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Mechanical Design of Shell and Tube Type Heat Exchanger as per ASME Section VIII Div.1 and TEMA Codes for Two Tubes

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***Abstract*— In this paper we are designing two tube shell and tube type heat exchanger as per ASME Section VIII Div. 1, TEMA codes and IS 4503:1967. Main aim is to check the mechanical stability of the device by calculating parameters such as thickness of the equipment. ASME Section VIII Div. 1 is especially for designing of pressure vessels and boilers. Since heat exchanger is also a type of pressure vessel we used the codes.**

***Index Terms*— mechanical design of shell and tube heat exchanger;, ASME Section VIII Div. 1; TEMA Codes; (key words).**

I. INTRODUCTION

A. *Introduction to shell and tube heat exchanger*

In a shell and tube heat exchanger two fluids circulate in a different temperature conditions exchange heat through the walls of the tubes without direct contact between the fluids.

The fluid flowing inside the heat transfer tubes that belongs to the tube bundle they finds the tube side of a shell and tube heat exchanger. On the contrary the fluid flowing inside the shell of the exchanger defines shell side of a shell and tube heat exchanger.

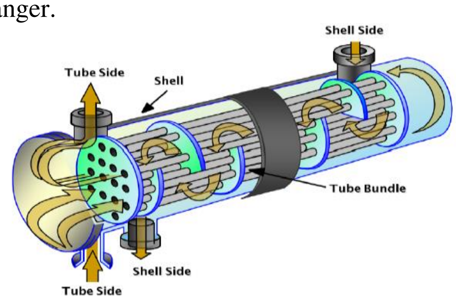

Fig. 1 Shell and Tube Heat Exchanger

Mechanical design of heat exchanger consists of designing various pressure and non-pressure components. The structural rigidity and adequate service of heat exchanger depends on the proper mechanical design. Mechanical design is commonly performed according to the design standards and codes. Following are the some mechanical design standards and pressure design codes used in heat exchanger design are:

Mechanical Design

1. ASME Section VIII Div. 1
2. TEMA Codes
3. HEI Standards
4. API

Pressure Design

1. ASME Section VIII Div. 1

ASME Section VIII Div. 1 and TEMA Codes are the most widely used standards for the mechanical design of shell and tube type Heat Exchangers. Since a HX is also a pressure vessel each mechanical design codes relates with the pressure vessel codes.

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II. METHODOLOGY

A. *Introduction to Nomenclature*

Depending on the many different configurations available shell and tube heat exchangers are formed by different elements. The fig. 1 shows the main parts of floating tubesheet type shell and tube heat exchangers.

As per TEMA (Figure N-1.2) there are various types of configurations of Heat Exchangers based on different applications [1]. Here we used 'A' type front head stationary head and 'E' type single pass shell.

Our main aim is the analysis of different finned tubes like Integral fin tubes, High fin tubes, Corrugated Tubes, etc. To get it done we need to remove the tube bundle each time to replace new tube bundle. This can be achieved by 'S' and 'W' type rear end head type. Here we are using 'S' type rear end head.

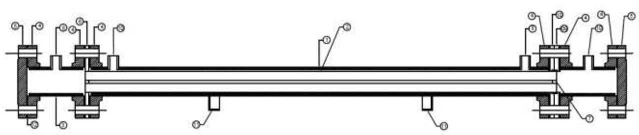

Fig. 2 Two Tube Shell and Tube Heat Exchanger

1. Shell
2. Fin Tube
3. Stationary Head
4. Lap Jointed Flange
5. Channel Cover
6. Stationary Tubesheet
7. Floating Tubesheet
8. Nozzle
9. Front Backing Device
10. Rear Backing Device
11. Saddle Support
12. Gasket

B. *Terminology*

a) Design Pressure:

Design pressure plays vital role to determine minimum thickness required for pressure parts. Generally design pressure is 5% greater than the maximum allowable working pressure [2]. In our case we are given with an design pressure of 40 kg/cm^2 for shell side and 10 kg/cm^2 for tube side.

b) Design Temperature:

The temperature is also a criterion for determining minimum thickness required for pressure parts. Usually we take 10°C higher than the maximum temperature of any component in the heat exchanger [2]. In our case maximum fluid temperature will be near to 85°C not beyond this and therefore we are taking 95°C as design temperature which is 10°C greater than maximum working temperature.

c) Maximum Allowable Stress Value:

The maximum allowable stress values for different materials can be determined by referring Subpart 1 of ASME Section II, Part D.

Table 1A
Section I; Section III, Classes 2 and 3;* Section VIII, Division 1; and Section XII
Maximum Allowable Stress Values S for Ferrous Materials
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Maximum Allowable Stress, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding													
	−20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
1	11.4	11.4	11.4	...	11.4	11.4	10.9	10.2	9.9	...	...	...	...	...
2	11.4	11.4	11.4	...	11.4	11.4	10.9	10.2	9.9	...	...	...	...	...
3	12.9	12.9	12.9	...	12.9	12.8	12.2	11.5	11.1	10.7	10.4	9.2	7.9	5.9
4	12.9	...	12.9	...	12.9	...	...	...	...	...	...	...	...	...
5	12.9	12.9	12.9	...	12.9	12.9	12.9	12.3	11.9	...	...	...	...	...
6	12.9	12.9	12.9	...	12.9	12.9	12.9	12.3	11.9	11.5	10.7	9.2	7.9	5.9
7	12.9	...	12.9	...	12.9	12.9	12.9	12.3	11.9	11.5	...	...	...	...
8	12.9	12.9	12.9	...	12.9	12.9	12.9	12.8	12.4	11.9	10.7	9.2	7.9	5.9
9	13.4	...	13.4	...	13.4	13.4	13.4	13.3	12.8	12.4	10.7	9.2	7.9	5.9
10	11.4	11.4	11.4	...	11.4	11.4	11.4	11.3	10.9	10.5	9.1	7.8	6.7	5.0
11	13.4	13.4	13.4	...	13.4	13.4	13.4	13.3	12.8	12.4	10.7	9.2	7.9	5.9
12	13.4	13.4	13.4	...	13.4	13.4	13.4	13.3	12.8	12.4	10.7	9.2	7.9	5.9
13	11.4	11.4	11.4	...	11.4	11.4	11.4	11.3	10.9	10.5	9.1	7.8	6.7	5.0
14	13.4	13.4	13.4	...	13.4	13.4	13.4	13.3	12.8	12.4	10.7	9.2	7.9	5.9
15	11.4	11.4	11.4	...	11.4	11.4	11.4	11.3	10.9	10.5	9.1	7.8	6.7	5.0
16	11.7	...	11.7	...	11.7	11.7	11.7	11.7	11.7	10.6	9.1	7.7	6.1	4.3
17	13.7	...	13.7	...	13.7	13.7	13.7	13.7	13.7	12.5	10.7	9.0	7.1	5.0
18	11.7	11.7	11.7	...	11.7	11.7	11.7	11.7	11.7	10.6	9.1	7.9	6.7	5.5
19	8.2	...	8.2	...	8.2	8.2	8.2	8.2	8.2	7.5	6.4	...	...	...
20	13.7	...	13.7	...	13.7	13.7	13.7	13.7	13.7	12.5	10.7	9.0	7.1	5.0
21	13.7	13.7	13.7	...	13.7	13.7	13.7	13.7	13.7	12.5	10.7	9.3	7.9	6.5
22	13.7	13.7	13.7	...	13.7	13.7	13.7	13.7	13.7	12.5	10.7	9.3	7.9	6.5
23	11.7	11.7	11.7	...	11.7	11.7	11.7	11.7	11.7	10.6	9.1	7.9	6.7	5.5
24	13.7	...	13.7	...	13.7	13.7	13.7	13.7	13.7	12.5	10.7	9.0	7.1	5.0
25	13.7	...	13.7	...	13.7	...	...	...	...	...	...	...	...	...
26	11.7	11.7	11.7	...	11.7	11.7	11.7	11.7	11.7	10.6	9.1	7.9	6.7	...

Fig. 3 Table for Determining Maximum Allowable Stress Value from ASME Section II Part-D [4]

As shown in the above table, Line numbers shows the different materials and maximum allowable stress values are given for the respective design temperature.

d) Material Selection:

As per [2] we can select suitable material as per the fluid temperature flowing inside exchanger. Hot fluid is flowing maximum at maximum temperature of 95°C.

TABLE 1 FLUID TEMPERATURE LIMITATIONS FOR PRESSURE PARTS

Material	Maximum Permissible Temperature, °C
Carbon steel	540
C-Mo steel	590
Cr-Mo steel	650
Low alloy steel (less than 6 percent chromium)	590
Alloy steel (less than 17 percent chromium)	590
Austenitic Cr-Ni steel	650
Cast iron	200
Brass	200

Fig. 4 Fluid Temp. Limitations for Pressure Parts [2]

As shown in the table we can use all of the materials for construction but as per availability of materials in industry we used different grades of carbon steels for different pressure parts. Here we cannot use cast iron for any Heat Exchanger part which has a primary function to avoid failure or corrosion of the parts.

C. *Design of Components*

Following are the major mechanical design components of S & T type heat exchanger:

a) Tubesheet thickness
b) Shell thickness under internal pressure
c) Flanges/Flat Cover
d) Nozzle
e) Gasket

a) Tubesheet Thickness:

Tubesheets are generally flat circular plates drilled with circular holes where tubes are inserted.

The purpose of tubesheet is firstly it devides flow between the shell & tube side preventing direct contact between the fluids, secondly it constitutes the most important structural element withstanding the shell and tube side pressure and third it supports all tubes of the bundle.

Tubesheet is a key element in heat exchanger for that reason the design and calculation carried out taking into consideration all factors that can increase the thickness such as slots, corrosion, coating, etc. Here tubesheet is designed and calculated according to the TEMA and ASME codes. Tubesheet is designed for the most critical condition due to the complex load system acting within the heat exchanger.

Shear and bending stresses are present in all types of tubesheets.

The required thickness due to bending can be calculated by:

$$T = \frac{FG}{3}\sqrt{\frac{P}{\eta S}}$$

Where, F = 1 for floating tubesheet [1]

T = Required thickness

G = Gasket mean diameter = 52.5 mm = 2.067 inches

P = Effective design pressure = 10 kg/cm^2 = 569 psi

S = Allowable stress = 20600 psi [1]

η = Coefficient per pass

$$\eta = 1 - \frac{0.785}{\left[\frac{pitch}{tube\ OD}\right]^2}$$

…For square tube pattern

Pitch = 25.4 mm

Tube OD = 19.05 mm

η = 0.4976

Therefore,

T = 0.16233 inches = 4.1232 mm

The required thickness due to shearing can be calculated by:

$$T = \frac{0.31 \text{ x } D_L}{\left[1 - \left(\frac{d_o}{P_o}\right)\right]}\left[\frac{P}{S}\right]$$

D_L = 4A/C = Equivalent Diameter
C = Tube Arrangement Perimeter

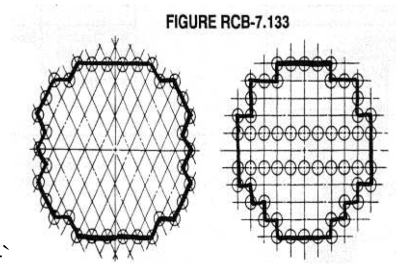

Fig. 5 Method to calculate ‘C’

A = Cross section, from tube center to tube center

P_o = Tube spacing, center to center

Therefore,

T = 0.0154 inches = 0.391 mm

Thickness calculated by shearing is less than thickness calculated by bending calculation. For the safer design of tubesheet we must consider the higher thickness. Therefore thickness will be 4.123 mm.

b) Shell Thickness under Internal Pressure:

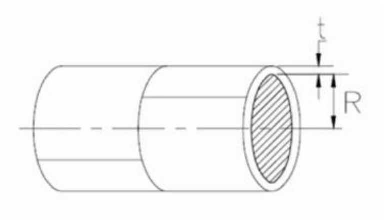

Fig. 6 Shell under Internal Pressure

Shell ID = 2 inches

Design Pressure = 40 kg/cm^2

Design Temperature = 95°C

Material = SA 312 TP 304

The minimum required thickness of shell under internal pressure can be calculated by following formulations under ASME Section VIII,

$$t = PR/(SE - 0.6P)$$

Where,

t – Minimum required thickness of shell

P – Internal design pressure

R – Inner radius of shell

E – Ligament Efficiency = 0.6289 [3]

S – Allowable stress value = 13700 psi [4]

Above formula is applicable if P < 0.385 SE
Here, above formula is applicable since it is meeting the criterion.

t = 0.0688 inches = 1.75 mm

From above calculations we come to know that the thickness of shell should not be less than 1.75 mm, it should be greater than 1.75 mm.

c) Flat Cover Design

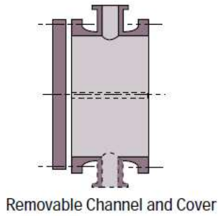

Fig. 7 Removable Channel and Cover

Flat cover may be known as channel cover which covers the stationary head. Thickness required can be calculated by referring ASME Section VIII UG-32:

$$T = G\left[\frac{CP}{S} + 1.9\frac{Wh_g}{SG^3}\right]^{1/2}$$

Where,

C = 0.25 (for bolted joint)

S = Flat cover material allowable stress at design temp. 95°C

W = Bolting Load

T = Flat cover Thickness

P = Design Pressure

G = Gasket mean diameter

h_g = Radial distance between the gasket mean dia. And the bolt circle dia. = 2 mm

We know all the parameters except bolting load (W) therefore to find bolting load we have,

$$W = 0.785G^2P + [2b \text{ x } 3.14GmP]$$

Table no. 2.5.1 and 2.5.2 given in ASME Section VIII Div. 1 helps to determine the values of above mentioned parameters like *b* and *m*.

Therefore,

W = 276.822 N and thickness will be,

T = 0.0092347 inches = 0.2345 mm.

According to above calculations thickness of flat cover must be greater than 0.2345 mm.

III. Results And Discussion

Actual fabrication details like thickness of shell and channel cover can be compared in this section.

As per the calculations done in above sections we got different values of thicknesses under designed pressure and designed temperature. These values are the minimum required values which means the values should be greater to have safer design and to prevent any failure of design therefore we keep some factor of safety to avoid these circumstances.

Table 1 Result Table

	Thickness Obtained by ASME Codes (mm)	Thickness Considered (mm)
Tubesheet	4.123	8
Shell	1.75	5
Flat Cover	0.2345	19

IV. CONCLUSION

Mechanical design includes checking the physical stability of heat exchanger to have a safe design without any failure.

Components like tubesheets, shell, channel cover, etc. which comes in contact with both shell side and tube side fluids, should be designed considering pressure acting on each side or the combination of pressure, whichever results in higher thickness of components.

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