CLASSIFICATION OF CLAMP-INDUCED STRESSES IN THIN-WALLED PIPE

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without inducing large changes in the bolt or spring loads on the contact pressure at the pipe/clamp interface. The design considerations for the CRBRP in-containment piping and clamps are described in detail in [1 and 2]. A typical support arrangement for the CRBRP in-containment horizontal piping runs is also shown in Figure 1.

A considerable amount of analytical work to determine the effect of these clamps on the piping has been completed and was reported in [3]. A combined 2-D and 3-D approach was utilized to perform this previous analysis. Briefly, this approach consisted of using a 2-D interaction model of the clamp and pipe wall, as shown in Figure 2, to obtain the circumferential interface load distribution. These circumferential loads were then applied to a 3-D shell model of the pipe in a uniform manner over the width of the clamp to obtain the corresponding stress state induced in the pipe wall. This approach was chosen because of the ease of changing the clamp or pipe geometry for the various clamp/pipe configurations. The 2-D interaction model also made it simple to examine a large number of load combinations. In addition, various design iterations and parametric studies were easy to investigate. It was previously judged that this procedure was conservative with respect to the stresses induced in the pipe wall since the 2-D equivalent bending stiffness derived for the ring model of the pipe wall should be stiffer than the actual pipe wall.

The following two questions arose as a result of this previous work;

1. Was the combined 2D/3D approach as conservative as expected? Of particular interest here was the effect of the actual axial interface load distribution on the resultant pipe wall stress distribution and the combined clamp/pipe stiffnesses calculated.

2. How should the pipe stress components due to the clamp effects be classified (i.e., as primary or secondary as defined by the ASME Boiler and Pressure Vessel Code [4] and Code Case 1592 [5])?

The purpose of this paper is to provide the answers to these questions through detailed 3-D nonlinear clamp/pipe interaction analysis.

QUALIFICATION OF THE COMBINED 2-D/3-D ANALYTICAL APPROACH

The method selected to answer the first question and thereby qualify the combined 2-D/3-D approach used to evaluate the clamp effects on the pipe was to develop a 3-D nonlinear interaction model of one clamp/pipe combination. A general purpose finite element code named WECAN [6] was used for this analysis. Since a large, 3-D, nonlinear model is relatively expensive to use, the initial interest was to analyze only the preload case. However, after setting up the model and completing the preload case successfully, a couple of external load cases were also analyzed to confirm the combined clamp/pipe stiffnesses calculated previously from the 2-D interaction model approach.
The particular clamp pipe combination chosen for modeling was a 0.607 m diameter by 0.304 m wide clamp on a 0.607 m diameter pipe with a wall thickness of 12.7 mm. This particular clamp has a specified preload of 176 kN at cold conditions. A 3-D shell model of the clamp and pipe was set up as shown in Figure 3. A quarter section model could be used because the axial plane through the clamp split line and the plane through the center of the clamp band are symmetry planes for the preload case. The clamp and the pipe meshes were connected to each other by four rows of spring-gap elements, also shown in this figure. As in the previous work discussed in [3], the springs represent the equivalent insulation stiffness and the gaps are given an initial distribution based on the worst case tolerance stackups between the maximum/minimum radii of the pipe O.D., the insulation band I.D./O.D. and the clamp band I.D. The clamp bolt was modeled with a spar element, and the Belleville spring stackup was modeled with gap elements also. Modeling the Belleville spring stackup with gap elements allows the preload to be induced into the model by specifying an appropriate negative gap value. To avoid interaction between the clamp effects and the boundary conditions for the end of the pipe, the proper pipe length was established by analyzing the preload case with both fixed and free end conditions on various pipe lengths. To determine if a sufficient length of pipe had been reached, two criteria were used. One criterion was based on the differences between pipe wall maximum stress intensities at four critical locations. The other criterion was based on a general comparison of the interface loads both in terms of magnitude at specific locations and in terms of the overall distribution. The later criterion is difficult to illustrate in a simple manner since a large number of interface loads are involved, but the first criterion is illustrated in Figure 4. As can be seen from this figure, the maximum stress intensities approach each other as the pipe length is increased as they should. A sufficient pipe length of four pipe diameters from the center of the clamp band was established by this procedure.

The interface load distribution for the preload case is shown in Figure 5. It can be seen from examination of this figure that the axial interface load distribution is not uniform across the width of the clamp as assumed in the previous combined 2-D/3-D approach. However, examination of the maximum stress locations revealed that the 3-D interaction model, which did include the axial effect, produced significantly lower maximum pipe wall stresses. Thus, the previous combined 2-D/3-D approach was conservative, as anticipated. A comparison of the integrated (over the clamp width) circumferential interface load distribution from the 3-D interaction model for the preload case to the corresponding case from the 2-D interaction model is shown in Figure 6. This figure indicates that another difference between the two approaches is that the interface loads from 3-D interaction model are spread over a larger circumferential distance than the corresponding 2-D interaction model results. This explains why the maximum pipe wall stresses from the 3-D interaction model are significantly lower than for the combined 2-D/3-D approach. This also confirmed previous parametric studies which had indicated that maximum stress component in the pipe is sensitive to the circumferential interface distribution but not to the form of the axial distribution.

A couple for external load cases were also analyzed to check the combined clamp/pipe stiffnesses calculated from the 2-D interaction model results, [3]. The two load cases chosen have either a 176 kN tensile or compressive load acting at the bolt flanges and directed along the split line of the clamp. Two load cases were picked because they could be used with the quarter section model since the symmetry planes are the same for the preload case. The average flange deflection at the point of load application due to the applied external load was extracted and used to calculate the combined clamp/pipe stiffness for the two loads. The stiffness values obtained were within 15 percent of those calculated from the 2-D interaction model. This confirmed that the stiffness values calculated previously were adequate for seismic analysis of the piping loops. In summary, the 3-D interaction model results verified that the combined 2-D/3-D approach produced conservative values for the maximum pipe wall stresses and also gave reasonable estimates of the clamp/pipe stiffnesses.

**CLASSIFICATION OF LOAD CONTROLLED AND DISPLACEMENT CONTROLLED QUANTITIES**

The design of reasonably conservative piping systems requires careful classification of induced stresses into primary (P) and secondary (Q) stress categories. The basic characteristic of a primary stress is that it is not self-limiting (see NB-3213.8 of [4]). Primary stress is not redistributed or relieved by inelastic deformation. The behavior is equilibrium-controlled or load-controlled and primary stresses may be alternatively described as equilibrium-controlled or load-controlled stresses. The basic characteristic of a secondary stress is that it is self-limiting (see NB-3213.9 of [4]). Secondary stress is caused by displacement constraints and is redistributed or relieved by inelastic deformation. This behavior is compatibility-controlled or deformation-controlled and secondary...
stresses may be alternatively described as compatibility-controlled or deformation-controlled stresses.

The importance attached to the classification of induced stresses derives from their relationship to modes of failure. Primary stresses which can result in failure from a single application of load are limited to relatively low allowable values. Secondary stresses which can result in failure only from repeated load cycles are less severely limited. It is the range and number of cycles of secondary stress (rather than the level of stress) that are of particular importance in preventing failure by exhaustion of ductility or gross distortion.

Figure 5. Interface Load Distribution for 178,000 N Preload

The following numerical example illustrates the importance of appropriate classification of a total stress into primary and secondary categories. The provisions of Code Case 1592 T-1324 [5] are used to satisfy the limits on inelastic strain accumulation. Consider a normal startup, hold for 104 hours at 593°C and normal shutdown. Assume that a primary-plus-secondary stress intensity of \( S(P_L + P_B + Q) \) = 172 MPa is reached initially. The yield stress is \( S_y = 102 \) MPa according to Figure I-14.5A of [5]. The accumulated inelastic strain can be calculated from Figure T-1800-A-7 and paragraph T-1324 of [5] for alternative classifications of the \( S(P_L + P_B + Q) \) stress intensity into primary and secondary stress categories. The limiting strain value is 1%. The results for two divisions of the 172 MPa stress are:

- \( S_L \) max \( S(Q) \) range \( \epsilon_{inelastic} \) \( \text{Result} \)
- 34 MPa 136 MPa 0.2% Pass
- 69 MPa 103 MPa 1.0% Fail

Thus, it is clear that development of a reasonably conservative design is dependent upon appropriate classification of stresses into primary, and secondary stress categories.

3-D INTERACTION MODEL MODIFICATIONS

To resolve the stress classification question both inelastic analysis and nonlinear, elastic parametric studies were required. However, since the full 3-D interaction model shown previously was expensive and time consuming to use, it was desired to cut both computer run time and core space requirements without a significant loss of modeling detail. Therefore, the linear portions of the 3-D interaction model were incorporated into super elements or substructures [6]. This procedure required three separate steps to complete. Step 1 creates the individual super elements. Step 2 assembles the super elements with other specified elements and solves for boundary displacements for each super element and forces/stresses in the nonlinear elements. Finally, Step 3 utilizes the boundary displacements to solve for forces/stresses within each super element.

Although this approach requires extra effort to set up and several computer runs to execute, the potential savings in overall computer charges and the ease of examining the results often make it worthwhile. Actually, Step 1 is completed only once for a given modal, and Step 3 is completed only if one or more of the super elements contains some desired displacement/stress information. This method does require some insight on the part of the analyst, however. The analyst must be able to select the proper master degrees of freedom (dfs) to retain the basic response of the complete model. In addition, the analyst must be able to pick enough appropriate super elements to avoid large solution wavefronts which sometimes occur. *\( S_m \) is replaced by \( S_0 \) in Code Case 1592.
This can happen because all of the master dofs for a specific super element enter the solution simultaneously. Therefore, it is possible to produce a larger wavefront using super elements than for a well-ordered mesh of standard elements.

Figure 7. Typical Super Elements

3-D Interaction Model With Super Elements

The 3-D interaction model shown in Figure 3 was broken into 12 super elements consisting of 6 super elements for the clamp mesh and 6 for the pipe mesh. Each super element spanned a 30° circumferential segment of the clamp or pipe. Two typical super elements are shown in Figure 7. A 0.304 m section of pipe elements in the immediate vicinity of the clamp were not included in the pipe super elements since this region would subsequently have inelastic material properties. The preload case was then rerun, and the interface load distribution was compared to that obtained from the full 3-D interaction model preload case to qualify the selection of dofs for the super elements.

It was found that the above model would cut the run time and computer charges to about one-half of that for the full 3-D model. Since inelastic analysis typically requires that a large number of load steps and substeps be completed, it was judged that inelastic analysis costs would still be prohibitive at this level. Therefore, it was desired to reduce the run time and charges even more. Upon examination of the previous preload results, a judgement was made that the interface load distribution was also essentially symmetric about the plane perpendicular to the axial plane containing the split line of the clamp. Therefore, a new model covering only an eighth (90 degree) section of the pipe and clamp was set up for inelastic analysis. Of course, this model could not be used for any of the external load cases, but the intent was to perform inelastic analysis of the preload case only. Again, the interface loads and pipe stresses for the preload case were compared to the previous preload results to qualify this model. Since only minor differences were found, it was judged that this model would be adequate for inelastic analysis of the preload case.

After the inelastic analysis was started, it was then found that the solution would not converge when the bolt preload was being generated internally through the use of a negative gap for the Belleville spring stackup model. Therefore, the bolt and Belleville spring elements were removed from the model and replaced by an external load acting on the bolt flange. The primary effect of this change is that the bolt load could not relax due to inelastic deformations of the pipe wall. Therefore, the inelastic results should be conservative. This approach worked as long as the load was applied in sufficiently small increments.

For the inelastic analysis a bilinear stress-strain curve was utilized. The proper elastic and plastic moduli were used but an artificial yield point was chosen to produce plastic flow at the full preload. The reason for this approach was to determine how the interface loads and pipe wall stresses would redistribute if a local section of the pipe wall yielded.

ELASTIC REDISTRIBUTION EFFECT

During the early attempts at inelastic analysis it was discovered that redistribution of the pipe stresses and interface load distribution occurred in the elastic regime. As indicated in Figure 8, which is a plot of the maximum hoop stress and axial stress components versus clamp preload at the critical location in the pipe wall the bending portion of the stress peaked and then fell off as the preload was increased. This occurred because of a redistribution in the interface loads as shown in Figure 9. This indicates a deformation-controlled situation since the bending stress would increase proportionally with a change in the preload in a load-controlled situation. Figure 8 also shows that both membrane components change almost linearly with increasing preload after an initial nonlinear response. This indicates a load-controlled situation for the membrane stresses. Because of the elastic redistribution effect, a significant amount of plasticity was not produced by simply continually increasing the applied preload.

INELASTIC ANALYSIS

To evaluate the effects of material nonlinearity only, the initial gap distribution was removed from the model by setting all of the gaps equal to zero. In the WECAN finite element code [6], a gap value of zero converts the gap element into a linear spring element because the gap status cannot change. This meant that all of the spring forces had to be checked after each load step to ensure that the springs remained in compression.
The elastic stress solution then becomes a linear function of the preload as shown in Figure 10. This zero gap approach has the added advantage of separating the elastic redistribution effect from any redistribution due to the presence of a local plastic zone in the pipe wall. Also, before the inelastic analysis was restarted, a lower yield point was chosen than that used for the previous analysis. The new yield point was picked to ensure that both the outer and inner surface of the pipe wall would be plastic at the full preload.

The elastic and inelastic pipe wall stress distributions along the outer surface of the pipe are shown in Figure 11 for the hoop and axial stress components. As can be seen from this figure, a plastic zone exists for approximately 15 degrees on each side of the clamp split line. This caused a significant redistribution of stresses even in the regions that remained elastic. The inner surface stress distribution showed a similar pattern but not as dramatically because the stresses at the inner surface were not as high as...
Because it was difficult to determine a specific primary to secondary split based on the inelastic analysis and since it is very expensive to perform inelastic parametric studies, a simplified equivalent elastic analysis technique was utilized to simulate the inelastic results as discussed in the following section.

**STRESS CLASSIFICATION USING EQUIVALENT ELASTIC ANALYSIS**

The equivalent elastic analysis is based on the concept that the local material flexibility in the plastic region can be simulated by lowering the elastic modulus of the pipe. This is illustrated schematically in Figure 12 which shows a plot of the generalized local load versus the local deformation experienced by a pipe subjected to clamp preload. If the clamp preload were a purely load controlled situation, then a change in elastic modulus from $E_i$ to $E_g$ would not affect the initial local load $P_1$, but the initial local deformation would increase from $\delta_1$ to $\delta_2$ represented by a horizontal line AB in Figure 12. On the other hand, if the preload were primarily a displacement-controlled loading, then the change in $E$ would not affect the initial local deformation $\delta_1$, but the initial local load would drop from $P_1$ to $P_2$ represented by a vertical line AC in Figure 12. Finally, in a mixed load and displacement-controlled situation the change in $E$ would not only increase the load deformations but also reduce the local load at a specified preload. This condition is represented by the line AD in Figure 12. However, the actual CRBRP clamp and pipe interaction is much more complex than this.

**Figure 11. Comparison of Elastic and Inelastic Stress Distributions**

on the outer surface of the pipe. At the center of the pipe wall the change in the stress components was minimal. Therefore, the bending components are being reduced substantially but the stresses near the mid-surface are not. This further indicates that the bending components are primarily secondary and the membrane components are primary. However, an extension of the inelastic analysis to higher preloads did not indicate that the bending component would completely disappear. Therefore, a small portion of the bending should be considered as primary to be conservative. This is due to the variations in the hoop bending stiffness of the clamp band such as the split line, the tapered gussets and separate center gussets and load lugs.

**Figure 12. Generalized Load - Deflection Response**

Figure 13 shows the inelastic results, where the applied preload is plotted against the local effective strain, $\epsilon_e$, at the most highly loaded location in the pipe. At loads above 80 kN the material nonlinearity increases pipe flexibility, hence $\epsilon_e$ increases significantly more than the increase in load. This inelastic response can also be simulated by using an equivalent elastic (secant) modulus, $E_g$, as illustrated in Figure 14. Two preload cases (180 N and 270 N) were analyzed using appropriate secant modulii and the 3-D interaction model with no initial gap.
Four equivalent elastic analysis covering an order of magnitude range in elastic modulus were examined for both cases with an initial gap distribution and without gaps. The results are listed in Table 1. In this table, the generalized local load is represented by the effective stress $\sigma_e$, and the generalized local deformation is represented by the effective strain $\varepsilon_e$. The values listed occur at the critical pipe location which is at the center of the clamp band and under the split line of the clamp. These results show that the local generalized stress, $\sigma_e$, is not constant but decreases with the increase in the pipe flexibility. On the other hand, the local generalized strain, $\varepsilon_e$, is also not constant but increases with the increase in pipe flexibility. Consequently, the applied preload cannot be classified as a purely load-controlled or as a purely deformation-controlled quantity. The normalized effective stresses predicted by the four
elastic analyses are plotted against the normalized effective strains in Figure 15. The load-controlled and deformation-controlled situations are represented by a horizontal line and a vertical line, respectively. These lines correspond to the lines AB and AC in Figure 12. One possible technique for classification of primary and secondary stresses is illustrated in Figure 15. The portion that can be designated as load-controlled is represented by the relative position of the actual stress-strain curve with respect to the load-controlled or deformation-controlled lines.

An alternate procedure for separating the primary and secondary portions of the pipe stress is illustrated in Figure 16, which is a plot of the normalized equivalent elastic modulus for the case with an initial gap distribution. The percentage of the load-controlled portion of the preload is the ratio of ε_p and the total increase in strain, ε_f, that would have occurred if the preload effects were not self-limiting. For example, for E_i/E = 3, 16% of the preload is load-controlled, whereas the rest of the preload is deformation controlled at the top surface. A similar load-controlled percentage for the bottom surface is 48%. More importantly, at higher pipe flexibility E_i/E = 10, the corresponding primary preload percentages are 0% and 31%. The variation of primary stresses with respect to the normalized equivalent elastic modulus is further illustrated in Figure 17. The increase in pipe flexibility reduces the percentage of the preload that has to be considered as load controlled. As the figure indicates, the primary fraction of the preload may approach zero in the

**Table 1. Effective Stresses and Strains from Equivalent Elastic Analysis Cases**

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<th>ε_eff</th>
<th>E(70°F)/E</th>
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CONCLUSIONS

Previous analytical work reported in [3] utilized a combined 2-D interaction model to evaluate the effect of preloaded, insulated clamps on thin-walled liquid metal piping. While it was judged at that time that this approach was adequate, there were a couple of questions which required a more detailed examination of the clamp/pipe interaction effect. These questions concerned the magnitude of conservatism in the combined 2-D/3-D approach and the proper classification of clamp-induced pipe wall stress components. A 3-D interaction model of one clamp/pipe combination was set up to investigate these questions. This model was later modified using a number of super elements (substructures) to reduce the computational costs and problem turnaround times for subsequent analysis.

The 3-D nonlinear interaction model results confirmed that the previous combined 2-D/3-D approach was conservative, as expected, without being overly conservative and gave reasonable estimates of the combined clamp/pipe stiffnesses for piping loop seismic analysis.

Subsequent inelastic and equivalent elastic analysis indicated that the membrane portion of the clamp induced pipe stresses should be treated as a primary stress as defined in [4] and [5] and the bending portion was essentially secondary. However, to ensure that a moderate amount of conservatism remained in the treatment of the pipe stress due to clamp effects a judgement was made to classify the bending component as 1/3 primary and 2/3 secondary. Ultimately, this classification procedure will be utilized when the pipe stress components due to clamp effects are combined with the global pipe stress state due to operating conditions to satisfy the ASME requirements of [4] and [5].

REFERENCES


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