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Case Study on Double Tube Sheet Failure in Heat Exchangers

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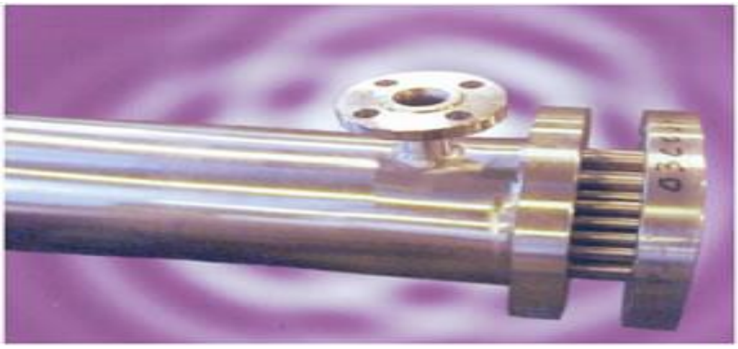
Abstract: **This case study was conducted at M/s Aker Solutions Pvt ltd as a part of Summer Training under the guidance of Shri Murali Mohan, Chief Engineer, Vessels. In the normal operations of a Double Tube Sheet, due to the presence of pressure and temperature differences of the working fluids, stresses tend to develop in the tube. This case study is about the failure of the tubes in between the tube sheets due to the above factors. Now TEMA (Tubular Exchanger Manufactures Association) also has suggested a design for the case, so we first use TEMA and find the respective stress values and then on the basis of the theoretical knowledge and with the help of reference books another or similar design is suggested and the values which were calculated using TEMA Standards are calculated and a comparison is drawn in between the two values.**

Keywords: **TEMA, Double Tube Sheet Failure in Heat Exchangers.**

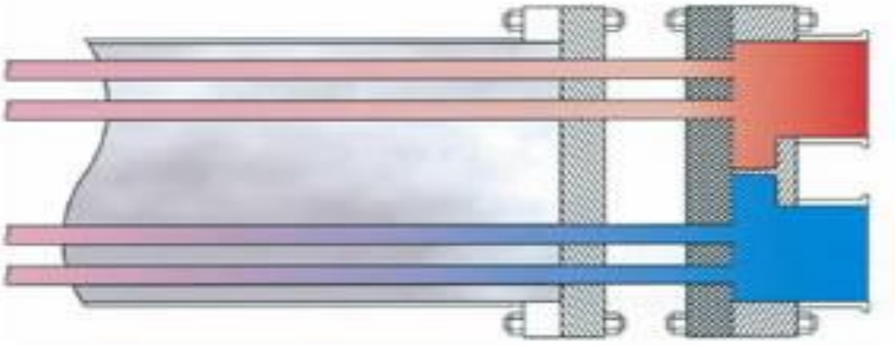
1. INTRODUCTION

About Double Tubesheets:

Double tubesheets are used for applications in heat exchangers where the mixing of the tube-side fluid and the shell-side fluid must be avoided. In the event of leaks occurring where the extremities of the tubes are expanded into the tubesheet, the tube-side fluid would leak between the two tubesheets instead of leaking into the shell.



The gap between the two tubesheets is sometimes open to the atmosphere so any leakage of either fluid should be visually and quickly detected.



2. A BRIEF DESCRIPTION OF THE CASE STUDY

In the normal operations of a Double Tube Sheet, due to the presence of pressure and temperature differences of the working fluids, stresses tend to develop in the tube. This case study is about the failure of the tubes in between the tube sheets due to the above factors. Now TEMA (Tubular Exchanger Manufactures Association) also has suggested a design for the case, so we first use TEMA and find the respective stress values and then on the basis of the theoretical knowledge and with the help of reference books another or similar design is suggested and the values which were calculated using TEMA Standards are calculated and a comparison is drawn in between the two values.

The nature and cause of Stresses being developed has been listed below:

Tube To Tube Sheet Stresses

» Differential pressure stress as determined by the difference in operating pressure between tube side and shell side fluids.

» Axial stresses which may be the result of tension or compression of the tubes. Differential thermal expansion between shell and tubes or between tube passes will develop the axial stress. Similarly tube side pressure will tend to pull the tubes out of tube shell.

» Differential diametric thermal expansion between the tubes and tube sheet.

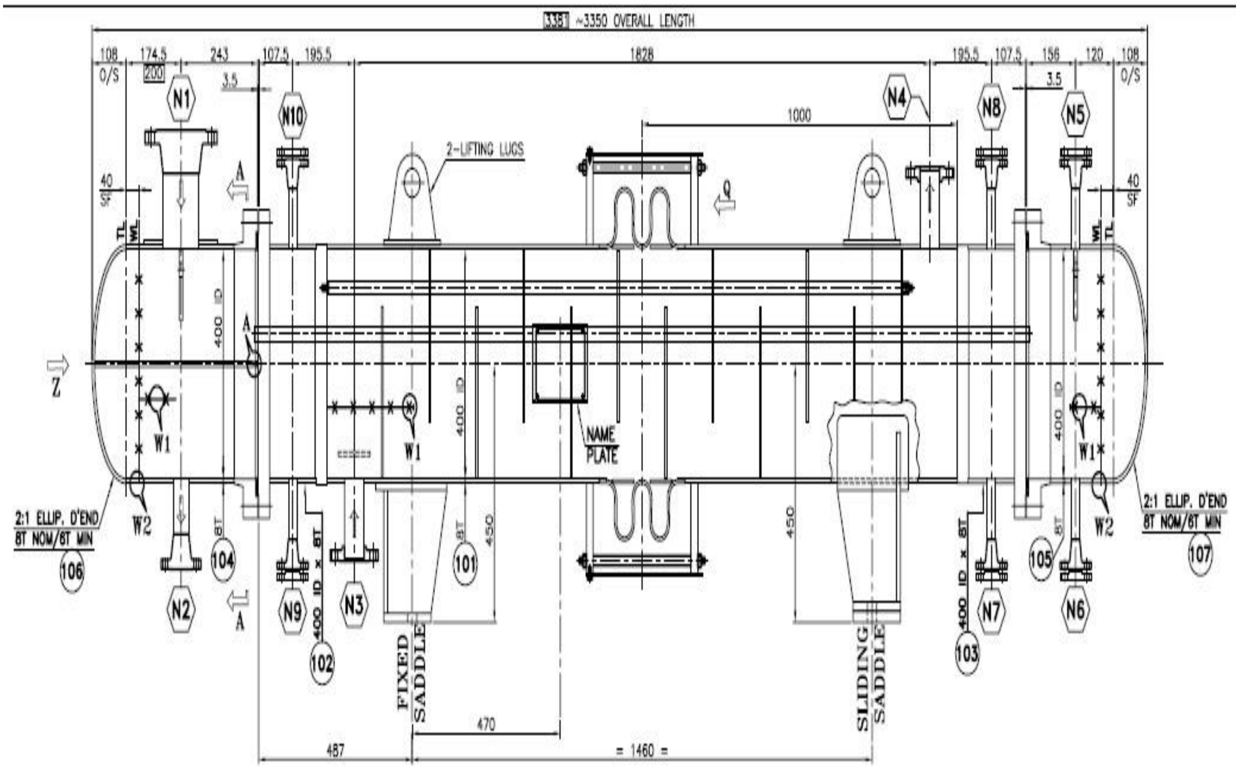
» Differential axial thermal expansion between the tube and the tube sheet with certain tube to tube sheet joint construction.

The procedure that is followed is that without considering the temperature effects, the pressure effects are studied and stress values calculated and then the total stresses are calculated by involving the temperature effects. Also while considering the effect of temperature, the shell or interconnecting element temperature is changed of each calculation and a range for which it can be used is generated. While controlling the temperature may not be very feasible or possible every time and hence the shell thickness is also changed to study its effect on the stresses and failure of tubes in between the double tube sheets.

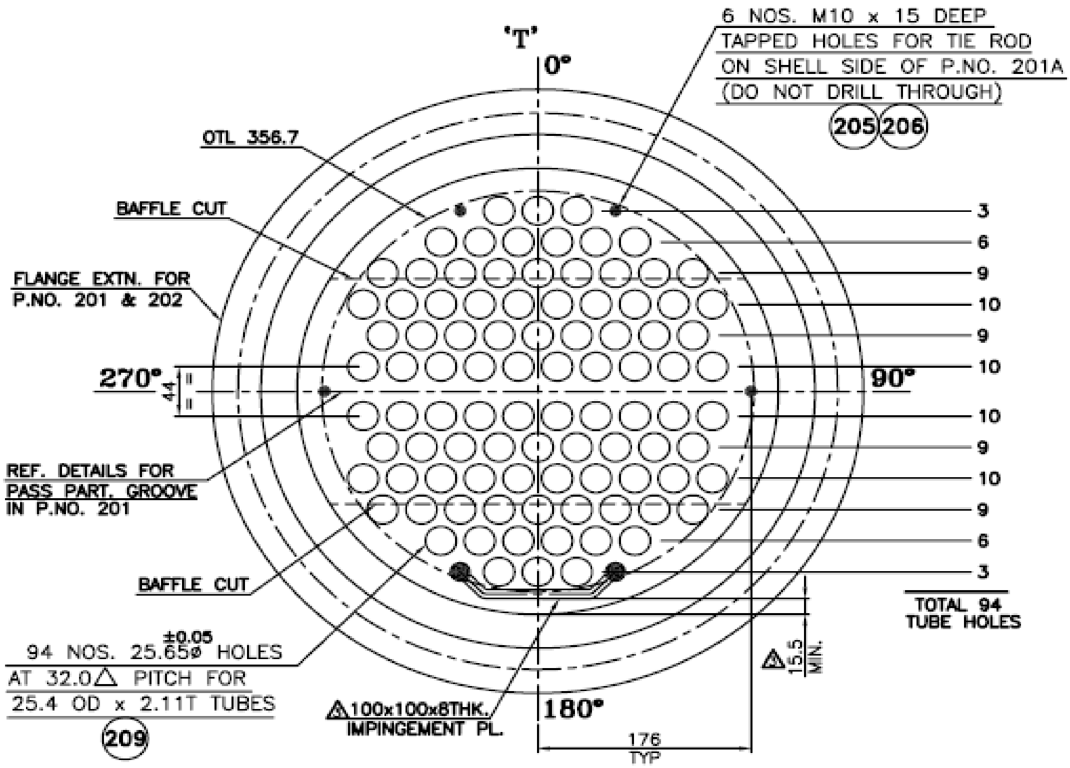
All calculations are also done taking in consideration the design temperatures as well as the pressures and it's indeed impossible to think about the design without considering the Upset Conditions.

3. THE DRAWINGS FOR THE CASE STUDY TAKEN UP

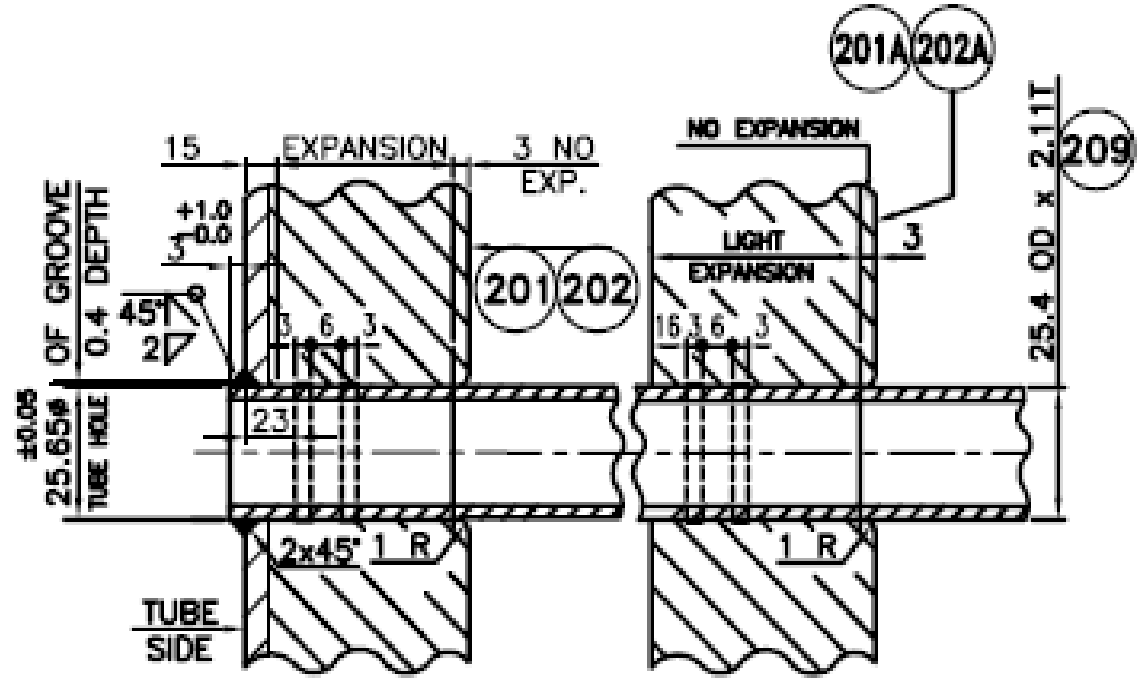
Design Data		
Design Code: ASME Sec VII, Div.1 ED-2004, TEMA CL-'B'/ED.1999		
Service: Non Lethal		
Particulars	Shell Side	Tube Side
Fluid Circulated	Cooling Water	CMS
Design Pressure kg/cm2 g	7	13
Design Temperature deg C	100	175
Working Pressure kg/cm2 g	4	7.4
Working Temperature deg C	34/39	148.3/45
Mean Metal Temperature (IN/OUT) deg C	36.3	112
Hydro test Pressure kg/cm2 g	9.1	16.9
Corrosion allowance mm	3	3
Stress Relieving		sr of dished end
Impact Testing Required	NO	NO
Joint efficiency	1.00/1.00	1.00/1.00
No of passes	One	Two
Type	BEM(H)	
Surface Area/ Shell (gross/effective) m2	15.0/14.8	
Duty/ unit	100.9	
Shells per unit	ONE	
Upset Condition for TUBE SHEET DESIGN		
Design Pressure kg/cm2 g	ATM.	7.4
Mean Metal Temperature deg C	40	148



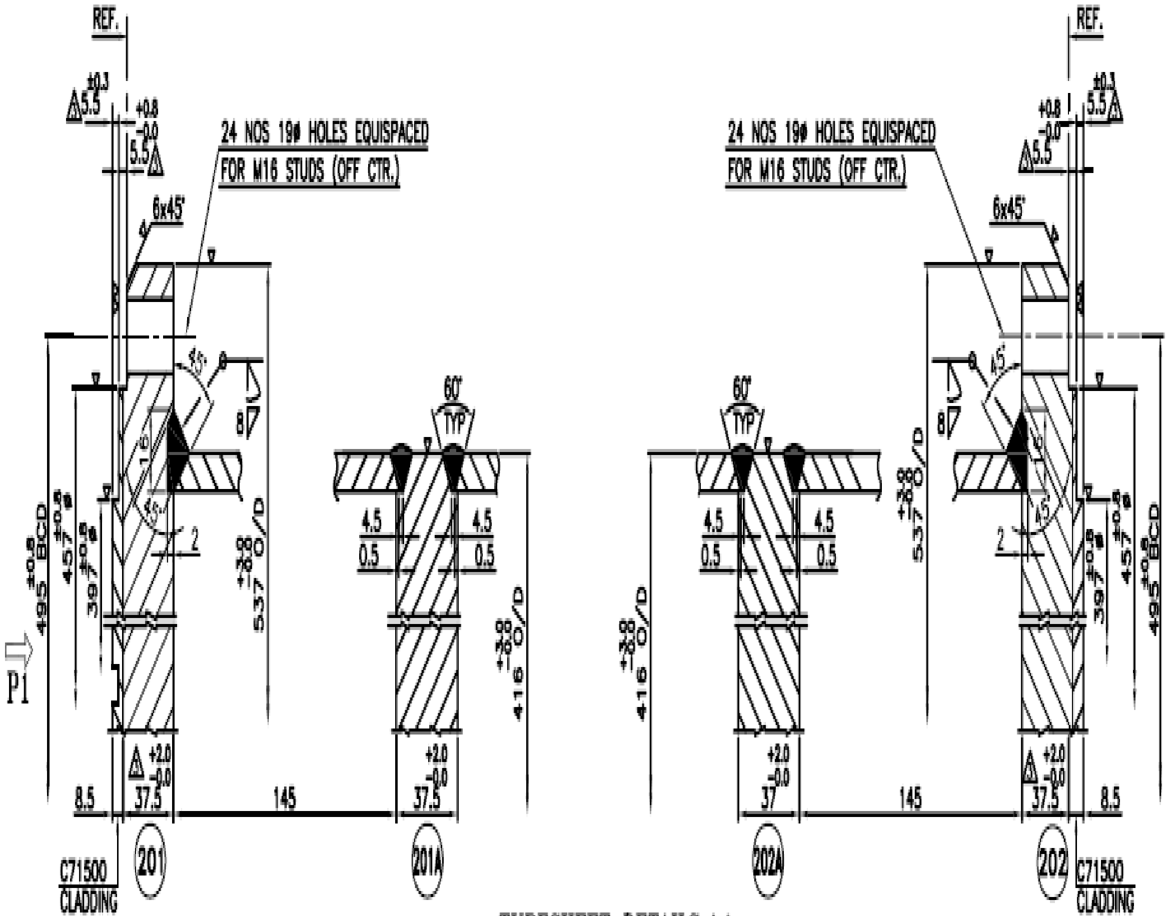
ELEVATION (SCALE 1 : 10)



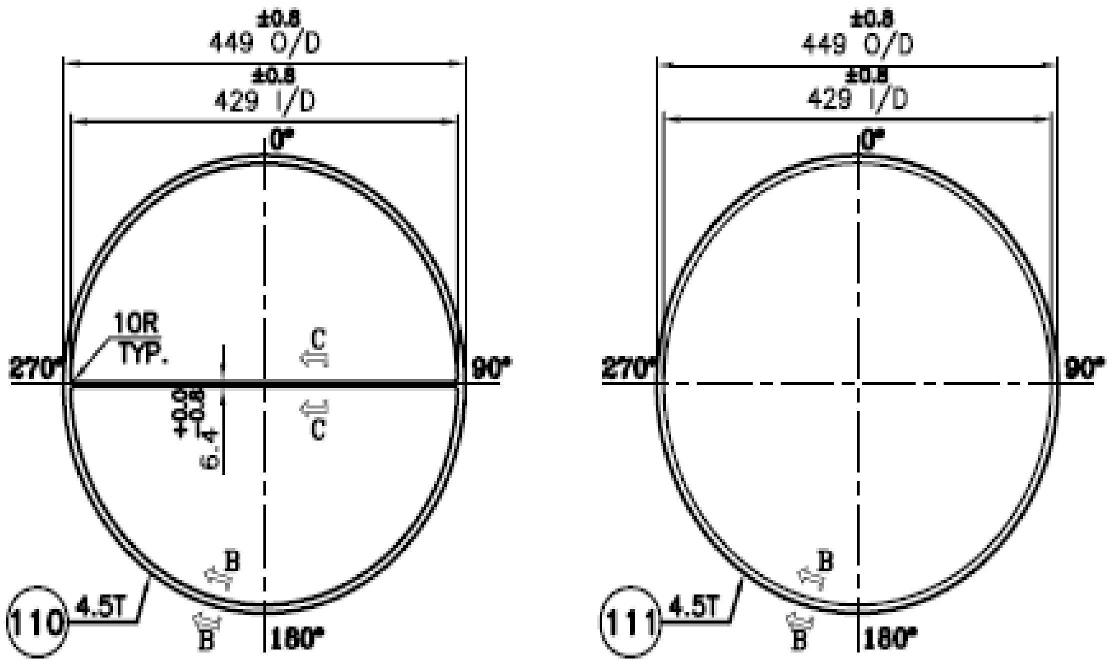
VIEW 'P1'
TUBESHEET DETAILS (201)(202)
(SCALE 1 : 5) (201A)(202A)



DETAIL OF TUBE HOLE &
& TUBE TO TUBE SHEET JOINT



TUBESHEET DETAILS



DETAIL OF GASKETS

TEMA Design for connected Double Tube Sheets:

Sections RCB-7.155-7.1551, 7.1552, 7.1553 and 7.1554 are used for the Code calculations below.

Tables for the Calculation of Modulus of elasticity and Co-efficient of Expansion as required by the Code as used namely Table D-10 Modulus of Elasticity and Table 11-D Mean co-efficient of expansion.

Above is given the design for the Connected Tube Sheets and our interest lies majorly in RCB 7.1554.

(A) Calculations Based on pressure:

The axial stress due to pressure (σp), psi (kPa) is defined as:

$$\sigma p = \frac{P\pi(G^2 - Ndo^2)}{4AT}$$

Where the symbols are as defined above.

Hence,

P = 13 kg/ cm2

= 1.275 * 10^3 kPa

N = 94

G = $\frac{Gasket\ OD + Gasket\ ID}{2}$ = 439 mm

Do= 25.4 mm

σ_p= 9112.72 kPa

(B) Now also considering the temperature effects we first try to find the minimum operating temperature for the shell by calculating the stress values at various temperatures to avoid failure of tubes:

The stress due to axial thermal expansion of tubes (σTT), psi (kPa) is determined by (σTT=FTE/AT) where $FTE = \frac{(\alpha T\Delta TT - \alpha e\Delta Te)(ETAT(EeAe)}{(ETAT)+(EeAe)}$

Where the symbols are as defined above.

Since the geometry does not change

At = 14504.72 mm2

Ae = 10248.96 mm2

Temp of Element (Te)	**Temp of tubes(avg) (Tt)**	**ΔTe**	**ΔTt**	**Co-efficient of expansion of element**	**Co-efficient of expansion of tube**	**Young's Modulus of element**	**Young's Modulus of tube**
93.3	96.3	72.3	75.3	12.01*10^(- 6)	15.3*10^-(- 6)	196.5*10^6	148.2*10^6
45	96.3	24	75.3	11.74*10^(- 6)	15.3*10^-(- 6)	199.45*10^6	148.2*10^6
44	96.3	23	75.3	11.73*10^(- 6)	15.3*10^-(- 6)	199.52*10^6	148.2*10^6
42	96.3	21	75.3	11.72*10^(- 6)	15.3*10^-(- 6)	199.64*10^6	148.2*10^6
41	96.3	20	75.3	11.71*10^(- 6)	15.3*10^-(- 6)	199.70*10^6	148.2*10^6
40	96.3	19	75.3	11.71*10^(- 6)	15.3*10^-(- 6)	199.76*10^6	148.2*10^6
39	96.3	18	75.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.82*10^6	148.2*10^6
37.8	96.3	16.8	75.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.9*10^6	148.2*10^6
37	96.3	16	75.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.95*10^6	148.2*10^6
35	96.3	14	75.3	11.68*10^(- 6)	15.3*10^-(- 6)	200.00*10^6	148.2*10^6
30	96.3	9	75.3	11.65*10^(- 6)	15.3*10^-(- 6)	200.38*10^6	148.2*10^6
29	96.3	8	75.3	11.65*10^(- 6)	15.3*10^-(- 6)	200.38*10^6	148.2*10^6
21	96.3	0	75.3	11.53*10^(- 6)	15.3*10^(- 6)	201.3*10^6	148.2*10^6

Temp of Element (Te)	Temp of tubes(avg) (Tt)	Thermal Axial Stress in tube	Axial Pressure Stress	Total Stress	Allowable Stress
93.3	96.3	20.34*10^3	9112.72	29.45*10^3	72.39*10^3
45	96.3	62.86*10^3	9112.72	71.97*10^3	72.39*10^3
44	96.3	63.75*10^3	9112.72	72.85*10^3	72.39*10^3
42	96.3	65.46*10^3	9112.72	74.57*10^3	72.39*10^3
41	96.3	66.33*10^3	9112.72	75.44*10^3	72.39*10^3
40	96.3	67.19*10^3	9112.72	76.30 *10^3	72.39*10^3
39	96.3	68.00*10^3	9112.72	77.11 *10^3	72.39*10^3
37.8	96.3	69.00*10^3	9112.72	78.11*10^3	72.39*10^3
37	96.3	70.18*10^3	9112.72	79.29*10^3	72.39*10^3
35	96.3	71.50*10^3	9112.72	80.61 *10^3	72.39*10^3
30	96.3	75.82*10^3	9112.72	84.93 *10^3	72.39*10^3
29	96.3	76.67*10^3	9112.72	85.78*10^3	72.39*10^3
21	96.3	83.61*10^3	9112.72	92.32*10^3	72.39*10^3

Here we see that for the tube temperature fixed, the operating temperature of shell is 45˚C

(C) Changing the thickness and maintaining the operating conditions at the given values as per design:

The stress due to axial thermal expansion of tubes (σTT=FTE/AT) where $FTE = \frac{(\alpha T \Delta TT - \alpha e \Delta Te)(ETAT(EeAe)}{(ETAT)+(EeAe)}$

Here the geometry of the whole thing changes, only the operating conditions remain same hence

Te(°C)	**Δ**Te(°C)	**α**e(mm/mm/°C)	**E**e(kPa)	**T**t(°C)	Δ**T**t(°C)	**α**t(mm/mm/°C)	**E**t(kPa)
21	0	11.53*10^(- 6)	201.3*10^6	96.3	75.3	15.30*10^(- 6)	148.2*10^6

At(mm2)	**A**e(mm2)	**h**e(mm)
14504.72	10248.96	8
14504.72	8945.86	7
14504.72	7649.04	6
14504.72	7390.43	5.8
14504.72	7364.58	5.78
14504.72	7261.22	5.7
14504.72	7132.07	5.6
14504.72	7002.99	5.5
14504.72	6358.50	5
14504.72	6229.79	4.9
14504.72	6101.14	4.8

Thermal Stress In Tubes	Stress Due to pressure	Total Stress	Allowable Stress	Thickness Of Shell	Thermal stress in shell	Allowable stress
83.61*10^3	9112.72	92.72*10^3	72.39*10^3	8	118.33*10^3	137.89*10^3
77.83*10^3	9112.72	86.94*10^3	72.39*10^3	7	126.20*10^3	137.89*10^3
71.26*10^3	9112.72	80.37*10^3	72.39*10^3	6	135.12*10^3	137.89*10^3
70.90*10^3	9112.72	80.01*10^3	72.39*10^3	5.8	137.69*10^3	137.89*10^3
70.44*10^3	9112.72	79.55*10^3	72.39*10^3	5.5	140.71*10^3	137.89*10^3
63.62*10^3	9112.72	72.73*10^3	72.39*10^3	5	145.12*10^3	137.89*10^3
62.91*10^3	9112.72	72.02*10^3	72.39*10^3	4.9	146.47*10^3	137.89*10^3
62.08*10^3	9112.72	71.19*10^3	72.39*10^3	4.8	147.58*10^3	137.89*10^3

Here we see that changing the thickness of shell is not a very good option and also practically it may find difficulties in application as in being compatible with Shell design for Internal Pressure, for example.

(D) The Upset Condition:

The Upset condition is basically a case where, the flow of the cold fluid does not happen due to some failure. Hence the hot fluid flows completely from the inlet of tubeside to its outlet and consequently the temperature being extreme the thermal stresses developed are much higher too. For the upset condition too we try to find a minimum operating temperature for the shell to prevent failure of tubes by the differential expansion.

The formulae given above are applicable here too.

Temp of Element (Te)	Temp of tubes(Tt)	Co-efficient of expansion of element	Co-efficient of expansion of tube	Young's Modulus of element	Young's Modulus of tube
120	148.3	12.18*10^(- 6)	15.66*10^(- 6)	194.9*10^6	145.5*10^6
115	148.3	12.15*10^(- 6)	15.66*10^(- 6)	195.1*10^6	145.5*10^6
113	148.3	12.14*10^(- 6)	15.66*10^(- 6)	195.3*10^6	145.5*10^6
110	148.3	12.11*10^(- 6)	15.66*10^(- 6)	195.5*10^6	145.5*10^6
105	148.3	12.09*10^(- 6)	15.66*10^(- 6)	195.8*10^6	145.5*10^6
100	148.3	12.05*10^(- 6)	15.66*10^(- 6)	196.0*10^6	145.5*10^6
98	148.3	12.04*10^(- 6)	15.66*10^(- 6)	196.2*10^6	145.5*10^6
97	148.3	12.03*10^(- 6)	15.66*10^(- 6)	196.3*10^6	145.5*10^6
93.3	148.3	12.01*10^(- 6)	15.66*10^(- 6)	196.5*10^6	145.5*10^6
37.8	148.3	11.70*10^(- 6)	15.66*10^(- 6)	199.9*10^6	145.5*10^6
21	148.3	11.53*10^(- 6)	15.66*10^(- 6)	201.3*10^6	145.5*10^6

Temp of Element (Te)	Temp of tubes(Tt)	Thermal Axial Stress in tube	Axial Stress-Pressure	Total Stress	Allowable Stress
120	148.3	55.72*10^3	9112.72	64.83*10^3	72.39*10^3
115	148.3	60.29*10^3	9112.72	69.39*10^3	72.39*10^3
113	148.3	62.08*10^3	9112.72	71.12*10^3	72.39*10^3
110	148.3	64.89*10^3	9112.72	74.00*10^3	72.39*10^3
105	148.3	69.36*10^3	9112.72	78.42*10^3	72.39*10^3
100	148.3	73.69*10^3	9112.72	76.59*10^3	72.39*10^3
98	148.3	75.46*10^3	9112.72	78.36*10^3	72.39*10^3
97	148.3	76.39*10^3	9112.72	79.29*10^3	72.39*10^3
93.3	148.3	79.69*10^3	9112.72	82.59*10^3	72.39*10^3
37.8	148.3	128.79*10^3	9112.72	131.69*10^3	72.39*10^3
21	148.3	143.38*10^3	9112.72	146.29*10^3	72.39*10^3

Here we see the shell temperature to prevent failure of tubes reaches as high as 113˚C.

4. THEORETICAL DERIVATIONS FOR DESIGNING THE TUBE AGAINST FAILURE

We see when we derive equations similar to the ones given by TEMA; the first step involves finding out the governing case out of the seven cases for the pressure conditions that can happen wrt to the design

(A)There is hot the fluid only in the tube side; hence here pressure is tube side pressure only on the tube side.

(B)There is the cold fluid only in the shell side and so pressure is only the shell side pressure on the shell side.

(C)This case is the normal operating case with fluids on both sides and both sides pressure exist.

(D)Taking case (A) and assuming that the hot fluid leaks into the intermediate space between the tube sheets, here tube side pressure will exist on tube side and in the intermediate space.

(E) Taking case (B) and assuming that the cold fluid leaks into the intermediate space between the tube sheets, here shell side pressure will exist on shell side and in the intermediate space.

(F) Taking case (C) and assuming that the hot fluid leaks into the intermediate space between the tube sheets, here tube side pressure will exist on tube side, so also shell side pressure on shell side tube side pressure and in the intermediate space.

(G)Taking case (C) and assuming that the cold fluid leaks into the intermediate space between the tube sheets, here tube side pressure will exist on tube side and shell side pressure will exist in shell side and in the intermediate space.

Once the governing case has been found this can be used as the equation for pressure conditions and the procedure adopted in the case of design using TEMA is followed here for a comparison.

* The pressure on the intermediate space is given by

$$P = \frac{P1}{(1+\frac{V2}{V1})}$$

Where,

P1 is the higher of pressure of the tube side and shell side.

V1 is the volume of the side from which leak occurs

V2 is the volume of the side in which leak occurs

Hence assuming that the pressure attained in the intermediate space attains that of the leaked side gives a conservative solution and the pressure is approximately equal to the leaked pressure if V1 is very large as compared to V2.

Even though our case study involves in its geometry flanges and gaskets too however for a better understanding of the stresses involved we ignore their effect and draw the Axial Force Diagrams for the seven cases:

Now the table below gives us the values of the stresses calculated by using the above Axial Force Diagrams:

Shell Side	Tube Side	Space btwn TubeSheets	Leakage From	Stress in Tube
Ps(kg/cm2)	Pt(kg/cm2)	P(kg/cm2)	(kg/cm2)	(kg/cm2)
atm	13	Atm	None	17.38
7	atm	Atm	None	0
7	13	Atm	None	17.38
atm	**13**	**13**	**Tube Side**	**58.34**
7	atm	7	Shell Side	22.05
7	**13**	**13**	**Tube Side**	**58.34**
7	13	7	Shell Side	39.43
atm	7	Atm	None	9.36
13	atm	Atm	None	0
13	7	Atm	None	9.36
atm	7	7	Tube Side	31.4
13	atm	13	Shell Side	40.96
13	7	7	Tube Side	31.4
13	**7**	**13**	**Shell Side**	**50.32**

D = ID Shell = 400mm	Thickness of tubes = 2.11mm
d = OD Tube = 25.4mm	Thickness of shell = 8mm
A = At + As = 24753.68mm2	N = Number of Tubes = 94
At = C/S area of tubes = 14504.72mm2	
As = C/S area of shell = 10248.96mm2	

In the above table it is seen that we also interchange the shell side and tube side pressures to prevent the effect of one being greater than the other exclusively.

So our governing equation must be from Case (D) and (F)

$$\sigma = \frac{Pt(Di2 - N(do2 - di2)}{(Do2 - Di2) + N(do2 - di2)}$$

Where,

Pt is the tube side pressure which is higher in the given case study

N is the number of tubes

D0 is the O/D of the Shell Element

Di is the I/D of the Shell element

do is the O/D of tube

di is the I/D of tube

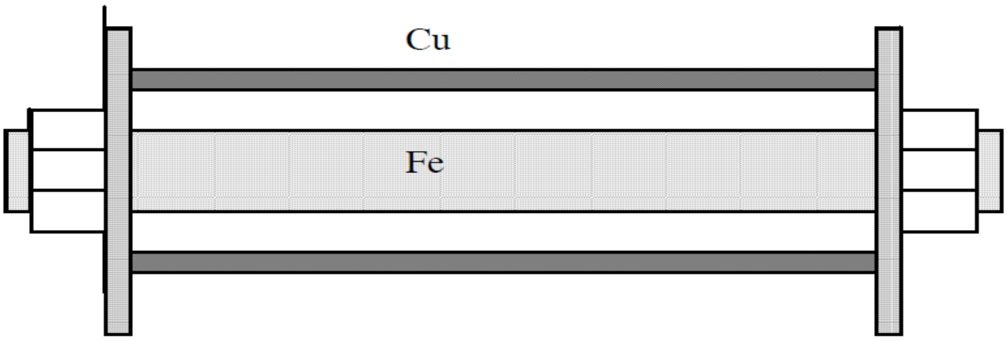
Now the observation for the geometry given to us has to involve the gaskets and the flange too.:

The result that we get here in terms of the equation is the same as that we have derived for the previous case, this is when we assume that the whole system is rigid which is a reasonable assumption since the flange and tube sheet are bolted together.

Also when an AFD is drawn for each case we get the same result and hence the governing case for the previous case is applicable here too so is the equation.

Next lets take the effect of the differential expansion on the stress developed in the tube with the following case:

Analysis – relevant to Heat Exchangers



assembled at room temperature

increase ΔT

Data: $\alpha_{cu} > \alpha_{Fe}$: copper expands more than steel, so will generate a TENSILE stress in the steel and a compressive stress in the copper.

Balance forces:

Tensile force in steel $|F_{Fe}| = |F_{cu}| = F$

Stress in steel $= F/A_{Fe} = \sigma_{Fe}$

" " copper $= F/A_{cu} = \sigma_{cu}$

Steel strain: $\varepsilon_{FE} = \alpha_{Fe}\,\Delta T + \sigma_{Fe}/E_{Fe}$ (no transvere forces)

$= \alpha_{Fe}\Delta T + F/E_{Fe}A_{Fe}$

copper strain $\varepsilon_{cu} = \alpha_{cu}\Delta T - F/E_{cu}A_{cu}$

Strains Equal: $F\left[\frac{1}{Afe\ Efe} + \frac{1}{Acu\ Ecu}\right] = (\alpha cu - \alpha fe)\Delta T$

Copper strain $\in cu = \alpha cu\ \Delta T - F/EcuAcu$

The tubes being welded to the tubes, this analysis gives us the force that is responsible for the stress and it's not very difficult to show that this is the same as that TEMA has used for its design.

Now that we are equipped with the equations for finding the stresses in the tube we shall follow the procedure followed as in TEMA design and design the case again.

(A) Calculations Based on pressure.

This case is just the case we used for deriving the equation and hence

σ = 5721.2 kPa from the table

(B) Now also considering the temperature effects we first try to find the minimum operating temperature for the shell by calculating the stress values at various temperatures to avoid failure of tubes:

Here,

At = 14504.72 mm2

Ae = 10248.96 mm2

Temp of Element (Te)	Temp of tubes(avg) (Tt)	Co-efficient of expansion of element	Co-efficient of expansion of tube	Young's Modulus of element	Young's Modulus of tube
93.3	96.3	12.01*10^(- 6)	15.3*10^-(- 6)	196.5*10^6	148.2*10^6
45	96.3	11.74*10^(- 6)	15.3*10^-(- 6)	199.45*10^6	148.2*10^6
44	96.3	11.73*10^(- 6)	15.3*10^-(- 6)	199.52*10^6	148.2*10^6
42	96.3	11.72*10^(- 6)	15.3*10^-(- 6)	199.64*10^6	148.2*10^6
41	96.3	11.71*10^(- 6)	15.3*10^-(- 6)	199.70*10^6	148.2*10^6
40	96.3	11.71*10^(- 6)	15.3*10^-(- 6)	199.76*10^6	148.2*10^6
39	96.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.82*10^6	148.2*10^6
37.8	96.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.9*10^6	148.2*10^6
37	96.3	11.70*10^(- 6)	15.3*10^-(- 6)	199.95*10^6	148.2*10^6
35	96.3	11.68*10^(- 6)	15.3*10^-(- 6)	200.00*10^6	148.2*10^6
30	96.3	11.65*10^(- 6)	15.3*10^-(- 6)	200.38*10^6	148.2*10^6
29	96.3	11.65*10^(- 6)	15.3*10^-(- 6)	200.38*10^6	148.2*10^6
21	96.3	11.53*10^(- 6)	15.3*10^(- 6)	201.3*10^6	148.2*10^6

Temp of Element (Te)	Temp of tubes(avg) (Tt)	Thermal Axial Stress in tube	Axial Pressure Stress	Total Stress	Allowable Stress
93.3	96.3	20.34*10^3	5721.2	26.06*10^3	72.39*10^3
45	96.3	62.86*10^3	5721.2	68.58*10^3	72.39*10^3
44	96.3	63.75*10^3	5721.2	69.47*10^3	72.39*10^3
42	96.3	65.46*10^3	5721.2	71.18*10^3	72.39*10^3
41	96.3	66.33*10^3	5721.2	72.05*10^3	72.39*10^3
40	96.3	67.19*10^3	5721.2	72.91*10^3	72.39*10^3
39	96.3	68.00*10^3	5721.2	73.72*10^3	72.39*10^3
37.8	96.3	69.00*10^3	5721.2	74.72*10^3	72.39*10^3
37	96.3	70.18*10^3	5721.2	75.90*10^3	72.39*10^3
35	96.3	71.50*10^3	5721.2	77.22*10^3	72.39*10^3
30	96.3	75.82*10^3	5721.2	81.54*10^3	72.39*10^3
29	96.3	76.67*10^3	5721.2	82.39*10^3	72.39*10^3
21	96.3	83.61*10^3	5721.2	88.93*10^3	72.39*10^3

Here the minimum operating temperature of shell element comes out to be lesser and equal to 40˚C.

(C) The Upset Condition:

Here again we take the case of the upset conditions and find out the temperature of shell.

Temp of Element (Te)	Temp of tubes(Tt)	Co-efficient of expansion of element	Co-efficient of expansion of tube	Young's Modulus of element	Young's Modulus of tube
120	148.3	12.18*10^(- 6)	15.66*10^(- 6)	194.9*10^6	145.5*10^6
115	148.3	12.15*10^(- 6)	15.66*10^(- 6)	195.1*10^6	145.5*10^6
113	148.3	12.14*10^(- 6)	15.66*10^(- 6)	195.3*10^6	145.5*10^6
110	148.3	12.11*10^(- 6)	15.66*10^(- 6)	195.5*10^6	145.5*10^6
105	148.3	12.09*10^(- 6)	15.66*10^(- 6)	195.8*10^6	145.5*10^6
100	148.3	12.05*10^(- 6)	15.66*10^(- 6)	196.0*10^6	145.5*10^6
98	148.3	12.04*10^(- 6)	15.66*10^(- 6)	196.2*10^6	145.5*10^6
97	148.3	12.03*10^(- 6)	15.66*10^(- 6)	196.3*10^6	145.5*10^6
93.3	148.3	12.01*10^(- 6)	15.66*10^(- 6)	196.5*10^6	145.5*10^6
37.8	148.3	11.70*10^(- 6)	15.66*10^(- 6)	199.9*10^6	145.5*10^6
21	148.3	11.53*10^(- 6)	15.66*10^(- 6)	201.3*10^6	145.5*10^6

Temp of Element (Te)	Temp of tubes (Tt)	Thermal Axial Stress in tube	Axial Stress-Pressure	Total Stress	Allowable Stress
120	148.3	55.72*10^3	5712.2	61.43*10^3	72.39*10^3
110	148.3	64.89*10^3	5712.2	70.60*10^3	72.39*10^3
105	148.3	69.36*10^3	5712.2	75.02*10^3	72.39*10^3
100	148.3	73.69*10^3	5712.2	73.19*10^3	72.39*10^3
98	148.3	75.46*10^3	5712.2	74.96*10^3	72.39*10^3
97	148.3	76.39*10^3	5712.2	82.10*10^3	72.39*10^3
93.3	148.3	79.69*10^3	5712.2	85.40*10^3	72.39*10^3
37.8	148.3	128.79*10^3	5712.2	134.50*10^3	72.39*10^3
21	148.3	143.38*10^3	5712.2	149.09*10^3	72.39*10^3

Here the tubes passes at 105˚C itself, which is lesser as expected than TEMA design.

(D) Since the tubes are surrounded by external pressure in the case of a leak we design and check for elastic failure due to buckling also.

This is done using the **ASME Code Section VIII, Division 1**

UG – 28 (c) defines the calculation of critical pressure

As per the code,

P = 13kg/cm2

do = 25.4 mm

t = 2.11 mm

L = 145 mm

L/do = 5.7

do/t = 12.04

Thus from ASME SECTION 2 Part D, Subpart 3, fig G

A = 0.008

From fig NFC-4

B = 421.8 kg/cm2

Hence Pa = 46.72 kg/cm2 < 13 kg/cm2

Also we can theoretically as per Jawad and Farr [2] find the critical pressure for buckling is given by:

$$P = KE\left(\frac{t}{Do}\right)^3$$

Where,

P = buckling pressure

E = modulus of elasticity

t = thickness

Do = outside diameter

K is found from the graph for valves of K v/s Values of L/r (ref. page 72, Jawad and Farr, Edition 2)

Thus P = 1546.8 kg/cm2

This is much greater than what the Code has to define, this is because from the graph we can see that our pressure calculation is for N = 6, while the code does it for N =2 and hence such a huge difference in pressure.

However the Tubes will not buckle in the normal and upset conditions both for the given design conditions.

5. CONCLUSIONS DRAWN FROM THE CASE STUDY

1. Double Tube Sheets indeed are a good way to prevent the mixing of the tube side and shell side fluids.
2. One major observation being that it's the thermal effect and the differential expansion which predominates on the stress developed in the tubes, over the similar effect of pressures.
3. It is observed that changing the thickness of shell is not a very good option and also practically it may find difficulties in application as in being compatible with Shell design for Internal Pressure, for example.
4. Also the minimum temperatures found out in the above study are quite high and may be difficult to obtain practically.
5. Failure due to buckling in the above case requires a very huge pressure which may never even be obtained and hence failure of tubes due to buckling is not expected.
6. TEMA Design is very conservative in nature and limited to its use in particular cases only.

REFERENCES

[1] Theory of pressure vessel design, J. Harvey.

[2] Design of pressure vessels, Jawad and Farr.

[3] ASME Sec VII, Div.1 ED-2004.

[4] TEMA CODES (8^{th} edition).