.14} = \left(\frac{5.8 \times 10^{-4}}{7.1437 \times 10^{-4}} \right)^{0.14} = 0.9712$$

Based $\phi_{t,new}$ and $\phi_{o,new}$, $h_i = 5035.2$ and $\frac{h_o D_{io}}{h_i D_o} = 0.75$. This is a marginal case when one need not go for finned tubes and the same is opted for ease of fabrication.

$$T_{w,new} = \frac{h_i T_{w,avg} + h_o \left(D_o / D_i \right) T_{c,avg}}{h_i + h_o \left(D_o / D_i \right)} = 47.9941\,^o\text{C}$$

Recalculated $\phi_{t,new}$ and $\phi_{o,new}$ does not show significant change and the iteration is stopped.

$U_c = 1/(1/h_o + D_{io}/(h_i.D_i) + D_{io}.\ln(D_{io}/D_i)/(2.k_{wall})) = 1700.3$

$U_D = 1/\{1/U_c + R_c + R_w.(D_{io}/D_i)\} = 865.98$

Total heat transfer area based on outer surface of inner tube, $A = Q/(U_D.LMTD) = 5.8752$ m^2.

Minimum length of tube = $A/(\pi Dio) = 35.63$ m.

We adopt standard tube length of 6 m and provide 3 hairpins that make the total tube length to be 36m.

N_{HP} =3, $L_{total} = 36$m

Pressure drop

Pressure drop in pipe straight length

$\mathrm{Re}_t = 96497$, is turbulent flow and Eqn. 3.16a is applicable.

$$f_i = 0.0035 + 0.246 / (Re_t)^{0.42} = 0.0055$$

$$\Delta H_{f,i} = \frac{4 f_i G_i^2 L_{total}}{2g\rho_i^2 D_i} = 3.8759\text{m}$$

$\mathrm{Re}_o = 12895$, is turbulent flow and Eqn. 3.16a is applicable.

$$f_o = 0.0035 + 0.246 / (Re_o)^{0.42} = 0.0081$$

$$\Delta H_{f,o} = \frac{4 f_o G_o^2 L_{total}}{2g\rho_o^2 (Do - D_{io})} = 1.59\text{m}$$

Pressure drop in bends

Annulus: $\Delta H_{f,o,bend} = \dfrac{(2N_{HP} - 1)V_o^2}{2g} = 0.1189\text{m}$

Inner fluid: Neglected

Pressure drop in nozzles

The nozzle and the pipe sizes are chosen to be same and hence the flows are turbulent in the nozzles.

$$\Delta H_{i,n} = \frac{2(N_{HP})V_{i,n}^2}{2g} = 1.03\text{m}$$

$$\Delta H_{o,n} = \frac{2(N_{HP})V_{o,n}^2}{2g} = 0.1426\text{m}$$

Total pressure drop

Inner fluid (Cooling water) = $(1.03 + 3.8759) \times 1000 \times 9.81 = 48126.9$ Pa = 0.48 kg/cm^2

Annular fluid (Caustic) = $(0.1426 + 0.1189 + 1.59) \times 1055 \times 9.81 = 19162$ Pa = 0.20 kg/cm^2

Pressure drop for both fluids being sufficiently within limit, these are not corrected for the tube wall temperature using Eq. 3.17.

- Total pressure drop limits are met for both fluids.

The summary sheet in Table 3.1 can be filled with the data arrived at above. The exchanger can be fabricated from bought out components with 150 lbs pressure rating.

Further Reads

Kern, Donald Quentin., *Process heat transfer*. Tata McGraw-Hill Education, 1950.
Kern, Donald Quentin, and Allan D. Kraus., "Extended surface heat transfer." (1972).
Serth, Robert, W., *Process heat transfer- Principles and Applications,* Elsevier, 2007.