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Stress Analysis of Inlet Pigtail Pipe Bend under Internal Pressure due to Ovality

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Abstract
Pipe bends are considered the critical components in a piping system. In the manufacturing process of pipe bends, it is challenging to avoid thickening on the inner radius and thinning on the outer radius. The cross section of the bend becomes non circular and its acceptability is based on the induced level of shape imperfections. Considering the shape imperfections it is necessary to estimate the pipe bend limit load under internal pressure. Ovality is the shape imperfection in the pipe bend considered for analysis. The three dimensional finite element method is used to model and analyse a standalone, long radius inlet pigtail pipe bend which is used in hydrogen reformer application. It is shown that the effect of ovality is significant in pipe bend with attached straight pipe due to internal pressure. The induced stresses in the pipe bend geometry also increases with the percentage of ovality.

Keywords: Ovality, Inlet pigtail pipe bend, hydrogen reformer

Introduction

Pipe bends are more flexible than similar length of straight pipe, due to the intricate deformation they exhibit under various loading conditions. Inlet pigtail pipe bends used in hydrogen reformer application experiences sustained, expansion and other occasional loads during operation. Limit loads for the bends have been of importance for integrity assessment [1-6] and many authors have proposed limit load solutions for the pipe bends subject to collective internal pressure and in plane bending established on detailed finite element limit analysis using elastic perfectly plastic materials under small geometry change assumptions [7]. The results suggested that existing analytical solutions for pipe bends [1-4] were much lower than the finite element results. One important point is that, for the finite element limit analysis, a piping system consisting of 90° pipe bend considering attached straight pipe to remove the effect instigated by the applied bending moment, the straight pipe length was set to be sufficiently long (five times the bend radius) [7]. Note that such practice is typical in finite element analysis of pipe bends [5-8] and furthermore is realistic in any plant that the bend is always joined to straight pipe.

Stress analysis in pipe bends were performed rely on the assumptions that the constant wall thickness along the contour of the pipe bend cross section and no ovality [1-8]. However, the majorities of pipe bends are made using a forming process and as a result, have variable wall thickness along the surfaces of the pipe bend cross section. The pipe wall is thinner than nominal on the convex side and is thicker on the concave one. The bend section may be a potential source of failure during service, particularly in cases where significant wall thickness and ovality variation (thinning/ thickening) exist in the piping system. The aim of this study is to investigate the effect of ovality in inlet pigtail pipe bend under internal pressure..

Definitions

i. Percent Ovality

During the bending operations, the cross section of the bend frequently assumes an oval shape whose major axis is vertical to the plane of bend, as shown in Fig.1

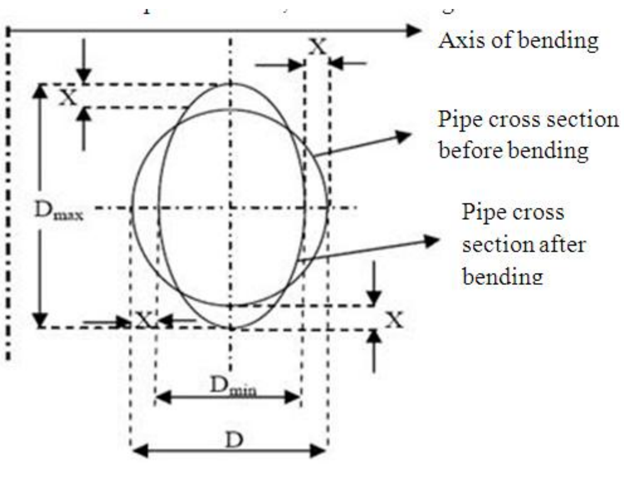

Fig.1. Bend cross section before and after bending

The percent ovality is resolute by the deviation of major and minor diameters divided by the nominal diameter of the pipe bend [9]

$$C_o = \frac{(D_{max} - D_{min})}{(D_{max} + D_{min})/2} x100, \qquad \left(D = \frac{D_{max} + D_{min}}{2}\right) \tag{1}$$

$$D_{max} = D + 2[X] \tag{2}$$

$$D_{min} = D - 2[X] \tag{3}$$

$$C_o = 400[X]/D \tag{4}$$

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ii. Internal Pressure

Under Internal pressure loading conditions Goodall [3] used following limit pressure equation without considering the shape imperfection.

$$P_o = \left(\sigma_o \frac{t}{r}\right)\left[\frac{1 - {}^{r}/_{R}}{1 - {}^{r}/_{2R}}\right] \quad (5)$$

The right hand side of Eq.(5) is the limit pressure for a straight portion of the pipe and the second term is the correction factor for the bend profile curvature considering the uniform thickness of pipe bend. When applying mises yield conditions for the first term in the Eq.(5) which corresponds to a lower bound based on tresca yield conditions, a factor of $2/\sqrt{3}$ could be applied.

$$P_o = \left(\frac{2}{\sqrt{3}}\sigma_o \frac{t}{r}\right)\left[\frac{1-{}^{r}/_{R}}{1-{}^{r}/_{2R}}\right] = P_o^s\left[\frac{1-{}^{r}/_{R}}{1-{}^{r}/_{2R}}\right] \quad (6)$$

Where P_o^s indicates the plastic limit pressure for straight portion of pipe.

Geometry

The piping system considered a long radius bend with two attached equal length straight pipe runs as shown in Fig.2 and the cross section of the bend is assumed to become a perfect ellipse after bending as shown in Fig.1. The mean cross sectional radius of the bend (R) considered as 250 mm, the bend factor (h) incremented from 0.1 to 0.5 and the percentage of ovality varied from 0 to 20 (Fig.3). The internal pressure incremented for each model until it reaches 27×10^8 N/mm^2.

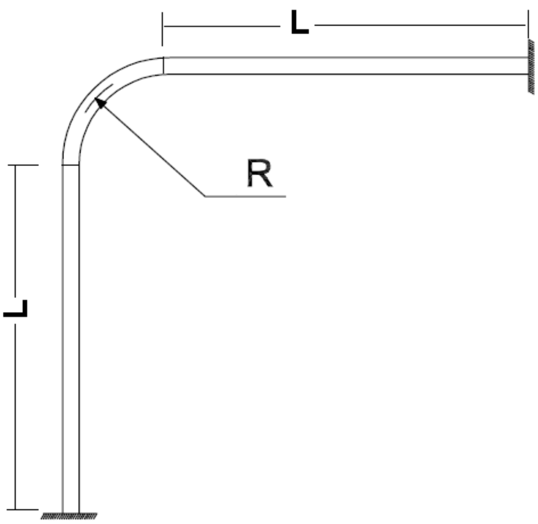

Fig.2. Schematic Illustration on Pipe Bend Geometry

The length of the attached straight piping was chosen to ensure that the bend response was independent. The major axis of pipe bend is assumed to be perpendicular to the plane of bending (out plane) and the minor axis of pipe bend is assumed to be in the plane of pipe bend (In plane). The pipe bend is assumed to be smooth, without ripples and flattening.

Finite Element Modelling and Analysis

The finite element modeling and analysis were carried out in Pro ENGINEER and ansys 14.5 workbench, a general nonlinear finite element package [10]. The inlet pigtail piping system configuration in hydrogen reformer has two planes of symmetry (vertical and horizontal) and as such can be modeled by a quarter finite element meshes, with suitable proportion boundary conditions applied.

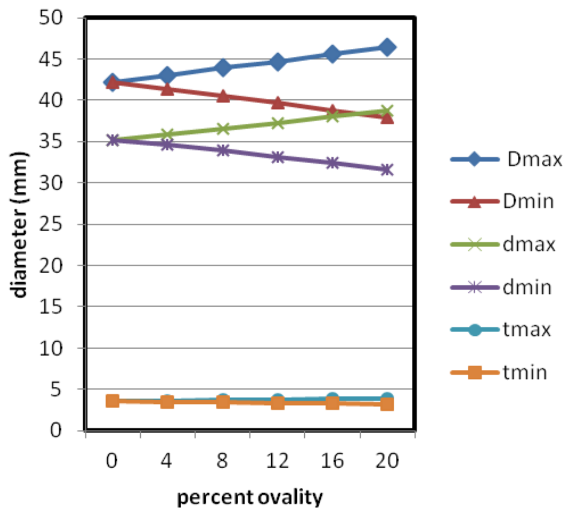

Fig.3. Dimensional change due to ovality

The geometry of inlet pigtail pipe bend modeled in Pro ENGINEER and imported in to ANSYS 14.5 as an IGES file. To establish a suitable mesh density for the FE model, a convergence study was performed shown in Fig.4. The bend was described by 352670 nodes and 66060 elements along the Pipe. The pipe bend is subjected to internal pressure loading and the same was applied as a surface load along the bend profile and for attached pipe surface.

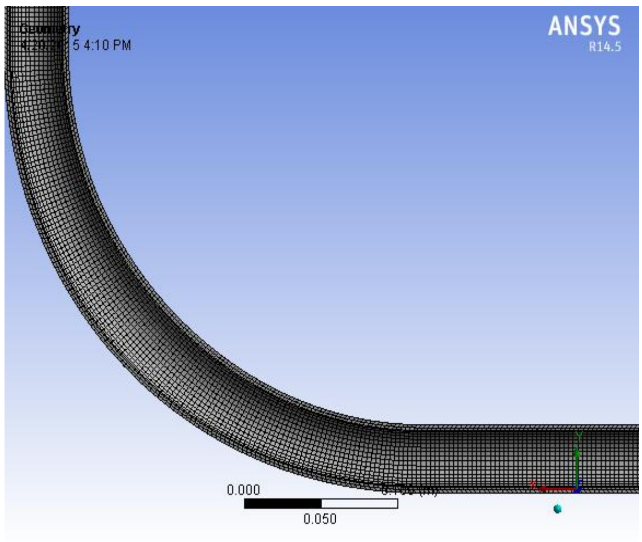

Fig.4. Pipe Bend Mesh element

The piping system was assumed to be fixed at both ends, such that the axial thrust was generated due to internal pressure in the system. Finite element analysis of the pipe bend was performed for inlet pigtail pipe bend, INCOLOY 800 ASTM B407 material and the following values of material properties used in present calculations. Young's modulus E=1734x10^{8}N/mm^2, Tensile strength 4589x10^{8}N/mm^2, Maximum allowable stress 1118 x10^{8}N/mm^2 and Poisson's ratio γ=0.3. Internal pressure applied as a distributed load to the inner surface of the model. To avoid problems associated with convergence in elastic- perfectly plastic calculations, the workbench option within ANSYS was invoked. It should be noted that the present work consider only the pipe bend with a sufficiently long attached straight pipe. A typical way to apply pressure is to constrain the nodes at the end of the pipe through the multi- point constraint option within ANSYS, and to directly apply sufficiently large deformation

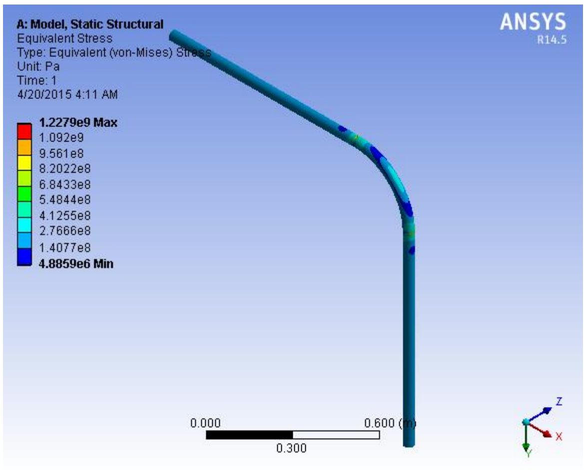

Fig.5.1. Von-mises stresses (Ovality 0%)

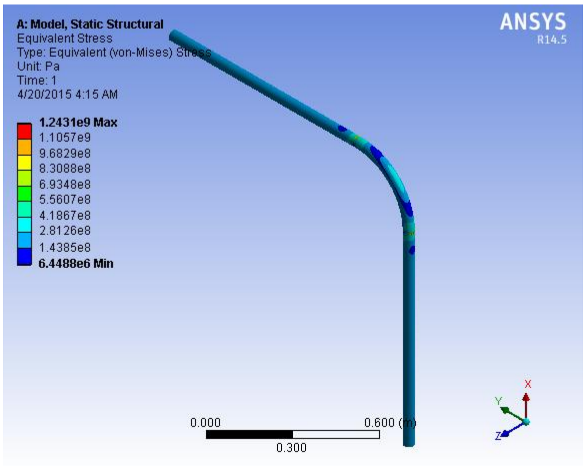

Fig.5.2. Von-mises stresses (Ovality 4%)

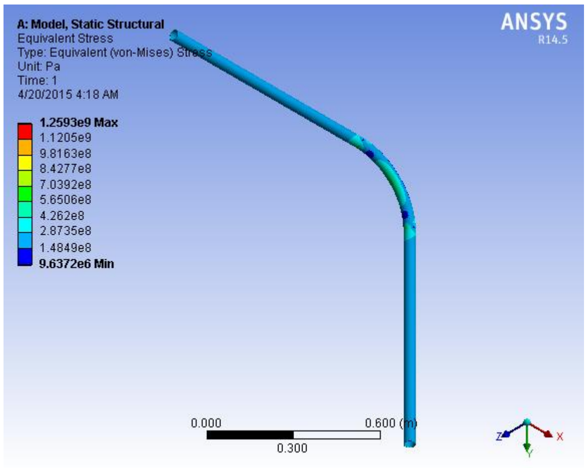

Fig.5.3. Von-mises stresses (Ovality 8%)

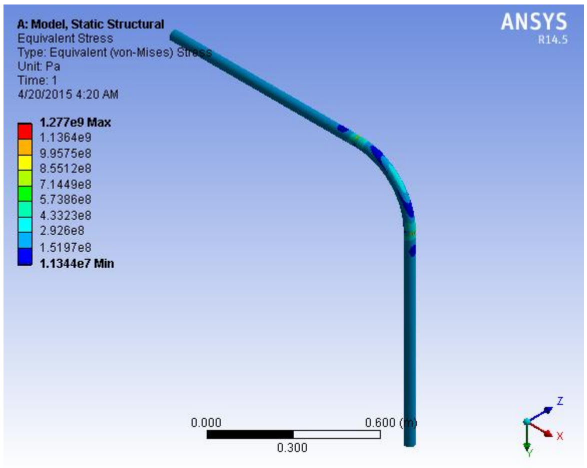

Fig.5.4. Von-mises stresses (Ovality 12%)

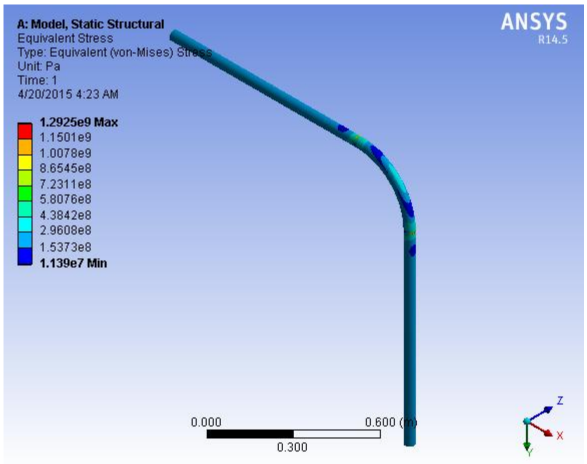

Fig.5.5. Von-mises stresses (Ovality 16%)

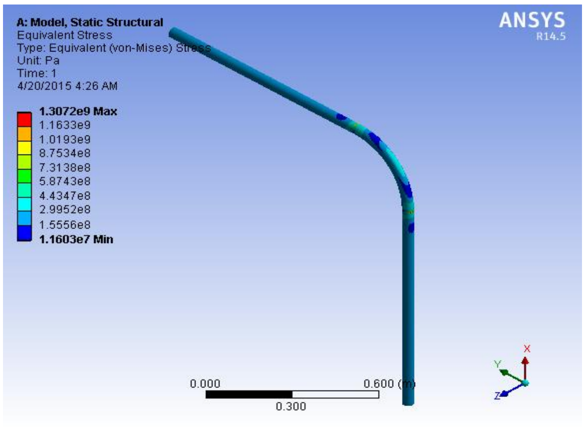

Fig.5.6. Von-mises stresses (Ovality 20%)

Results and Discussions

The application of internal pressure changes the way of pipe bend behaves under internal pressure loading, not only in terms of its load-deflection behavior, but also in terms of distribution of stresses and strains. The axial distribution of mises stress at the inside wall, the point where yielding starts to take place in the models as shown in Fig.5.1 to Fig 5.6 for various percentage of ovality.

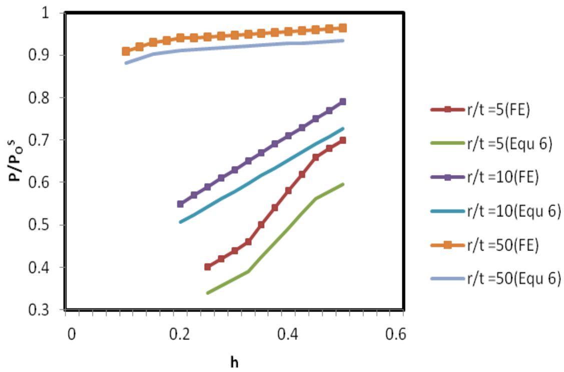

Fig.6 Comparison of theoretical limit pressure solutions with FE results

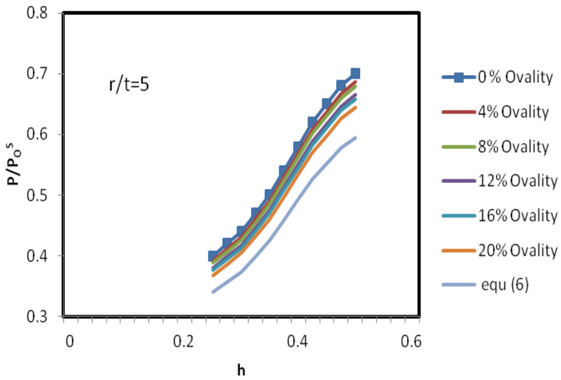

Fig.7 Limit pressure with FE results (r/t = 5)

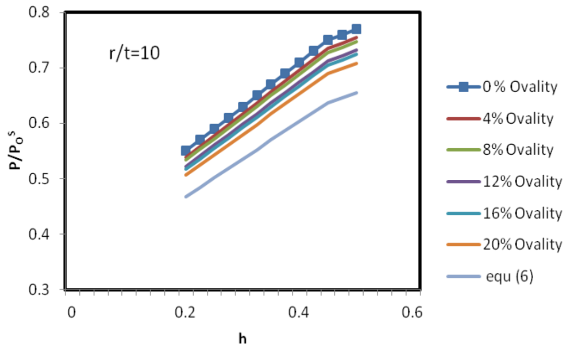

Fig.8 Limit pressure with FE results (r/t = 10)

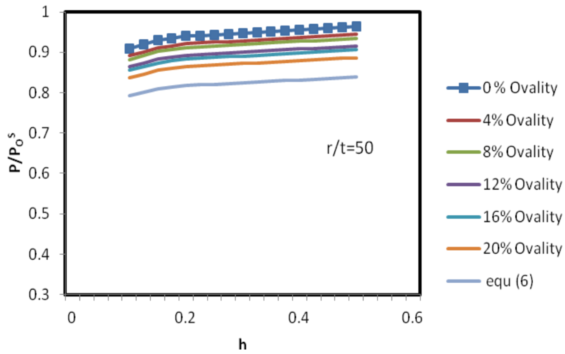

Fig.9 Limit pressure with FE results (r/t = 50)

Normalized limit pressure of pipe bend (P/P_o^s) without ovality, analytical results were compared with finite element results(Fig.6) obtained in the present study for various r/t ratio (r/t= 5, 10 and 50). The results are good agreement (within 3%) with the results from analytical solution Eq.(5) for pipe bend. The deviation between analytical and finite element results increases with increasing bend factor (h) and r/t. It can be seen that the induced stresses in bend geometry are higher due to the percent ovality increases and the results having significant effect due to attached straight pipe with pipe bend which is most practical case in the piping system.

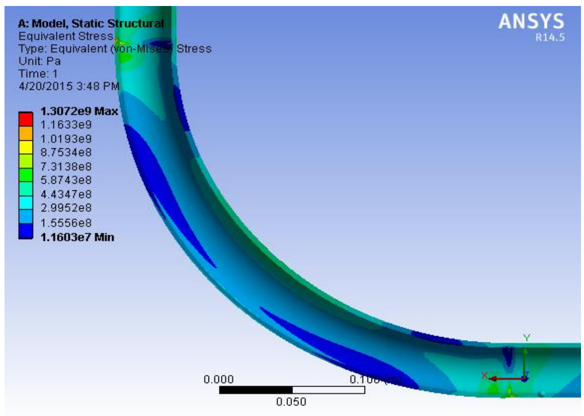

Fig.10 Contours of Von-mises stress in ovality region

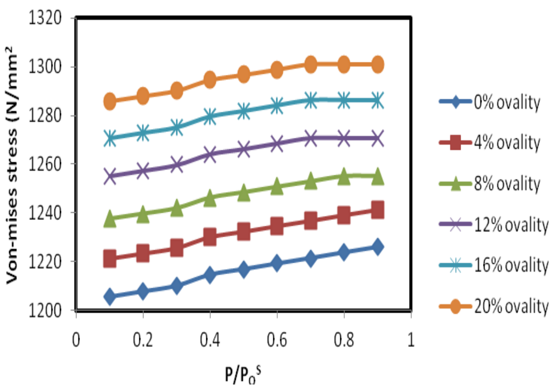

Fig.11 Normalized limit pressure Vs Von-mises stress

Under internal pressure loading, first yielding occurs in the middle of the bend at the inside surface of the intrados. As the pressure is increased the plastic zone spreads axially towards the junction with straight run and circumferentially towards the extrados. Fig.7 to Fig.9 indicates the effect of internal pressure for various percent ovality (0 to 20) for r/t =5, 10 and 50 and is found that the limit pressure limitation is almost linear due to internal pressure loading and the solution is much higher bound than the analytical one as expected.
Careful examination suggested in finite element results motivating plastic yielding patterns in pipe bends with attached straight pipes Fig.10 and Fig.11. For lesser r/t, an inside bend region yields first and the yielding region spreads to a different part of attached straight pipe. Consequently a piping system can withstand more loads due to internal pressure. At a elevated loading condition, the crown region of the inlet pipe bend finally yields and load reaches its limiting value indicated, which corresponds to the limiting pressure.

Conclusions

The effect of ovality for inlet pigtail pipe bend used in the hydrogen reformer application under internal pressure has been quantified, based on the finite element analysis using elastic perfectly plastic materials with small geometry change option. Internal pressure has an obvious effect on this behavior, which is more significant in pipe bends for smaller bend factor.
Higher values of pressure have a detrimental effect on the strength, due to the additional stresses induced by the pressure itself and further the effect of ovality in pipe bends on plastic limit load can be significant.

From the results, the following conclusions were derived.

1. The finite element results are consistent with those from analytic solutions, indicating that the proper modeling method is adopted and calculation of limit load of pipe bend is effective under internal pressure with considering the shape imperfections.
2. Increasing internal pressure, the displacement controlled bending load was relaxed by plastic deformation at the extrados in pipe bends.
3. The results of finite element analysis show that the limit pressure of pipe bends increases with increasing the value of h.
4. The result of the investigation show that geometric non linearity is a substantial consideration when calculating the stresses of the pipe bends subject to internal pressure loading. Geometric weakening in pipe bend is observed when ovality increases.
5. The presence of ovality produces much effect on limit load of the pipe bend, the limit load decreases with increasing the percentage of ovality.

Furthermore, effort should take dissimilar material properties into consideration to attain more general conclusion.

Nomenclature

c_o	Percent ovality
P_o	Limit pressure of a pipe bend
P_o^s	Limit pressure of straight pipe
P	Internal pressure
σ_o	Limiting stress of an elastic-perfectly plastic model
h	Bend characteristics $(= Rt/r^2)$
D_{max}	Maximum outside pipe diameter (mm)
D_{min}	Minimum outside pipe diameter (mm)
t	Wall thickness of pipe
R	Bend radius to neutral axis (mm)
r	Mean pipe radius (mm)

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