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A DESIGN CONCEPT FOR MINIMISING TUBESHEET STRESS AND TUBE JOINT LOAD IN FIXED TUBESHEET HEAT EXCHANGERS

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ABSTRACT

An expansion joint is normally employed to alleviate the tube axial load in fixed tubesheet heat exchangers. Incorporation of an expansion joint usually results in increased tubesheet thickness and hence, increased cost of hardware. A design strategy to simultaneously reduce both tubejoint load and tubesheet stress in the heat exchanger (without the usual recourse of expansion joints) is herein presented. The range of application and practical feasibility of this concept is also discussed.

NOMENCLATURE

M_r: Tubesheet radial bending moment at radius r

M_1: Discontinuity moment per unit circumference between shell and tubesheet (Figure 2)

M_2: Discontinuity moment per unit circumference between channel and tubesheet (Figure 2)

N_1: Shell axial load (tension positive) per unit circumference

N_2: Channel axial load (tension positive) per unit circumference

P_s: Shell side pressure

P_t: Tubeside pressure

Q_r: Tubesheet shear load per unit circumference at radius r

Q_1: Discontinuity shear per unit circumference between shell and tubesheet (Figure 2)

Q_2: Discontinuity shear per unit circumference between channel and tubesheet (Figure 2)

q_r: Maximum tubesheet radial shear stress

t: Tubesheet thickness

t': Normalized tubesheet thickness

σ_t: Maximum axial tube stress (tensile)

σ_{tc}: Maximum axial tube stress (compressive)

δ: Differential thermal expansion between tubes and shell

INTRODUCTION

In an integral tubesheet heat exchanger, the two tubesheets together withstand shellside and tubeside pressure loadings. The coupling between the deformation and stress fields in the two tubesheets is provided by the tubes and the shell. In effect, the staying action of the tubes and the shell helps reduce the thickness of the tubesheet. The tubes act as an elastic foundation. For this reason, the thickness of the tubesheet in a fixed tubesheet heat exchanger is invariably less than the thickness of an otherwise identical but unstayed design (such as a U-tube construction). The integral tubesheet design is ideally suited for pressure loadings (shellside or tubeside); however, it is subject to an additional stress field due to differential thermal expansion between tubes and shell. In some cases, stresses due to thermal expansion are so severe that an expansion joint is added in the shell to reduce tube-tubesheet joint load. Addition of the expansion joint, however, eliminates the shell as an axial load carrying member, leading to an increased tubesheet thickness. This is clearly an uneconomical design remedy for commercial heat exchangers. The object of this paper is to propose a far more economical and powerful design remedy which is effective in many situations. We will refer to this concept as the "pre-stressed shell design." Before introducing this concept it is necessary to examine some basic structural characteristics of the integral tubesheet construction. Such a discussion will lead us to the pre-stressed shell concept in a natural manner.

A STANDARD DESIGN ANALYSIS

The analytical tool utilized to establish the fixed tubesheet structural behavior is a computer program based on reference [1]. Therefore, we will limit ourselves herein to describing the basic quantities of interest and their significance. The reader is referred to ref. [1] for the details of mathematical analysis.

we consider the following numerical example:

Shell and channel mean radius = 20" *(508mm)*
Shell and channel corroded thickness = 0.5" *(12.7mm)*
Young's modulus = 30 x 10^6 psi *(206.85 x 10^6 kPa)*
Tubesheet equivalent perforated median radius - 18.5" *(469.9mm)*
1500 tubes, 5/8" diameter x 18BWGX 400" long;
Young's modulus of tubes = 20 x 10^6 psi *(137.9 x 10^6 kPa)*

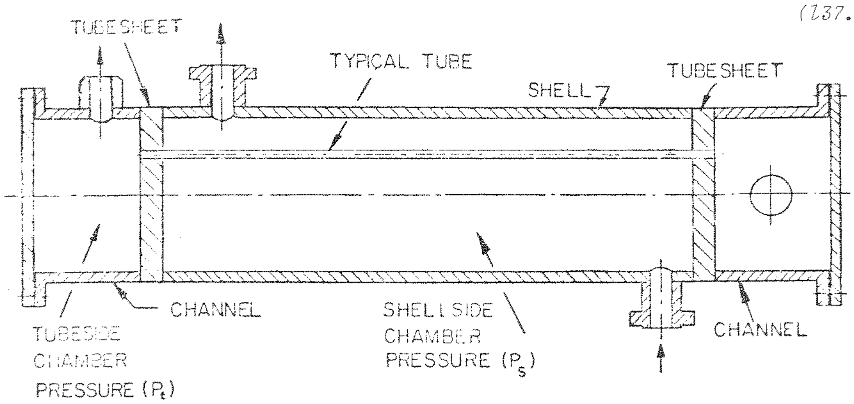

FIG. 1: INTEGRAL TUBESHEET HEAT EXCHANGER (THROUGH-TUBE DESIGN)

Figure 1 shows a typical fixed tubesheet exchanger. This exchanger is subject to three principal types of loadings, namely

(a) Tubeside pressure, P_t

(b) Shellside pressure, P_s

(c) Differential thermal expansion between tubes and shell, δ. Differential expansion is taken as positive when tube thermal expansion exceeds that of the shell.

Other relatively unimportant loadings for commercial hardware, such as gravity effects, seismic loads, etc. are not included here to avoid unnecessarily complicated solutions which might obscure the central theme of the paper.

The principal design quantities of interest are:

(a) Maximum bending moment in the tubesheet, M_r

(b) Maximum axial tubeload (or maximum tube axial stress (σ_t, σ_{tc}))

(c) Shell axial load, N_1

(d) Maximum tubesheet shear load, Q_r, or shear stress q_r

Somewhat less important design data are:

(e) Edge moment, M_1 and shear, Q_1 between shell and tubesheet

(f) Edge moment, M_2 and shear, Q_2 between the channel and the tubesheet.

Figure 2 shows these quantities pictorially to identify their assumed positive directions. We note that the channel end load N_2 is determined from equilibrium directly. The shell longitudinal load N_1, however, depends on a complex interaction between the structural members involved; namely the tubesheet, shell, channel and tubes. It is instructive to examine the variation of N_1 and other quantities of interest as the tubesheet thickness is varied. For this purpose,

The equivalent Young's modulus and Poisson ratio of the tubesheet perforated interior are found from reference [2] to be 10 x 10^6 psi *(68.95 x 10^6 kPa)* and 0.35, respectively.

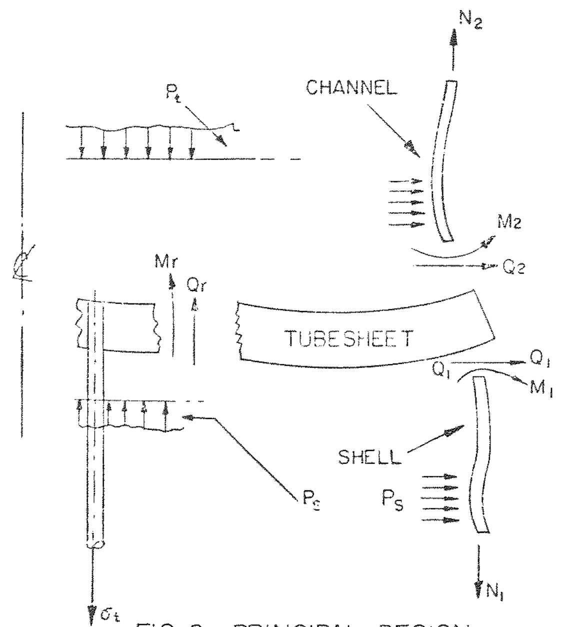

FIG. 2: PRINCIPAL DESIGN VARIABLES AND THEIR NOTATION

Corresponding to a tubeside design pressure, P_t of 300 psi,† the TEMA [3] design rules yield a thickness of 1.760"† for an assumed tubesheet allowable stress of 17500 psi. † The thickness of a corresponding unstayed tubesheet (integral U-tube design) according to TEMA is 2.61". † The difference between the two thicknesses is due to the effect of the supportive action of tubes and shell. Let us now examine the reduction in the tubesheet stress (bending moment) as its thickness is increased. In an unstayed tubesheet design, the result is quite uncomplicated; the bending stress drops as the inverse of the square of the tubesheet thickness. Figure 3 shows the fractional drop in tubesheet maximum bending stress as the tubesheet thickness is increased. The basis for deriving the fraction is 1"† thick tubesheet. For comparison purposes, the equivalent drop, if the construction were unstayed, is also shown. We note that increasing the tubesheet thickness has a minimal effect on reducing the bending stress in a stayed tubesheet design. Actually, in a small range the maximum bending moment even undergoes a small increase as the thickness is increased. This apparent paradox is explained by referring to Figure 4 which shows N' and σ_t' plotted against the thickness ratio t'.

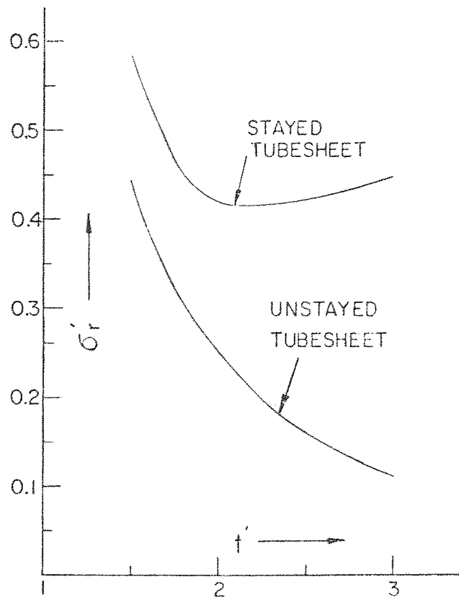

FIG. 3: DIMENSIONLESS MAXIMUM TUBESHEET BENDING STRESS VS. DIMENSIONLESS TUBESHEET THICKNESS

The primed quantities are the ratio of their values at the tubesheet thickness, t, to that at tubesheet thickness equal to 1". † It is seen that N_1' and σ_t' (ratio of maximum tube axial tensile stresses) drop as t' is increased indicating that more of the pressure load is taken up by the tubesheet. The maximum tube compressive stress rises at first, and then begins to drop off. The drop in peak tensile and compressive stresses indicates that the tubesheet is carrying more of the pressure load, and the participation of the tubes is diminished. Furthermore, the bump in the tube

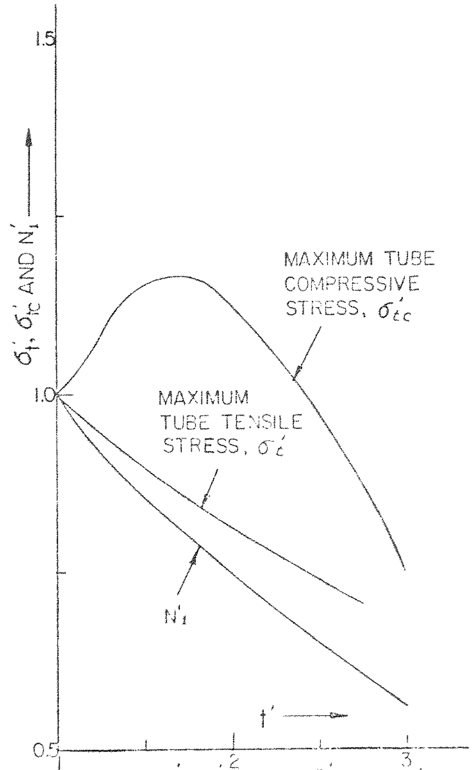

FIG. 4: N_1', σ_t' and σ_{tc}' - vs. t'

compressive stress curve suggests that increasing tubesheet thickness may indeed increase the tube joint load! Keeping the tube joint load to a minimum is an important design goal in tubular heat exchanger design. Increasing the tubesheet thickness, as this case suggests, may actually thwart this objective in some cases. Thus increasing the thickness of the tubesheet does little to alleviate its stress level; it merely relieves other load carrying members. This conclusion is further reinforced by examining Figure 5 which shows the maximum value of normalized tubesheet shear stress as a function of normalized thickness t'. The value of q_r for 1"† thick tubesheet is used as the normalizing parameter. For the purpose of comparison, the equivalent curve for an unstayed tubesheet is also given. The shear stress in the stayed tubesheet design does not fall as rapidly as that in an unstayed tubesheet, indicating that the tubesheet assumes a larger portion of the total pressure load as its thickness is increased. Thus, increasing the tubesheet thickness has the primary effect of reducing stresses in other members, namely the shell and tubes. Since the shell, in particular, has a great deal of reserve strength in the axial direction, reducing its stress level at the expense of an increased tubesheet thickness is not a suitable strategy for an efficient design. On the contrary, the design goal should be directed towards tapping the axial strength of the shell to reduce the stress levels in the tubesheet.

With this objective in mind, we will now examine the details of the stress field in the tubesheet.

† 1 inch = 25.4 mm
1 psi = 6895 Pa

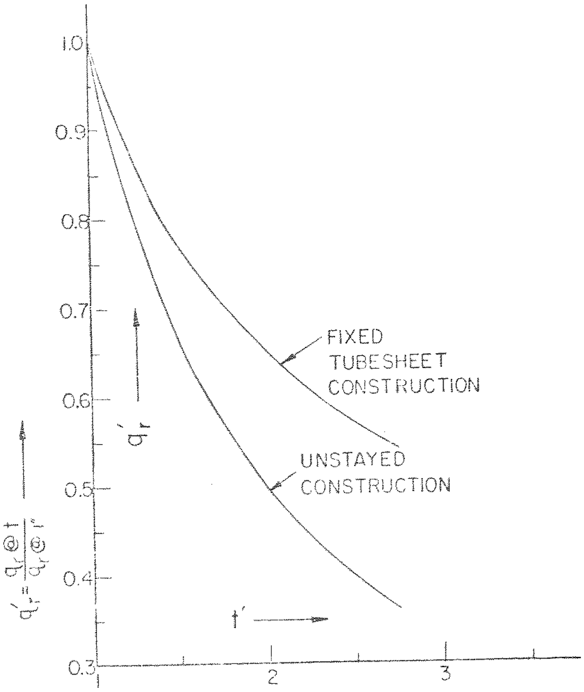

FIG.5: NORMALIZED TUBESHEET SHEAR STRESS VS. NORMALIZED THICKNESS

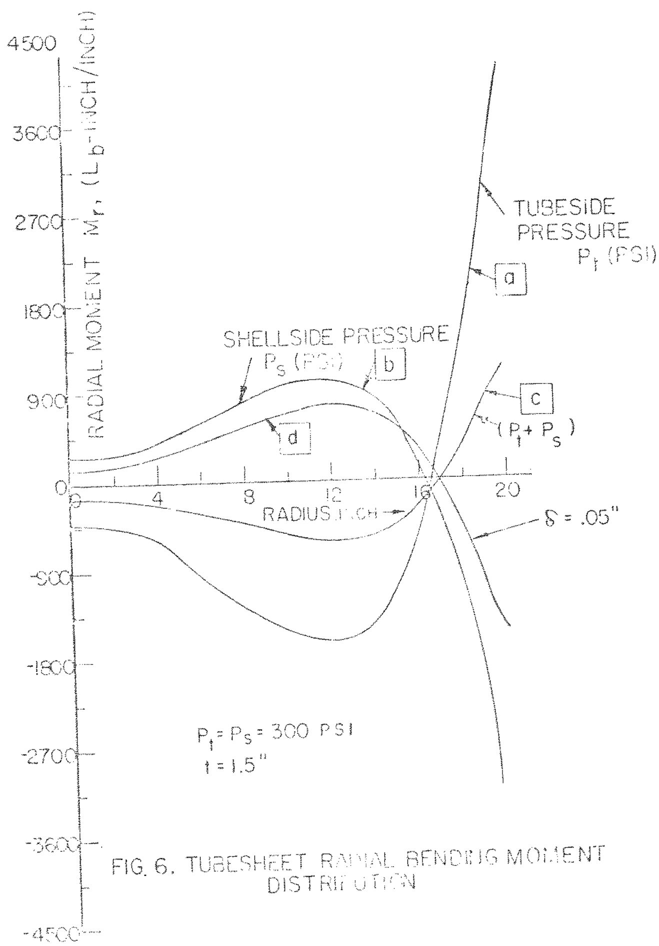

FIG. 6. TUBESHEET RADIAL BENDING MOMENT DISTRIBUTION

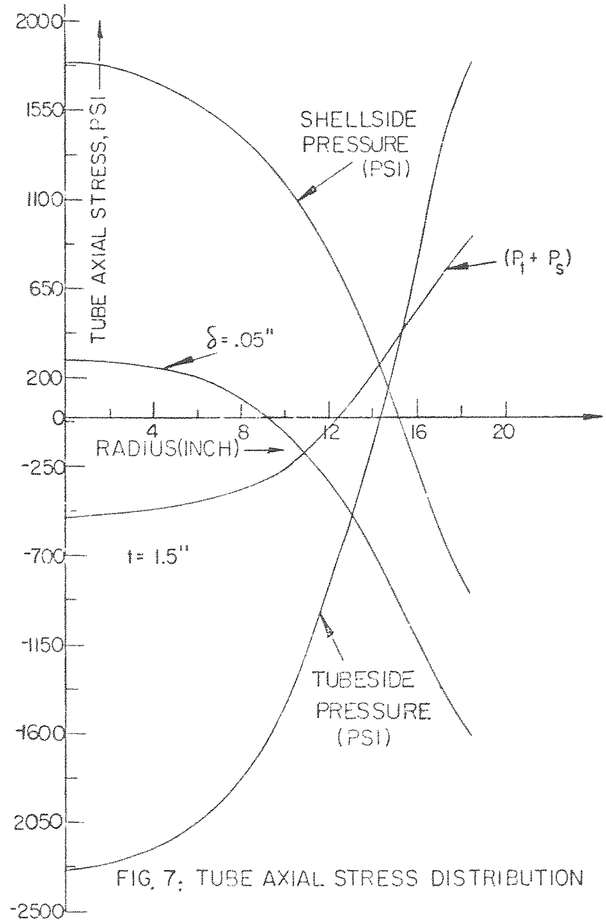

FIG. 7: TUBE AXIAL STRESS DISTRIBUTION

TUBESHEET STRESS FIELD IN A FIXED TUBE UNIT

In order to illustrate the state of stress in the tubesheet, we consider the above example with 1.5" thick tubesheet subject to the following loadings:

a. Tubeside pressure only (P_t = 300 psi)

b. Shellside pressure only (P_s = 300 psi)

c. P_t and P_s active concurrently

d. Differential thermal expansion only (δ = .05")

Figure 6 shows the tubesheet radial bending moment M_r as a function of the radial location r for loading conditions (a) to (d). We note that the general shape of the radial moment curve for case (a) is opposite to that for case (b). The same holds true for tube axial stress distribution (Fig. 7). Thus, the controlling design condition is one where only one chamber is pressurized (either P_t or P_s zero). We also observe that the shape of the radial moment curve and tube axial stress curves for expansion δ and P_s (shellside pressure only) are similar. This fact is found to hold true in the meaningful range of heat exchanger geometry parameters (i.e., shell and channel thickness, tubesheet thickness, shell diameter, etc.) Table 1 shows shell axial load N_1, shell/tubesheet discontinuity moment M_1 and shear Q_1, and channel/tubesheet discontinuity moment M_2, shear Q_2 for the above loading cases. We note that all quantities except N_1 are opposite in sign for tubeside and shellside pressures. The effect of δ is similar in sign to that of P_s. Hence, from a purely structural viewpoint, a positive value of the differential thermal expansion, δ, produces a stress field which adds to the shellside pressure induced stress field and subtracts from the tubeside pressure induced stress field for all design quantities except N_1. This information can be harnassed to improve the stress levels in a wide range of integral tubesheet heat exchangers, which we will now proceed to demonstrate.

THE PRE-STRESSED SHELL CONCEPT

The fluid stream in a majority of tubular heat exchangers are arranged such that the high pressure fluid is on the tubeside; i.e. $P_t > P_s$. Such an arrangement leads to a lower equipment cost and hence is preferred unless other considerations (corrosion, two phase flow, etc.) warrant otherwise. As was shown in the foregoing, the tubesheet moment distribution and tube axial stress fields for tube and shellside pressures counteract each other over most of the region of the tubesheet. Even if P_t and P_s are equal, the stress field due to P_t dominates, as illustrated in Fig. 6 (curve C) for 1.5" thick tubesheet. A similar plot for the axial stress is given in Fig. 7. Thus, in most practical cases, where P_s is smaller than P_t the resultant stress field is similar to that due to P_t acting alone. If the operating condition leads to a net differential thermal expansion between tubes and shell to be positive, then the corresponding stress field will subtract from the resultant stresses due to pressures. However, if the thermal expansion δ is negative, then the thermal stress field will add to the pressure induced stresses. In order to reduce the combined stress levels, it would be advantageous to superpose a stress field which is generally disposed to opposing "tubeside pressure type" loadings. Fortunately, it is possible to achieve this by prestressing the shell in the axial direction such that the shell is δ units shorter than the tubes in its stress free state. Figure 8 shows the tubesheet radial moment distribution (a) due to P_t only (b) due to P_t and δ_1 where δ_1 = .05" (c) due to P_t and δ_2, where δ_2 = .08".

Figure 9 shows similar curves for tube axial stress distribution. We observe that the net effect of δ is to reduce tube bending stress as well as tube joint load. As a matter of fact all other design variables: maximum tubesheet shear stress, discontinuity moments M_1 and M_2, shear Q_1 and Q_2 are also reduced. The only stress which increases is the shell axial stress. Since usually large margins of safety exist in shell axial stress, such an increase does not derate equipment reliability. In essence, this method exploits the available axial strength in the shell to reduce stresses in all other members, including the vital region of concern such as the tube/tubesheet joints.

The procedure to accomplish a prestress in the shell is quite straightforward, and relies on the weld shrinkage phenomenon. It is well known that butt welds between two members produce a net shrinkage upon completion of welding. Two points A and B located a distance S apart in two plates approach each other subsequent to welding by an amount Δ, where Δ depends on the bevel detail, welding process type, voltage, amperage, filler wire; and several other variables. It is possible to control the precise amount of shrinkage by monitoring the essential parameters. The weld shrinkage is a universal bane in pressure vessel fabrication; yet we propose to use this phenomenon to our advantage. In order to obtain the precise amount of pre-strain in the shell, it is necessary to use test coupons which simulate the plate thickness, bevel detail and welding process for actual shell course girth seams. The coupons are used to determine the precise setting of all the welding variables to obtain desired shrinkage. Having calibrated the welding procedure, it is a simple matter to apply it in welding the shell girth seam. It is clear that some changes in the fabrication sequence are required to lock the stresses in the shell. The two tube ends should be roller expanded and welded (wherever specified) prior to welding the last girth seam in the shell. The last seam can be the joint between a tubesheet and the shell; or any of the girth seams joining two courses in the shell. Rolling the tubes into the tubesheet sets the shell length, welding the girth seam using a calibrated welding procedure produces the desired pre-stress in the shell. No further complications in the fabrication steps are involved.

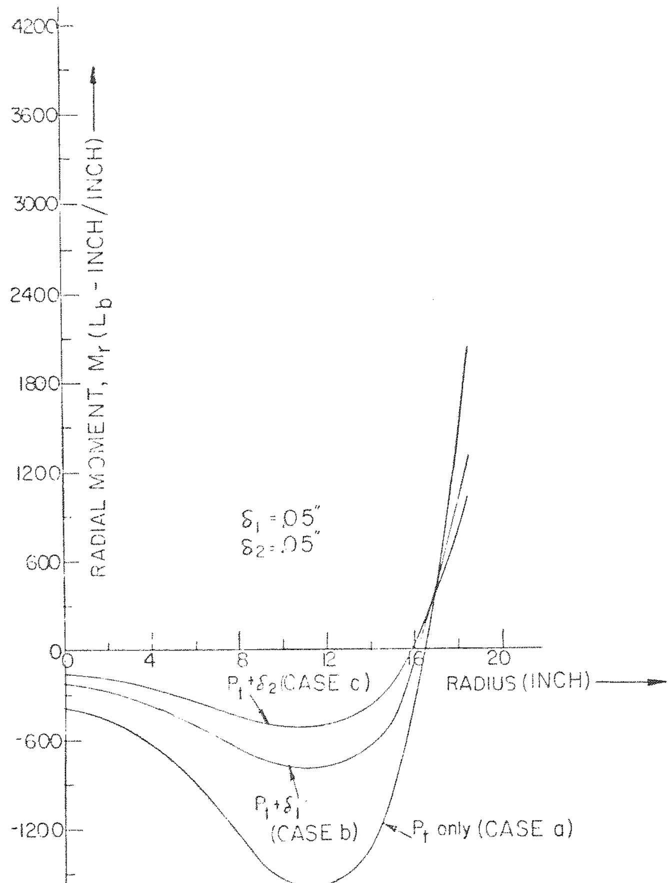

FIG.8. TUBESHEET RADIAL MOMENT DISTRIBUTION FOR VARIOUS PRE-STRESS CONDITIONS

CONCLUSIONS

Controlled weld shrinkage in a heat exchanger shell is suggested as a means of inducing a favorable state of pre-stress in a fixed tubesheet heat exchanger. Numerical examples are used to illustrate the concept and to give indication of the possible tubesheet and tube stress level reductions. This concept is particularly appealing in the design of "off-the-shelf" heat exchangers where tubesheet thickness has a substantial bearing on the equipment cost. This concept is of value only in those situations where the operating

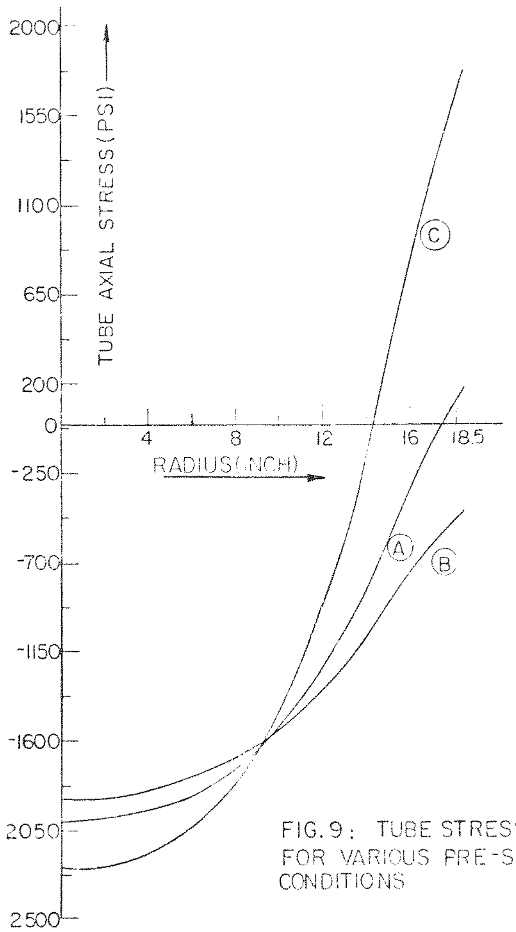

FIG.9: TUBE STRESS DISTRIBUTION FOR VARIOUS PRE-STRESS CONDITIONS

condition of the heat exchanger produces stresses which are counteracted by the built-in pre-stress. Such a situation normally exists in heat exchangers where the coolant medium is inside the tubes. The pres-stress technique suggested here hinges on the ability to control weld shrinkage precisely. This can only be accomplished in a highly controlled automated welding operations. Therefore, this concept is recommended for mass manufacturing of commercial heat exchangers. Its viability as a practical idea in customized one-of-a-kind designs is somewhat doubtful.

REFERENCES

[1] K. P. Singh, "Analysis of Vertically Mounted through-tube Heat Exchangers," Trans. of the ASME, Journal of Engineering for Power, vol. 100, No. 2, 1978, pp. 380-390.

[2] ASME Boiler and Pressure Vessel Code, Section III, Appendices, Fig. A-8131-1, 1980.

[3] Standard for Tubular Exchanger Manufacturers Association, 6th Edition, New York, 1978, p. 36.

TABLE 1* - STRESS RESULTANTS FOR 1.5"

Thick Tubesheet - Various Loadings

Case Designation	Loading	Axial Shell Load N_1 (lb/inch)	Shell/Tube Sheet Moment M_1 (lb-inch/inch)	Shell/tube Sheet Edge Shear Q_1 (lb/inch)	Channel/tubesheet Moment M_2 (lb-inch/inch)	Channel/tubesheet Edge Shear Q_2 (lb/inch)
a	P_t=300 psi	1220	-1090	565	1810	-1050
b	P_s=300 psi	1340	1460	-995	-610	306
c	δ = 0.5"	587	542	-263	-569	285

* Conversion of Engineering Units to SI Units

1 inch = 2.54 cm
1 psi = 6895 Pa
1 lb/inch = 175.1 N/m
1 lb-inch/inch = 4.448 N