Fig. 1

Computer input form

(TEMA fixed-tube-sheet design)

Item No.: SAMPLE

Shell side

Shell design pressure	Metal temp. minus 70° F.	Elastic mod. x 10^{-6}	Coeff. of exp. x 10^6	Shell o.d.	Wall thickness (corroded)
250	362.5	27.3	6.9	18.0	0.219
		(26.8 for steel at 500° F.)	(7.0 for steel at 500° F.)		

Tube side

Tube design press.	Metal temp. minus 70° F.	Elastic modx. x 10^{-6}	Coeff. of exp. x 10^6	Tube o.d.	Tube wall	Tube length, in.
250	337.5	27.6	6.8	0.75	.060	288

Tube sheet

Tube count	Tube pitch	Elastic mod. x 10^{-6}	Stress	G	M1	M2
260	0.9375	28.0	17500	17.58	119144	116716
				Inside diameter of integral pressure part	Operating moment	Bolting moment

Expansion joint

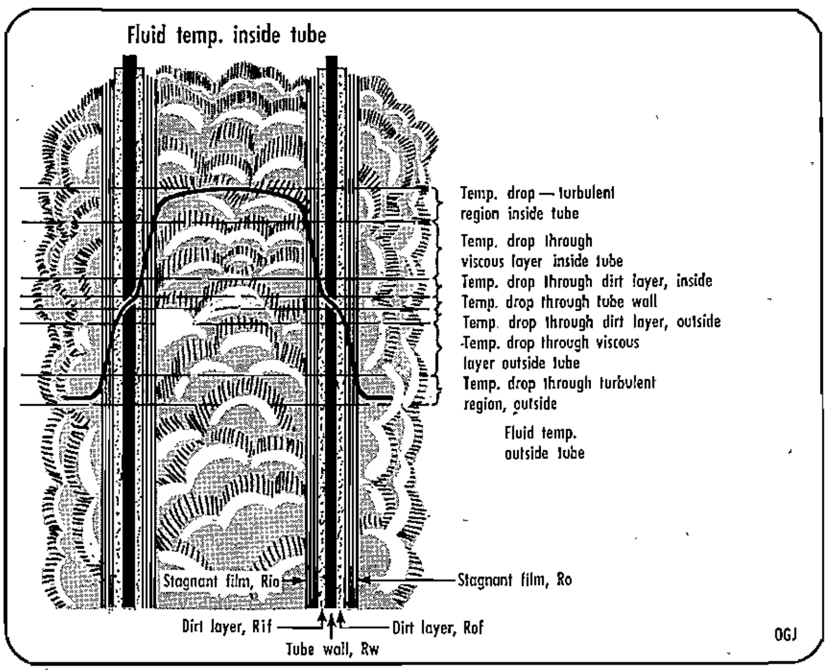
J	Inside diam. of exp. joint
	28
1.0 without joint 0 with joint	

OGJ

Computers help design tube sheets

Fig. 2

Heat flow through tube walls



Dale L. Gulley
Gulley Computer Associates
Tulsa

IN THE LATEST issue of TEMA, the calculation procedures for designing tube sheets in a fixed-tube-sheet exchanger are greatly expanded. The new calculation procedure takes into account differential thermal expansion. It also takes into account the interaction of the various components in a fixed-tube-sheet exchanger and their effect on the tube sheet.

The result is a more accurate analysis of the tube sheet, but a more lengthy calculation procedure. It also provides the basis to check to see if an expansion joint is needed. This new trial-and-error procedure is a good computer application.

Instead of using the design pressure for calculating the tube-sheet thickness, special effective design pressures are calculated for both the shell and tube sides. For TEMA R procedures, methods can be found on p. 40 and 41.[1]

Input

Fig. 1 is a computer input form. There are four main categories: shell side, tube side, tube sheet, and expansion joint. It is necessary to know the metal temperatures of both sides. It is tempting to simplify things and use the average fluid temperature on each side, but this would give you a thicker tube sheet than you really need.

If the thermal resistance of the tube-side fluid is relatively high, the two metal temperatures will be much closer together than the average fluid temperatures indicate.

Normally, the heat exchanger is designed for operating conditions, but if the nature of the process would give unusual start-up conditions, then these should certainly be taken into account.

For metal temperature on the shell side, use the average shell fluid tempertaure. The calculation of the tube-side metal temperature requires the knowledge of the heat-transfer coefficients.

For the **clean** condition an average tube-side metal temperature is calculated (refer to Fig. 2):

1. If shell side is hottest

$$TM_{avg} = T_{avg} + \frac{LTMD_c \times Rio}{(Rio+Ro+Rw)} \quad (1a)$$

2. If shell side is coldest

$$TM_{avg} = T_{avg} - \frac{LTMD_c \times Rio}{(Rio+Ro+Rw)} \quad (1b)$$

WHERE:

$LTMD_c$ = Corrected log means temperature difference, °F.

Rio = Inside heat-transfer resistance referred to outside surface.

Ro = Outside heat-transfer resistance.

Rw = Tube-wall resistance.

T_{avg} = Average tube-fluid temperature, °F.

TM_{avg} = Average tube-metal temperature, °F.

The resistances used in these equations are simply reciprocals of the heat-transfer coefficients. They have been put in the form of resistances to more easily understand the equations.

Fig. 3

Computer output

```
TUBE SHEET THICKNESS FOR FIXED TUBE SHEET

INPUT   SAMPLE

                                          SHELL         TUBE

DESIGN PRESSURE=                          250.          250.
METAL TEMPERATURE-70.                     362.          337.
ELASTIC MODULUS X 10 -6                    27.30         27.60
COEFFICIENT OF EXP. X 10 +6                 6.90          6.80
O. D.                                      18.0000        0.7500
THICKNESS                                   0.2190        0.0600
I. D. OF EXPANSION JOINT                   28.00

TUBESHEET
   TUBECOUNT                              260.
   PITCH                                    0.9375
   ELASTIC MODULUS X 10 -6                 28.00
   ALLOWABLE STRESS                     17500.
   G                                       17.5800
   OPERATING MOMENT                    119144.
   BOLTING   MOMENT                    116716.


OUTPUT ------------------------------------------------

   J= 0.     K= 0.357     FQ= 2.399
PRESSURE OF DIFFERENTIAL THERMAL EXPANSION
          0.

SHELL BOLTING PRESSURE    TUBE BOLTING PRESSURE
        133.                      136.
PS'= -192.
PT'=  250.
EFFECTIVE SHELL DESIGN PRESSURE-BENDING
        -192.
EFFECTIVE SHELL DESIGN PRESSURE-SHEAR
        -192.
EFFECTIVE TUBE DESIGN PRESSURE
         578.


TUBE SHEET THICKNESS-SHEAR
          0.364
TUBE SHEET THICKNESS-BENDING
          1.598
```

OGJ

If the inside and outside fouling are equal, it doesn't matter if you evaluate the exchanger in the clean or dirty condition. Under certain conditions the difference in metal temperature will be greater when the exchanger is fouled. This will be when the ratio of the shell-side fouling to the tube-side fouling is greater than 1.

For the **foul** condition the average tube-side metal temperature is calculated:

1. If shell side is hottest

$$TM_{avg} = T_{avg} + U\ (Rio+Rif)\ (LMTD_c \quad (2a)$$

2. If shell side is coldest

$$TM_{avg} = T_{avg} - U\ (Rio+Rif)\ (LMTD_c) \quad (2b)$$

WHERE:

Rif = Inside fouling resistance

U = Overall heat-transfer coefficient

After the metal temperatures are calculated, 70° is subtracted from them for input data.

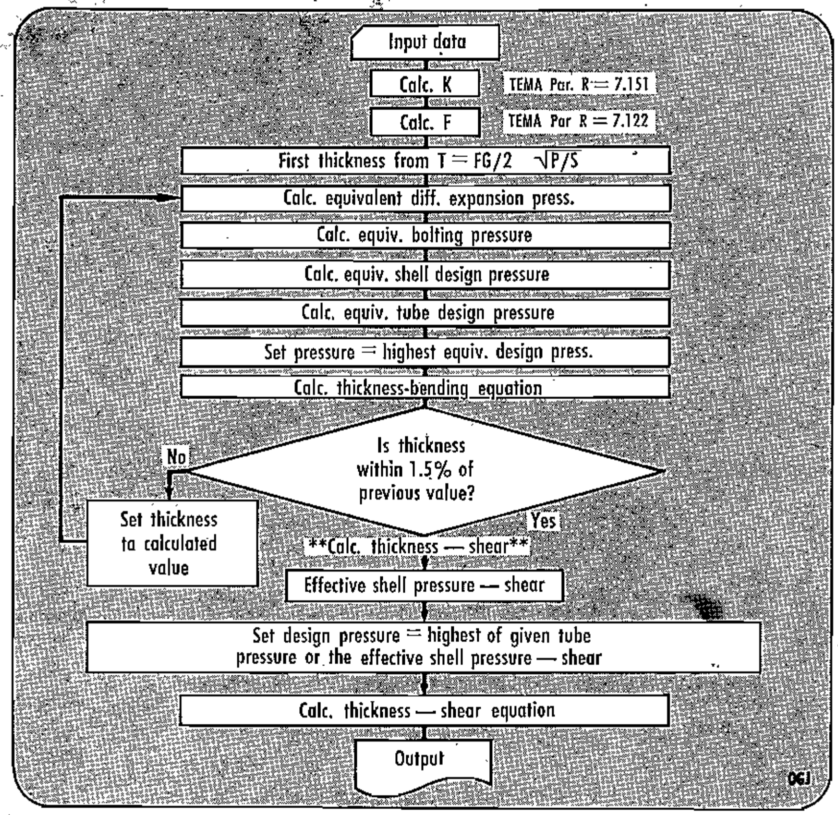
The modulus of elasticity can be found on p. 192.[1] For the shell wall thickness, the thinnest condition should be used. Therefore, from the wall thickness when new subtract the corrosion allowance. If the shell is constructed from pipe, also subtract the mill tolerance of 12½%.

The tube-wall thickness and the length are expressed in inches.

G is defined as the inside diameter of the integral pressure part. For TEMA type BEM, use the shell i.d.

Fig. 4

Flow chart of computer logic



For TEMA type CEN, use the channel i.d.

M1 and M2 are the operating and bolting moments, respectively. For an integral type tube sheet (CEN), this will be zero. For other types this comes from the corresponding flange calculation.

Fig. 1 is a completed input sheet for this example:

Type TEMA BEM

Shell design pressure, 250 psi

Shell 18-in. std. wt steel pipe with ⅛ in. C.A.

Channel design pressure, 250 psi

Channel 18 in. std. wt steel pipe with ⅛ in. C.A.

Tubes, A-214 steel with 0.060 in. wall, 260° ¾ in. on 15/16 by 24 ft long

Shell temp. in = 525° Temp. out = 340° F.

Tube temp. in = 320° Temp. out = 445° F.

Shell heat-transfer coefficient = 231 Fouling = 0.002

Tube heat-transfer coefficient = 132 Fouling = 0.001

28 in. i.d. expansion joint in shell

Calculation of metal temperatures:

Shell

For shell use average temp.: (525 +340)/2 = 432.5°

For computer input: 432.5 — 70° F. = 362.5°

Tube

1. Calculate MTD

   GTD = 525 — 445 = 80

   LTD = 340 — 320 = 20

   Difference = 60

   MTD = 43.2

2. Calculate resistances

   Rio (1/tube h) = 1/132 = 0.00758

   Ro (1/shell h) = 1/231 = 0.00433

   Rif = 0.001 × 0.75/0.63 = 0.00119

   Rof = 0.00200

3. Since shellside is hottest, use Equation 2a.

$$TM_{avg} = T_{avg} + U\ (Rio + Rif)\ (LTMDC)$$
$$= (320 + 445)/2 + 66.1\ (0.00758 + 0.00119)\ 43.2$$
$$= 407.5° \text{ F.}$$

For computer input: 407.5—70° F. = 337.5° F.

Shell wall thickness.

Subtract out corrosion allowance 0.375 — 0.125 = .25

Allow for mill tolerance for piping 0.25 × 0.875 = .219

Metal temperature for tube sheet.

To be conservative, average shell fluid temperature and tube fluid temperature at coldest tube sheet. (Large values of E, give thicker tube sheets).

Temp. = 330°

G—For TEMA type BEM this would be the Shell i.d.

Calculate this using above shell wall thickness

I.D. = O.D. — 2 × TK. = 18 — 2 (0.219)

= 17.58

J = 0 since there is an expansion joint.

Output

Fig. 3 shows the computer output. The top part reflects the values used on input. This provides a record for verification. The bottom part of the output shows the important values computed.

The last two variables shown are the calculated values for thickness using the two different equations. The designer uses the largest value.

Interpretation

Fig. 4 is a flow chart for the computer logic. The tube-sheet thickness is determined by trial and error. The starting thickness is determined from TEMA equation R: 7.122. The pressure used is the highest of the two design pressures.

Usually the thickness calculated according to the bending equation will be controlling. The shear equation controls at high design pressure.

The bolting moments imposed on the tube sheet can substantially increase the calculated tube-sheet thickness. At high design pressures where the attached flange gives high moments, it may be better to use an integral tube sheet such as a TEMA type CEN. The integral tube sheet, which has zero bolting moment, calculates to only a fraction of the thickness that a combined flange and tube-sheet unit would.

With a few modifications, this program can be made to analyze for expansion-joint requirements. It uses the same effective design pressures and pressure of differential thermal expansion to calculate the stresses.

Reference

1. Standards of TEMA, Fifth Edition, 1968 and 1970 addenda.