DISCUSSION

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The author has developed an attractive method for the stress analysis of vertically mounted through-tube heat exchangers under the unsymmetrical loading conditions (the two tube sheets are loaded unsymmetrically). These unsymmetrical loads, the dead weight, for example, normally are analyzed separately from the symmetrical loads (such as the tube-sheet longitudinal differential thermal expansion load) and the final result is obtained by superposition. The author's method provides a single solution for both the symmetrical and the unsymmetrical loadings. However, for completeness of the analysis, as not mentioned in the paper, the loadings induced by the poisson's ratio effect of the tubes, the shell rotations at the sheet-shell junctions due to thermal gradients along the shell, and the thermal bow effect of the sheet may be included in the calculation. The first two of these loads may readily be included in equation (39) and the boundary condition of the annular ring portion of the sheet. The thermal bow load of the sheet may be treated in the same manner as in reference [14].[4] The effect of these loads on the results of the analysis depend on the design conditions specified and may not be ignored in all cases.

In this paper, since the iteration method is used and many coefficients and numerical values are involved, deviations and errors are apt to occur. A comparison of Fig. 4 and Table 1 of the paper with the closed form method [14, 15], using the same data as the paper provided, indicates that the magnitudes and distribution patterns of the sheet moments and the tube stresses shown in Fig. 4 and Table 1 are at variance with the closed form solution. The results for the closed form solution for the combined case of the shell and tube side pressures and the tube-shell longitudinal differential thermal expansion are respectively shown in Figs. A and B. The maximum values in Fig. 4 and Table 1 of the paper are about two times higher than those of Fig. A and Fig. B and the location of the maximum moment is at r = 12.7 in., rather than r = 13.1 in. as the paper calculated. Also the closed form solution shows that all of the tubes are in tension, whereas this paper indicates that the tubes in the edge region are in tension and those in the center region are in compression.

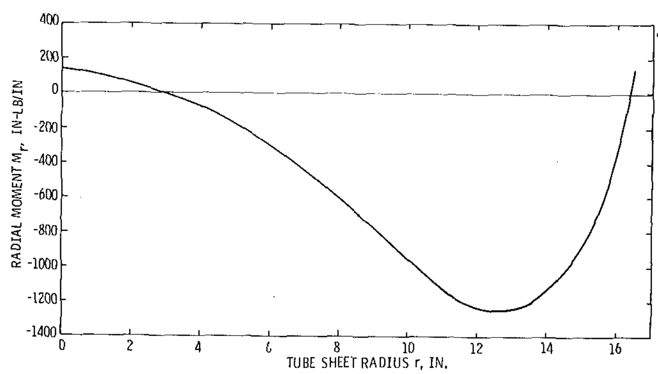

Fig. A Distribution of radial bending moment of tube sheet due to shell side and tube side pressures and longitudinal differential thermal expansion between the shell and the tube bundle

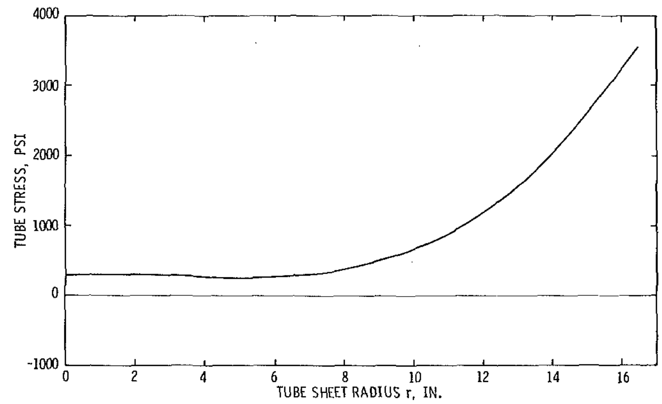

Fig. B Distribution of tube stress due to shell side and tube side pressures and longitudinal differential thermal expansion between the shell and the tube bundle

Additional References

14 Chiang, Chia Chi, "Closed Form Design Solutions for Box Type Heat Exchangers," ASME publication 75-WA/DE, 1975

15 Chiang, Chia Chi, "Structural Design Optimization of Once-through Type Heat Exchangers," ASME publication 77-JPGC-NE-3, 1977.

Author's Closure

The author is grateful to Mr. Chiang for making some pertinent comments on this paper. The principal object of this paper was to present a unified analysis for fixed tubesheet exchangers with unequal pressure loadings at the tubesheet locations. To the best of our knowledge, there is no published solution which adequately treats the unsymmetrical component of pressure loadings, as may be inferred from the discusser's remarks.

Mr. Chiang is correct in pointing out that the longitudinal deformation of the tubes due to pressure loadings is not included in the analysis. For the sake of consistency it should be included, although hoop stresses in the tubes are generally quite small. This is best accomplished by modifying equation (6) to read as follows:

$$\Delta = \frac{\mu_t \ell_1^2}{2E_t} - \frac{\mu_s(\ell_3^2 - \ell_2^2)}{2E_s} - \frac{\nu_t(p_{12} + p_{22} - p_{11} - p_{21}) r_t \ell_1}{2t_t E_1} \quad \ldots \quad (6)$$

where ν_t, r_t and t_t denote Poisson's ratio, means radius, and wall thickness of the tube, respectively. Remainder of the equations given in the paper remain unchanged.

"Thermal bow effect" of the tubesheet is not included, because in typical exchangers there is no thermal bow as envisaged in the references cited by the discusser. The temperature profile through the tubesheet is generally flat, and undergoes a skin drop on the shell side surface [16].[1] Similarly, the temperature distributions in the shell and channel regions proximate to the tubesheet are highy complex. A sweeping assumption as to their distribution may be in gross error. Furthermore, temperature induced stresses belong to the "secondary" and "peak" category as defined in the ASME Codes [17]. Because of their self limiting nature, the design codes permit relatively high limits on such stresses, and many do not even require their evaluation. The task of analyzing such relatively unimportant stress components is best accomplished by a suitable finite element program.

Finally, the results given for the numerical example in this paper were based on an earlier version of the computer program, which neglected tube as well as shell poisson effect (last two terms in the r.h.s. of equation (39)) and tubesheet surface rotation term in displacement continuity (right hand side in equation (36)). These

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[4] Numbers in brackets designate Additional References at end of discussion.

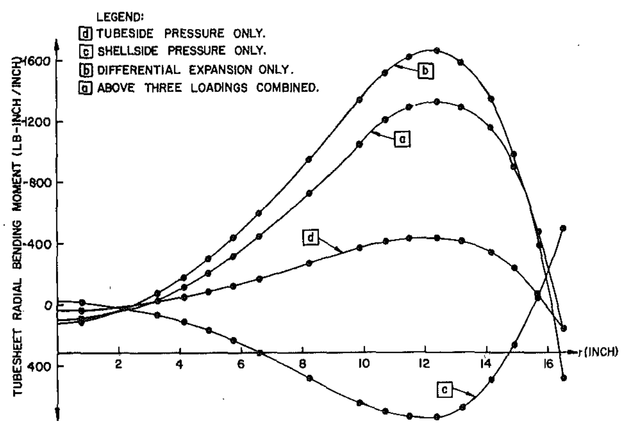

Fig. 4A Radial bending moment in the tubesheet as a function of radius

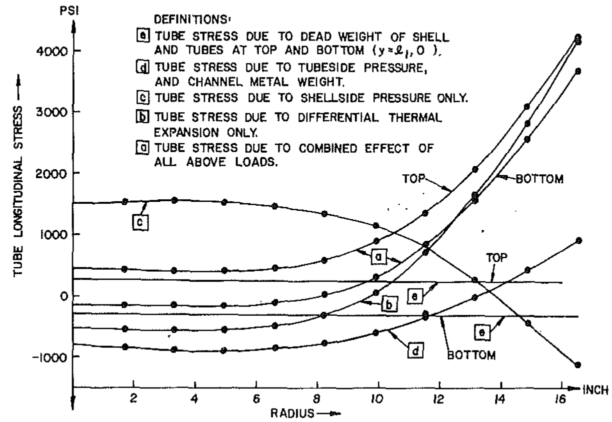

Fig. 5A Variation of tube longitudinal stress with radius

Table 1A Tube longitudinal stress, σ_t, psi (kg/cm^2), as a function of radius; nominal design pressures and differential thermal expansion

Loading Case \ Radius r	0	3.3″ (6.38 cm)	6.6″ (16.76 cm)	9.9″ (25.15 cm)	13.2″ (33.53 cm)	16.5″ (41.91 cm)
d. Tubeside pressure only	−613. (−43.1)	−618. (−43.5)	−596 (−41.9)	−439 (−30.9)	−11. (−.8)	677 (47.6)
c. Shellside pressure only	1374 (96.6)	1378 (96.9)	1322 (93.)	1011 (71.1)	216 (15.2)	−996 (70.1)
b. Differential expansion only	−532 (−96.6)	−553 (−38.9)	−486 (−34.2)	67 (4.7)	1621 (114)	4188 (294.5)
a. Combination of above loads	229 (16.1)	207 (14.6)	240 (16.9)	639 (44.9)	1826 (128.4)	2869 (272.1)

Table 2A Tubesheet radial bending moment, M_r, lb-in./in. (kg-cm/cm), as a function of radius

Loading Case \ Radius (inch)	LOWER TUBESHEET 0	3.3	6.6	9.9	13.2	16.2	UPPER TUBESHEET 0	3.3	6.6	9.9	13.2	16.5
(e) Deadweight only	1349 (612)	1261 (572)	997 (452)	556 (252)	−65 (−30)	−878 (−398)	−1351 (−613)	−1261 (−572)	−991 (−450)	−542 (−246)	83 (38)	872 (396)
(d) Tubeside pressure & channel weights, only	−668 (−303)	−685 (−311)	−738 (−335)	−772 (−350)	−512 (−232)	712 (323)	733 (333)	623 (283)	290 (132)	−211 (−96)	−604 (−274)	−220 (−100)
(c) Shell side pressure only	512 (232)	572 (260)	750 (340)	945 (429)	724 (328)	−924 (−419)	−576 (−261)	−442 (−201)	−45 (−20)	517 (235)	809 (367)	−178 (−81)
(b) Differential expansion only	104 (47)	−69 (−31)	−593 (−269)	−1338 (−607)	−1582 (−718)	497 (225)	104 (47)	−69 (−31)	−593 (−269)	−1338 (−607)	−1582 (−718)	497 (225)
(a) Combination of above heads	1297 (588)	1080 (490)	416 (189)	−609 (−276)	−1436 (−651)	−593 (−269)	−1090 (−494)	−1149 (−521)	−1339 (−607)	−1573 (−714)	−1294 (−587)	971 (440)

omissions, individually quite unimportant, combine in the numerical example problem to produce significant difference in the results. Corrected results consistent with the analysis given in this paper (including tube poisson effect) are given herein. The tables and figures correspond to their numerical counterparts in the main text of the paper. Comparison of the results show the importance of these neglected terms. The influence of shell poisson effect is found to be especially important. Ironically, this term was neglected in nearly all previously published papers on fixed tubesheet design.

Additional References

16 Gardner, K. A., "Heat Exchanger Tube Sheet Metal Temperatures," *The Refiner and Gasoline Manufacturer*, March 1942.

17 "ASME Boiler and Pressure Vessel Code," Section III, Div. 1, subsection NB, Par, NB3213, P. 55, *ASME (1977)*