STRESS ANALYSYS IN THE TUBES-TUBESHEET JOINT OF THE HEAT EXCHANGER UNDER HYDRAULIC EXPANSION

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ABSTRACT

In the present work, we are presenting the stresses and displacement occured in the tube/tubesheet joint of a heat exchanger under hydraulic expansion process. During this process a great amount of tubes cracked.
An elasto-plastic finite element calculation was carried out in order to determine the exact deformations of the tube-tubesheet joint. The most important conclusions are presented and compared with the obtained by analytical procedures.

1 INTRODUCTION

Hydraulic expansion, a method of anchoring tubes in the tubesheet has been used for many years and has in the meantime been applied in over a millon cases.
The process is characterized by the careful treatment of the material to be expanded under favourable stress conditions. The process is extremely accurate as a result of the hydraulic application of pressure.
This accuracy not only ensures that all joints are uniform and thus reproducible, it also permits the use of informative calculations. Nowadays it is possible to calculate and check the suitability of the selected tube-tubesheet joint before it is made.
The hidraulic method provides a clear basis for calculating the expansion pressure required to achieve a desired adhesion pressure between the tube and tubesheet.
The adhesion pressure is the pressure necessary to guarantee the joint under the changes of the loads operation for the whole service life of the vessel.

2 DESCRIPTION OF THE HYDRAULIC EXPANSION METHOD

Hydraulic expansion differs from all other known anchoring methods because of tube is plastically deformed by means of a pressurized liquid.
Under the pressure exerted by this liquid, the tube material flows until it reaches the wall of the bore.
By increasing the pressure further, the tube is pressed against the wall of the receiving bore.
The pressure is increased until the bore has been so deformed that the tube disc will lock permanently around the tube as a result of the elastic recovery of the disc when the pressure

is removed.
Figure 1 illustrate the principle of hydraulic expansion.
The different slopes of the elastic lines show the differing geometries of the tube and the tubesheet. It can be seen from figure 1 that a tight and secure tube to tubesheet joint depends on the material pairing, i.e, the relationship between the two yield points, and the geometry.
The tube expander can be seen from Figure 2.

3 TUBE ANCHORAGE CALCULATION BY ANALYTIC PROCEDURE

The hydraulic method provides a clear basis for calculating the expansion pressure required to achieve a desired adhesion pressure (p_h).
The adhesion pressure is the residual pressure between the tube and the plate which remains as a result of the greater recovery of the plate compared with the recovery of the tube after the pressure has been removed from the liquid, i.e, the adhesive pressure is the radial stress in the contact area of the tube and tubesheet.
It is a defination value which quantitatively describes the quality of a tube-tubesheet joint and allows comparisions to be made between two different joints.
The adhesion pressure required to secure the tubes using the hydraulic expansion method can be calculated.
The method was developed in Reference [1].
To obtain a simple analytical solution, the tubesheet geometry is simplified as shown in figure 3. It is assumed that deformation is limited to that part of the plate which is located within the inscribed circle Ra. Strain measurements performed on a test block have confirmed that this assumption yields to satisfactory results.
We shall introduce some terms that will be used below.
The limit pressure is the expansion pressure of the liquid at which, after removing the pressure, the elastic recovery of the plate is equal to the elastic recovery of the tube, and thus is the limit at which no residual adhesion can be obtained between the tube and the tubesheet.

$$p_0 = \sigma_{ft} \frac{2(U_p^2-1)}{\sqrt{3U_R^4+1}\,[U_p^2(1+\mu)+(1-\mu)]} + \frac{U_R^2-1}{2}\sigma_{ft} \qquad (1)$$

The final pressure is the maximum expansion pressure of the liquid that should be obtained and it must be greater than the limit pressure. The final pressure is set on the overflow valve of the equipment before performing the expansion operation.

$$p_i < \sigma_{fp}\frac{(U_p^2-1)}{(U_p^2+1)} + \sigma_{ft}\frac{(U_R^2-1)}{2} \qquad (2)$$

This last requirement must be satisfied if we wish to avoid plastic deformation of the material between the bores in the tubesheet.

4 CALCULATION DATAS

MATERIAL PROPERTIES

The following material properties of the tube and tubesheet are:

* Tubes X1CrNiMoNb (1.4575) (welded tube)

$\sigma_{ft} = 633$ MPa
$\sigma_{rt} = 776$ MPa

* Tubesheet 15 MnNi63 (1.6210)

$\sigma_{fp} = 385$ MPa
$\sigma_{rp} = 559$ MPa

These values were specified from test. The plastic material stress-strain curves used in this analysis are shown in Figures 4 a and b.

5 DESCRIPTION AND DETAIL OF COMPONENT DESIGN

The drawing in figure 5 shows the geometry detail of the heat exchanger.
The main dimension are:

Outer diameter of the tube = 23 mm
Thickness of the tube = 1.1 mm
Diameter of the tubesheet = 1230 mm
Thickness of the tubesheet = 330 mm
Pitch = 25.11 mm (triangular)
Diameter of the borehole = 23.35 mm

6 LOADING

According to the process datas of the hydraulic expansion the tubes were successfully expanded using a expansion pressure of 3000 bar. This test is assumed as is shown in figure 6.
The maximum pressure is reached at 3 second and the discharge at 6 second.

7 ANALYTICAL CALCULATION

According to the expression (1) and (2) above describe, the limit pressure is

$$p_o = 2530 \text{ bar}$$

The maximum or final expansion pressure must be lower than

$$p_i < 2150 \text{ bar}$$

We want to point out the following conclusions:

* The limit pressure is greater than the maximum pressure,which isn't correct because a plastic deformation between the bores in the tubesheet will be reached.
* The applied pressure, 3000 bar, is greater than the mentioned pressures.
* According to works of Reference [1] and these results, it is recommended to change the tubesheet material by other with yield stress limit higher. Thus we avoid a plastic deformation of the material between the bores in the tubesheet.
  In other case, a more complex and conservativeless analysis would be required in order to determine the exact deformation of the tube-tubesheet joint.

8 NUMERICAL CALCULATION

According to the explained in the above paragraph, an elasto-plastic analysis by using finite element method was carried out. The mathematical model was prepared to be used by Finite Element Program **NISA** (Reference [2]).
Figure 7 shows the section of the tubesheet and the tube idealized with 2D-plane stress element, with the limiting conditions resulting from the symmetry.
A dense mesh was used in the vecinity of the tube-tubesheet joint.
The material behaviour of the tube and tubesheet were assumed by elastic piecewise linear hardening.
In the initial condition there is a radial clearance of 0.175 mm between the sheet boreholes and the outer diameter of the tube.
A long-lasting contact force cannot be built up until this clearance has been reduced to nought by plastic deformation of the tube. In order to realize this condition GAP elements are inserted between the tube and the borehole.
The load is built up incrementally.

9 STRESS ANALYSIS RESULTS

The results of the two different stages of the process are shown.

a) Maximum expansion pressure, corresponding to 3000 bar at 3 seconds.

Figures 8 a and b show the curves of the efective stress of Von Mises and the radial displacement respectivelly during the expansion process when the pressure reachs the maximum value of 3000 bar. In this condition there are zones in the tube and the tubesheet which undergo plastic deformation. The maximum efective stress and the average displacement calculated in the tube-tubesheet joint are the following:

* TUBE

$\sigma_t = 694$ MPa $< \sigma_{rt} = 776$ MPa

$> \sigma_{ft} = 633$ MPa

$\delta_t = 0.2844$ mm

* TUBESHEET

$\sigma_p = 462$ MPa $< \sigma_{rp} = 559$ MPa

$> \sigma_{fp} = 385$ MPa

$\delta_p = 0.0956$ mm

b) The expansion pressure is relieved, in a period of 6 seconds.

Figures 9 a, b and c show the stresses and the displacements present in the joint.
Figure 9a : Efective stress of Von Mises
Figure 9b : Principal stress in circunferential direction.
Figure 9c : Radial Displacement.

The zone in contact presents compression stresses of average value:

$\sigma_c = -136$ MPa

The maximum stress and the average displacement in the tube-tubesheet joint are the following:

* TUBE

$\sigma_t = 135$ MPa

$\delta_t = 0.2516$ mm

* TUBESHEET

$\sigma_p = 315$ MPa

$\delta_p = 0.07$ mm

The final elastic recovery of the tube and tubesheet are:

$$\delta_{re} = \delta_{t=3\ \text{sec.}} - \delta_{t=6\ \text{sec.}} = 0.2844 - 0.2516 = 0.0328\ \text{mm}$$

The radial displacement of the tube at the end of the process is 0.2516 mm, and the initial radial clearence is 0.175 mm,therefore the contact between the tube and the borehole is insured.
Figures 10 a and b represent the time history of the displacement and the efective stress of Von Mises respectivelly for a radial direction.
In this graphics can be seen the non-linearities changes caused by contact between tube and tubesheet and by elastoplastic behaviuor of materials.
The contact between the tube and borehole is reached at 0.7 seconds as is illustrated in figure 10 a.
During the expansion process, in the instant 2.6 seconds, the tubesheet undergo plastic deformation.
When the expansion pressure is removed the elastic recovery of the tube and the plate provides compression stresses sufficient to maintain the contact surface closed.
The maximum displacements, reached on the plate after 3 seconds, are about 0.0956 mm while on the tube is 0.2844 mm. When the pressure is relieved, at 6 seconds, the remanent displacement in the plate is 0.07 mm and the tube 0.2516 mm
The idealized structure and the deformed structure are shown in figure 11. It can be seen that the radial deformation of the stressed hole hardly changes over the extent of the hole, and it is therefore very similar in its behaviour to a tube under internal pressure.

10 CONCLUSION

The present work describes the main results using analytical and numerical procedures in order to determine the structural integrity of the tube-tubesheet joint under hydraulic expansion process.

According to the analytical results, a large hoop stress, greater than the yield point value of the tubesheet is necessary.
In this case, if we wish to avoid the tubesheet plastic deformation is recomended to change the material by other with higher limit.

The most important conclusions obtained by numerical procedures are summarized in the following points.

a) The remanent circunferential compressive stresses obtained in the joint during the expansion hydraulic process are enough to fix the tube.

b) The plastic deformation of the tubesheet is confirmed. It might be avoided if we choose other material with higher yield stress limit.

c) The contact between tube and tubesheet is insured.

d) The stresses obtained in the tube at maximum pressure condition, 3000 bar after 3 second, are higher (694 MPa) and close to the rupture stress of material.
Information tests on yield/rupture stress value in other tubes are available. A large dispersion of these values with respect to the one used in this work were found.
The incidence of these values in the results will have to be studied.

e) The final radial displacement of the tube is 0.2516 mm while the initial radial clearence is 0.175 mm (44 % lower). If the expansion pressure is lower, the field stress in the joint will be lower too.
Recalculation should be done to determine wether the contact is ensured with a lower expansion pressure.

f) The influence of welds in the welded tube and the fractomechanic behaviour would be analized on view that the 90 % of the cases the crack appears in the weld. They aren't the objetives of this work.

SYMBOLS AND ABBREVIATIONS

σ	stress
p	pressure
U	ratio of radii (outside/inside)
μ	Posson's ratio
δ	displacement

INDICES

ft	yield limit of tube
fp	yield limit of tubesheet
t	tube
p	tubesheet
c	circunferential direction

REFERENCES

[1] Balcke-Durr. Sonderdruck A31e. Hydraulic Expansion, a new method for anchoring of tubes.

[2] FINITE ELEMENT PROGRAM NISA
Version 92.0 EMRC

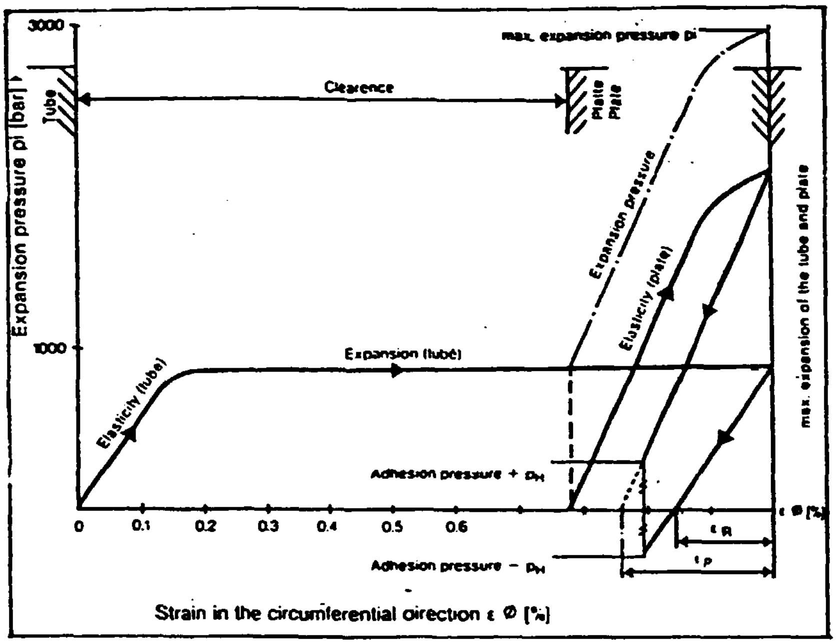


FIGURE 1 Principle of hydraulic expansion

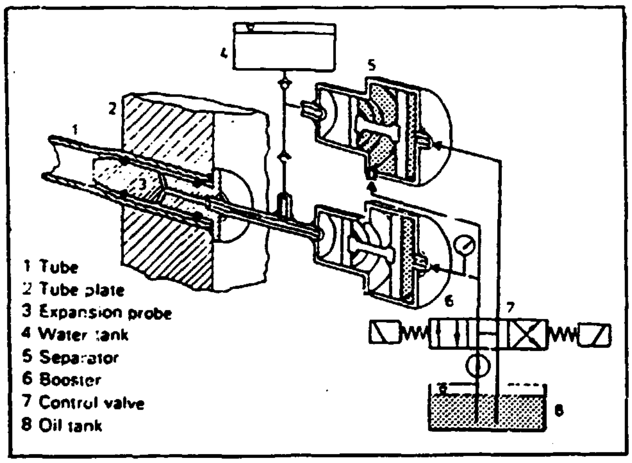


FIGURE 2 Layout of tube expander

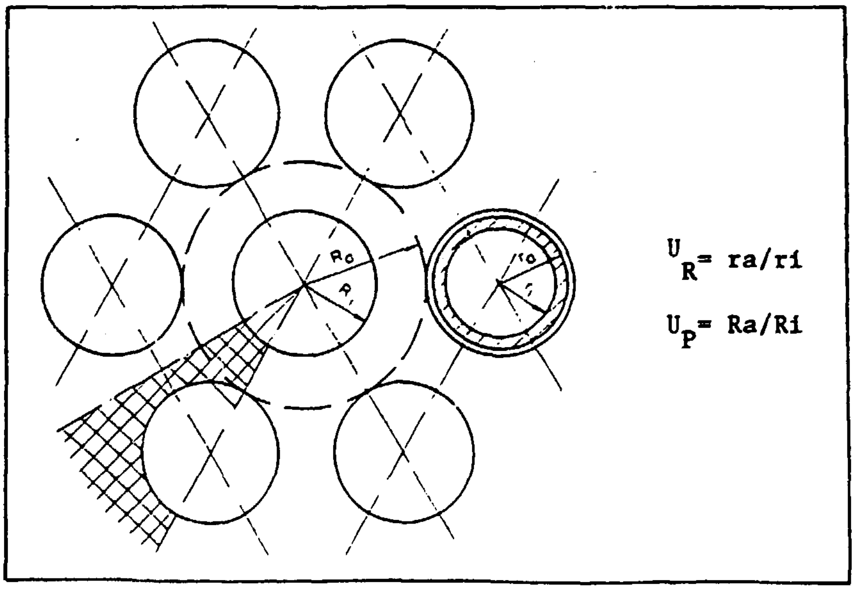
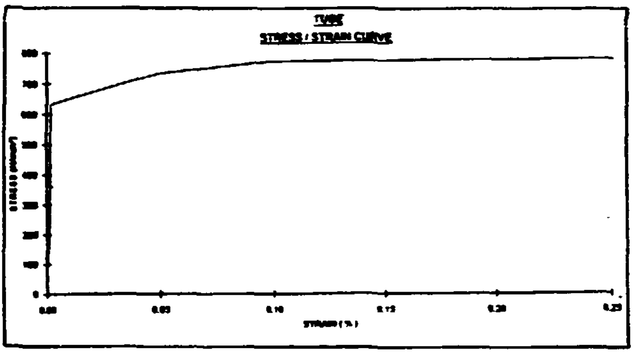


FIGURE 3 Plate geometry



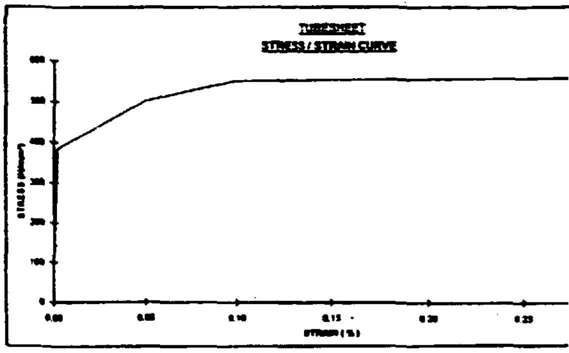


FIGURE 4a Stress-strain curve for tube

FIGURE 4b Stress-strain curve for tubesheet

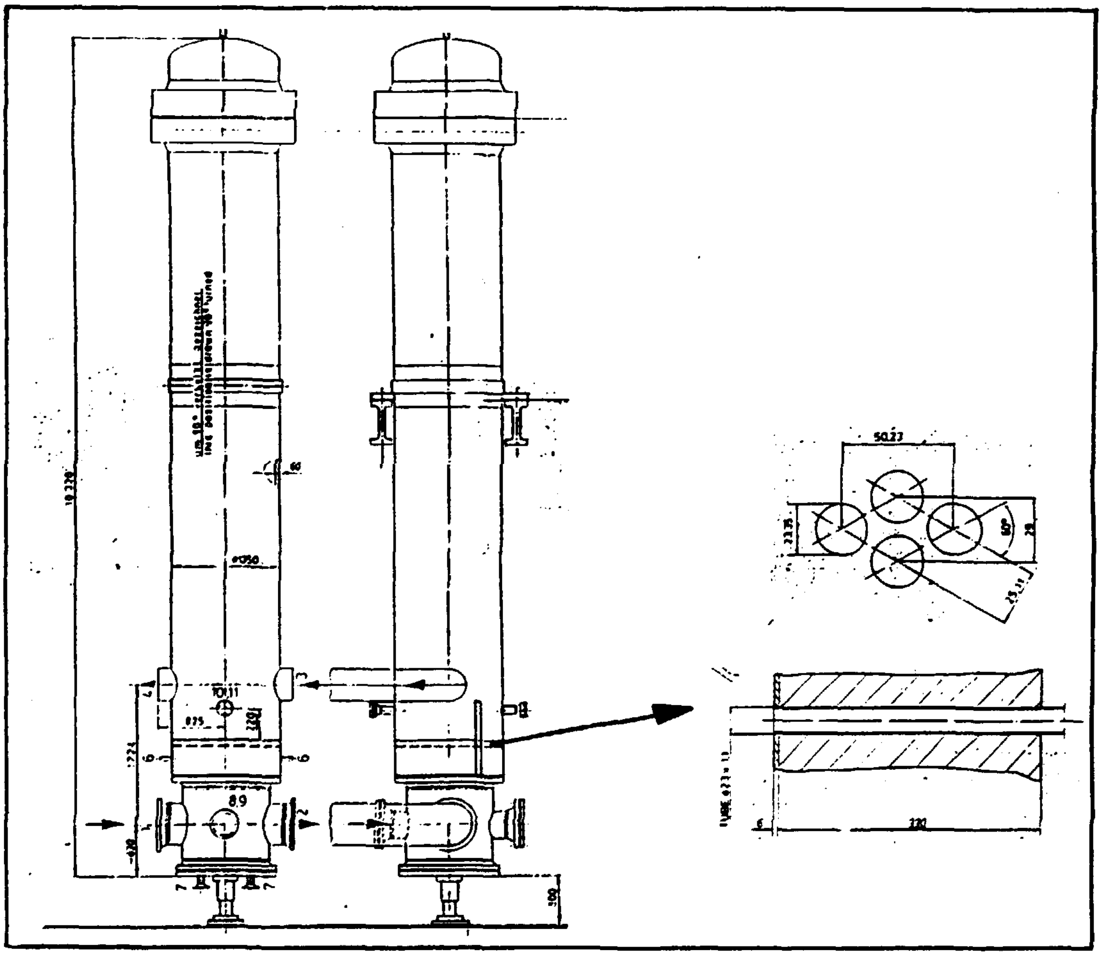


FIGURE 5 Geometry detail of component

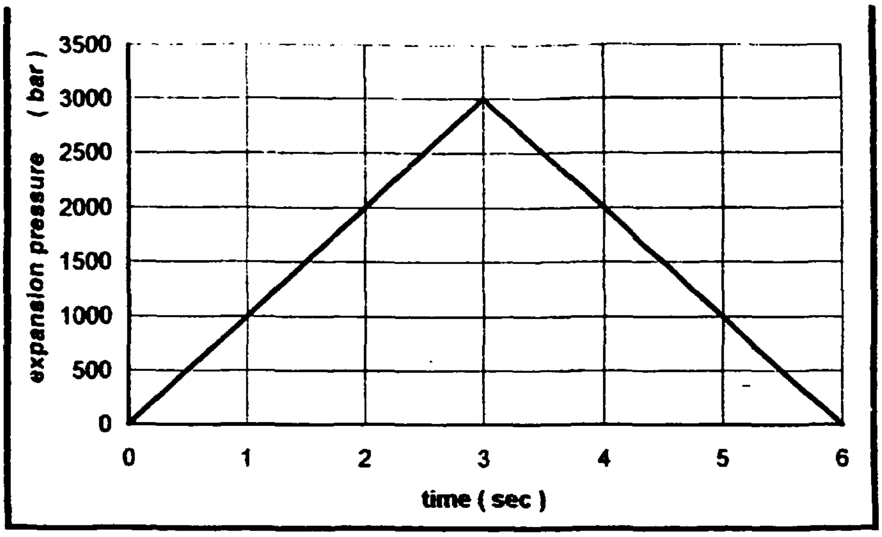


FIGURE 6 Curve of expansion pressure vs. time

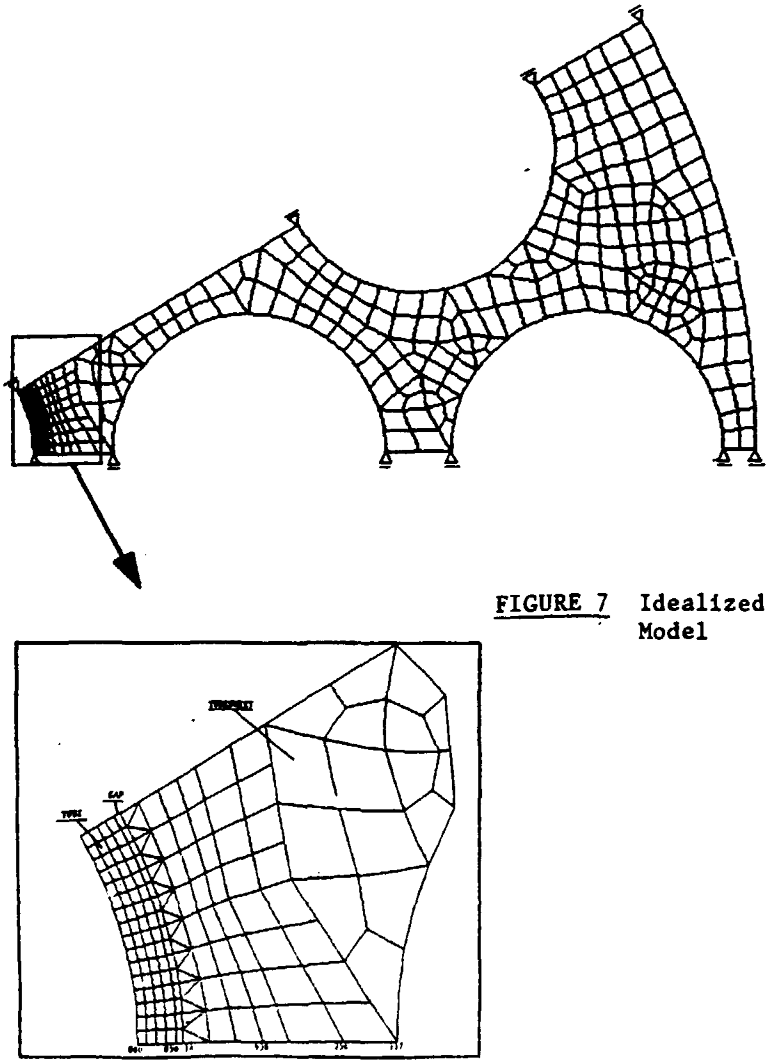


FIGURE 7 Idealized Finite Element Model

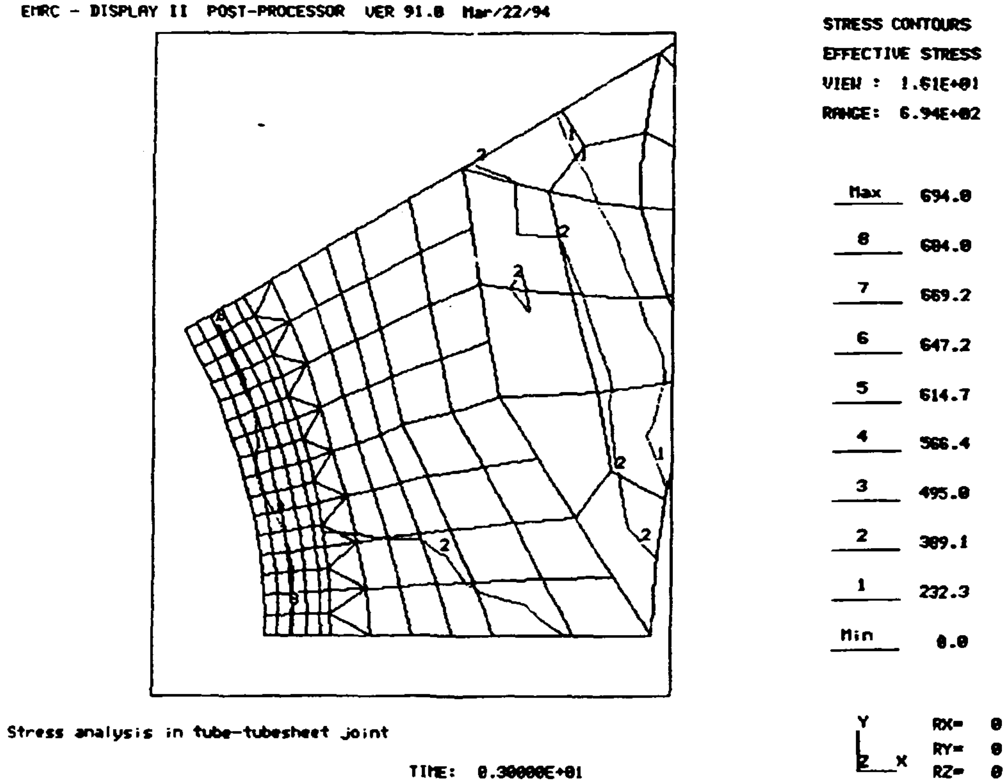


FIGURE 8a Efective stress of Von Mises (maximum pressure, t= 3 sec.) MPa.

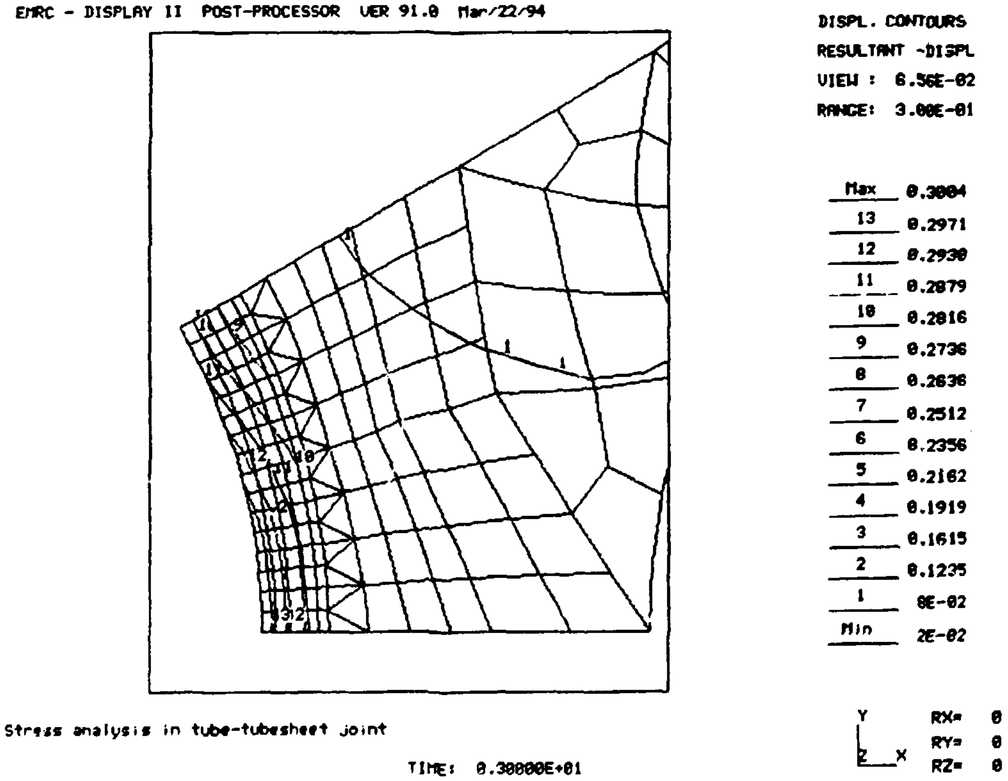


FIGURE 8b Radial displacement (maximum pressure, t= 3 sec.) mm.

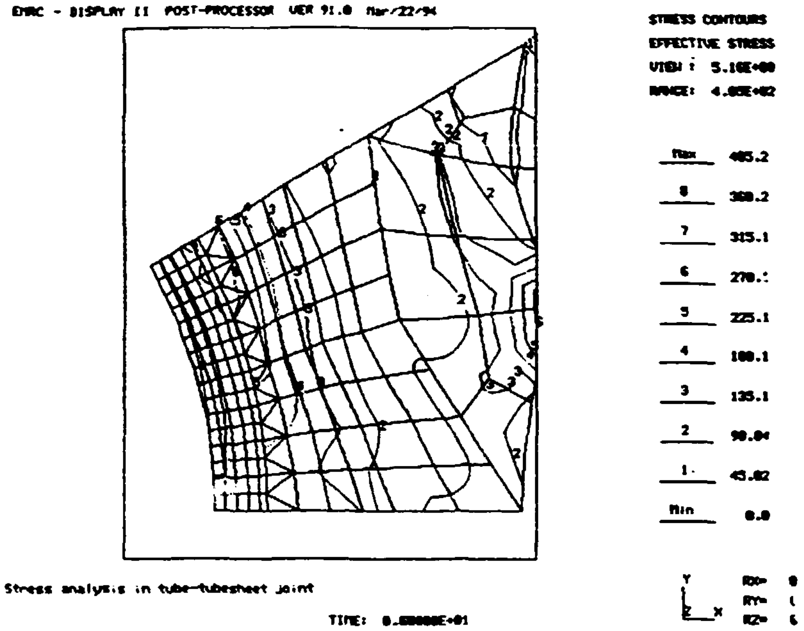


FIGURE 9a
Efective stress of Vo
Mises (t= 6 sec.) MPa

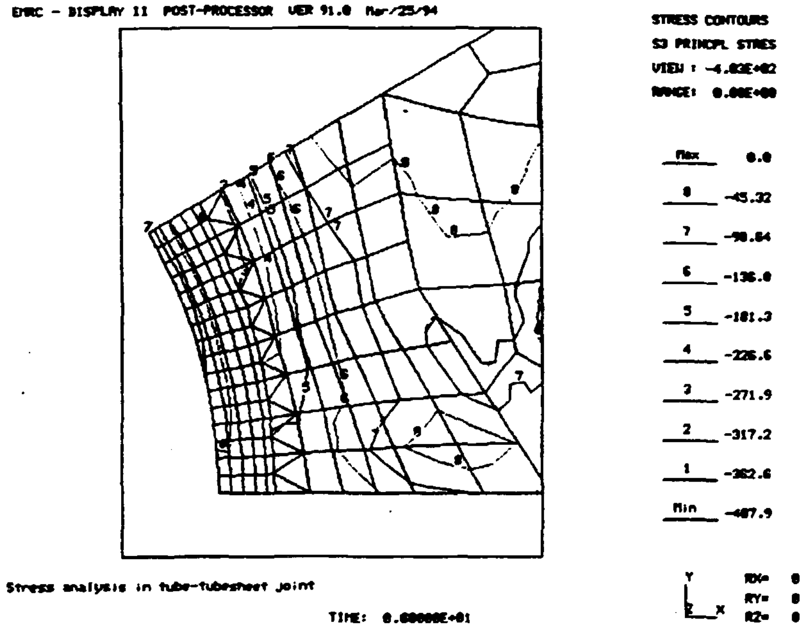


FIGURE 9b
Circunferential stre
(t= 6 sec.) MPa

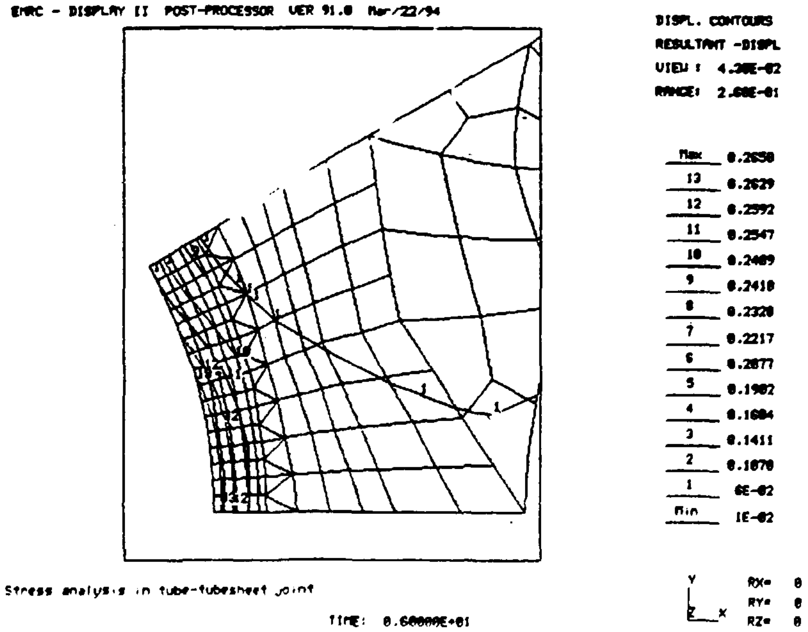


FIGURE 9c
Radial displacement
(t= 6 sec.) mm

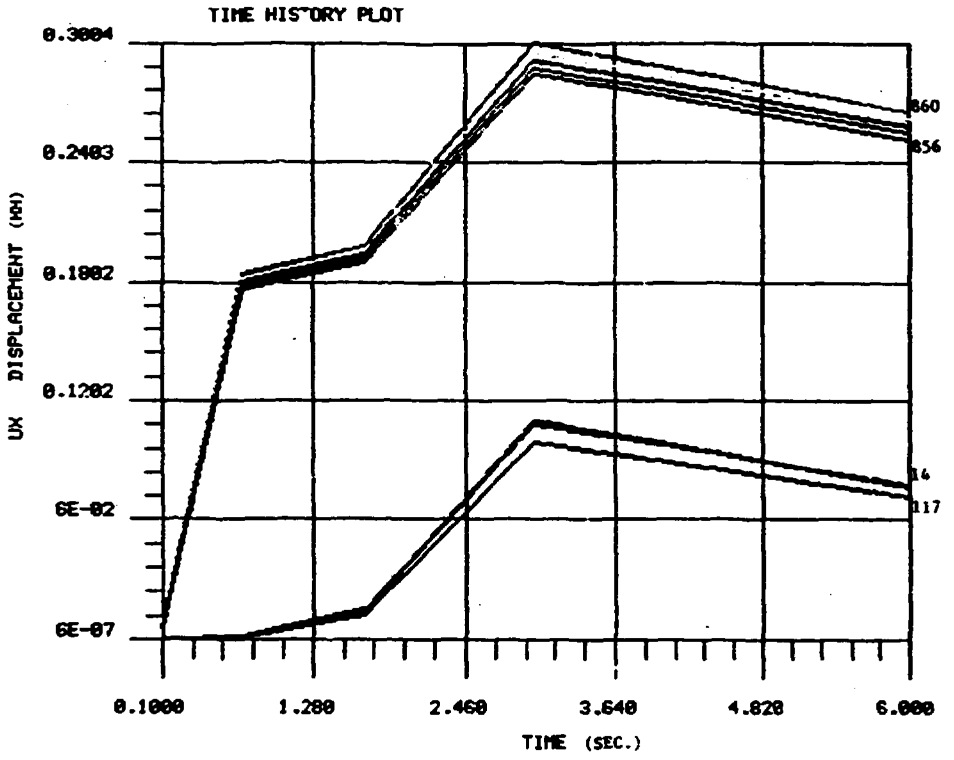


FIGURE 10 a Time history of the radial displacement (The numbers indicated in the curves correspond to the nodes illustrated in figure 7)

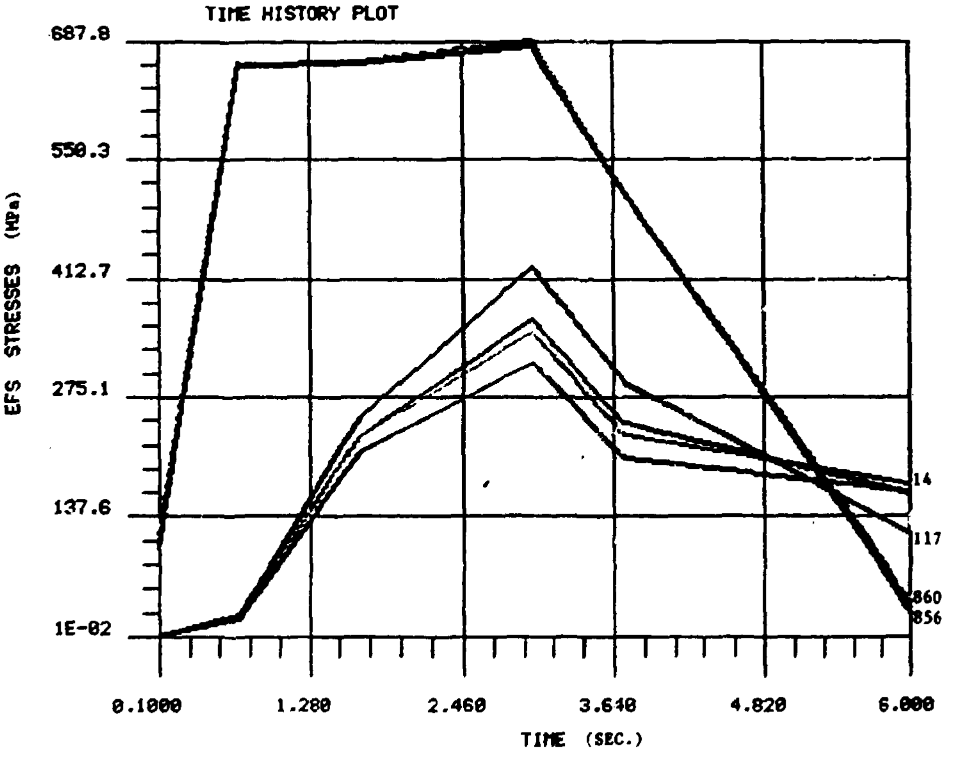


FIGURE 10b Time history of the efective stress of Von Mises (The numbers indicated in the curves correspond to the nodes illustrated in figure 7)

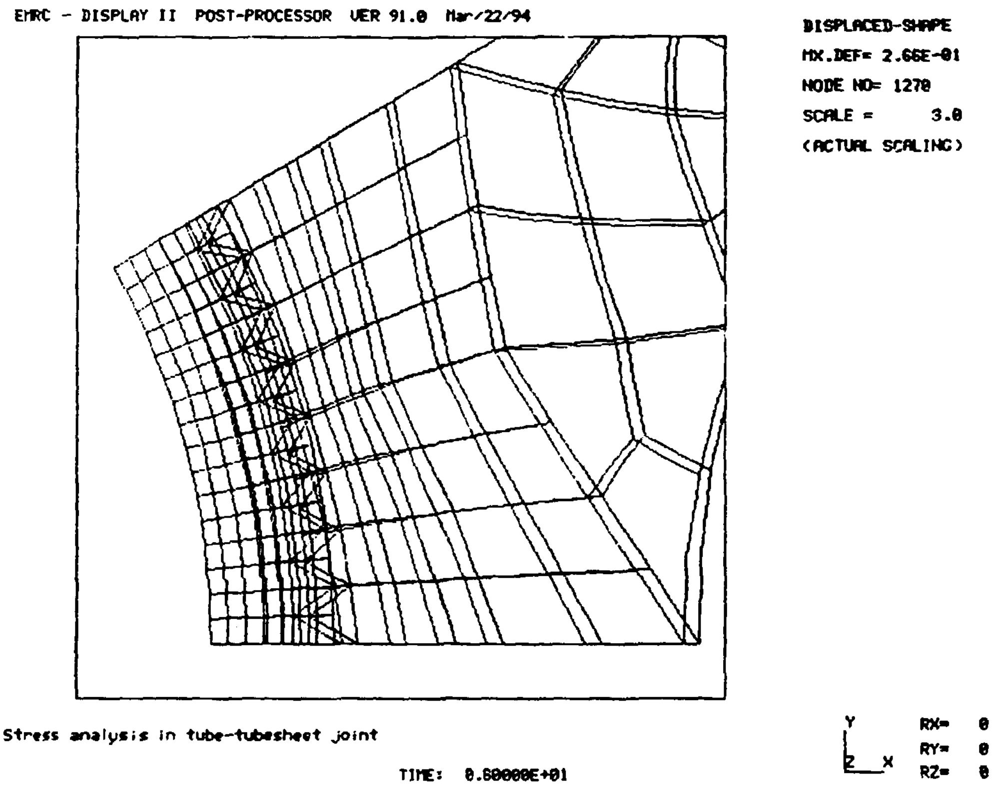


FIGURE 11 Idealized and deformed structure