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Burst pressure of super duplex stainless steel pipes subject to combined axial tension, internal pressure and elevated temperature

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ABSTRACT

The burst pressure of super duplex stainless steel pipe is measured under combined internal pressure, external axial tension and elevated temperature up to 160 °C. The experimental results are compared with existing burst pressure prediction models. Existing models are found to provide reasonable estimate of the burst pressure at room temperature but significantly over estimate the burst pressure at elevated temperature. Increasing externally applied axial stress and elevated temperature reduces the pressure capacity.

1. Introduction

An understanding of the burst pressure of pipes and pressure vessels is of particular importance to the oil and gas, chemical and nuclear industry. Consequently the determination of the burst pressure of tubulars has been examined over the years by many researchers. Law and Bowie [1] provided a summary of existing burst pressure prediction models and compared the accuracy of the models with test results on some API X grade steel used in oil and gas production system. Many of the existing models are for tubulars subjected to only internal pressure, with very few incorporating the effect of external axial force on burst pressure.

Production tubulars in oil and gas wells and subsea pipelines are, in general, subjected, in service, to a combination of internal pressure, external pressure, external axial force and elevated temperature. The failure mechanism of a tubular subject to a combination of internal pressure and axial load can be different from that subject to just internal pressure [2]. For example, a tubular without any pre-existing cracks or defects and subject to internal pressure and external axial tension can fail by burst (i.e. rupture) or necking depending on the relative pressure to external axial stress. In order to improve the design and ensure better selection of materials, there is a need for detailed evaluation of burst pressure in the presence of axial stress and elevated temperature.

The analysis of the deformation and failure of metal tubes under combined axial tension and internal pressure is provided in Refs. [3–5] and a detailed analysis of the rupture and necking of tubulars is provided by Klever [2], Stewart et al. [6] and Paslay et al. [7]. None of these analyses include the effect of elevated temperature.

With increasing discovery of new oil and gas fields in high pressure – high temperature and corrosive environment, super duplex stainless steel is a suitable material for the transportation of the hydrocarbon due to its better corrosion resistance in comparison to low-alloy carbon steel. In the present paper, the burst pressure of super duplex stainless steel pipe subject to a combination internal pressure, external axial tension and elevated temperature is examined. Experimental tests are conducted using a mini pipe, which has been shown to produce reliable and accurate result and to be a cost effective alternative to burst tests on full-size pipes [8]. The experimentally determined failure envelope is compared with predictions by existing theoretical models.

1.1. Burst pressure estimates of tubulars under combined loading

Various burst pressure models exist in literature. Law and Bowie [1] have provided a summary of some of the models for tubulars subject to only internal pressure. Experimental results for pipes

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made from steel grades with high tensile to yield strength ratio shows that only a few of the models are reliable in predicting the burst pressure. A summary of some of the relevant burst pressure equations for tubulars subject to only internal pressure loading as well as for tubulars subject to combined internal pressure and external axial stress is provided.

Consider a circular cylindrical pipe with inner diameter D_i, outer diameter D_o, wall thickness t and subject to a combined internal pressure P_i, and external axial stress σ_a. We assume the pipe has closed ends, the wall thickness is much less than the diameter for thin-walled conditions to hold, i.e. $D_i/t > 20$, and that the tubular is made from a material with uniaxial 0.2% offset yield stress σ_Y and tensile strength σ_{uts}. The burst equations are sometimes expressed in terms of the mean diameter $D_m = 0.5(D_i + D_o)$,. Hereafter, the terms burst and rupture are used synonymously.

1.2. Burst pressure equation for pipes subject to only internal pressure

For a close-ended pipe made from an elastic/ideally plastic material obeying von Mises yield criterion, Hill [3] proposed the burst pressure

$$P_{b,1} = \frac{2}{\sqrt{3}}\sigma_Y \ln\left(\frac{D_o}{D_i}\right), \tag{1}$$

while pipeline design code by DNV [9] gave the burst equation as

$$P_{b,2} = \frac{2}{\sqrt{3}}\frac{2t}{D_m}\sigma_Y \tag{2}$$

which, surprisingly, is widely used in the design of subsea pipelines for assessing the pressure capacity. In fact, neither of these equations actually defines the burst pressure: Eq. (1) is the pressure required for through-wall plastic yielding while Eq. (2) is the pressure at the onset of plastic yielding at the pipe's inner wall. However, for a thin-walled pipe made from an ideally plastic material, once the wall thickness has completely undergone plastically yielding, the plastic strain increases rapidly with a very small increase in pressure leading to burst. This is why (1) gives a good estimate of burst pressure for thin-walled pipes, but not accurate for thick-walled pipe. In order to correct the anomaly, Nadai [10] suggested the use of tensile strength instead of the yield stress and thus proposed:

$$P_{b,3} = \frac{2}{\sqrt{3}}\sigma_{uts} \ln\left(\frac{D_o}{D_i}\right) \tag{3}$$

The effect of strain hardening is not included in any of the burst Eqs. (1)–(3).

Guided by experimental data on pressure vessels made from Q235-D and 20R mild steel, Faupel [11] proposed a burst pressure equation which incorporates the ratio between the yield stress and the tensile strength:

$$P_{b,4} = \frac{2}{\sqrt{3}}\sigma_Y\left(2 - \frac{\sigma_Y}{\sigma_{uts}}\right)\ln\left(\frac{D_o}{D_i}\right) \tag{4}$$

Asser Brabin et al. [12,13] have suggested a modified version of Faupel's burst equation:

$$P_{b,5} = \frac{2}{\sqrt{3}}\sigma_Y\left\{1 + \lambda\left(1 - \frac{\sigma_Y}{\sigma_{uts}}\right)\right\}\ln\left(\frac{D_o}{D_i}\right) \tag{5}$$

where λ is a material dependent constant; $\lambda = 0.65$ for steel vessels. Both Eqs. (4) and (5) provide better comparison with experimental data. However, none of the above equations follow from finite strain formulation and do not adequately describe the effect of the work-hardening material response on the burst pressure.

Detailed finite deformation analysis of internally pressurised pipe (without any externally applied axial stress) made from a power-law hardening material with strain hardening index n, shows that the burst pressure can be expressed based on Tresca yield criterion by Ref. [2,6,14]

$$P_{bT} = \frac{1}{2^n}p_{ref}, \tag{6}$$

and for von Mises yield criterion by Ref. [2,6,15,16]

$$P_{bM} = \frac{2}{\left(\sqrt{3}\right)^{n+1}}p_{ref} \tag{7}$$

where subscripts T and M are used to indicate burst equation based on Tresca and von Mises, respectively, and p_{ref} is a reference load given by

$$p_{ref} = \frac{2t}{D_m}\sigma_{uts} \tag{8}$$

Comparison of the predicted burst pressure by (6) and (7) with experimentally measured values for API X grade tubulars showed that the von Mises based model over-predicts the burst pressure by about 9% while the Tresca based model (6) under-predicts the burst pressure by about 7% [2]. The average of the two predicted values is therefore recommended as this provides a better correlation between predicted and measured values. The average burst pressure in the absence of externally applied axial stress is [2,6]

$$P_{bC} = \left[\left(\frac{1}{2}\right)^{n+1} + \left(\frac{1}{\sqrt{3}}\right)^{n+1}\right]p_{ref} \tag{9}$$

None of the above models includes the effect of externally applied axial load on the burst pressure.

1.3. Burst pressure equations for pipes subject to combined internal pressure and external tensile stress

A ductile circular pipe that is subject to only a uniaxial tension without any internal or external pressure, will fail by necking when the externally applied axial stress equals the tensile strength σ_{uts} of the material. Necking is the only possible failure mode when no internal or external pressure is applied, as this is identical to conventional uniaxial tensile test. However, under a combined internal pressure and external axial tension, the pipe can fail either by rupture or necking.

Using a combination of Tresca and von Mises yield criterion together with incremental flow theory, Klever [2] and Stewart et al. [6] determined the combination of loads required for the initiation of either of these failure mechanisms for a pipe with constrained axial deformation. With the constrained axial deformation, the effective axial stress on the pipe's wall is due to a combined effect of the externally applied stress σ_a and the Poisson's effect of the hoop stress; recall the pipe is thin-walled, so the radial component of stress is negligible. The effective axial stress in a thin-walled pipe first subject to an external axial stress σ_a and subsequently fully axially restrained while an internal pressure P_i is applied is given by

$$\sigma_{eff} = \sigma_a - \frac{P_i D_m}{4t} \tag{10}$$

Combined axial stress and internal pressure loading is normally implemented experimentally following one of two loading paths: