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PREDICTION AND TESTING (ASTM E-837-89) OF RESIDUAL STRESSES IN STRUCTURES (OPERATED IN SOHIC ENVIRONMENT) CAUSED BY WELDING

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ABSTRACT

The study and testing of the structural behavior of a failed pipe with a spiral weld seam was conducted. Our study and testing was part of a larger failure investigation to determine the influence of residual stress for initiation of hydrogen stress cracking. Finite element and structural testing methods were used to guide the study for determination of residual stresses. All properties used in our study were measured per ASTM A-370. The validation of the finite element model of the pipe was conducted against the test results per ASTM E-837-89. The difference in predicted and measured residual stresses was 3% at the ID of the structure. Due to the small differences in the results from analysis and testing, no correlation was required, and the test-validated finite element model was used to predict the total stresses due to the manufacturing processes and operational loads. Based on the validation results, one can conclude that the finite element technique is accurate for predicting residual welding stresses. The finite element technique, however, is far less time-consuming and thus less expensive method than mechanical testing for determining the residual stresses.

INTRODUCTION

A section of 24-in off-gas piping located in an NGL plant failed in service due to stress oriented hydrogen induced cracking (SOHIC). Figure 1 shows the unzipping of the pipe directly adjacent to the heat-affected zone of the spiral weld. Any rupture or failure of a large gas-carrying pipe is considered to be major, since the consequences (i.e. injury to personnel, loss of equipment, and down time) are great.

SOHIC is a specific form of hydrogen-induced cracking (HIC) that results in development of through wall cracking in areas of elevated stress. The stress necessary to initiate cracking can be either residual, operation stress, or externally applied. In this specific case the pipe base material had been tested and shown to be resistant to HIC damage and therefore suitable for "sour" service. Limited laboratory data indicate that stress levels as low as 40% of a materials specified minimum yield stress can initiate SOHIC in severe hydrogen charging environments. One additional point regarding SOHIC is that cracking initiates (as small blisters) within the material and not as surface cracking as in other "environmental cracking" morphologies. The initiation of subsurface blistering (cracking) in a material not susceptible to 'normal' HIC damage makes this one of the harder failure mechanisms to predict. This study does not include data for the evaluation of the effects of the hydrogen gas in the initiation and propagation of cracking in either the base material or weldment. However, the knowledge gained from this study will assist the material scientists in understanding the direction and potential magnitude of induced stresses, and to formulate ways to improve the fabrication of the pipe (e.g. welding the material along the edge parallel to the rolling direction, or cold expanding the material) in order to mitigate the risk of hydrogen damage. The source of the residual stresses in a pipe similar to the one under study is the bending of the surfaces caused by thermal contraction of the weld. The resultant residual tension located adjacent to the spiral weld produced conditions within the pipe body that allowed accumulation of molecular hydrogen and subsequent SOHIC propagation.

The purpose of this study is to quantify the total stresses due to the manufacturing processes and operational conditions in the structure of a pipe with a spiral weld seam, similar to the pipe that failed. The pipe considered for this study is made of a flat plate, which is first bent to have a circular cross section (spiral-like), and then welded at its seam. The operational environment consists of pressure loads, axial forces caused by pressure, and thermal-mechanical loads. The steps taken to predict the total stresses are: 1) first to apply the thermal-mechanical loads due to the welding; 2) validate and if required correlate the finite element model; 3) use the test-validated finite element model to apply the operational loads.

STRUCTURAL TESTING

The primary objective of the testing was to measure, evaluate and compare the residual stresses on the ID and OD surfaces of the pipe, at several locations. These locations included the weld metal, heat-affected zone, and other locations some distance from the weld [2].

The test procedure followed in our study was ASTM E-837-89. This type of testing is semi-destructive and involves attaching strain gages to the surface, drilling a hole in the vicinity of the gages, and measuring the relieved strains. The measured strains are then related to relieved principal stresses through a series of equations [1].

Several measurements were taken on the pipe [2]. As each hole was being drilled, the data acquisition system produced three "strain & hole depth" versus time curves, per graph, in real time. The three final strains were later corrected for hole diameter, and produced two principal stresses in addition to their angle with the first element of the rosette. These two principal stresses were also calculated and expressed in relation to the weld's longitudinal and transversal directions [2].

The vast majority of the residual stresses were compressive (negative values) at the OD, and tensile (positive values) at the ID of the pipe [2]. In general, peak residual stress values located at or near the heat-affected zone, were of the order of 80% to 100% of the estimated material's yield strength, estimated at 52 ksi at the time of testing [2].

Residual stresses were measured with Blind Hole Drilling Techniques [1]. A series of measurement locations were made at the centerline of the spiral weld, and at each side of the weld, near the heat-affected zone, and 1/2" away from the toe of the weld, totaling 5 measurements per weld location. Additionally other measurements were made away from the weld [2].

Strain gages for blind hole drilling were rosettes. A mill guide with high-speed air turbine (Figure 2) was used to drill the holes with a bit diameter of 0.062 inches and a depth of the hole 0.080" [2].

Five holes were drilled on the OD of the pipe, one at the centerline of the weld, two at 0.469" at either side of the weld, and two at 0.844" at either side of the weld. Measurements revealed compressive residual stresses. The maximum residual stress (least negative) was in the direction parallel to weld. Five more holes were drilled on the ID of the pipe, one at the centerline of the weld, two at 0.469" at either side of the weld, and two at 0.844" at either side of the weld. Measurements revealed tensile residual stresses. The maximum residual stress (least negative) value was in the direction transverse to weld [2].

FINITE ELEMENT ANALYSIS

In order to determine the factors that contributed to the failure, a finite element technique was used to mathematically model the pipe. The pipe in our study goes through two manufacturing processes. The first process is the bending of a flat plate until a spiral shape is achieved and the two opposite edges (free edges) of the plate meet. The first process introduces a permanent deformation in the entire pipe material. This process does not, however, introduce any permanent deformation at the free edges of the bent plate. The second manufacturing process is the welding of the free edges. The welding process also introduces permanent deformation local to the weld metal and the heat-affected zone.

Since the welded region of the pipe goes through plasticity due to the welding, we did not use the property values from ASME standards. We measured all the material properties and dimensions in our Materials Lab. ASTM A-370 standards were used to fabricate the test samples of the welds and base plate and test for their elastic and mechanical properties, Table 1. The dimensions of the pipe and the weld were measured and the results have been tabulated in Tables 2 and 3.

Due to the high assembly temperatures produced by the welding process, there will be nonlinear effects in the structure of the pipe. The nonlinearity is due to the large deformations (translations and rotations) caused by the permanent set in the material when the weld region cools from the temperature of metal solidification to room temperature [3]. Therefore, the temperature load of the weld material was set equal to -2,900 Degrees F which represented the solidification temperature of the weld metal, while the temperature load of the remainder of the structure was set equal to the room temperature, i.e. 70 Degrees F. Solution sequence 106 of the MSC.Nastran [4] was used to solve the nonlinear problem. This particular solution sequence can solve problems that contain both the material and geometric nonlinearities.

Quadratic and triangular (CQUAD4 and CTRIA3) plate elements were used to model the pipe. These elements are of isoparametric types and are capable of predicting the bending, shear, and membrane stresses [4]. These elements are also capable of predicting the stresses in the plastic region of the material. The finite element model of the pipe contains 121 triangular plate elements, 3,169 quadratic plate elements, 3,288 grid points, and approximately 19,700 degrees of freedom. The boundary conditions for the finite element model of the pipe consist of the thermal loads due to the welding, the internal pressure loads, and the axial forces due to the internal pressure.

Although the focus of this study was to test-validate a pipe with a spiral weld against ASTM E-837-89, a second finite element model of a pipe with a straight weld was also built and exercised. The purpose for this second model was to study the differences in residual stresses and displacements, and furthermore determine if a pipe with a straight weld is structurally superior to a pipe with a spiral weld.

VERIFICATION OF FINITE ELEMENT MODEL

The verification methodology followed in [5] was used in our study to verify the finite element models of the pipes, the following:

1. The finite element model of the pipe was fixed at one end and subjected to a unit gravity load. A static solution (Solution Sequence 101) was then exercised. The single-point-constraint forces were recovered. The total single-point-constraint forces were calculated to be equal to the weight of the model. This procedure proved that the geometry, mass, and stiffness matrices of the finite element models are correct.

2. The finite element model of the pipe was freed (all single point constraints were removed). A normal mode solution (Solution Sequence 103) was then exercised. The solution resulted in producing six rigid body modes, and the maximum strain energies were all in the order of 1E-05. This procedure proved that there were no improper mathematical constraints or connectivties in the finite element model.

3. The last step in the verification process was to set the weld thickness and pipe thickness equal, and apply a unit internal pressure. The predicted stresses using Solution Sequence 101 were within 1% of the stresses from a hand calculation. This check proved that the sense of pressure loading was correct.
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RESULTS AND DISCUSSION

Figures 3 represents the displacement plots of the pipe model with a spiral weld seam. In this model the mechanical loads (internal pressure and axial forces due to the internal pressure) have been removed. As can be observed, the contraction of the weld causes the pipe to displace inward at the weld seam resulting in bending of the weld and the heat-affected zone.

Figures 4 and 5 represent the stress plots of the pipe at the ID and OD respectively. In this model the mechanical loads have been removed. As can be observed, the stresses at the weld are close to the measured yield strength of the material. The stresses in the heat-affected zone, adjacent to the weld area, are in the plastic region. The bending stresses at the ID of the pipe are in tension, thus producing conditions that allowed hydrogen damage initiation.

Figure 6 shows the displacement plots of the pipe model. In this model the mechanical loads (internal pressure and axial forces due to the internal pressure) are present. As expected, the contraction of the weld causes the pipe to displace inward at the weld seam. Furthermore, when comparing the displacements of this figure to the ones from Figure 3, one can conclude that the pressure loads tend to straighten the displaced region of the weld, and thus reduce the bending/residual stresses.

Figures 7 and 8 represent the stress plots of the pipe at the ID and OD respectively. In this model the mechanical loads (internal pressure and axial forces due to the internal pressure) are present. The stresses at the weld seam in this model are close to the yield strength of the material. The stresses in the pipe material, adjacent to the weld area, are in the plastic region.

For our type of operation, the differences in stresses from both loading cases (Case 1: mechanical loads have been removed and Case 2: mechanical loads are present) are very small, Table 4. Therefore, the measured stresses in the weld could be accepted as the total stresses that are experienced in the field.

The magnitudes of stresses in the pipe with a straight weld are higher than the ones from the pipe with a spiral weld (Table 4). The highest stresses in the pipe with a straight weld seam are experienced in a wider area when compared to the highest stresses in a pipe with a spiral weld seam. The reason for the lower stresses in the pipe with a spiral weld is the curvature of the weld that acts as a natural and additional stiffener. The displacements of the spiral weld are higher than the displacements of the straight weld (approximately twice).

VALIDATION OF FINITE ELEMENT MODELS

The validation of the finite element model of the pipe consisted of calculating the equivalent stresses from [2], and comparing their values to the stresses from the finite element analysis. The maximum measured principle stresses at the ID, on the weld, were 71,881 psi and 59,971 psi. The minimum measured principle stresses at the OD, on the weld, were –74,618 psi and –31,315 psi. The resulting equivalent stresses based on these measured stress values were 66,728 psi and 64,898 psi respectively. The predicted stresses at the respective locations were 62,700 psi and 63,000 psi. The stresses in two orthogonal directions at 0.469" and at 0.844" at either side of the weld were also measured [2]. The equivalent stress based on measurement at the OD was 57,684 psi versus the predicted equivalent stress of 58,500 psi from the finite element analysis. The equivalent stress based on measurement at the ID of the pipe was calculated to be 30,858 psi. This measurement could not be accurate, since the stresses at the OD and ID of the pipe should be of approximately the same magnitude with different directions

The difference between the equivalent stresses (measured and predicted) at the ID, on the weld, was calculated to be 5.5%. The difference between the equivalent stresses (measured and predicted) at the ID, on the weld, was calculated to be 3.4%. The difference between the equivalent stresses (measured and predicted) at the OD, and away from the weld, was calculated to be 1.4%.

ASTM E-837-89 test is accurate for measurements of stresses that are below 50% of the yield. When the stresses are above 80% of the yield, the accuracy in stress measurement diminishes rapidly [1]. Therefore, the measured stresses on the base metal should not be accurate due to the first manufacturing process of the pipe, and no attempt was made to validate the finite element model to any of these results.

CONCLUDING REMARKS

The successful validation of the finite element model of the pipe against the test conducted proved that finite element technique is accurate to predict residual stresses due to the welding.

The accuracy of the ASTM E-837-89 test will diminish as the stress approaches the yield strength. The measurements of stresses in the base metal at locations remote from the weld are indicative of this fact.

When operation pressure is relatively small, the stresses in the welded regions increase slightly. Therefore, the residual stresses measured per ASTM E-837-89 can be looked upon as the total stresses.

The stresses due to the welding in a pipe with a straight weld seam are higher than the ones in a pipe with a spiral weld seam. These larger stresses, however, have not caused failure due to the SOHIC, since this class of pipes is normally cold expanded, and the welding is done parallel to the rolling direction (i.e. normal to the short grains).

ACKNOWLEDGMENTS

We thank the staff of MT&MG for their efforts to obtain the elastic and mechanical properties of the material needed for this study. Machining test samples from the weld and the base of the pipe per ASTM A-370 standards was very difficult.

REFERENCES

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[3] Armand, S., Liao, M.H., Morris, R.W., "A Transient Plasticity Study and Low Cycle Fatigue Analysis of the Space Station Freedom Solar Array Blanket", NASA Technical Memorandum, March 1990

[4] The MacNeal-Schwendler Corporation, "MSC.Nastran Handbook for Nonlinear Analysis", Version 67

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Table 1: Material Testing of Pipe and Weld per ASTM A-370

SAMPLE NO.	***1***	***2***	***3***	***4***
Yield Load (Klb)	**3.07**	**3.02**	**3.23**	**2.70**
Yield Stress (0.2% offset)(Ksi)	**62.56**	**61.48**	**65.80**	**55.01**
Ultimate Tensile Load (Klb)	**3.88**	**3.81**	**4.16**	**3.48**
Ultimate Tensile Stress (Ksi)	**79.07**	**77.64**	**84.73**	**71.0**
Elongation 2" G.L. (%)	**40.00**	**30.10**	**38.00**	**32.50**
Reduction of Area (%)	**72.96**	**75.00**	**74.60**	**78.84**

Note:

Sample # 1:Transverse to the weld, Sample # 2: Parallel to the weld, Sample # 3:On the weld, Sample # 4: Transverse to the rolling direction

Table 2: Dimensions of Pipe

SAMPLE NO.	*Inside Diameter(In)*	*Outside Diameter (In)*	*Thickness (In)*
1	**23**	**24**	**0.488**
2	**23**	**24**	**0.495**
3	**23**	**24**	**0.495**
4	**23**	**24**	**0.497**

Table 3: Dimensions of Weld

SAMPLE NO.	***Thickness(In)***	***Width (In)***
1	**0.630**	**0.775**
2	**0.625**	**0.770**
3	**0.632**	**0.772**

Table 4: Equivalent Stress at Weld

SAMPLE NO.	*Test OD (ksi)*	*FEM OD (ksi)*	*% Diff.*	*Test ID (ksi)*	*FEM ID (ksi)*	*% Diff.*
Spiral-weld pipe without mechanical load	**64.9**	**62.7**	**3.4**	**66.7**	**63.0**	**5.5**
Spiral-weld pipe with mechanical load	-	**62.7**	-	-	**63.2**	-
Straight-weld pipe without mechanical load	-	**67.1**	-	-	**69.6**	-

Figure 1: Picture Of Pipe With a Spiral Weld

Figure 2: Pipe Under Structural Testing

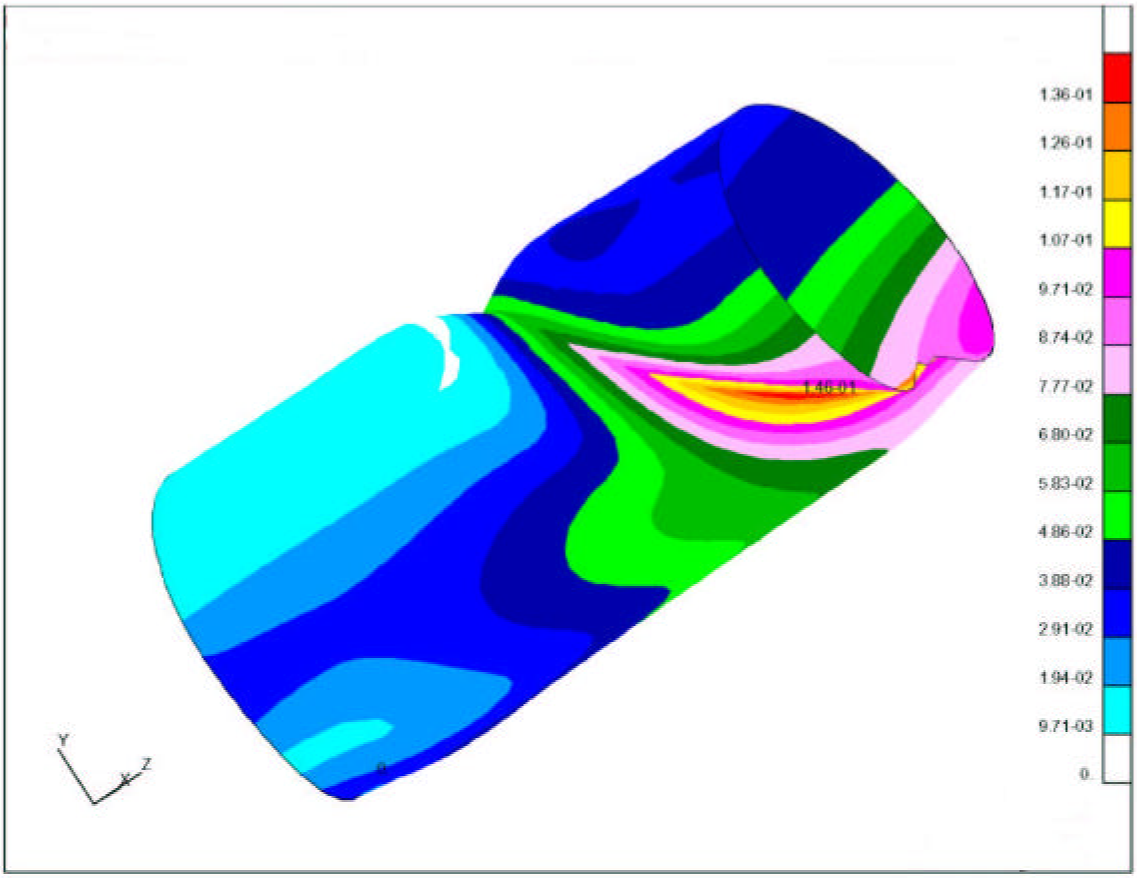

Figure 3: Displacement Plot of Pipe With a Spiral Weld, No Mechanical Loads Present

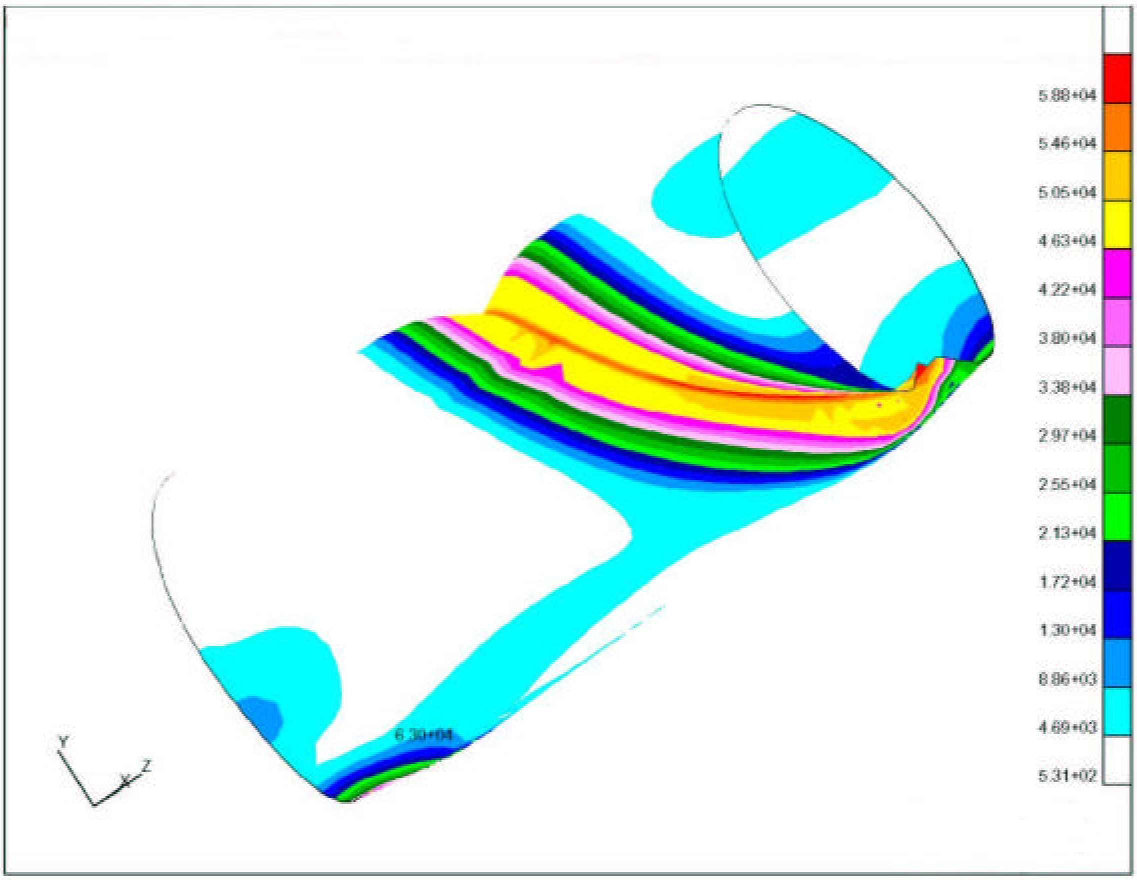

Figure 4: Stress Plot of Pipe With a Spiral Weld at ID, No Mechanical Loads Present

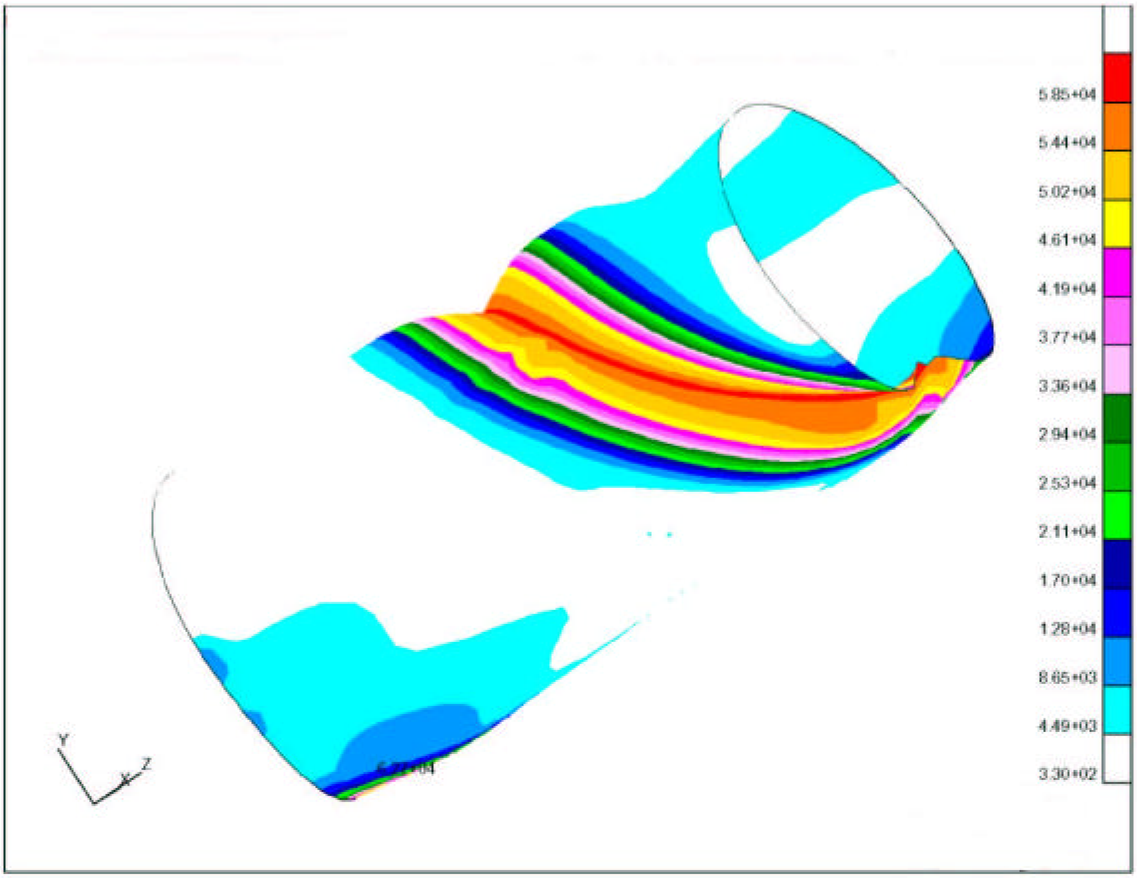

Figure 5: Stress Plot of Pipe With a Spiral Weld at OD, No Mechanical Loads Present

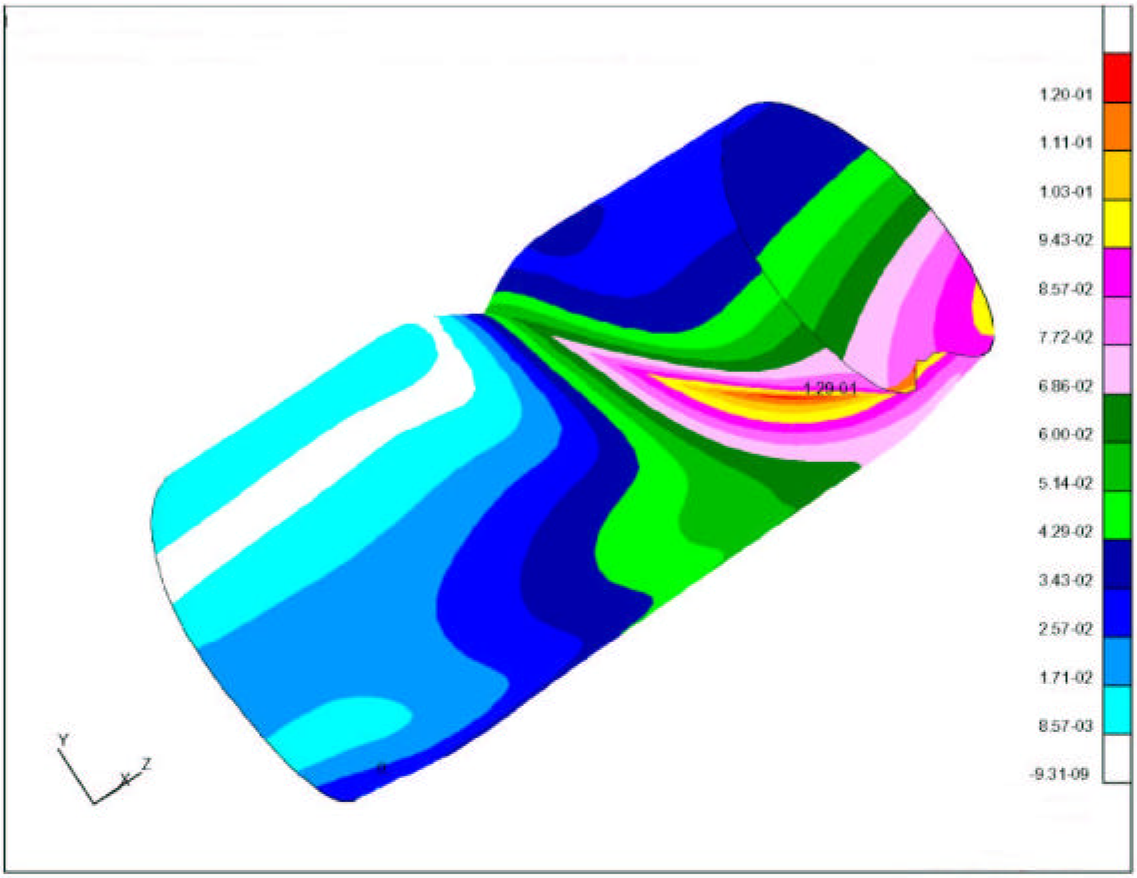

Figure 6: Displacement Plot of Pipe With a Spiral Weld, Mechanical Loads Present

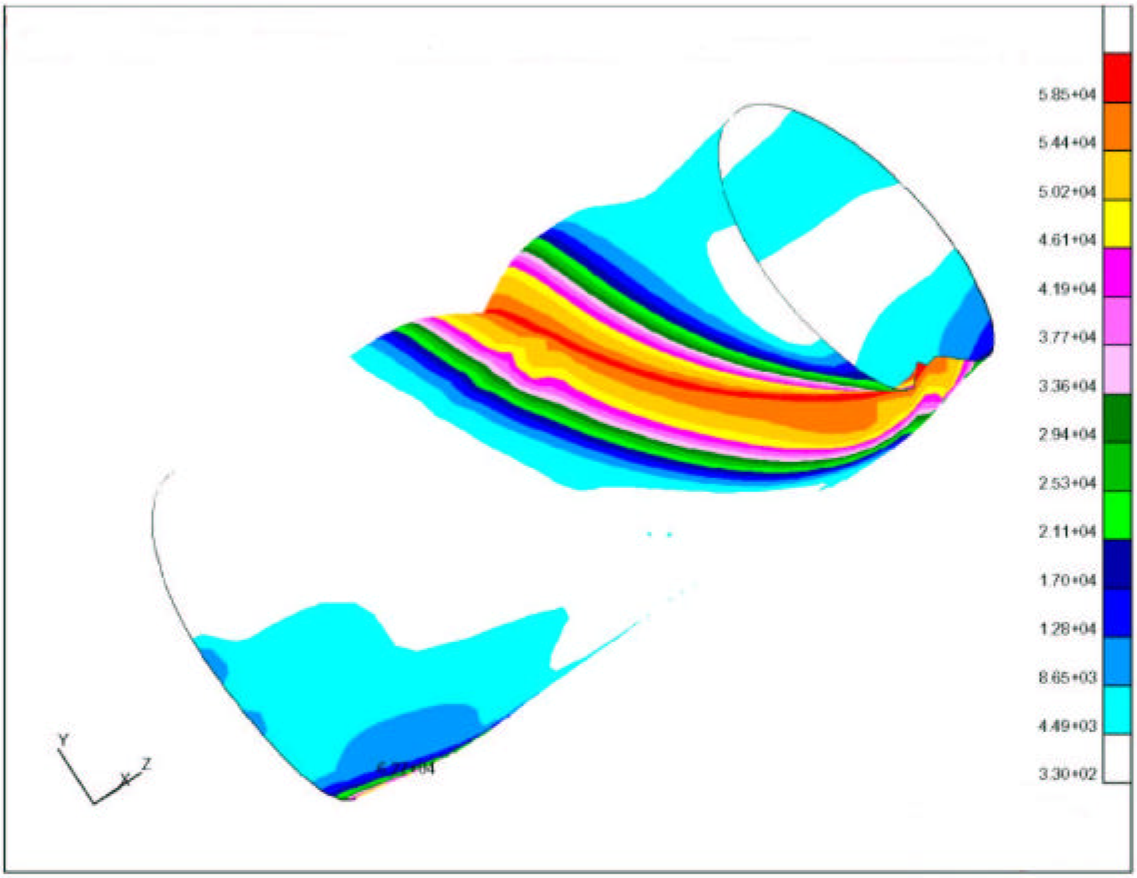

Figure 7: Stress Plot of Pipe With a Spiral Weld at ID, Mechanical Loads Present

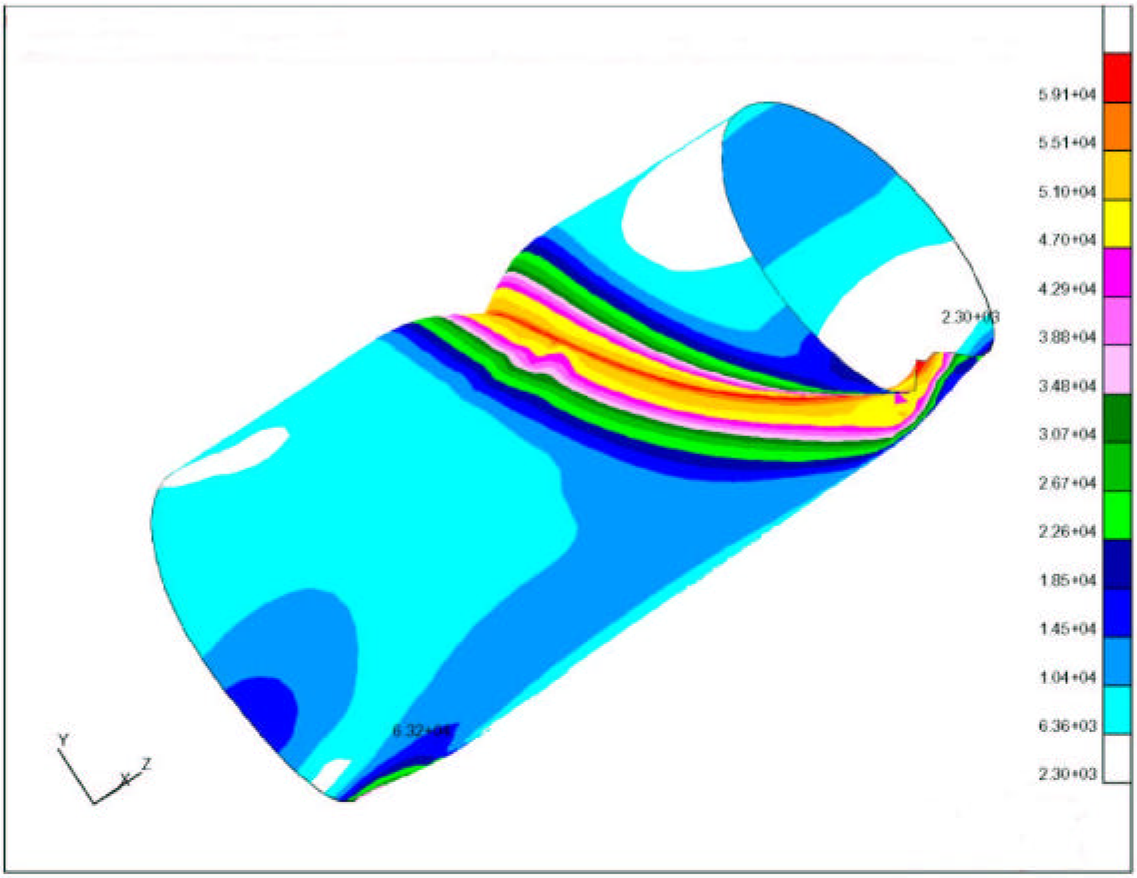

Figure 8: Stress Plot of Pipe With a Spiral Weld at OD, Mechanical Loads Present