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TOPICS IN VIBRATORY STRESS RELIEF OF WELDMENTS

Michael E. Robbins

A seminar submitted to the faculty of Rensselaer at Hartford in partial fulfillment of the requirements for the degree of Master of Science

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Topics in Vibratory Stress Relief of Weldments

by

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A Seminar Submitted to the Faculty of Rensselaer at Hartford in
Partial Fulfillment of the Requirements for the
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Major Subject: Mechanical Engineering

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Prof. Ernesto Gutierrez-Miravete

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Abstract

Welding processes are used for many applications and are being applied to even more as companies try to manufacture lighter and simpler products. The welding process can join two similar materials with a bond that has similar mechanical properties to the original material. Unfortunately, this process leaves residual stresses in the weldment, which if left untreated can cause distortion of the part, premature failure in fatigue, or cracking along the weld. A post-weld heat treatment is the traditional method of relieving these stresses, but is costly and a time consuming process. For weldments, which see heavy fatigue loading or are used for critical operations the expense of the heat treatment is required, since the post-weld heat treatment method reduces the residual stresses in the weldment, generates more uniform mechanical properties, and can change the crystal size and structure of the material. The post-weld heat treatment is an unnecessary expense for weldments where the only concerns are dimensional stability and light fatigue loads. Vibratory stress relief techniques could be used to substitute the heat treatment for these types of weldments and save time and money.

It is the purpose of this paper to give a brief overview of the creation, measurement, and reduction of residual stresses. In addition, an overview of the vibratory stress relief method and its applications is discussed. Finally, a computer model was developed using the finite element modeling program ANSYS. Using ANSYS, a cantilever beam was pre-stressed and then subjected to longitudinal cyclic displacements at the beams natural frequency and several random frequencies to see the affect on the residual stresses. It was shown that the amplitude of the displacement of the beam had the greatest affect on the reduction of the residual stresses and not the frequency of loading as originally thought.

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Symbol	Definition	Dimension
δ_x	Change in length in the x direction	in
$\sigma_{r,i}$	Initial residual stress	lb/in^2
$\sigma_{r,n}$	Residual stress after n cycles	lb/in^2
σ_x	Stress in the x-direction	lb/in^2
ε_x	Strain in the x direction	in/in
ν	Poison's Ratio	in/in
L_0	Original length of the beam	in
L	Length of beam after any displacement	in
E_x	Young's Modulus in the x direction	lb/in^2
t	Time	sec
ρ	Density	lb/in^3
S_y	Yield strength	lb/in^2
S_{ut}	Ultimate tensile Strength	lb/in^2
f_n	Natural Frequency	Hz

1. INTRODUCTION

1.1 Background

Welding processes are used in the manufacturing of products ranging from simple products, such as toasters and chairs, to complex products, such as ships and skyscrapers. The welding process uses heat to fuse two pieces of metal together to create a metallic bond with properties similar to those of the original material. According to James Brumbaugh, there are seven widely used methods of generating the heat for welding: electricity, gas flame, forge or furnace heating, electron beam, chemical reaction, ultrasonic vibration, and laser beam (13-15). This study will be looking at the general fusion welding of low-carbon steel; the melting together of the base metal or base metal and a filler metal creating a union between parts. Fusion welding is commonly used in building large weldments for tooling and ships.

The problem with this manufacturing technique is that heat generated for the fusion has the adverse affect of generating residual stresses in the metal, which can cause the welded piece to deform over time or can cause early failure of the weldment, as shown below in Figure 1.1.1. In addition, the residual stresses can lead to stress cracks forming near the welded. Residual stresses are stresses, which exist without loading or restraints. It is the goal of this paper to discuss the mechanics behind the vibrational stress relief of weldments, different methods for measuring stress in metals, and show a computer model of a simple vibrational stress relief problem.

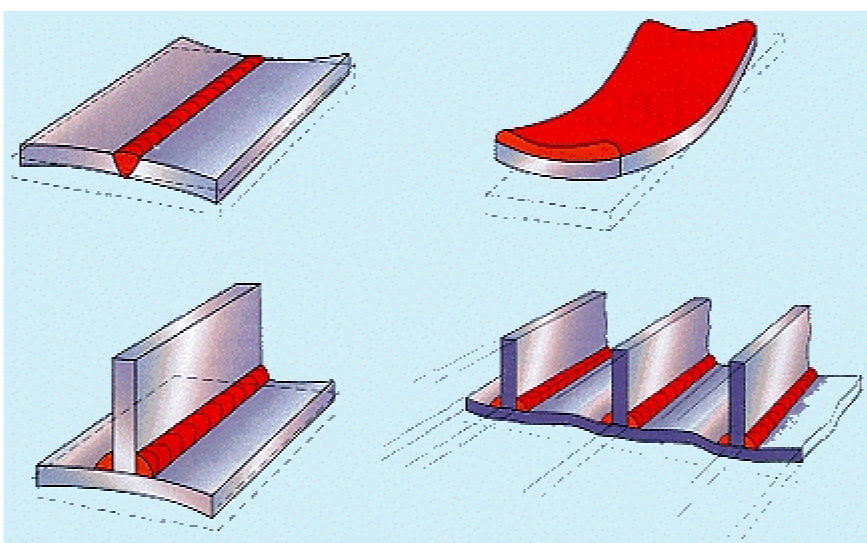

Figure 1.1.1: Examples of Deformations Due to Welding (TWI 2004).

1.2 Welding Stresses

Regardless of the type of fusion welding used, residual stresses are introduced into the welded piece. These residual stresses are caused by the rapid heating and cooling of the material close to the heat source. The temperature changes vary through the material being welded in relation to the distance from the center of the weld and are categorized into three zones, as shown in Figure 1.2.1.

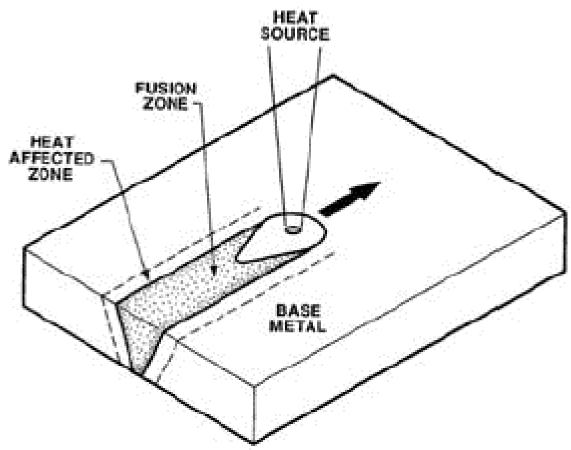

Figure 1.2.1: Diagram Showing the Three Weld Affected Zones (Lampman 4).

The first is the fusion zone (FZ), which is where weld metal resides. Next is the heat-affected zone (HAZ), which sees the temperature gradient. The American Welding Society defines the HAZ as the “portion of the base metal which has not been melted, but whose mechanical properties or microstructures have been altered by the heat of welding or cutting” (Welding Handbook 9). The final zone is the unaffected base metal.

According to Helmut Wohlfahrt, residual stresses due to welding are due to both plastic and elastic deformations (40). A quick look at the welding process can show how this occurs. As the material is heated and locally liquefies, it tries to expand, but is constrained by the cooler base metal surrounding the heated area, creating compressive stresses in the material. Then the heated area non-uniformly contracts as it quickly cools, while the cooler stronger base metal surrounding the fusion zone stays in place, creating tensile stresses, microscopic deformations, transient stresses, and phase changes which all contribute in the creation of the residual stresses. In simpler terms, the weld is being “stretched” by the base metal (Molzen & Hornbach, 39).

Figure 1.2.2 shows the effect of a simple butt-weld with a residual stress distribution loaded by uniform external loads. Curve 0 shows the distribution of the longitudinal residual stress in the as-welded condition. Curve 1 shows the stress distribution when σ_1 is applied and curve 1’ is the stress distribution after

σ_1 has been applied and released. Curve 2 shows the tensile stress increase to σ_2 and curve 2' is the stress distribution after σ_2 has been applied and released. Curve 3 shows that the stress distribution eventually becomes even and the effects of the residual stress decreases (Conner, 226-227). Over time, residual stresses will naturally relieve themselves, causing distortions in the welded part.

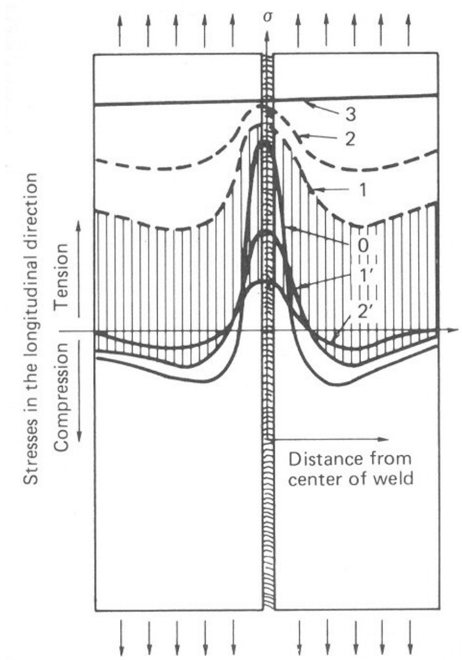

Figure 1.2.2: Diagram of Typical Longitudinal Stresses in a Butt Weld (Lampman, 237).

Residual stresses can be both harmful and beneficial to a part. Tensile residual stresses contribute to cracks developing in the HAZ. These cracks can occur during the solidification process, due to the inability of the hardening material to handle the thermal shrinkage strains (Lampman, 71). The tensile residual stresses reduce the effective fatigue life of the weldments as well, which leads to premature part failure. Figure 1.2.2, shows that the stresses near the center of the weld and portions of the HAZ are stressed in tension.

On the other hand, some residual stresses are beneficial to a part, such as compressive residual stresses that can increase the fatigue strength of a part or weldment (Hilley, 3). In either case, post weld stress relief methods are used to reduce the effective residual stress in the part, which will be discussed later in this paper.

2. MEASURING THE RESIDUAL STRESSES

It is important to know how large the residual stresses are in a part to predict the performance of the part or weldment. An example of the importance of knowing the residual stress in part would be if an engineer assumed the service load would not exceed 30,000 psi of tension and there is an 8,000 psi residual stress in tension in the part. This would mean that the safe service load would be reduced to 22,000 psi of tension. On the other hand, if the residual stress was in compression, the same service load would be increased to 38,000 psi in tension (Hilley, 3). Residual stresses cannot be measured using the standard displacement or strain-gage measurements since these methods only measure change in stress due to applied loads. There are several different methods of measuring the residual stresses in a part or weldments; including the slitting method, x-ray diffraction, holographic method, and strain-gage method. Figure 2.1 shows the depth ranges for each of the methods.

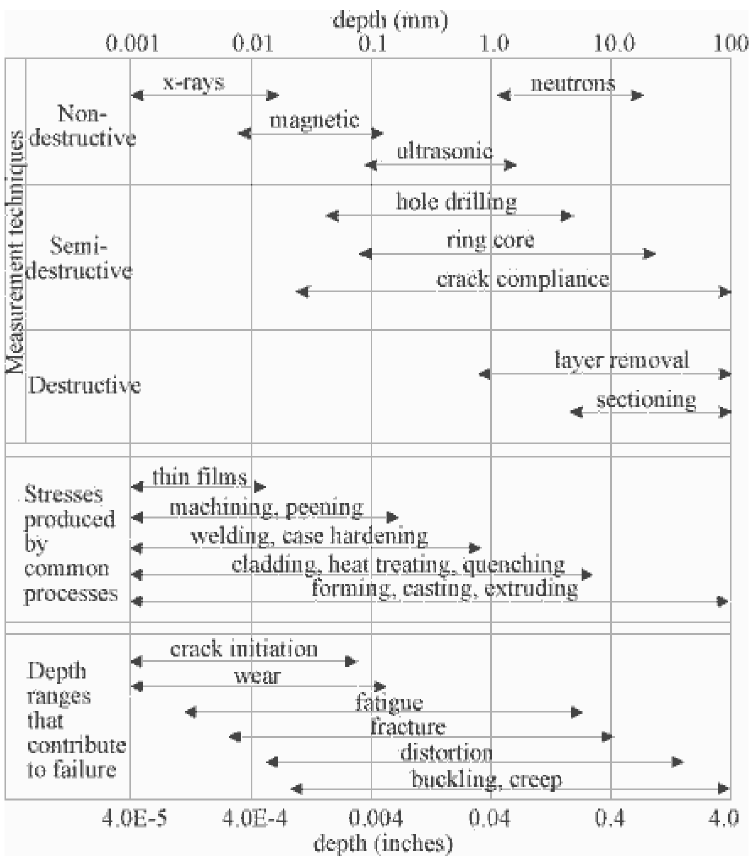

Figure 2.1: Comparison of Residual Stress Measurement Methods (Los Alamos, Slitting Method, 2004).

2.1 Stress Relaxation Technique

The stress relaxation technique is a commonly used method for measuring the residual stress in a part. This technique is accomplished by drilling a hole into the material and measuring the surface strains that result. This can be accomplished by the Mather-Soete Drilling Technique, which places three strain gages equally spaced around a center point, as shown in Figure 2.1.1 (Connor, 232-233). A hole is the carefully drilled in the center and measurements of the changes in the stress are recorded. This method is reliable, but requires a smooth flat surface to mount the strain gages and mounting strain gages is time consuming (Steinzig, 1). There are several other methods used for solid cylinders and tubes and for three dimensional solids, which follow similar procedures as the Mather-Soete Technique, refer to (Conner, 231-234).

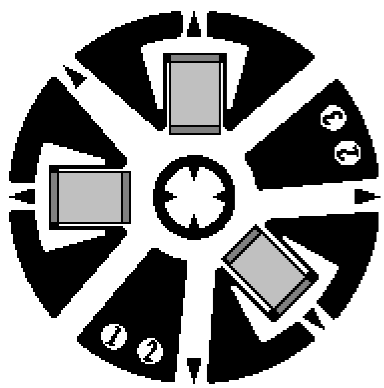

Figure 2.1.1: Strain Gage Arrangement for Measuring Residual Stress (Micro Measurements, 2004).

Other methods using the relaxation technique include optical and holographic methods. These methods use a laser beam to measure the displacement around the hole. The advantage of this method is that the setup time is minimal and the displacements are converted by software into the residual stresses (Steinzig, 1). The displacements are compared to known states of stress (Steinzig, 1). For more information on this topic, refer to (Steinzig, 1).

2.2 Slitting Method

The slitting method is similar to the relaxation method in that it cuts the material and measures the resulting displacement using strain-gages. The slit is cut by wire electric discharge machining (EDM), figure 2.2.1, milling cut, or by a saw blade. The wire EDM method is preferable because it will introduce only a small amount of stress and it makes a narrow slot about 0.0035” wide (Los Alamos, Slitting Method, 2004).

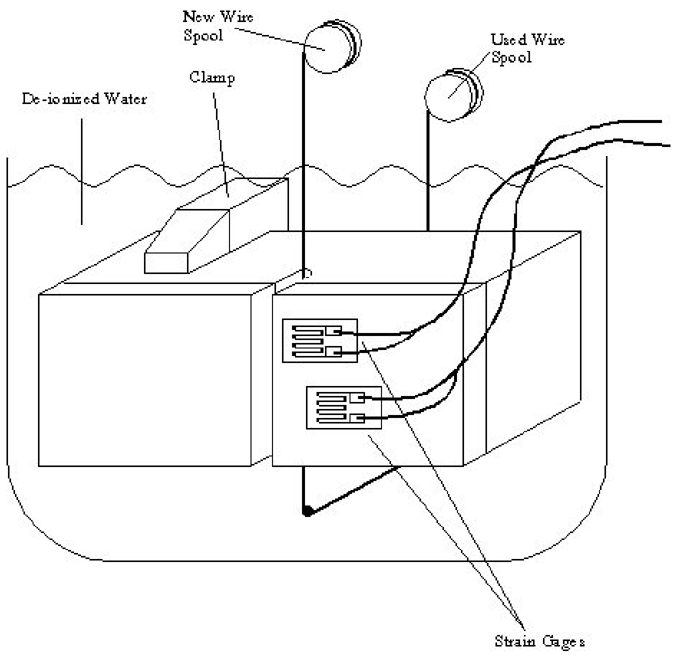

Figure 2.2.1: Wire EDM setup for Measuring Residual Stress (Los Alamos, Slitting Method, 2004).

2.3 X-Ray Diffraction

According to (Molzen & Hornbach, 40) "x-ray diffraction is the most accurate and best developed method for quantifying residual stress due to various mechanical/thermal treatments…" The x-ray diffraction technique looks at the displacements in the atomic lattice arrangements, which are altered by stress. The measurement is taken "by measuring the angular position of a diffracted x-ray beam" (Hilley, 3). The change in the measurement, which is the change in the spacing of the atomic planes, is a strain that can be converted into stress, for more information and derivation of equations refer to (Hilley, 12-16).

The stress for any direction can be calculated if strain is measured in two different directions. This method is non-destructive only if the surface stresses are measured. To get the sub-surface stresses, material is ground away to expose the layer to be measured (Hilley, 3). The disadvantages of this process include being slow and this method is not very accurate for parts that have been heat-treated.

3. BASICS OF STRESS RELIEF

There are two general techniques for decreasing or removing residual stresses caused by welding, which are by annealing (post-weld heat treatment) or through a mechanical treatment. There are many factors that effect the creation of the residual stresses and the stress relieving process, which include the material type, heat input, heat transfer rate, thickness of the material, type of fusion welding process, how the material is restrained, speed of the process, etc. The numerous factors in dealing with residual stress makes it difficult for an accurate prediction of stress relief process, so companies rely on their historical results or on experimental data in predicting results.

The annealing or heat treatment of welded materials is the standard method for reducing residual stresses in welded materials. Weldments are slowly heated to an elevated temperature, such that temperature gradients and minimized to prevent the creation of new residual stresses. The weldment is then held at that temperature or “soaked” for a predetermined length of time (Callister, 329). At this elevated temperature, the yield point of the base material of the weldment is low enough for the stressed areas to plastically relax to a lower stress state (Conner, 219). At the end of this period, the part is slowly cooled, such that new stresses are not introduced. The main problem with the heat treatment of parts is the cost in time and energy. Large weldments require an oven large enough to accommodate their size and there are only a few such ovens spread across the country. In addition, large weldments could require being “soaked” 40 hours or more to reduce the residual stresses to acceptable levels. Another heat treatment method is locally heat-treating the weld with an acetylene torch. This method will slightly reduce the residual stresses, but is time consuming and is done by hand.

There are several mechanical methods to reduce or modify the residual stresses. The first method is shot peening, which involves bombarding a part with small spherical pellets. The pellets impact the surface creating small deformations, which induce compressive stresses on the surface and tensile stresses in the sub-surface of the part. These surface compressive stresses are less likely to initiate cracking and increases resistance to fatigue failure. Cold deformation, also known as cold working, is used to stress relieve hand forgings. The part is first solution treated and quenched. Then the cold part is placed between cold flat dies and is reduced in thickness and finally the part is aged. Proof testing a weldment will also reduce the residual stresses by slightly plastically deforming the weld area, leading to a drop in residual stresses. This method is mainly used on pressure vessels and pipes.

4. OVERVIEW OF VIBRATORY STRESS RELIEF

Vibratory stress relief (VSR) has been used commercially for over thirty years, but only recently have scientists and engineers quantitatively and mechanically analyzed this method of stress relief. “The main purpose of VSR is to lower and redistribute stresses to safe levels such that the component’s accuracy and long term stability are assured” (Claxton, 1999). VSR does not work for every application, as will be discussed later in this section, but it does work well for applications that do not require the change in crystal structure or mechanical properties. In addition, there are many benefits in using VSR over the heat treatment of weldments including savings in cost and time.

According to (Claxton, 1999) there are three different methods of VSR, which are resonant (R-VSR), modal sub-resonant (SR-VSR) and sub-harmonic (SH-VSR) vibratory stress relief techniques. Resonant VSR, as the name implies, is achieved by vibrating the part at its natural frequency and large changing loads. For large complex parts, this can only be accomplished using high-force exciters with stable characteristics at high speeds (Beatyheat, 2004). If the equipment is not capable of reaching the resonant frequency of the part, then sub-resonant vibratory stress relief can be used. This is done such that the base of the resonant peak is achieved (Claxton, 1999) “Optimum results” occur when using this method, “if up to ten times the number of cycles required for the resonant VSR are applied in inverse proportion to the magnitude of the cyclic response” (Claxton, 1999). It is also stated in (Claxton, 1999) that results comparable to annealing is achievable if a combination of resonant and sub-resonant frequencies are used. Sub-harmonic VSR is the cyclic loading of the part at frequencies below the resonant frequency range. (Claxton, 1999) states that this range will result in minor stress relief, but according to (Gnirss, 1988) and (Munsi et al, 2000) this method will give the same results as annealing. They state that the amplitude of the applied cyclic loads that control the reduction in residual stress, not the frequency.

There are many benefits in using VSR compared to the traditional heat treatment of weldments; among them are savings in cost and time, as well as environment benefits. With VSR, the largest reduction in residual stress occurs with in the first ten cycles. This means that the VSR process will take ten minutes if the resonant or sub-harmonic methods are used, or just a few hours if the sub-resonant method is applied. In addition, the actuators used for VSR are small and transportable as seen in Figure 4.1 on the next page. Figure 4.1 shows a single vibrational actuator, Meta-Lax, on a base for a dynamometer that had been milled for 80 hours after welding with a distortion of less than 0.002 in after manufacturing (Meta-Lax, 2004). Meta-Lax is a VSR process developed by Bonal Technologies, Inc., which they use for both post-weld stress relief and for weld conditioning during the welding process.

Figure 4.1: Single Vibratory Actuator on a Dynamometer Base (Meta-Lax, 2004).

Vibratory stress relief can be used on any application the does not require a change in the mechanical properties or the crystal structure of a part. The VSR technique would be good for applications which require dimensional stability and applications requiring the relief of residual tensile stresses that cause early fatigue failure and stress cracking (Gnirss, 1988). Applications include the stress relief of tooling fixtures, welded frames, and machined parts. Figure 4.2 shows some more examples of VSR applications using the Meta-Lax equipment.

Figure 4.2: Meta-Lax VSR Applications (Meta-Lax, 2004).

5. FININITE ELEMENT MODEL

“The finite element method is a numerical analysis technique for obtaining approximate solutions to a variety of engineering problems” (Huebner et al, 3). T There exists an infinite number of unknowns in a problem looking at the displacement, stress, strain, or other quality. The finite element method works by simplifying the analysis. First, the shape being analyzed is divided into elements, which are connected by nodes. An interpolation function is then chosen to represent the different parameters across the element. The interpolation function is usually a polynomial, since they are easy to integrate and differentiate. The number of nodes assigned to the element dictate the degree of the polynomial. Next, matrix equations define the properties of the individual elements. The element properties are then combined to create the equations of the system. *Huebner et al* states “the basis for the assembly procedure stems from the fact that at a node, where elements are interconnected, the value of the file variable is the same for each element sharing that node” (8). The boundary conditions are defined for the problem and the system equations, which are comprised of a set of simultaneous equation, are solved.

In this study a finite element-modeling program, ANSYS, was chosen to create a computer model of the vibratory stress relief of pre-stressed beam. The beam was setup as a cantilever beam, with the left end connected to ground and the right end in space. A 2-D structural solid, the PLANE42 element in ANSYS, was chosen to represent the beam. The PLANE42 element is a 2-D element defined by four nodes each having two degrees of freedom, which are the translations in the x and y directions. In addition, the PLAIN42 element has the capabilities to show plasticity, creep, swelling, stress stiffening, large deflection, and large strain.

The shape of the beam was defined by three square elements connected by eight nodes, as shown below in Figure 5.1. The undeformed length of the beam is fifteen inches, the height is five inches, and the depth was set as one unit deep. The depth was set as an arbitrary measurement since the frequency response section of the code requires a value.

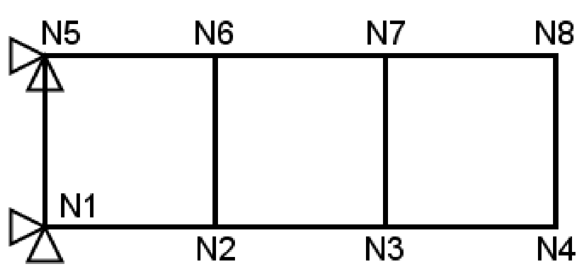

Figure 5.1: ANSYS Model used for this Analysis.

The boundary conditions for the problem were that nodes 1 and 5 were fixed and had zero degrees of freedom. The material properties, shown in Table 5.1, were chosen to be those for the aluminum alloy 6061-T6 to compare data to several other papers, which used the same material.

Table 5.1: Material Properties of Aluminum, 6061-T6 (Callister, 793-798.)

Density, ρ	Modulus of Elasticity, E	Poison's Ratio, ν	Yield Strength, S_y	Tensile Strength, S_{ut}
0.0975 lb/in^3	10x10^6 lb/in^2	0.33	40x10^3 lb/in^2	45x10^3 lb/in^2

To take in account the bilinear kinematic hardening of the aluminum as it plastically deformed, several points were taken from the stress-strain curve for aluminum 6061-T6, as shown in Figure 5.2.

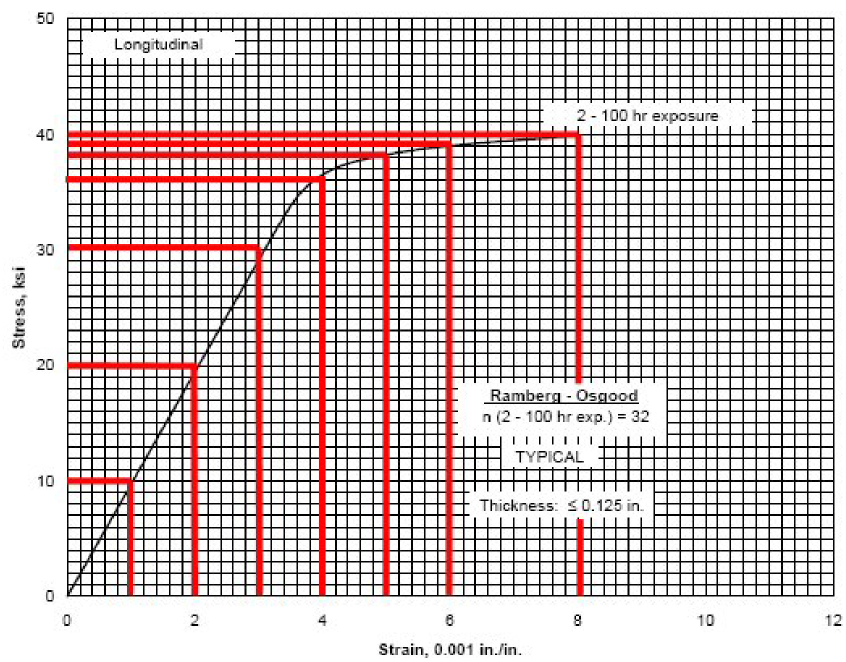

Figure 5.2: Aluminum Alloy 6061-T6 Tensile Stress-Strain Curve (MIL-HDBK-5H, 3-267).

6. DISCUSSION

The finite element computer simulation was split into two separate programs. The first part of both programs contained the material properties, defining the beam with eight nodes, and the longitudinal tensile loading of the beam. The tensile loading of the beam was used to create the residual stresses in the beam through the plastic deformation of the beam and releasing the beam so all external forces would go to zero. The distances of the longitudinal tensile displacements were chosen by looking at the stress-strain curve, shown in Figure 5.2 on the previous page, of the aluminum alloy 6061-T6 and using the strain equation to get rough estimates of values to use. Figure 6.1, below, shows a sketch of a cantilever beam and the dimensions used in this study were $L_0 = 15\,in$ and $h = 5\,in$.

$\varepsilon = \dfrac{L - L_0}{L_0} = \dfrac{\delta}{L_0}$, then solving for δ gives $\delta = \varepsilon L_0$.

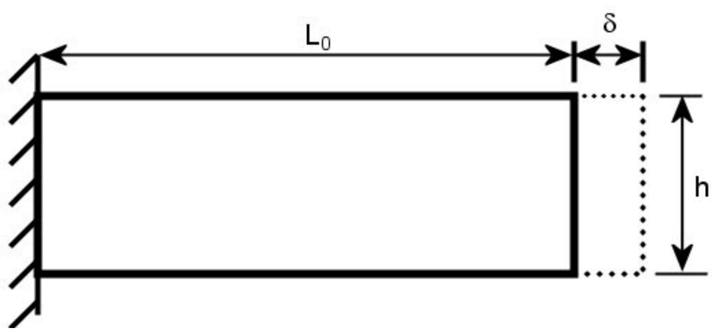

Figure 6.1: Cantilever Beam Used for in the Analysis.

Using this information, three preload cases were developed, as shown below in Table 6.1. The natural frequency was found using a modal analysis by ANSYS after the beam was pre-stressed, the ANSYS code for the frequency response is in Appendix A.

Table 6.1: Preload Parameters for the Three Test Cases.

	Case 1	Case 2	Case 3
Displacement, δ (in)	0.060	0.070	0.0725
Residual Stress, $\sigma_{r,i}$ (psi)	317.860	950.163	1364.00
Natural Frequency, f_n (Hz)	34.04	34.34	34.41

The residual stress distribution for Case 2 can be seen in Figure 6.2. Note that the highest residual stresses occur at the root of the beam. This could be considered comparable to a plate with the weld bead at the same location. Also, note that the beam was beginning to neck and remains deformed due to the plastic deformation.

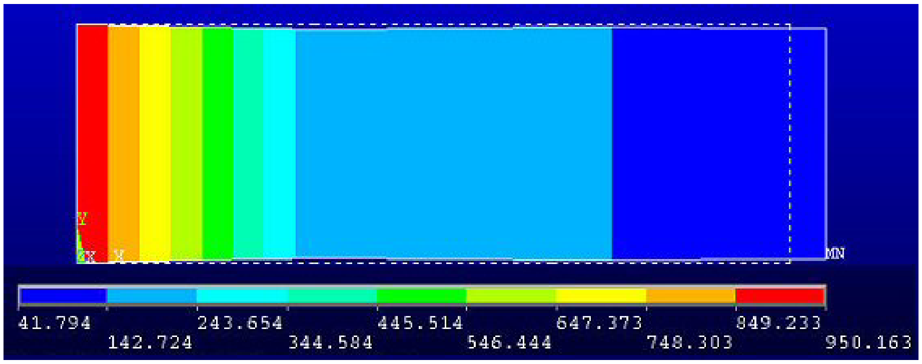

Figure 6.2: The Residual Stress Distribution for Case 2.

The cyclic loading was chosen to be in the longitudinal direction to reduce the residual stresses throughout the beam. If the cyclic loading had been tip loading in the vertical direction, then the greatest stress relief would have been at the root of the beam. The cyclic loading or rather the cyclic displacements were in the positive and negative directions and were applied using a square function, as shown below in Figure 6.2. Similar results can be achieved by just negative displacements, but the residual stresses decrease after a few cycles and then began to increase again. The cyclic displacements were applied at the natural frequency. The cyclic displacements were also tried at various frequencies, but with the same results as those at the natural frequency. In the figure below, one cycle ends at (a) on the figure and takes $t = 1/f_n$ seconds to occur, which at 34 Hz is 0.029 seconds. The ANSYS code for the cycling of the beam can be seen in Appendix B.

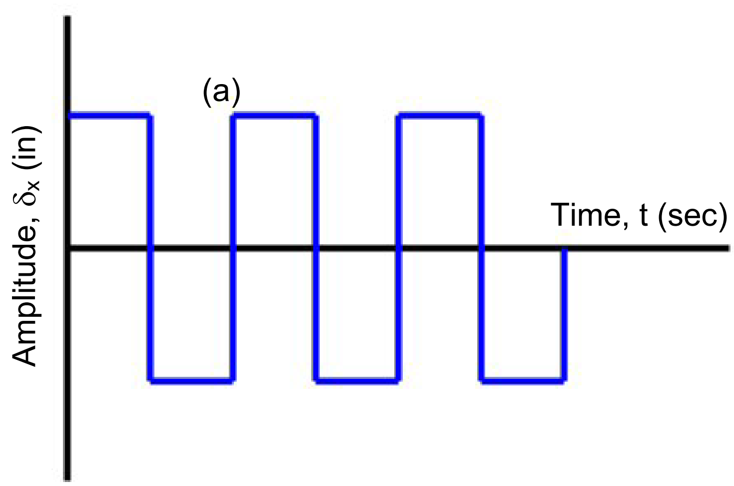

Figure 6.3: The Cyclic Displacement Function Curve.

The results were not as expected. Originally, it was thought that the greatest stress relief would occur at the natural frequency of the beam. It turned out with the longitudinal loading that the amplitude of the displacements had the greatest impact. Appendix D shows the stress distribution results for the Case 2 residual stress after 2 cycles and 20 cycles at a cyclic amplitude of 0.0575-in. This shows that after 2 cycles, the residual stress dropped 69.5% and after 20 cycles, it dropped 71%. This shows that most of the stress relief occurred in the first and second cycles and almost no additional stress relief occurred after 100 345 cycles, as can be seen below in Figure 6.4 for Case 3.

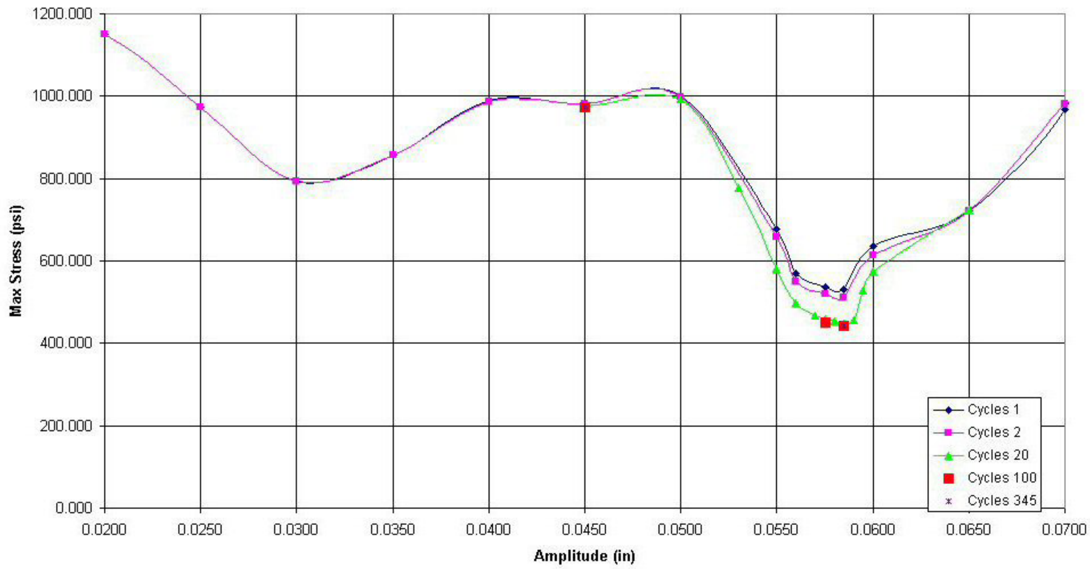

Figure 6.4: Max Stress vs. Amplitude Displacement Results for Case 3.

Note that the curve is not linear; there is a slight dip at 0.030 and then dips again at 0.0575. The low point at 0.0575 is expected due to the requirement that the summation of the external cyclic and residual stresses exceed the elastic limit of the material. (Hahn, 2002) also found this requirement to be true. Using this method of axial cyclic displacements reduced the residual stresses by 67% at the 0.0585-in amplitude in Case 3.

7. CONCLUSION

The results of this study show that vibratory stress relief (VSR) can indeed be used to reduce the residual stresses creating during the welding process. This was shown through the modeling of a simple residual stress problem using the finite element program ANSYS and by reviewing available research on the subject. The ANSYS computer model of a pre-stressed cantilever beam undergoing longitudinal cyclic displacements showed that the amplitude of the displacements controlled the amount of residual stress removed, not the frequency as originally thought. This result agrees with those by (Gnirss, 1988) and (Munsi et al, 2000), which state that the amplitude of vibration is the controlling factor in VSR. Further work should look at comparing the computer model to lab tests to verify the results. In addition, more complex finite element models should be developed with a variety of vibratory loading schemes to see if shape and complexity affect the results.

It should be noted that VSR is not for every stress relief application. VSR is ideal for parts that do not require a change in material properties or crystalline structure. For these changes only a heat treatment process, such as annealing, will work. A few examples of applications that VSR is ideal for, include the stress relief of tooling fixtures, welded frames, and machined parts where dimensional stability and a longer fatigue life are the key concerns.

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Appendix List

Appendix A
ANSYS Code for Finding the Natural Frequency of a Pre-Stressed Beam

```
/COM,ANSYS MEDIA REL. 60 (090601) REF. VERIF. MANUAL: REL. 60
/VERIFY,mer_modal ! NAME OF COMPUTER FILE
/PREP7
/TITLE, NATURAL FREQUENCY OF A PRE-STRESSED FIXED-FREE BEAM
ANTYPE,STATIC
ET,1,PLANE42,,,,,,2     ! DEFINE MODEL
!
! MATERIAL PROPERTIES - ALUMINUM 6061-T6
!
MP,EX,1,10E6      ! MODULUS OF ELASTICITY (PSI)
MP,NUXY,1,0.33    ! POISON'S RATIO
MP,DENS,1,0.1     ! DENSITY (LB/IN^3)
MP,DAMP,1,0.003   ! DAMPING RATIO
!
R,1,1       ! REAL CONSTANT, THICKNESS OF 1
!
! DEFINING THE STRESS-STRAIN CURVE
!
TB,MKIN,1
TBTEMP,,STRA
TBDATA,1,.001,.002,.003,.004,.005,.006,.008   ! STRAINS (IN/IN)
TBTEMP
TBDATA,1,10000,20000,30000,36000,38000,39000,40000 ! CORRESPONDING
!STRESSES (PSI)
!
! DEFINING THE GEOMETRY
!
N,1,0,0,0   ! DEFINING THE NODES
N,2,5,0,0
N,3,10,0,0
N,4,15,0,0
N,5,0,5,0
N,6,5,5,0
N,7,10,5,0
N,8,15,5,0
E,1,2,6,5
EGEN,3,1,1
!
D,1,ALL,,,  ! END CONSTRAINTS - FIXED
D,5,ALL,,,
!
FINISH
!
! PLASTICALLY STRESSING THE PART TO CREATE RESIDUAL STRESSES
!
/SOLU
D,4,UX,0.06 ! APPLY INITIAL DISPLACEMENTS, D1
D,8,UX,0.06
TIME,0.001  ! TIME TO COMPLETE DISPLACEMENT
NSUBST,10
```

```
SOLVE
!
OUTPR,NSOL,1
OUTPR,ESOL,1
!
DDELE,4,ALL ! REMOVING D1
DDELE,8,ALL
TIME,0.002
NSUBST,10
SOLVE
FINISH
!
! MODAL SOLUTION
!
/SOLU       ! MODE-FREQUENCY ANALYSIS
ANTYPE,MODAL
MODOPT,LANB,3
MXPAND
PSTRES,ON
SOLVE
FINISH
!
! ENTER GENERAL POSTPROCESSOR TO DISPLAY RESULTS
!
/POST1
*GET,FREQ1,MODE,1,FREQ
*GET,FREQ2,MODE,2,FREQ
*GET,FREQ3,MODE,3,FREQ
*DIM,LABEL,CHAR,3,2
*DIM,VALUE,,3,3
LABEL(1,1) = '      f1','      f2','      f3'
LABEL(1,2) = ',  (Hz) ',',  (Hz) ',',  (Hz) '
*VFILL,VALUE(1,1),DATA,FREQ1,FREQ2,FREQ3
/COM
/OUT,mer_modal,vrt
/COM,------------------ VM61B RESULTS COMPARISON --------------
/COM,
/COM,                  |   ANSYS   |
/COM,
*VWRITE,LABEL(1,1),LABEL(1,2),VALUE(1,1)
(1X,A8,A8,'   ',F10.2,'  ',F10.2,'   ',1F5.3)
/COM,---------------------------------------------------------

/OUT
FINISH
*LIST,mer_modal,vrt
```

Appendix B
ANSYS Code for the Cyclic Longitudinal Loading of Pre-Stressed Cantilever Beam

```
/COM,ANSYS MEDIA REL. 60 (090601) REF. VERIF. MANUAL: REL. 60
/VERIFY,mer02d
/PREP7
/TITLE, MICHAEL ROBBINS - SEMINAR - VERSION 03
ANTYPE,STATIC
ET,1,PLANE42,,,,,,2     ! DEFINE MODEL
!
! MATERIAL PROPERTIES - ALUMINUM 6061-T6
!
MP,EX,1,10E6      ! MODULUS OF ELASTICITY (PSI)
MP,NUXY,1,0.33    ! POISON'S RATIO
MP,DENS,1,0.1     ! DENSITY (LB/IN^3)
MP,DAMP,1,0.003   ! DAMPING RATIO
!
R,1,1       ! REAL CONSTANT, THICKNESS OF 1
!
! DEFINING THE STRESS-STRAIN CURVE
!
TB,MKIN,1
TBTEMP,,STRA
TBDATA,1,.001,.002,.003,.004,.005,.006,.008
TBTEMP
TBDATA,1,10000,20000,30000,36000,38000,39000,40000
!
! DEFINING THE GEOMETRY
!
N,1,0,0,0   ! DEFINING THE NODES
N,2,5,0,0
N,3,10,0,0
N,4,15,0,0
N,5,0,5,0
N,6,5,5,0
N,7,10,5,0
N,8,15,5,0
E,1,2,6,5
EGEN,3,1,1
!
ESIZE,,11
LMESH,1
OUTPR,BASIC,1
D,1,ALL,,,  ! END CONSTRAINTS - FIXED
D,5,ALL,,,
!
FINISH
!
! PLASTICALLY STRESSING THE PART TO CREATE RESIDUAL STRESSES
!
/SOLU
D,4,UX,0.070      ! APPLY INITIAL DISPLACEMENTS, D1
D,8,UX,0.070
```

```
TIME,0.001  ! TIME TO COMPLETE DISPLACEMENT
NSUBST,10
SOLVE
!
OUTPR,NSOL,1
OUTPR,ESOL,1
!
DDELE,4,ALL ! REMOVING D1
DDELE,8,ALL
TIME,0.002
NSUBST,10
SOLVE
!
! APPLYING CYCLIC LOAD TO PRE-STRESSED BEAM
!
LSSOLVE
!
TIEMPO=0.002      ! INITIAL TIME
NCYC=100    ! NUMBER OF CYCLES
FREQ=34.34  ! FREQUENCY OF CYCLES (FIRST MODE NAT. FREQ)
AMPLIT=0.0575     ! AMPLITUDE OF DISPLACEMENT
!
DTCYC=1/(2*FREQ)
!
*DO,ITIME,1,NCYC
D,4,UX,-AMPLIT                   ! APPLY INITIAL DISPLACEMENTS
D,8,UX,-AMPLIT
TIME,TIEMPO+DTCYC
NSUBST,5
SOLVE
TIEMPO=TIEMPO+DTCYC
!
D,4,UX,AMPLIT                    ! APPLY INITIAL DISPLACEMENTS
D,8,UX,AMPLIT
TIME,TIEMPO+DTCYC
NSUBST,5
SOLVE
TIEMPO=TIEMPO+DTCYC
*ENDDO
!
DDELE,4,ALL ! REMOVING CYCLIC LOADS
DDELE,8,ALL
TIME,TIEMPO+0.001
NSUBST,5
SOLVE
!
! THESE COMMANDS SHOW THE FINAL VON MISES STRESS
!
/POST1
SET,LAST
/EFACE,1
AVPRIN,.E+00, ,
!*
PLNSOL,S,EQV,2,1
FINISH
```

Appendix C
Results from ANSYS Simulations: Data

Case 1

Full Tension and Compression Cycles

Displacement	0.060	in
Residual Stress	317.860	psi
Natural Frequency	34.04	hz

Cycles	1
Amp (in)	Max Stress (psi)
0.020	317.860
0.025	317.860
0.030	195.674
0.035	33.420
0.0375	80.407
0.040	169.196
0.045	130.010
0.0475	131.701
0.049	285.868
0.050	388.034
0.055	328.193
0.060	296.751
0.065	548.690

Cycles	2
Amp (in)	Max Stress (psi)
0.020	317.860
0.025	317.860
0.030	195.631
0.035	33.389
0.0375	80.453
0.040	169.196
0.045	131.199
0.0475	130.949
0.049	285.158
0.050	387.215
0.055	320.285
0.060	296.658
0.065	547.822

Cycles	20
Amp (in)	Max Stress (psi)
0.030	195.631
0.035	33.389
0.0375	80.453
0.040	169.196
0.045	133.397
0.0475	134.429
0.049	287.368
0.050	388.546
0.055	291.644
0.060	296.493

Case 2

Full Tension and Compression Cycles

Displacement	0.070	in
Residual Stress	950.163	psi
Natural Frequency	34.34	hz

Cycles	1
Amp (in)	Max Stress (psi)
0.020	829.597
0.025	649.176
0.030	471.579
0.035	378.153
0.040	521.937
0.045	653.407
0.050	779.657
0.055	428.341
0.0575	290.051
0.060	396.635
0.065	641.897
0.070	889.804

Cycles	2
Amp (in)	Max Stress (psi)
0.020	829.597
0.025	649.170
0.030	471.537
0.035	378.095
0.040	522.040
0.045	652.595
0.050	777.977
0.055	418.070
0.0575	290.051
0.060	393.358
0.065	636.861
0.070	911.825

Cycles	20
Amp (in)	Max Stress (psi)
0.020	829.597
0.025	649.170
0.030	471.537
0.035	378.095
0.040	522.040
0.045	647.943
0.050	769.948
0.055	381.048
0.0575	276.975
0.060	388.505
0.065	644.997
0.070	934.359

Cycles	100
Amp (in)	Max Stress (psi)
0.045	651.758
0.050	769.911
0.055	380.406
0.0575	272.922
0.060	401.288

Case 3

Full Tension and Compression Cycles

Displacement	0.0725	in
Residual Stress	1364.000	psi
Natural Frequency	34.41	hz

Cycles	1
Amp (in)	Max Stress (psi)
0.0100	1364.000
0.0200	1151.000
0.0250	972.617
0.0300	793.059
0.0350	856.208
0.0400	988.902
0.0450	980.677
0.0500	1000.000
0.0550	678.521
0.0560	568.738
0.0575	536.246
0.0585	531.224
0.0600	637.206
0.0650	722.384
0.0700	968.555
0.0725	1250.000

Cycles	2
Amp (in)	Max Stress (psi)
0.0100	1364.000
0.0200	1151.000
0.0250	972.584
0.0300	793.059
0.0350	856.158
0.0400	986.986
0.0450	980.223
0.0500	998.538
0.0550	658.127
0.0560	550.540
0.0575	519.046
0.0585	512.537
0.0600	614.211
0.0650	722.935
0.0700	981.694

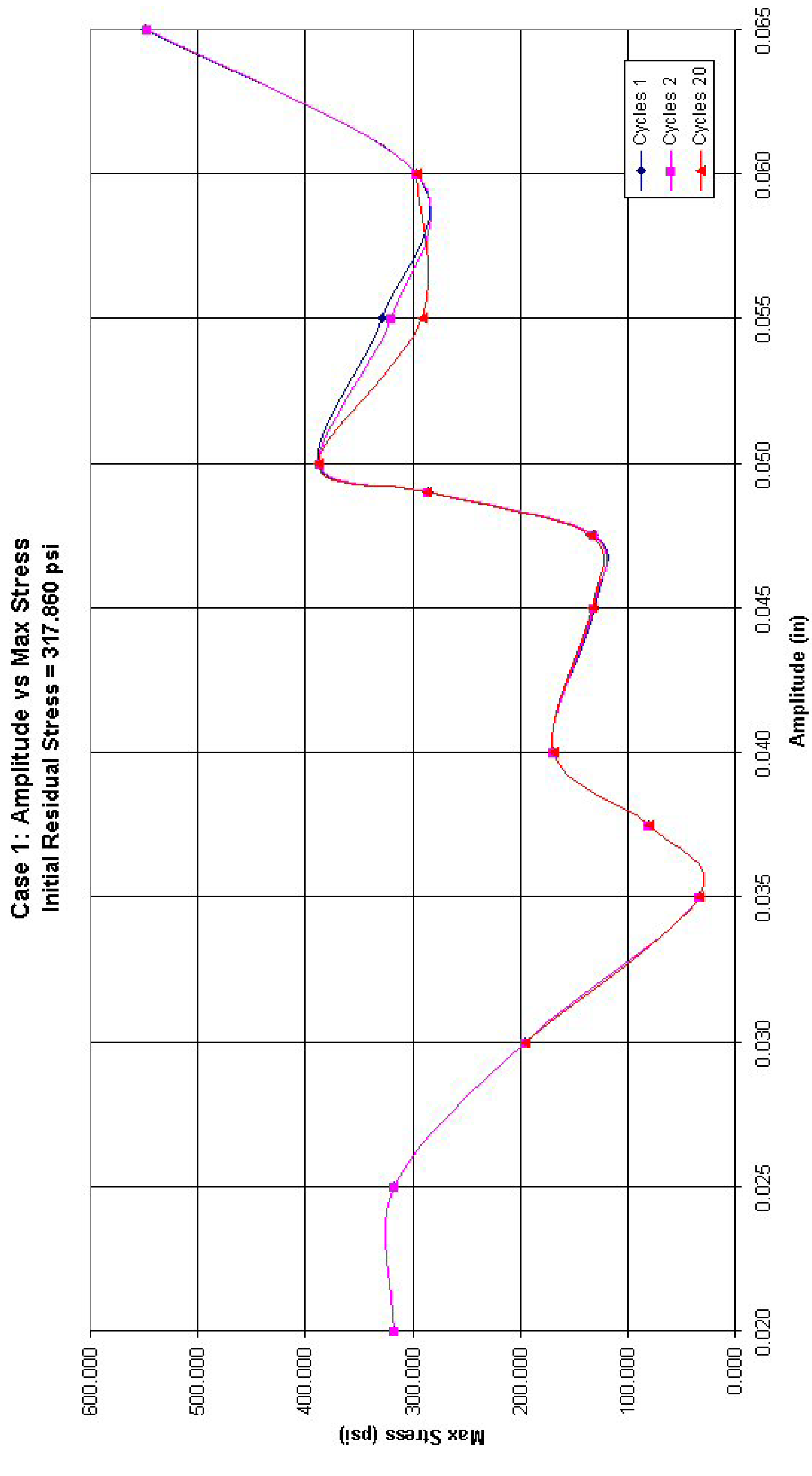
Cycles	20
Amp (in)	Max Stress (psi)
0.0450	976.227
0.0500	993.007
0.0530	777.907
0.0550	581.166
0.0560	498.251
0.0570	467.392
0.0575	460.233
0.0580	453.795
0.0585	446.722
0.0590	454.962
0.0595	527.483
0.0600	575.856
0.0650	724.456

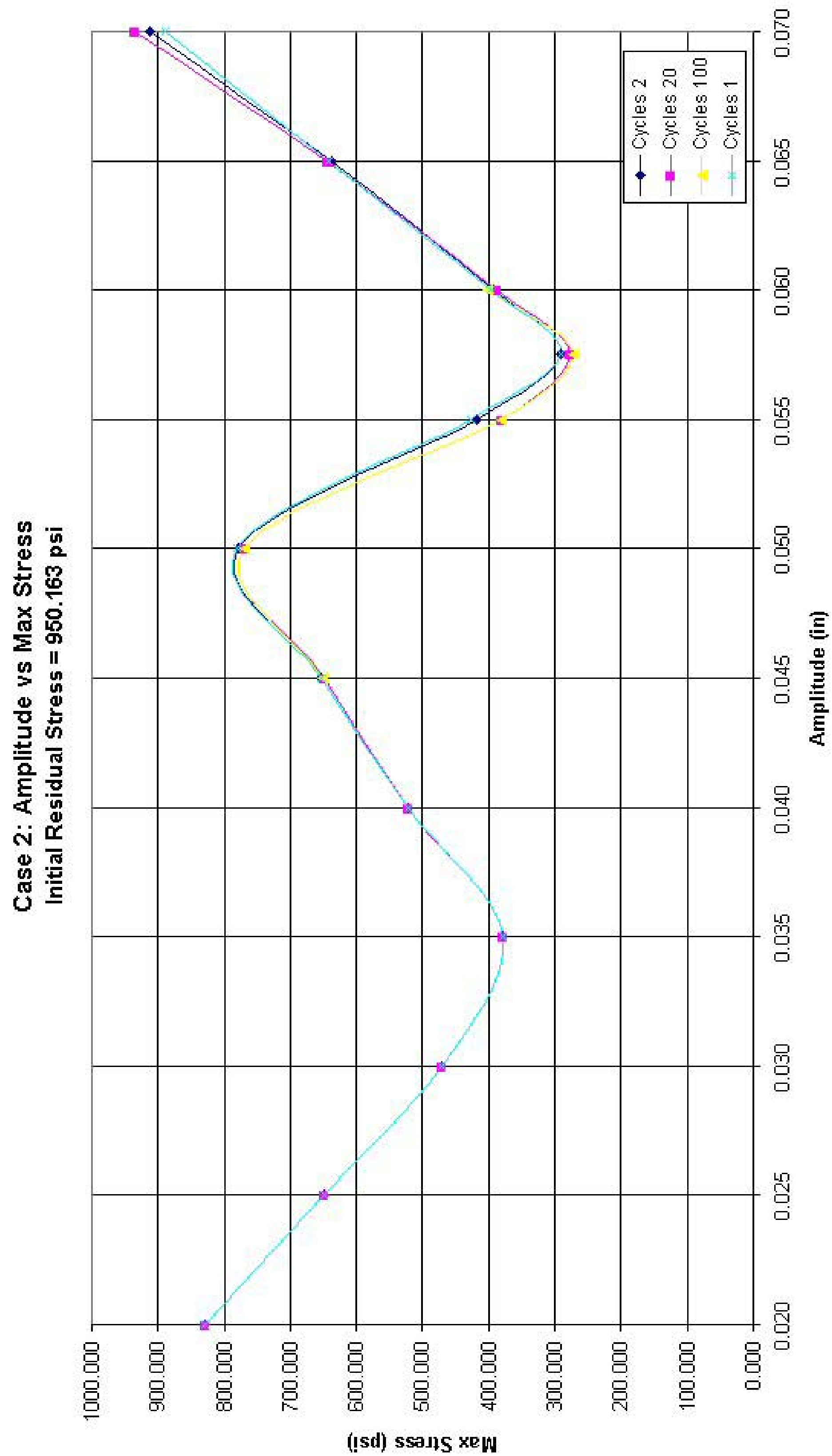
Cycles	100
Amp (in)	Max Stress (psi)
0.0450	973.017
0.0575	451.481
0.0585	442.342

10 sec

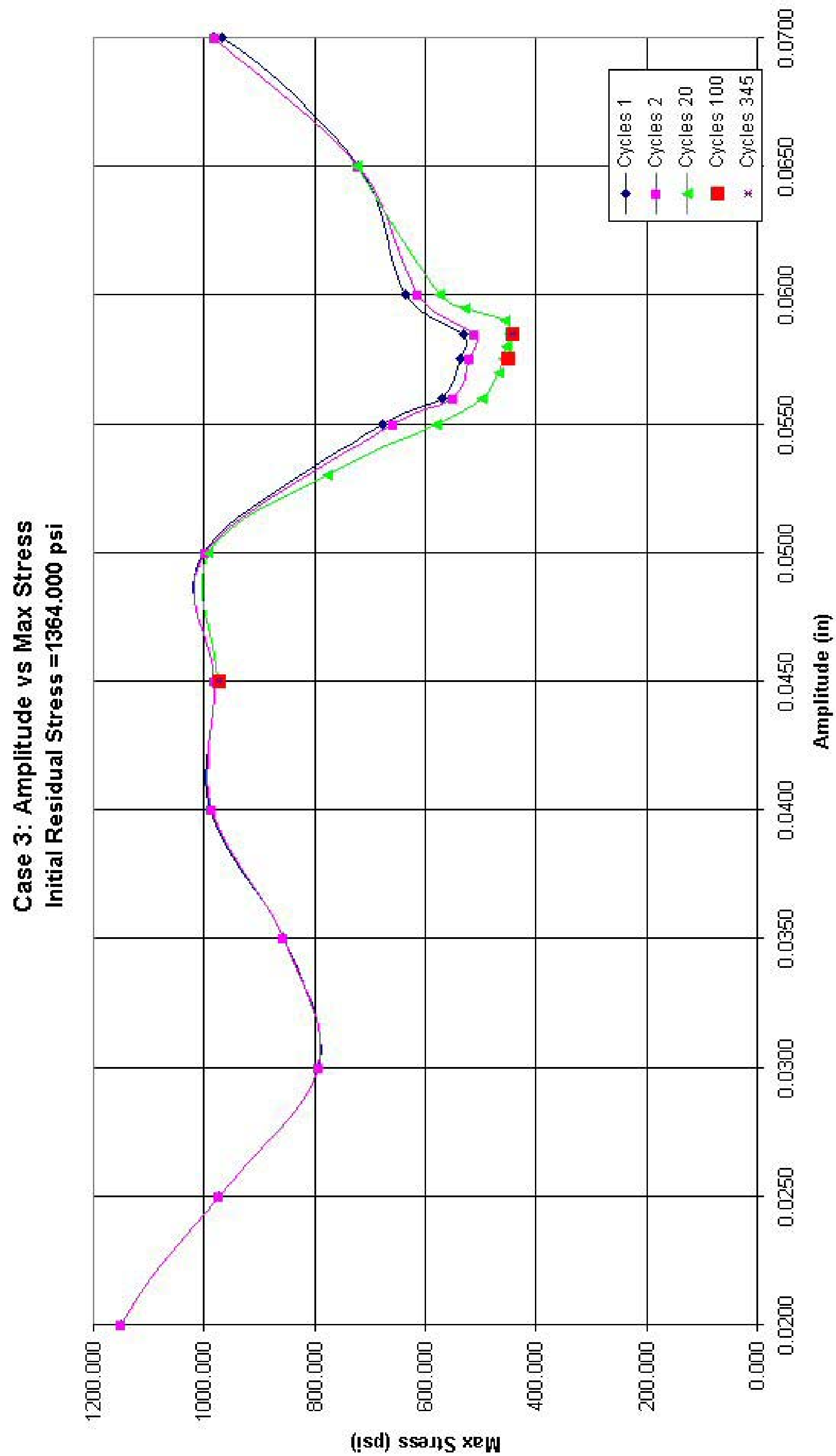
Cycles	345
Amp (in)	Max Stress (psi)
0.0450	972.764
0.0585	442.342

Appendix D
Results from ANSYS Simulations: Charts





Case 2: Amplitude vs Max Stress
Initial Residual Stress = 950.163 psi
Max Stress (psi)
1000.000
900.000
800.000
700.000
600.000
500.000
400.000
300.000
200.000
100.000
0.000
0.020
0.025
0.030
0.035
0.040
0.045
0.050
0.055
0.060
0.065
0.070
Amplitude (in)
Cycles 2
Cycles 20
Cycles 100
Cycles 1



Case 3: Amplitude vs Max Stress
Initial Residual Stress =1364.000 psi
Max Stress (psi)
1200.000
1000.000
800.000
600.000
400.000
200.000
0.000
0.0200
0.0250
0.0300
0.0350
0.0400
0.0450
0.0500
0.0550
0.0600
0.0650
0.0700
Amplitude (in)
Cycles 1
Cycles 2
Cycles 20
Cycles 100
Cycles 345

Appendix E
ANSYS Von Mises Stress Results

The screen grabs below are from ANSYS for the Case 2 condition, where a preload of 950.163 psi was induced on the beam.

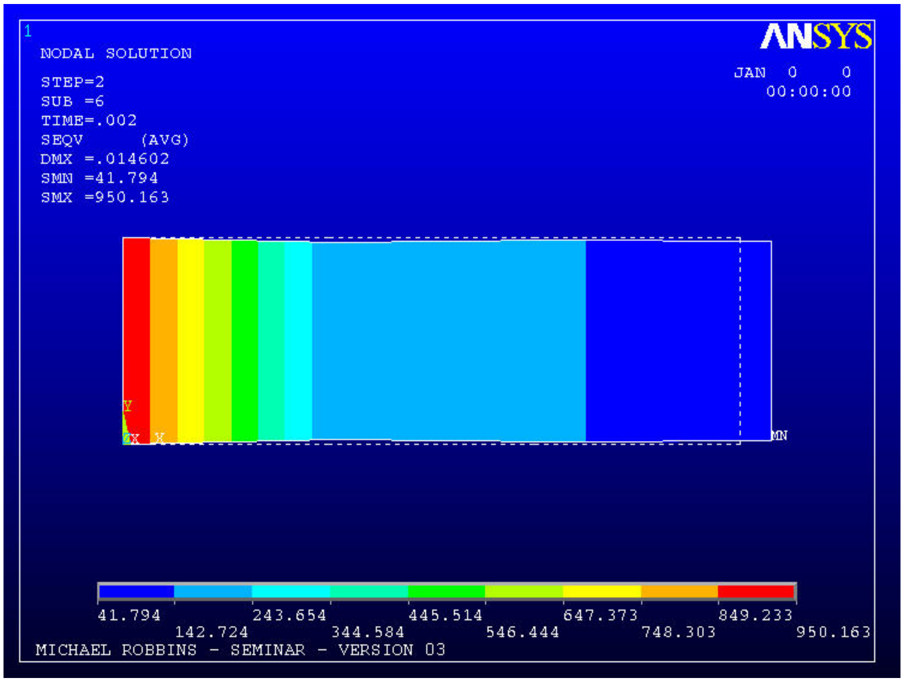


Figure E.1: The Initial Residual Stress Distribution for Case 2.

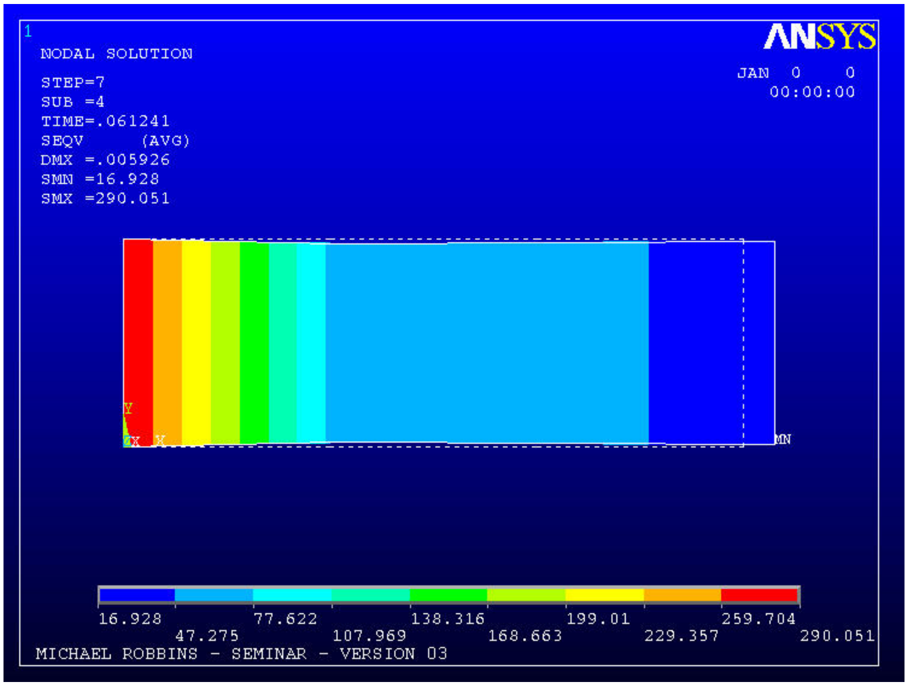


Figure E.2: The Stress Distribution after 2 Cycles of Amplitude 0.0575-in, Case 2.

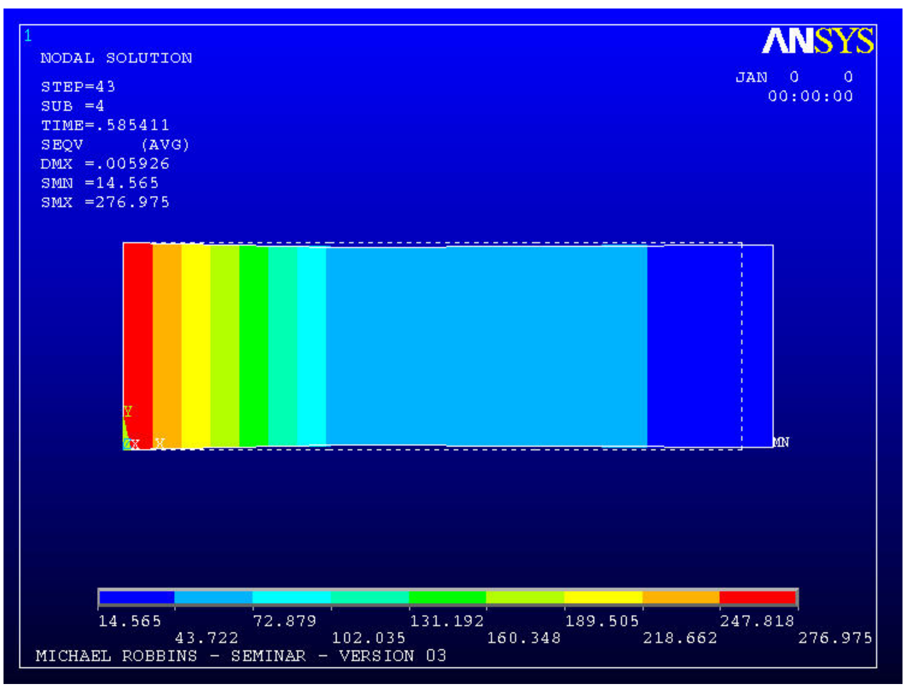


Figure E.3: The Stress Distribution after 20 Cycles of Amplitude 0.0575-in, Case 2.

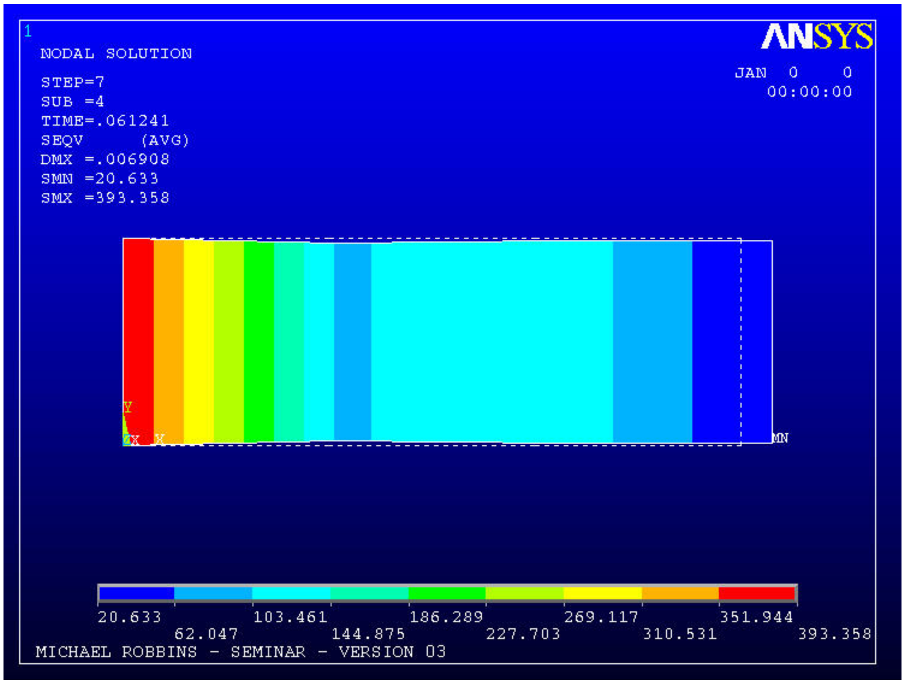


Figure E.4: The Stress Distribution after 2 Cycles of Amplitude 0.060-in, Case 2.

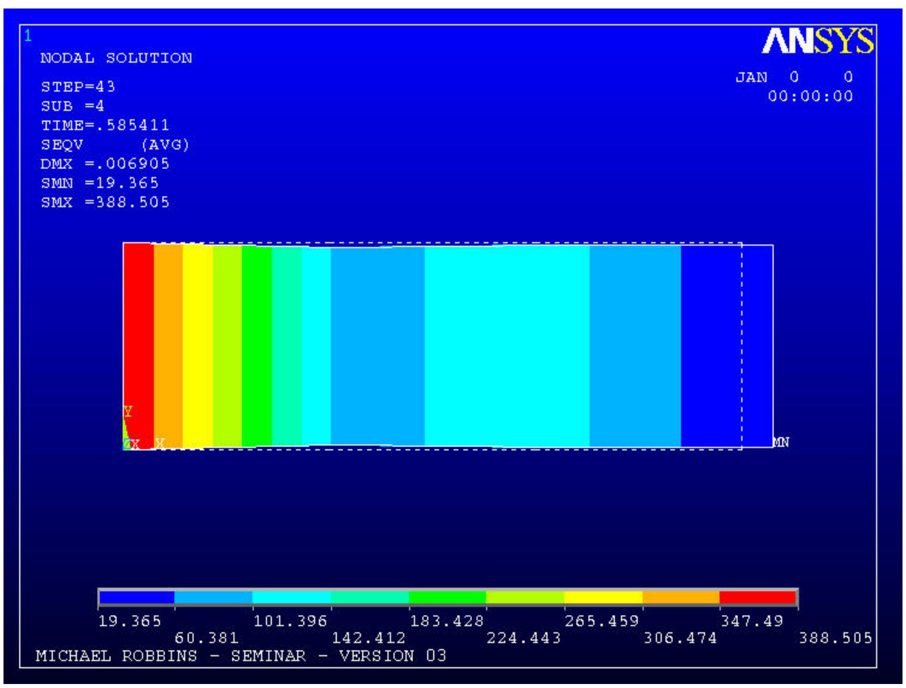


Figure E.5: The Stress Distribution after 20 Cycles of Amplitude 0.060-in, Case 2.