PIPING SHEAR LUG STRESS ANALYSIS

G. E. O. WIDERIA, G. T. KITZ, V. K. VERMA, J. S. LAN

Engineering Mechanics Division, Sargent & Lundy Engineers,
35 East Monroe Street, Chicago, Illinois 60603, U.S.A.

SUMMARY

Due to space limitations present in many of the current nuclear power plants, it is frequently necessary to apply restraints skewed to a piping system's axis in order to restrain the piping against hydraulic transient and seismic loads. In order to assure that the restraints function as required, shear lugs are often used to eliminate relative motion between the pipe and the restraint. The design of these lugs must be such that the stresses induced in the pipe are in compliance with the criteria of the ASME Code. According to the Code, stresses may be determined by either a detailed stress analysis or by use of given simplified analysis procedures employing stress indices. At present, stress indices are not provided for lugs. It is the purpose of this paper, to present detailed stress analyses of the pipe-lug system and from them deduce expressions for the needed indices.

The analyses are accomplished through use of three-dimensional finite-element modeling employing the computer program SAP IV. Eight node brick elements are used to represent both the pipe and the lug. The resulting mesh consists of 316 elements and 558 nodes and incorporates the symmetry aspects of the physical system. The mesh is a graded one with small elements employed in the area of the lug. A check on the expected accuracy is obtained through a comparison with both an approximate elasticity solution and a proprietary computer program ISOPAR-SHL. The latter allows local stress concentration regions to be modeled by using higher order three-dimensional elements (36 nodes) while the remaining structure is represented by curved shell elements, with transition elements connecting the two.

Numerical results are carried out for typical pipe diameters and loadings are then interpreted and recast in terms of the Code design criteria. They are furthermore compared with the results obtained by employing the procedure given in WRC Bulletin #198. This comparison illustrates the conservativeness of the WRC approach, especially for small lugs and lugs whose long dimension is in the circumferential direction. In particular, for a circumferential lug on a 12 in. diameter pipe subjected to loading representative of the upset condition, use of the WRC procedure yields results which are larger by a factor of four for the shear term and by 1.5 for the bending term.
1. Introduction

Piping systems have been described as "wet noodles" requiring a multitude of supporting hardware to keep them geometrically fixed when subjected to loads such as those arising from hydraulic transients, safety relief valve discharges and seismic motions. To prevent relative motion between the pipe and the often skewed restraints, integral attachments such as lugs are needed. A typical arrangement of such a skewed restraint is shown in Fig. 1. The lugs are welded to the pipe and the restraint is attached to a clamp. The load transfer takes place in a direction parallel to the pipe axis.

The ASME Boiler and Pressure Vessel Code [1] specifies the operating conditions that must be met and also defines a set of allowable stresses. With regards to attachments, it requires that their effects on the pressure restraining members be taken into account when checking against the allowable values. In carrying out this check the Code allows the use of a simplified analysis procedure which requires the use of stress indices. This simplified analysis procedure has built into it the underlying assumption that the maximum stress intensity (twice the maximum shear stress according to the Tresca failure criterion) which exists in a component is less than or equal to the sum of the maximum stress intensities due to the loads taken individually.

At the present time, the design equations of the Code are not directly applicable to lug-type attachments. A modification of these equations by adding new terms specifically for integral lugs was proposed in Welding Research Council Bulletin #198 [2] and then subsequently adopted as Code Case #1745 [3]. The Bulletin presents some simple formulas for computing stress indices for thrust, moment torque and shear loading. As such, the formulas are based in large part on the work of Dodge [4] whose work in turn is based on that of Bijlaard [5]. The conservativeness of these formulas is especially apparent for shear loaded lugs. For this case, the assumption is made that the axial tangential load is transmitted to the pipe by membrane shear forces whose magnitude is given by a simple strength of materials expression. When applied to lugs whose circumferential dimension is much larger than the axial one, it leads to results which are entirely inappropriate.

In the present paper an attempt has been made to analyze the state of stress produced in pipe due to a circumferential lug which is subjected to an axial shear load. The analysis is accomplished through the use of finite element models. Numerical results are presented for a typical pipe diameter, lug and loading and a comparison is made with those obtained through use of Reference [2], assuming it to be applicable. The results are furthermore interpreted with respect to design requirements according to Section III of the Code.

2. Finite Element Modelling

Due to the fact that exact solutions of elasticity problems are extremely difficult to obtain, the analysis of the integral shear lug-pipe system was performed by use of three-dimensional finite element modelling. In setting up the models, it was assumed that two opposite lugs carry the entire load and that this vertical load is distributed uniformly over the top horizontal surface of the lugs. The first of these assumptions allowed the modelling to be reduced to a 90° segment while the second enabled a further reduction to a one-half-the-length portion (See Fig. 2). In making this second reduction, use was also made of the fact that the given loading could be replaced by one which is symmetrical with respect to a horizontal plane through the center of the lug and one which is antisymmetrical.

Fixed support conditions were assumed at the pipe ends, with the distance between the ends taken far enough from the lug so that the ends could be expected not to exert any influence on the localized behavior in
the neighborhood of the lug.

Two separate computer codes, SAP IV and ISOPAR-SHL, were employed in the
analysis. SAP IV is a commercially available code and the mesh used is
shown in Fig. 2. As can be seen from the figure, 8-node brick elements were
used to represent both the pipe and the lug. The resulting mesh consisted
of 316 elements and 581 nodes. The same mesh was used for the whole range
of pipe sizes considered, thus necessitating only changes in the radial co-
ordinate specification. The ISOPAR-SHL can perform static analysis of thick
shells and three-dimensional solid structures using isoparametric elements.
It allows for the use of higher order three-dimensional elements in combina-
tion with shell and transition elements. The mesh employed for use with
this code is shown in Fig. 3. Inspection of the figure shows that higher
order elements were used in the lug area where high stress gradients can be
expected to be present while the remainder of the structure was modelled by
curved shell elements with transition elements connecting the two. The mesh
contained eleven 36-node solid elements, 35 shell elements, 7 transition
elements and 356 nodes.

3. Numerical Results

The finite element analysis was carried out for a range of pipe sizes
for typical loads and for lugs which had a circumferential extent of 50°.
In each case it was seen that the effect of the lug produced a rapidly
damped state of stress in the pipe with the maximum occurring in the imme-
 diate area of the lug.

The critical sections were the horizontal section through the pipe ele-
ments just above the lug and the vertical section through the pipe-lug inter-
face. For example, for a steel pipe having 12 in. O.D., a 0.375 in. wall,
a lug having a radial extent and a depth of 1 in. each and loaded by 20,000
lbs., the major stress component variations for these sections are shown in
Figs. 4 and 5, respectively. The usual designation of z for axial, θ for
circumferential and r for radial direction is implied.

An examination of these figures shows that the axial stress is the
dominant stress component and that it is the major influence on the stress
intensity. Its maximum value occurs in elements adjacent to the upper end
of the pipe-lug interface and its character is primarily that of a bending
stress. In Fig. 5, stresses $\sigma_z$, $\sigma_\theta$, $\sigma_r$, are antisymmetric with respect to
the z=0 axis while $\tau_{rz}$ and the maximum stress intensity are symmetric.

4. Code Analysis

The simplified design procedure of the Code is not directly applicable to
the stresses resulting from a finite element analysis. The results of
such an analysis must first be converted into stress resultants typically
associated with thin walled structural elements. For a cylinder under arbi-
trary loading and composed of elasticity elements, the resultants at an
arbitrary cut perpendicular to the axis are given by

$$F_z = \frac{1}{R} \int_{-t/2}^{t/2} \sigma_z R(x) \, dx,$$
$$F_\theta = \int_{-t/2}^{t/2} \sigma_\theta \, dx,$$
$$F_{rz} = \frac{1}{R} \int_{-t/2}^{t/2} \tau_{rz} R(x) \, dx,$$
$$M_1 = \int_{-t/2}^{t/2} \sigma_z x R(x) \, dx,$$
$$M_\theta = \int_{-t/2}^{t/2} \sigma_\theta x \, dx.$$

Here, t is the thickness, R the mean radius, the $F_i$ are the various membrane
resultants, $M_j$ the moment resultants and

$$x = r - R$$
Use of stress resultants as determined above in conjunction with the value of the radial stress, allows for the determination of the membrane and membrane plus bending stress intensities at critical sections. Examples of such critical sections would be locations of geometric stress concentration and discontinuities. For the stresses given in Fig. 4, the stated procedure yields a membrane stress intensity of 4,862 psi and a membrane plus bending stress intensity of 27,850 psi.

4. WRC Bulletin 198 Calculations

The lugs considered in this study were also analyzed according to the procedure given in WRC Bulletin 198. In the bulletin it is shown that the contribution to the primary stress intensity due to an axially loaded lug on a pipe is given by

\[(P_L + P_D)_{\text{lug}} = B_L \frac{M_L}{Z_L} + \frac{Q}{Z_L L_D}\]  

(3)

Here, \(M_L\) is the applied axial bending moment, \(Q\) the axial shear force, \(Z_L\) the section modulus of the lug at the pipe surface, \(L_D\) one-half the lug dimension in the axial direction, \(L_D\) the smaller of half the lug width and pipe thickness, and \(B_L\) the stress index for moment loading.

Use of the dimensions used for the previous sample calculations yielded \(B_L = 1.48\) and thus

\[(P_L + P_D)_{\text{lug}} = 33,931 + 53,333\]

\[= 87,264 \text{ psi}\]  

(4)

According to the finite element analysis, the maximum stress intensity is equal to 34,000 psi, of which 18,000 psi results from bending action while 16,000 psi is due to shear. This result would seem to indicate a value of \(B_L = 0.785\) rather than the 1.48 obtained from the WRC formula.

5. Discussion

In the above, a finite element analysis of circumferential lugs subjected to axial shear loads was presented. A comparison of the results obtained with those by use of the simplified design procedure employing stress indices shows this procedure to be highly over-conservative for the type of lugs considered.

The effect of lugs of small circumferential variation were also analyzed as part of a more general study of lug behavior. The results obtained can be summarized as follows: An increase in lug length or width reduces the maximum stress intensity, with a more appreciable reduction due to a length increase. The stress distribution in the pipe in the neighborhood of the lug was nearly antisymmetric in character. The stresses had primarily a bending character associated with them and thus load eccentricity is an important parameter. In each case the maximum stress intensity occurred in the outer pipe element adjacent to the loaded surface of the lug, showing the effect of the lug in increasing the rigidity in the area of the attachment. Finally, the incorporation of the effect of a weld at the unloaded lug-pipe interface can bring about a significant reduction in the value of the stresses.
References


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**Figure 1. Typical Lug Arrangement**
Figure 4. Stress Distribution (Through Wall, Above Lug)
Figure 5. Stress Distribution (Pipe-Lug Interface)