Burst Strength of Type 304L Stainless Steel Tubes Subjected to Internal Pressure and External Forces

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Thirteenth International Specialty Conference on Cold-Formed Steel Structures
St. Louis, Missouri U.S.A., October 17-18, 1996

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by

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ABSTRACT

The findings of an investigation concerning the burst strength of cold-formed Type 304L stainless steel tubes subjected to internal pressure and static external forces are reported in this study. The use of cold-formed stainless steel longitudinally welded tube in pressurised processes in industry are limited due to the belief that seamless tubes have superior resistance to internal pressure.

The primary objective of this study was to experimentally and theoretically describe the failure criteria for thin-walled longitudinally welded Type 304L stainless steel tubes subjected to internal pressure and static external point loads and torsion loads. Due to the diversity of the pipe manufacturing process, problem areas which were most likely to cause failures were identified. A microscopic study was done of the weld region where failure was expected in order to support the test results.

It was found that cold-formed longitudinally welded Type 304L stainless steel tubes could attain very high bursting pressure values and could compete with seamless tubes in this respect. It was also found that the internal pressure was the most important criteria in tube failure and that the effect of static external forces could be neglected to a certain extent.

GENERAL REMARKS

Longitudinally welded stainless steel tubes have many uses in various industrial applications. Certain perceptions regarding welded tubes pertain because of the weld integrity and changing material properties in the heat affected zone (HAZ).

Due to the difference in the mechanical behaviour of stainless steels compared to carbon and low alloy steels, research on the behaviour of longitudinally welded stainless steel tubes subjected to internal pressure and external forces is necessary. Aspects such as the gradual

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yielding and work hardening of stainless steel has an impact on the failure resistance of thin-walled tubes.

Local instability and excessive stresses in tubes often occur because of internal pressure spikes and external forces caused by process variations and movement in pipe networks. Such loads are applied to tubes during actual operation resulting in strain due to a combination of factors. Accumulation of these factors could lead to the failure of a pressurised system.

OBJECTIVE OF THE INVESTIGATION

The ANSI/ASME B31.3 design specification for Chemical Plant and Petroleum Refinery Piping[1] do not take into account the effect of direct forces on a straight metallic pipe subjected to internal pressure. It does allow for the calculation of moments in bends and in branch connections to be able to calculate flexibility stresses.

The objective of this research was to determine the effect of externally applied forces on the failure pressure of thin-walled longitudinally welded stainless steel tubes. Tubes used in networks are supported by pipe support systems and hangers. Investigating the effect of externally applied forces on pressurised tubes, results in structural strength determination of tubes tested to failure.

A method to reach the objective was developed which consisted of a series of destructive pressure tests done on Type 304L stainless steel tubes while subjected to different external force load conditions. These external load conditions simulates practical load conditions experienced in pipe networks.

Results obtained were then evaluated and compared to existing design criteria in a bid to analyse the effect of the different individual and combined forces.

THEORETICAL BACKGROUND

The theoretical analysis of tube failure due to pressure is very complex as it is based on various elastic and inelastic theories. Historical elasticity theories such as those of Rankine, Saint-Venant, Tresca, Guest, Haig, Von Mises and the Mises-Hencky theory do not coincide.[2,3] Inelastic theory based on these varying elasticity theories becomes important when doing destructive testing as failure occurs during inelastic instability. Failure of a structure because of large inelastic deformation takes place because of a progressive strain stage ranging from homogenous strain to heterogeneous strain.[4] Adding external forces to this already complex matter further elevates the problem. This can clearly be seen in the literature available on tube failure.[2,3]

Cooper[5] determined that yielding could be calculated from the tensile strength of the material. The maximum burst pressure could not be calculated from the tensile strength. Tubes under pressure are loaded in multi-axial directions which differs from the single axial tension test used to determine yield strength. Cold working properties of a material need to be taken into consideration especially where the forming process includes cold working as it has an effect on material characteristics.[6]

This investigation covered the following four loading conditions applied to thin-walled tubes where the wall thickness, t, is less than D/6:

* Tube subjected to internal pressure;
* Tube subjected to internal pressure with the ends constrained;
* Tube subjected to internal pressure and a point load applied to the midpoint of the tube;
* Tube subjected to internal pressure and torsional load applied to one end of the tube;

Loading condition 1: Tube subjected to internal pressure.

Maximum attainable pressure before failure in straight thin walled tubes should be predictable from first principals. When considering the first loading condition, stresses in the tangential direction are given by Equation 1.

$$\sigma_1 = \frac{pd}{2t} \tag{1}$$

while stresses in the longitudinal direction are given by Equation 2.

$$\sigma_2 = \frac{pd}{4t} \tag{2}$$

where:

σ_1	tangential stress
σ_2	longitudinal stress
p	internal pressure
d	outside tube diameter
t	tube wall thickness

Radial stresses were ignored because of the thin wall thickness of the tubes tested. The internal pressure, wall thickness, tube diameter as well as the tangential and longitudinal stress components are respectively defined in Figure 1 and Figure 2. Further manipulation of the tangential stress equation results in the Lamè equation as used for minimum wall thickness calculations in ANSI/ASME B31.3.[1] Graphical illustrations of the various load conditions are shown in Figure 3 while calculations of the maximum allowable working pressure, the calculated failure pressure and the burst pressure can be found in Table 1.

Loading condition 2: Tube subjected to internal pressure with end constraints.

Investigation on independent axial loads together with internal pressure on tubes by Dubey,[7] revealed the sensitivity of the critical stress value when considering variations at the point of local stress. Kitagawa[8] found that uniform strain could be achieved by applying the axial force and the internal pressure so that the relationship between the axial and circumferential stress components stay constant. Maximum stress as a result of pressure in the longitudinal direction is given by Equation 3.

$$\sigma_{maks} = \left(\frac{2G}{M\beta}\right)^{\frac{1}{M}} \left(\frac{\sigma_y}{\sqrt{1-\beta+\beta^2}}\right)^{\frac{(M-1)}{M}} \tag{3}$$

while the maximum stress as a result of the axial force in the longitudinal direction is given by Equation 4.

$$\sigma_{maks} = \left(\frac{6G}{M(2-\beta)}\right)^{\frac{1}{M}} \left(\frac{\sigma_y}{\sqrt{1-\beta+\beta^2}}\right)^{\frac{(M-1)}{M}} \tag{4}$$

where:

$\beta = \sigma_1/\sigma_2$
σ_1 = tangential stress
σ_2 = longitudinal stress
M = material dependant material constant
G = elastic shear modulus
G = E/3
E = elastic modulus
σ_y = yield stress

Bi-axial testing using an axial load and internal pressure by Lefebvre[9] revealed that Poisson's ratio differs in the elastic and inelastic region. It was found that satisfactory results could be obtained using a Poisson's ratio value of 0.48 for Type 304L in the inelastic region. True longitudinal stress (σ_a), tangential stress(σ_t), and the true radial stress (σ_r) are given by Equation 5 respectively.

$$\begin{aligned} \overline{\sigma}_a &= \frac{F}{A} + \frac{p\pi D_i^2}{4A} \\ \overline{\sigma}_t &= \frac{pD_m}{2t_v} \\ \overline{\sigma}_r &\approx -\frac{p}{2} \approx 0 \end{aligned} \tag{5}$$

If the load conditions dominate the test, failure will be determined by the maximum pressure. On the other hand, if displacement is controlled, failure will occur under reducing pressure and at strain levels exceeding the strain at maximum pressure levels.[10] This relates to the following: The tube will fail in the tangential direction except if the axial force in the longitudinal direction is larger than half of the tangential stress. This will result in failure in the longitudinal direction It is expected that the failure pressure will be less than that of loading condition 1 but that strain levels should be more because of the constraints at the end points. It was assumed that the constraint had no effect on the load conditions and the pressure calculations were done accordingly as can be seen from Table 3.

Loading condition 3: Tube subjected to internal pressure and a point load applied to the midpoint of the tube.

Work done on bent tubes subjected to different load conditions,[11] the bending of tubes[12] as well as the determining of critical bending moments[13] led to further investigation by Watanaba and Ohtsubo.[14] They investigated non-elastic flexibility and strain concentrations in pipe bends in creep with plasticity effects, while Calladine's investigation led to the use of the Raleigh method for calculations of thin walled elastic tubes.[15] Darlaston and Harrison[16] reviewed the above mentioned work together with investigations done by Thompson and Spence[17] as well as research done by Corona and Kyriakides[18] and came to the conclusion that linear elastic failure modes are not valid for structures with defects or non-homogenous inserts in the weld region.

Kiefner et al[19] suggested that Equation 6 be used for the determination of the failure pressure:

$$P_f = \frac{\sigma_f t}{R_m}\left[\frac{\frac{t}{d_f} - 1}{\frac{t}{d_f} - \frac{1}{m}}\right] \qquad (6)$$

where:

t	tube wall thickness
R_m	average tube radius
d_f	defect depth
σ_y	yield stress
σ_u	ultimate stress
σ_f	flow stress defined as:

$$\sigma_f = 0.5 \quad (\sigma_y + \sigma_u) \qquad (7)$$

and where the Folias expansion factor, m, is defined as:

$$m = \left(1 + 1.05\frac{C^2}{R_m t}\right)^{1/2} \qquad (8)$$

with 2c equal to the length of the defect.

It was found that the load deflection characteristics of the structure as a whole and the position of the defect in terms of the bending plane had very little effect on the results. The combined internal pressure and point load tests resulted in the following design postulate: If axial stress as a result of bending does not exceed half of the calculated tangential stress, then failure as a result of internal pressure can be calculated by using Equation 6.

Straight tubes and tubes subjected to bending perceive different stress conditions. Tubes subjected to a point load, result in local oval forming which influences tube strength. Oval forming and the flattening effect of tubes experienced during bending was first researched by Brazier and therefore called the Brazier effect.[20] An increase in internal pressure, together with a point load condition, result in changing sectional properties of the tube thus, leading to bending of the tube and ultimately failure.

Loading condition 4: Tube subjected to internal pressure and a torsional load applied to one end of the tube.

The stress conditions associated with tubes subjected to torsion and internal pressure is investigated. Tests by Chaudhuri and Abu-Arja[21] compare the effects of separate torsional loads and internal pressures to the effect of combined internal pressure and torsional load. Other researchers[22] used stress tensors to determine the precise plastic strain of thin-walled tubes when subjected to various load conditions. Failure is determined by both the effective stress and the maximum stress, where the effective stress determines the start of failure while the growth of the failure mode depends on the maximum tensile stress.[23] Research[24] showed that the reduction of strain resistance of a material could be attributed to the sudden change in stress conditions, that result as soon as the torsional stress is superimposed on the tensile stress.

Internal pressure and external torsion are independent quantities while stability may change to instability without any of the values reaching a maximum. Failure criteria for large inelastic deformations cannot only be based on the stress conditions but should be based on an analysis of the complete structure to determine when inelastic instability will occur for a specific load condition.[4]

When considering a tube of length l with a radius of r, subjected to an internal pressure p and a force applied to one end, it creates a moment M_d as a result of the force-couple $2Fr_0$ where r_0 is kept constant. It is assumed that $t \ll r$ and that the length to radius relationship is such that any end effects are neglected. Elastic strain is ignored so as to simplify the calculations.

The true stresses in the tubes are given by Equation 9.

$$\begin{aligned} \sigma_r &= 0 \\ \sigma_t &= \frac{pr}{t} \\ \sigma_a &= \frac{pr}{2t} \\ \tau_{ta} &= \tau_{at} = \frac{Fr_0}{\pi r^2 t} \end{aligned} \tag{9}$$

while the derived stresses are given by Equation 10.

$$
\begin{aligned}
S_r &= -\frac{pr}{2t} \\
S_a &= \frac{pr}{2t} \\
S_t &= 0 \\
S_{ta} &= S_{at} = \frac{Fr_0}{\pi r^2 t}
\end{aligned}
\tag{10}
$$

where F represents the external loads.

Strain increments are defined by Equation 11 as:

$$
\begin{aligned}
d\varepsilon_r &= \frac{dt}{t} \\
d\varepsilon_t &= \frac{dr}{r} \\
d\varepsilon_a &= \frac{dl}{l} \\
d\varepsilon_{ta} &= d\varepsilon_{at} = \frac{rd\phi}{2l}
\end{aligned}
\tag{11}
$$

When the tube is turned through an angle of ϕ while under load conditions p and F and external effect, δp and δF results. If the strain is relatively small, stability will be kept for small particles if:

$$dpdl\pi r^2 + dpdr2\pi r1 + dF2r_o d\phi > 0 \tag{12}$$

By making use of the Von Mises flow rules and substituting in Equations 9, 10, 11 and 12 it can be shown that:

$$\left(dp\pi r^2 - dp2\pi r1\frac{r}{21} + dF2r_o\frac{6Fr_o}{r}\right) dl > 0 \tag{13}$$

By rewriting the equation in terms of stresses it relates to Equation 14 at the point of instability.[4]

$$\frac{2}{3}\sigma_{eff} d\sigma_{eff} = \left(\frac{pr}{t}\right)^2\frac{dr}{r} + \frac{1}{2}\left(\frac{pr}{t}\right)\frac{dl}{l} - 2\left(\frac{Fr_o}{\pi r^2 t}\right)^2\left(\frac{dr}{r} - \frac{dl}{l}\right) \tag{14}$$

The fundamental relationship for the theoretical analysis of inelastic strain consist of relatively simplistic equations when compared to the complex behaviour of the actual material.

EXPERIMENTAL PROCEDURE

Longitudinally welded Type 304L tubes were manufactured from stainless steel sheets formed by cold rolling and tungsten inert gas welding in the longitudinal direction. The tubes were manufactured according to ASTM A269 specification.[25] Main production processes consist of weld treatment, annealing, pickling and passivating. Testing procedures include of both visual end Eddy current testing. Test specimens from two different tube manufacturers were used as their heat treatment process varies. One manufacturer uses in line heat treatment while the other batch heat treats the finished product.

Tube specimens were cut into 2 m lengths to eliminate strengthening end effects created by the end caps used. Two sets of tests were carried out on the longitudinally welded Type 304L tubes using a tube wall thickness of 1,2 mm and 1,6 mm. Square 100 mm by 100 mm end caps with a thickness of 5 mm were prepared to seal the ends of the test tubes. Pre-heated end caps were arc welded onto the tube ends using a Type 316L welding rod.

Test specimens were subjected to the four different loading conditions as mentioned before. Tubes were left free from any constraints during loading condition 1 with the only load consisting of internal pressure. Internal pressure was produced using a converted hydraulic hand pump using water as pressure medium. During load condition 2, both endplates were clamped to rigid structures to form a fixed end connection while the internal pressure was increased.

Load condition 3 was achieved by clamping the closed end plate while placing the other endplate on a roller. Point loads were applied using weights and a load frame. By using a reinforced bicycle wheel addapted to accept endplate clamps, a torsional load could be applied to achieve load condition 4. The specific loads applied to each of the specimens are tabulated in Table 1.

Pressure measurements were taken using a pressure transducer connected in line between the pump and the test specimen. Deflection measurements on critical points were made using displacement transducers. Wall thickness and diameter measurements were taken before testing was done using a Nikon profiloscope. The average of five measurements were taken. The same measurements were taken again after failure and the results compared to determine the increase in strain. Failure modes and failure position in relation to the weld was investigated on each specimen.

RESULTS

Results of the microscopic study revealed variations between the parent material and the heat affected zone.

Failure in general occured at higher than anticipated pressures. Experimental results for each of the loading conditions are given in Tabels 2 to 5. Comparisons made between design, anticipated failure and experimental failure pressure are given in these tables.

In Figures 4, 5, 6 and 7 pressure - time curves for a 76.2 mm x 1.2 mm tube are compared for each of the different loading conditions. These graphs were constructed from the original data obtained during the tests without any refinement.

Excessive deflection caused by the point load was recorded during load condition 3. Deflection occured only after commencement of pressurisation. The torsional load applied in load condition 4 caused rotation such an extent that full rotation of 360 degrees was found in some cases.

DISCUSSION OF RESULTS

The microscopic study done on tubes from two different manufactures revealed that in line annealing did not prove to be as efficient as batch annealing It was recommended that the specific manufacturer re-evaluates the annealing procedure used during the manufacturing process.

Failure occurred at between 30% to 35% higher than calculated for load condition 1, using the ultimate stress values for Type 304L stainless steel. in the calculations. Failure pressure calculations were based on Equation 1.

The end constraints used in load condition 2 had very little effect on the final failure pressure when comparing results to that of load condition 1. The failure pressures were in general 10% higher than the calculated failure pressures while strain levels were higher as predicted.

Load condition 3 produced good results as failure again occurred at higher than anticipated pressures. Excessive deformation and deflection occurred before failure. The load deflection characteristics had very little effect on the results.

Failure pressures obtained during load condition 4 proved that tubes subjected to internal pressure and external torsional loads would not fail prematurely. Specimen Lot 8-4 failed at a lower than expected value which was still double the design value.

CONCLUSIONS

The literature reveals different approaches in the theoretical analysis of tube failure subjected to internal pressure and various external forces. The investigation proved that longitudinally welded Type 304L stainless steel tubes could be used in situations where severe loading conditions were involved, resisting failure equal to the same levels expected from seamless tubing.

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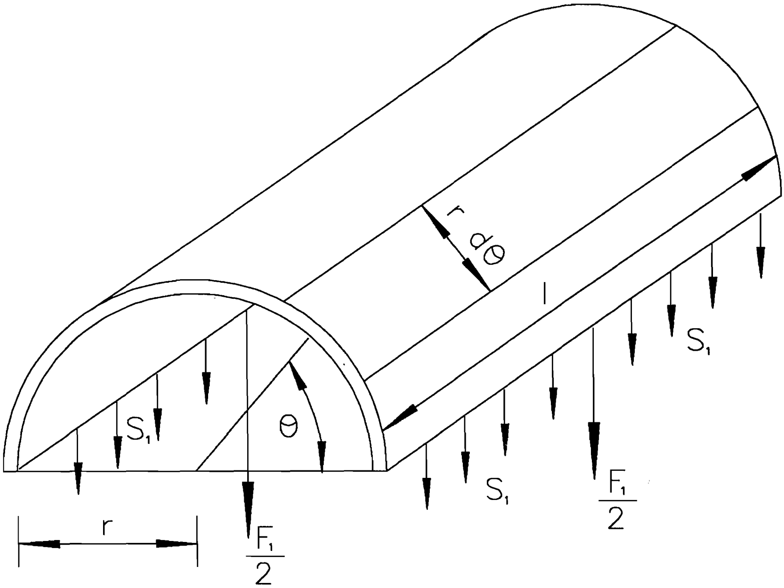

FIGURE 1 **Tangential stress condition.**

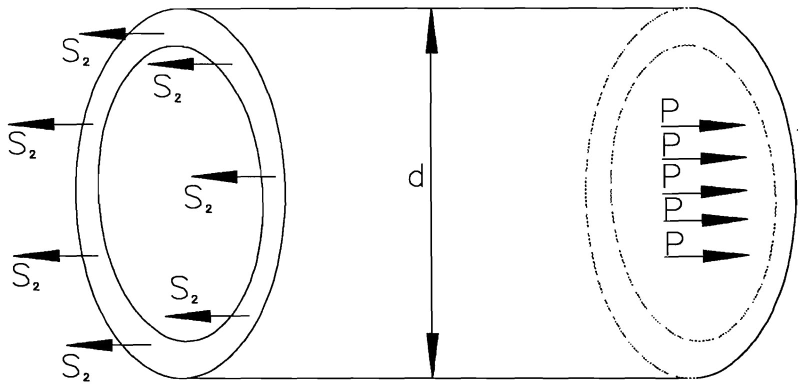

FIGURE 2 **Longitudinal stress condition.**

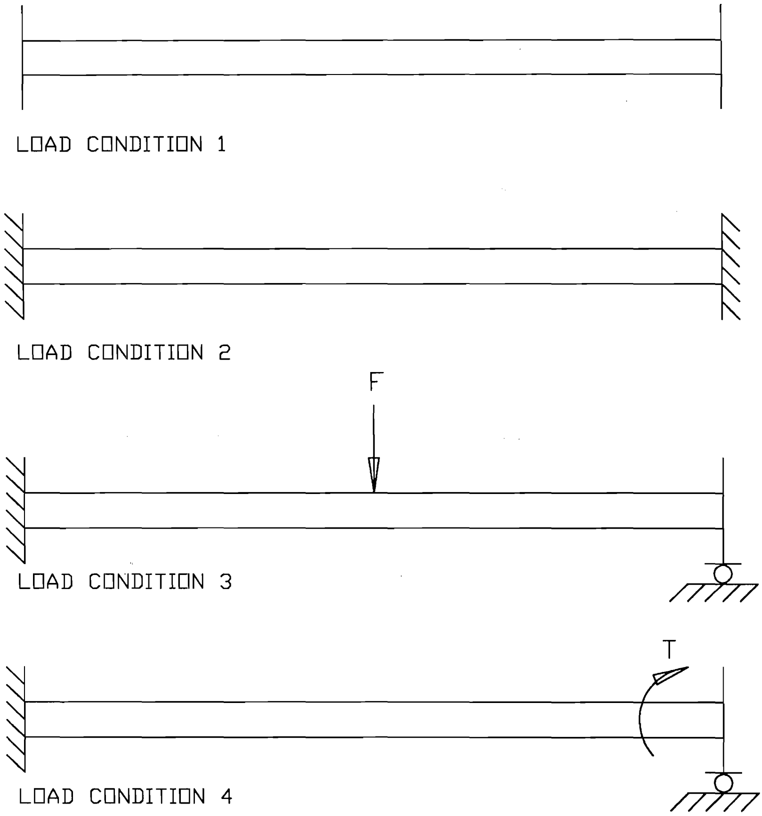

FIGURE 3 Load conditions.

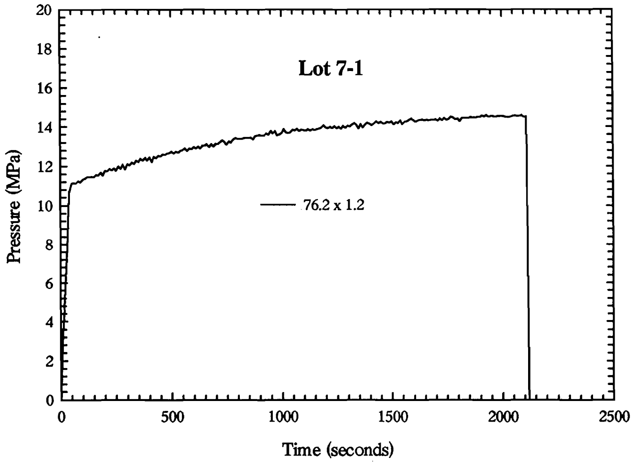

FIGURE 4 Pressure - Time Graph for load condition 1.

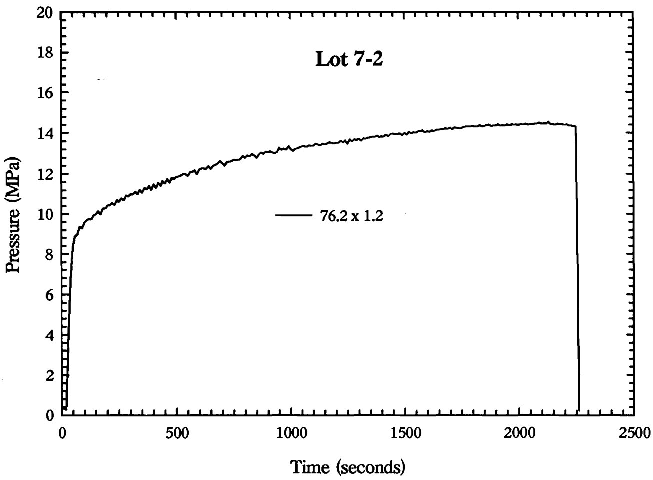

FIGURE 5 Pressure - Time Graph for load condition 2.

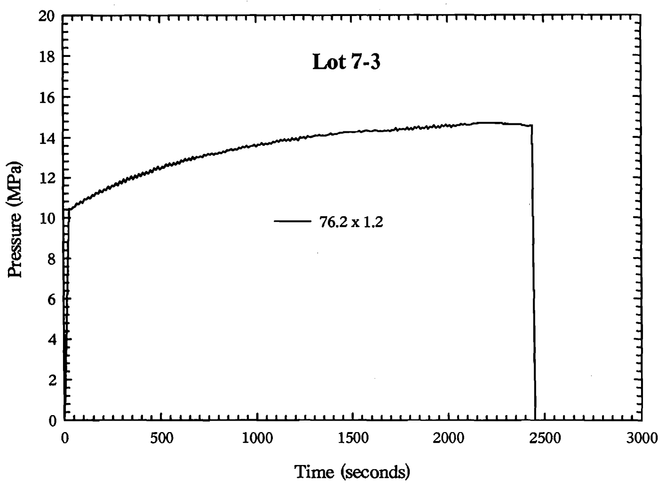

FIGURE 6 Pressure - Time Graph for load condition 3.

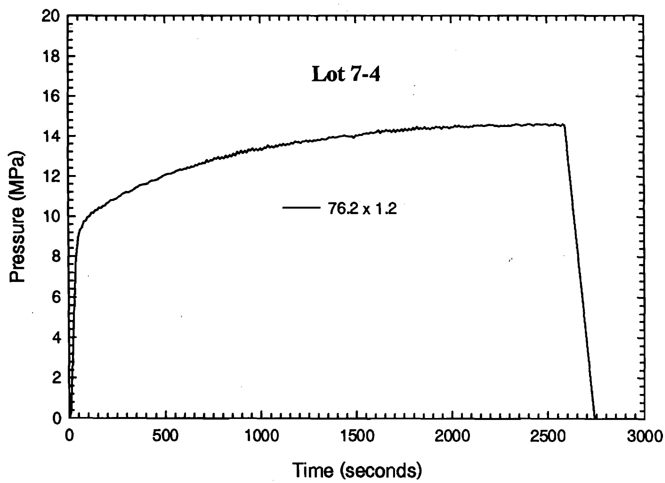

FIGURE 7 Pressure - Time Graph for load condition 4.

TABLE 1 Point loads and Torsional loads

TUBE SIZE (mm)	POINT LOAD (N)	TORSIONAL LOAD (Nm)
38.1 x 1.2	343	103
38.1 x 1.6	441	132
50.8 x 1.2	637	192
50.8 x 1.6	838	250
63.5 x 1.2	981	294
63.5 x 1.6	1226	368
76.2 x 1.2	1471	441
76.2 x 1.6	1717	515

TABLE 2 Pressure comparison between calculated and experimental burst pressure for load condition 1.

Specimen	Tube Size (mm)	Max Working Pressure (MPa)	Max Calculated Pressure (MPa)	Experimental Burst Pressure (MPa)
LOT 1-1	38.1 x 1.2	11	30.5	41.41
LOT 2-1	38.1 x 1.6	14.6	40.7	55.86
LOT 3-1	50.8 x 1.2	8.2	22.9	29.5
LOT 4-1	50.8 x 1.6	11	30.5	39
LOT 5-1	63.5 x 1.2	6.6	18.3	23.5
LOT 6-1	63.5 x 1.6	8.8	24.4	33.4
LOT 7-1	76.2 x 1.2	5.5	15.2	18.9
LOT 8-1	76.2 x 1.6	7.3	20.3	26.7

TABLE 3 Pressure comparison between calculated and experimental burst pressure for load condition 2.

Specimen	Tube Size (mm)	Max Working Pressure (MPa)	Max Calculated Pressure (MPa)	Experimental Burst Pressure (MPa)
LOT 1-2	38.1 x 1.2	11	30.5	33
LOT 2-2	38.1 x 1.6	14.6	40.7	43
LOT 3-2	50.8 x 1.2	8.2	22.9	23
LOT 4-2	50.8 x 1.6	11	30.5	22
LOT 5-2	63.5 x 1.2	6.6	18.3	26.5
LOT 6-2	63.5 x 1.6	8.8	24.4	27
LOT 7-2	76.2 x 1.2	5.5	15.2	15.9
LOT 8-2	76.2 x 1.6	7.3	20.3	22

TABLE 4 Pressure comparison between calculated and experimental burst pressure for load condition 3.

Specimen	Tube Size (mm)	Max Working Pressure (MPa)	Max Calculated Pressure (MPa)	Experimental Burst Pressure (MPa)
LOT 1-3	38.1 x 1.2	11	30.5	36
LOT 2-3	38.1 x 1.6	14.6	40.7	44
LOT 3-3	50.8 x 1.2	8.2	22.9	27.5
LOT 4-3	50.8 x 1.6	11	30.5	34
LOT 5-3	63.5 x 1.2	6.6	18.3	19.5
LOT 6-3	63.5 x 1.6	8.8	24.4	26
LOT 7-3	76.2 x 1.2	5.5	15.2	15
LOT 8-3	76.2 x 1.6	7.3	20.3	23

TABLE 5 Pressure comparison between calculated and experimental burst pressure for load condition 4.

Specimen	Tube Size (mm)	Max Working Pressure (MPa)	Max Calculated Pressure (MPa)	Experimental Burst Pressure (MPa)
LOT 1-4	38.1 x 1.2	11	30.5	34
LOT 2-4	38.1 x 1.6	14.6	40.7	42
LOT 3-4	50.8 x 1.2	8.2	22.9	31
LOT 4-4	50.8 x 1.6	11	30.5	33
LOT 5-4	63.5 x 1.2	6.6	18.3	20.5
LOT 6-4	63.5 x 1.6	8.8	24.4	25
LOT 7-4	76.2 x 1.2	5.5	15.2	14.5
LOT 8-4	76.2 x 1.6	7.3	20.3	15