Evaluation of an Aluminum Replaceable Pad Track for the M-1 Main Battle Tank

Contract Number DAAE 07-84-C-R054

September, 1988

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Evaluation of an Aluminum Replaceable Pad Track for the M-1 Main Battle Tank

Daniel F. Carbaugh and Mark A. Holtz

Final

FROM 11/84 TO 05/86

September, 1988

An aluminum replaceable pad track, capable of utilizing existing T-156 track hardware, was analyzed using both empirical and finite element analyses. The track was to be interchangeable with the current T-156 track at a minimum weight penalty. Such a track was analyzed and its load carrying capability in tension and torsional loads were predicted by correlating track load with block stress and comparing that with the strength levels of ingot metallurgy aluminum alloys, and powder metallurgy aluminum alloys. The former offering the best combination of material properties and economics, the latter offering the best combination of metal strength, ductility, and toughness.
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1.0. INTRODUCTION

1.1. Original Scope of Work

This final technical report prepared by the Aluminum Company of America, Forging Division, for the U.S. Army Tank Automotive Command under contract number DAAE07-84-C-R054, describes phase I in the evaluation of an aluminum replaceable pad track for the M-1 Main Battle Tank. Phase I was to include design evaluation through the use of engineering formulas, stress analysis, weight analysis and Stress Coat/strain gage analysis. Upon completion of this work, a limited number of track shoes were to be submitted to TACOM for laboratory analysis. A second phase was to produce track for field testing on M-1 Main Battle Tanks.

1.2. Work Completed

Only the engineering analysis portion of Phase I was completed. No track blocks were actually produced due to changes in M-1 Tank track philosophies by the U.S. Army.

1.3. Purpose

The integral pad T-156 used on the M-1 tank, though structurally efficient was useless after only several hundred miles of driving due to road pad failure. A replaceable pad track would extend track life by:

- Having thin road pads that would minimize heat build-up due to high internal friction in rubber pads.
- Having the capability to run on bare blocks even if road pad rubber failed.

The purpose of the work was to analyze track shoe block stresses and determine if aluminum could be used as the block material to help minimize weight.

1.4. Previous Work

Although aluminum track had been designed for the M-1 Tank in its development stages, designs were not optimized. Further, earlier aluminum track designs did not use advanced metallurgy alloys that offer significant property advantages.

1.5. Goals

Once the track design was evaluated the contractor was to recommend any relevant design changes and select alloy(s) that would offer the appropriate combination of material properties to best serve the needs of the M-1 Tank.
2.0. OBJECTIVES

The purpose was to design an aluminum replaceable pad (RP) track suitable for service on the M-1 Main Battle Tank. The track was to have the following characteristics:

- Replaceable road pads.
- Integral grousers.
- Be a substitutes for the T-156 track.
- Use the same drive sprockets, bushings, center guides, end connectors, drive pins as the T-156 track.
- Weigh no more than 9615 lbs./vehicle (approximately 800 lbs. more than the T-156 track).
- Utilize the best ingot metallurgy or powder metallurgy aluminum alloys.
- Be of lower life cycle cost than the T-156 track.

3.0. CONCLUSIONS

3.1. Empirical Analysis

The results of the experimental work are summarized on the three plots, Figures 5-9, 5-10 and 5-11, relating material stress to load. The tension plot in particular shows the load carrying abilities of the existing shoe design at the two highest stressed points, Gages four and six (Figure 5-2). The typical ultimate strength levels of several candidate alloys have been superimposed on the material stress vs load for tensile load case. The plot has been extrapolated assuming the loading would remain in the elastic region. Typical ultimate strengths are higher than guaranteed minima for the alloys presented, but are more representative of actual track behavior. These values were used because they best describe the ultimate strength of the material used in previous work by TACOM and others. The results of previous work can be directly compared with those achieved in this analysis.

When the typical ultimate strength of 2014-T6 (70 ksi) is placed on the material stress vs load graph for tensile loading, as seen from Figure 5-9, the track should show catastrophic failure at approximately 185,000 lbs. load. The U.S. Army Tank Automotive Command (TACOM) has analyzed the same track and alloy, 2014-T6, in tension and achieved similar results. When considering other alloy candidates of both ingot and powder metallurgy, materials reviewed not only must have high strength, but also excellent ductility,
toughness and resistance to stress corrosion cracking. This combination of properties implies superior damage tolerance and therefore, battlefield survivability.

3.2. Finite Element Analysis

The FE model results and conclusions are well documented in report #1. Three important points in that report need to be highlighted:

. The rubber bushing preload has a significant effect on the aluminum shoe stresses. When the rubber preload is exceeded or separation occurs, the shoe stresses are higher than they would be if adequate preload were maintained.

. A careful examination of the machine dynamics should be done. This would allow the FE model to be used to its fullest extent. Presently, without accurate boundary conditions, the FE model can only be used for comparison studies.

. Photographs of the photos of the failures enclosed in Report #2, reveal several failures at the three o'clock position or at the parting line of the binocular. This contradicts both the experimental and analytical analyses which indicates that failure should occur at Gage Six (one o'clock) or the Gage Four (fillet radii blending the end plate into the binocular tube) location in a pure tensile load case. The highest stressed area of binocular section is at approximately at one o'clock (see Report #1). Both the in-service dynamic loading and hardware behavior noted above could explain the difference in failure location between that found in field trials and that identified by the laboratory and FE analyses. Further, the effect of parting line location on the shoe block probably contributed to the three o'clock location of the previous track shoe failure described above.

3.3. Suitability of Aluminum

Figures 5-9, 5-10, and 5-11 show that significant increases in load capability of aluminum were possible by using track made of 7050-T74 or 7175-T74 material. Since the powder metallurgy alloys 7090, 7091, and CW67 did not become commercially available they should not be considered. The best alloy/temper candidate is 7175-T74. Further, if track design were not restricted to using T-156 hardware and drive arrangement, a track can be designed to more efficiently use aluminum yielding a lightweight, durable track. With the design restrictions applied an aluminum track could be designed that weighs about
9,200 lbs. or only 400 lbs. heavier than the T-156 track.

4.0. RECOMMENDATIONS

4.1. Existing Design Envelope

Due to the original scope constraints of using existing hardware and maintaining the existing envelope, redesign options of the block were somewhat limited. However, based on the work performed, the following areas should be changed to improve the load carrying abilities of the shoe, particularly in pure tension.

- Both analyses show the thin end plate of the shoe is a highly stressed area at Gage Six (see Figure 5-2). This section should be thickened to match the other side.
- The fillet radii between the binocular and the thick end plate Gage Four (Figure 5-2) should be increased. This area has high tension and torsion stresses.
- Change the material alloy to 7050-T74 or 7175-T74 to increase the overall load carrying abilities of the shoe.
- Determine if additional rubber bushing preload is required.

The above listing is not all inclusive since the effort to correlate the field failures to the analysis was not conclusive. A complete optimization design is not possible since a better definition of the loads which caused the track failures is required.

4.2. Connecting Hardware

A detailed study of the load carrying capabilities of the shaft and end connectors should be done. One theory of possible early track failures is that an end connector actually fails first, thus drastically changing the load path and causing high point loading where rubber prestresses are exceeded and separation occurs.

4.3. Dynamic Loading

A careful examination of the machine dynamics should be done. This would allow the finite element model generated to be used to its fullest extent. Understanding these machine dynamics will help to correlate both FE as well as analytical experimental data with actual track service loading. This may ultimately have an effect on design as well as alloy.
4.4. Unrestricted Design

If the design were not restricted to utilizing the T-156 hardware, then an aluminum track could be optimized to work on the M-1 Tank and survive the dynamic loading and severe service requirements of the vehicle. This might ultimately require different hardware and drive sprockets than the T-156 track. However, this would assure a lightweight replaceable pad track that would satisfy the service requirements of the M-1 Tank.

5.0. DISCUSSION

5.1. Background

The M-1 Main Battle Tank, due to its weight and high performance characteristics places severe demands on its track. The T-156 track currently installed on the M-1 Tank has an integral road pad bonded to a steel framework. Due to the high loads the road pads must withstand, heat readily builds up in the road pad from internal friction. This combined with dynamic loads destroys the road pads. Because the metal framework bonded to the road pad does not provide a good traction surface, when the bonded rubber pad deteriorates the remaining bare steel block presents an inadequate running surface for the tank.

A replaceable pad (RP) track allows the road pad to deteriorate without, due to block design, inhibiting tank mobility. Steel RP tracks have been considered for the M-1 Tank, but due to their solid block design require a severe weight penalty. By substituting aluminum for steel the block weight is significantly reduced. As a result an aluminum RP track can provide RP track benefits at a weight comparable to the weight efficient T-156 track.

5.2. Previous Aluminum Track Programs

Alcoa designed an aluminum RP track for the M-60 Tank to the T-142 design. This track was quite successful in testing at the Aberdeen Proving Grounds yielding track lives up to 8,000 miles. Also, aluminum RP track proved successful in lighter amphibious vehicle testing. An aluminum track design existed for the M-1 Tank. This design was part of the early development work on the M-1 by General Motors and Chrysler Defense (now part of General Dynamics). This track had limited success due to other vehicle drive problems and was dropped. The original track used alloy 2014-T6, a relatively high strength material.

The U.S. Army Tank Automotive Command had also done some laboratory analysis on this original M-1 aluminum track. This work concluded, among other things, that the tracks ultimate
tension load capability was about 185,000 pounds. That information was useful in the work of this report.

5.3. Laboratory Analysis

The objective of the analysis was to determine the load capabilities of a track shoe block and recommend possible improvements. Initially an experimental approach was pursued with the expectation of correlating the findings with actual field failures. This, however, was not conclusive, so a finite element analysis was done. It also did not agree entirely with field failures, but was in fair agreement with the experimental work.

In both analyses, the rubber pads on both faces of the block were not accounted for, since they add little structurally. The rubber bushings between the shaft and the block, however, were considered.

In Phase I, several track shoes were forged in alloy 6061-T6 and were then fitted with the standard T-156 hardware excluding pads and road wheel rubber. These parts were then subjected to a Stress Coat/Strain Gage (SC/SG) analysis to locate and quantify the high stress areas in the block. This method locates stresses in parts by first coating them with a brittle lacquer material and then subjecting them to a load. As the load is increased, the more highly stressed areas in the part begin to elongate first cracking the brittle lacquer coating. The cracked coating located the highly stressed areas which were then fitted with strain gages. With the strain gages in place, the part was placed under load again and the strain was measured. The measurements taken were converted into stress levels in the part. By monitoring the strains (and consequently the stresses) generated for given loads, a curve was set up to correlate track tension and torsion to parts stressed.

For this SC/SG analysis, the shoes were placed in a special track stressing fixture (shown in Figure 5-1) that tests three pitches, connected together, that applies tension and torsion loads both separately and in combination. This fixture was specifically designed to simulate track load and loading geometry. The shoes of the middle pitch were evaluated to minimize any end effects of the test setup.

Following the SC/SG analysis, computer modeling of the track using Finite Element (FE) analysis was done to completely understand the load/stress relationship in the track. The procedure for this work and the results there found are explained in detail below.

Once the track load/material stress relationship was estab-
lished, track tension and torsional load abilities were predicted based on the ultimate strengths of several candidate alloys. The results of this correlation are shown in Figure 5-9. Based partially on this prediction of track load capability, alloys were recommended. Other factors considered in the recommendation included stress corrosion cracking resistance, ductility, forgeability and toughness. The alloy selections are stated in RECOMMENDATIONS 4.0.

5.3.1. Experimental Analysis. The experimental portion included both Stress Coat and strain gage evaluation. The Stress Coat was applied to a block installed in the test fixture (see Figure 5-1) and then subjected to a tensile load. The high stress areas and direction of stress lines were noted. The block assembly was then loaded in torsion. The high stress areas and directions were again noted. (See Figures 5-2 and 5-3) From this information, the location and orientation of six single gages was determined. One rosette gage, Gage Seven, was also applied to confirm the method of gage orientation of the six single gages to the principal axes. (See Figure 5-3) The block assembly was installed in the test fixture and subjected to the following load cases:

1. Pure tension
2. Pure torsion
3. Combination tension/torsion (20,000 lbs. tension and varying torsion)
4. Combination tension/torsion (50,000 lbs. tension and varying torsion)

Figures 5-4, 5-5, 5-6 and 5-7 are plots of the test load vs strain for the above load cases. As can be seen from the plots, Gages Four and Six are of primary concern since they are the maxima.

Next a plot of material stress vs load on Gages Four and Six was generated for the tensile case. (See Figure 5-9) Superimposed on this plot were typical ultimate strengths of various alloys and the corresponding load necessary to achieve those strength levels. For all alloys shown, a modulus of elasticity of 1086 psi was assumed. Similar graphs were created for the pure torsion load case (Figure 5-10) and the combined tension (50,000 lbs.) and torsion case (Figure 5-11).

5.3.2. Analytical. Since the experimental stress result did not appear to agree with in-service failures previously reported (Report 2), an analytical approach was used to better understand the entire load distribution within the block. The stress pattern on the inside of the binocular was of particu-
SIMPLE TENSION, STRAIN VS. LOAD

FIGURE 5-4
SIMPLE TORSION, STRAIN VS. TORQUE

FIGURE 5-5
lar interest since strain gages could not be applied in this area. Finite Element (FE) analysis was chosen as the best approach to analyze the shoe. Because of the symmetry of the track shoe assembly, only ¼ of a block was modeled. (See Figure 5-8) The model was subjected to the following load cases:

1. Tensile load
2. Out of plane load
3. Twisting load

Checks were then done to compare analytical to experimental results. In general, the model corresponds with the experimental results. The FE model confirmed peak stress locations on the shoe. However, the experimental and analytical results did not agree very well with the failures photographed in Report #2.

The details of the model are covered in greater depth in report entitled: "3-D Finite Element Analysis of an Aluminum M-1 Tank Track Shoe" (Report #1).

5.4. Material Selection

For the purpose of the experimental analysis, aluminum alloy 6061-T6 was used since it is easy to forge, machine, and use in Stress Coat/strain gage analysis. The material does not have sufficient properties for service on the M-1 Tank, although it proved quite successful in testing on a P-7 Program (Report 3). The analytical analysis combined with this laboratory work and work by TACOM led to Figures 5-9, 5-10, and 5-11 which compare material stress at given loads for higher strength aluminum alloys that could be used in aluminum track. As these figures show, high strength 7XXX alloys far surpass the load carrying capability of alloy 2014-T6.

Since powder metallurgy alloys 7090, 7091, and CW67 (a 7091 derivative) were not commercialized they were dropped from consideration. Alloys 7050-T74 and 7175-T74 are commercially available and should offer good candidates for aluminum track material for the M-1. For strength reasons, 7175-T74 would be the best alloy for test purposes.

5.5. Second Tier Material Properties

Due to the demanding service conditions of a tank track, any material must combine strength with damage tolerance. The second tier properties of both 7050-T74 and 7175-T74 are excellent. These include ductility and toughness which are both indicators of damage tolerance.
3-D ANSYS Model of M-1 Track Shoe, Isometric View

FIGURE 5-8
LOAD VS. MATERIAL STRESS FOR PURE TENSION
(assuming a modulus of elasticity of 10E6)

FIGURE 5-9
LOAD VS. MATERIAL STRESS—FOR PURE TORSION
(ASSUMING A MODULUS OF ELASTICITY OF 10E6)

FIGURE 5-10
LOAD VS. MATERIAL STRESS FOR COMBINATION LOAD
CONSTANT TENSION AND VARIABLE TORSION FOR GAGE LOCATION 4
(ASSUMING A MODULUS OF ELASTICITY OF 1023)

FIGURE 5-11
5.6. **Weight**

Weight calculations indicate that an aluminum RP track of the original design would weigh 9,215 lbs., or only 400 lbs. heavier than the T-156 track. This is well within the weight bogey for an aluminum RP track for the M-1 Tank.

5.7. **Design**

The design evaluated used the T-156 hardware and drive sprockets so that if an aluminum track proved successful in analysis and lab testing it could be incorporated on the vehicle without requiring unique hardware. This also would ensure a continuous supply of spare parts common to both the aluminum track and the T-156 track. This of course limited design flexibility for the aluminum track. An aluminum track design without these restrictions could be readily optimized using the data collected. This would lead to more efficient aluminum designs for the M-1 Tank that would increase track life with a lightweight track.
Appendix A
3-D FINITE ELEMENT
ANALYSIS OF AN
ALUMINUM M-1 TANK
TRACK SHOE

Prepared For
Aluminum Company of America
1600 Harvard Avenue
Cleveland, OH 44105

Prepared By
Robert S. Joseph
Edward H. Long

November 1985
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1.0 SUMMARY

This report documents the finite element analyses that are performed on the aluminum M-1 tank track shoe using the ANSYS finite element computer program (Revision 4.1). The scope of work defined for this project by Alcoa is contained in Appendix A of this report for reference. The purpose of this project is to develop a three-dimensional finite element model of the track shoe including the steel pin and rubber bushing and to demonstrate part performance by analyzing three separate load cases.

Section 2.0 describes the 3-D finite element model of the track shoe that is developed to evaluate the track shoe. The model includes the steel pin and rubber bushing in order to develop the proper loading on the shoe binocular. The model is a one-half symmetry model of the shoe and contains 2838 ANSYS isoparametric solid elements. Detailed descriptions of the various element types and nodal point locations are given in Section 2.0.

Three load cases are analyzed with the 3-D model by applying the appropriate boundary conditions on the two symmetry planes. Sketches illustrating the three load cases, namely, pure tensile load, out-of-plane load, and twisting load, are contained in Section 3.0. The maximum loads assumed for these cases are somewhat arbitrary since the primary purpose of these demonstration runs is to qualify the analytical model. However, the maximum load for the pure tensile load is selected to correspond to the maximum load used in the Goodyear
test (Figure B-5). Additionally, the analytical model is linear and the results can be scaled as long as rubber preload is maintained in the binocular section of the shoe.

The track shoe assembly contains three different materials: the steel shaft, the rubber bushing, and the aluminum shoe body. The aluminum and steel material properties are readily available in the literature and the properties used in this analysis are listed in Section 4.0. However, rubber properties, especially a compressive stress-strain curve, is not easily found. Rubber does not follow Hooke's law and can be characterized by a nonlinear elastic behavior which becomes stiffer with increasing strain. The approach used in this analyses is to select an effective Young's modulus from a rubber stress-strain curve that will approximate the actual rubber stiffness of the assembly. A development of the rubber properties for initial use in the analytical model is presented in Appendix B. These rubber properties are further refined as a result of the tensile load calibration runs discussed in Section 5.2.

A supplemental parametric study using a 2-D interaction model is presented in Appendix C. The purpose of this study is to investigate the interaction of the shaft, rubber, and endplate due to a tensile pull load using an economical model. The sensitivity of the rubber modulus and the effect of rubber preload are investigated. These results were used to guide the 3-D model analysis presented in Section 5.0 and to gain some insight into the load paths of the assembly.
Section 5.0 presents the 3-D finite element stress results for the three demonstration load cases. The results are presented in the form of tabulated maximum stress summaries, displacement plots, and stress contour plots. The results demonstrate that the 3-D track shoe model behaves in a predictable and proper manner for the loads considered. Input listings for all final ANSYS runs discussed in this report are contained in Appendix D. A listing of all computer files for this project residing on Alcoa's DEC VAX 11/785 computer system is given in Section 6.0.
2.0 **FINITE ELEMENT MODEL DESCRIPTION**

This section describes the 3-D finite element model that was developed to evaluate the M-1 tank track shoe body. The model also includes the steel pin and rubber bushing in order to develop the proper loading on the shoe. Photographs of the track shoe body without pin and bushing are shown in Figures 2-1 and 2-2.

A one-half symmetry model of the track shoe was developed using ANSYS STIF45 isoparametric solid elements. Figures 2-3 through 2-7 show various isometric views of the 3-D finite element model. The model contains a total of 2838 solid elements with a breakdown of the elements as follows:

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<tr>
<td>Aluminum Shoe</td>
<td>1734</td>
</tr>
</tbody>
</table>

The specific dimensions used to construct this model are defined in Figure 2-8. These dimensions were obtained from drawings supplied by ALCOA and these reference drawings are listed in Figure 2-8. Since the steel shaft extends through a shoe pair, only one-half of the shaft was modeled. The gap between the shoe pair was given as 1.82", and therefore the shaft length on the inboard side was set to 0.91". The shaft extension on the outboard side was set to 1.25" which is the distance to the midpoint of the flat on the shaft.
Although the model is constructed of all solid elements, it was advantageous to separate the model into seven element types in order to facilitate model development and postprocessing of results. Figure 2-9 shows a sketch of the model identifying the various element types. The nodal point numbers for each element type are also listed.

Since the displacement and stress results are calculated and presented at nodal points, it is important to have a complete description of all the nodal points in the model. Figures 2-10 to 2-17 define the nodes in each element type of the model. In general, the model was developed by defining nodes on one plane and then incrementing the nodes in the third direction.
Figure 2-1 - Photograph of an Aluminum M-1 Tank Track Shoe, Top View
Figure 2-2 - Photograph of an Aluminum M-1 Tank Track Shoe, Bottom View
Figure 2-3 - 3-D ANSYS Model of M-1 Track Shoe, Isometric View
Figure 2-4 - 3-D ANSYS Model of M-1 Track Shoe, Isometric View
Figure 2-5 - 3-D ANSYS Model of M-1 Track Shoe, Isometric View
Figure 2-6 - 3-D ANSYS Model of M-1 Track Shoe, Isometric View
Figure 2-7 - 3-D ANSYS Model of M-1 Track Shoe, Isometric View
Model dimensions based on following reference drawings:

1. U.S. Army Tank Automotive Command,
   Dwg. No. XM67059

2. ALCOA Forging,
   Dwg. No. F-18292

Figure 2-8 - M-1 Track Shoe Dimensions Used in Analysis
1. Steel Shaft
2. Rubber Bushing
3. Alum. Cylinder (Binocular)
4. Alum. End Plate (Thick)
   Alum. End Plate (Thin)
5. Alum. Rib & Intersecting Wall
6. Alum. Web
7. Alum. Fillets

Node Range

$\{ \text{2-3949} \}$
4001-4451
4618-4955
5001-5543
6001-6657
7001-7019

Figure 2-9 - 3-D Model Isolating the Seven Element Types
Figure 2-10 - Shaft, Bushing and Cylinder Nodes
CROSS-SECTION AT \( z = 0.25 \)"

ALL ROWS (RINGS) INCREMENT NODES BY 150

Figure 2-11 - Steel Shaft Nodes (Type 1)
CROSS-SECTION AT Z = 1.5"

ALL ROWS (RINGS) INCREMENT NODES BY 150

Figure 2-12 - Rubber Bushing Nodes (Type 2)
CROSS SECTION AT z = 1.5""}

ALL ROWS (RINGS) INCREMENT NODES BY 150

Figure 2-13 - Aluminum Cylinder (Binocular) Nodes (Type 3)
Figure 2-14 - Aluminum Endplate Nodes (Type 4)
Figure 2-15 - Rib & Intersecting Wall Nodes (Type 5)
Figure 2-16 - Web Nodes (Type 6)
Figure 2-17 - Fillet Nodes (Type 7)
3.0 LOAD CASES

The 3-D finite element model was used to analyze three loading conditions on the shoe. The tensile and side loads will be evaluated with the quadrant model by applying the proper boundary conditions to the symmetry planes. The specific set of load cases that were analyzed are listed below:

- Case 1 - Pure Tensile Load ($F_{tensile}$)
- Case 2 - Out-of-Plane Load ($F_{side}$)
- Case 3 - Twisting Load ($F_{twist}$)

These three load cases are illustrated in Figures 3-1 to 3-3.

The maximum load assumed in this analysis for Case 1 is $F_{tensile} = 72,000$ lbs. for the shoe pair. This load corresponds to the Goodyear test (see Figure B-5) which loaded one shoe and one pin/bushing to 36,000 lbs. The maximum load assumed for Case 2 is $F_{side} = 72,000$ lbs. The maximum twisting load for Case 3 was assumed to be $F_{twist} = 18,000$ lbs. This corresponds to an applied twisting moment of $T = F_{twist} \times D = 18,000$ lbs. $\times 4.94$ in. $= 88,920$ in-lb. Since the analyses are linear, the results from any of the cases can be linearly scaled.
BOUNDARY CONDITIONS:

Plane 1 - Symmetry
Plane 2 - Symmetry

\[ F_{\text{tensile}} = \text{Total Applied Tensile Load} \]

Assume \( F_{\text{tensile}} = 72,000 \text{ lbs.} \) for Analysis

Figure 3-1 - Case 1 - Pure Tensile Load on Track Shoe
BOUNDARY CONDITIONS:

Plane 1 - Symmetry
Plane 2 - Symmetry
R = Reaction force required to balance out-of-plane component

\[ F_{\text{side}} = \text{Total Out-of-Plane Load at an Angle } \theta \]

Assume \( F_{\text{side}} = 72,000 \text{ lbs. and } \theta = 30^\circ \) for Analysis

Figure 3-2 - Case 2 - Out-of-Plane Load on Track Shoe
BOUNDARY CONDITIONS:
   Plane 1 - Anti-Symmetry
   Plane 2 - Anti-Symmetry

\[ T = F_{\text{twist}} \times D = \text{Total Applied Twisting Moment} \]

Assume \( F_{\text{twist}} = 18,000 \text{ lbs.} \) for Analysis

Figure 3-3 - Case 3 - Twisting Load on Track Shoe
4.0 MATERIAL PROPERTY DATA

The M-1 tank track shoe assembly contains three different materials: the steel shaft, the rubber bushing, and the aluminum shoe body. Two material properties are required for each material to perform a static, elastic analysis, namely: Young’s modulus, E

Poisson’s ratio, ν

The material properties used in the 3-D ANSYS model are listed below:

Steel Pin, ANSYS Material 1

\[ E = 30 \times 10^6 \text{ psi} \]
\[ ν = .3 \]

Rubber Bushing, ANSYS Material 2

This material is natural rubber (NR) with an ultimate tensile strength of 3000 psi and a Shore A durometer of 65-70. Refer to Appendix B for a development of the rubber properties. The first approximation rubber properties as developed in Appendix B are:

\[ E = 20 \times 10^3 \text{ psi} \]
\[ ν = .49 \]

These properties were modified as a result of the tensile load calibration runs performed in Section 5.2. The load-deflection test data obtained by Goodyear was used as a basis to scale the rubber modulus. The rubber properties used in the final analysis are:

\[ E = 4 \times 10^3 \text{ psi} \]
\[ ν = 0 \]

Aluminum Shoe Body, ANSYS Material 3

\[ E = 10 \times 10^6 \text{ psi} \]
\[ ν = .33 \]
5.0 **FINITE ELEMENT STRESS ANALYSIS**

This section presents the finite element stress results that were obtained with the 3-D ANSYS model described in Section 2.0. In addition to the three defined load cases in Section 3.0, a uniform temperature case was run to check out the connectivity of the model.

5.1 **Uniform Temperature Check Case**

As a final check of the 3-D model, a uniform temperature case was run. This is a very useful case to verify that all parts of the model are connected properly and that all displacement boundary conditions are correctly applied. The temperature of all nodes in the model was set to 1000°F and the coefficients of thermal expansion for all three materials were set to 10 \times 10^{-6}/^\circ F for this case only. These conditions allow free thermal expansion of the model and the resulting stresses for each element type should be essentially zero.

A postprocessing file using POST1 was set up to sort on element type and scan for the highest stresses in each element type or component. The stress summaries are based on both element data and nodal data, and the stresses are listed in order of decreasing values of stress intensity. The stresses listed in the tables are defined below:
SIG1 = \sigma_1 = \text{first principal stress value}

SIG2 = \sigma_2 = \text{second principal stress value}

SIG3 = \sigma_3 = \text{third principal stress value}

SINT = SI = \text{stress intensity} = |(\sigma_1 - \sigma_3)| = \text{twice the maximum shear stress}

SIGE = \text{Von Mises equivalent stress}

\[
\text{SIGE} = \frac{1}{\sqrt{2}} \left[ (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right]^{\frac{1}{2}}
\]

Tables 5-1 to 5-7 show the maximum stress summaries in the seven element types for the uniform temperature case. Refer to Figure 2-9 for a sketch of the various element types. The element stresses are lower than the nodal stresses because they occur at the element centroids. The tables also define the nodal points for each of the high stress elements to aid the reader in locating the elements. It is suggested that the nodal data be used as a basis for determining peak stresses since these stresses occur on the surface. Node points are described in Figures 2-10 to 2-17. The largest stress intensity was calculated to be 4.9 psi at Node 6439 which is in the shoe web (Table 5-6). Therefore, for a 1000°F temperature change, the maximum stress is essentially zero and the model is behaving as expected.
**TABLE 5-1**

Maximum Stress Summary

Type I - Steel Shaft

Uniform Temperature Case

**ERSE FOR LABEL TYPE FROM 1 TO 1 BY 1**

#### POST1 ELEMENT STRESS LISTING ####

<table>
<thead>
<tr>
<th>ELEM</th>
<th>SIG1</th>
<th>SIG2</th>
<th>SIG3</th>
<th>SINT</th>
<th>SIGE</th>
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<tbody>
<tr>
<td>1311</td>
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<td>0.3655043E-02</td>
<td>-0.5259150E-03</td>
<td>0.2944285E-01</td>
<td>0.2758717E-01</td>
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<tr>
<td>1370</td>
<td>0.2875893E-01</td>
<td>0.3654051E-02</td>
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<td>-0.2875254E-01</td>
<td>0.2857849E-01</td>
<td>0.2679848E-01</td>
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<td>1378</td>
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<td>-0.3785048E-02</td>
<td>-0.2889546E-01</td>
<td>0.2826705E-01</td>
<td>0.2682013E-01</td>
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<tr>
<td>1306</td>
<td>0.7727128E-01</td>
<td>0.3312730E-02</td>
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<td>0.2818022E-01</td>
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<tr>
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<td>0.2650513E-01</td>
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#### POST1 ELEMENT LISTING ####

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<th>STIF</th>
<th>MAT</th>
<th>NODES</th>
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</thead>
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<td>1</td>
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<td>1831 1681 1657 1807 1832 1682 1658 1808</td>
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<td>45</td>
<td>1</td>
<td>1832 1682 1658 1808 1833 1683 1659 1809</td>
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</tbody>
</table>

#### POST1 MODAL STRESS LISTING ####

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<th>SIG1</th>
<th>SIG2</th>
<th>SIG3</th>
<th>SI</th>
<th>SIGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1833</td>
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<td>-0.1079572E-01</td>
<td>-0.4258170E-01</td>
<td>0.4269354E-01</td>
<td>0.3900044E-01</td>
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<tr>
<td>1832</td>
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<tr>
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<td>0.3337794E-01</td>
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## TABLE 5-2
Maximum Stress Summary
Type 2 - Rubber Bushing
Uniform Temperature Case

**ERSE FOR LABEL: TYPE FROM 2 TO 2 BY 1**

### POSTI ELEMENT STRESS LISTING

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<thead>
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<th>ELEM</th>
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<th>SIG2 (ksi)</th>
<th>SIG3 (ksi)</th>
<th>SINT (ksi)</th>
<th>SIGE (ksi)</th>
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<tbody>
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<tr>
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### POSTI MODAL STRESS LISTING

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<th>SIG3 (ksi)</th>
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TABLE 5-3
Maximum Stress Summary
Type 3 - Shoe Binocular
Uniform Temperature Case

**POST1 ELEMENT STRESS LISTING**

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<th>SIGE</th>
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**POST1 MODAL STRESS LISTING**

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### TABLE 5-4
**Maximum Stress Summary**
Type 4 - Shoe End Plates
Uniform Temperature Case

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### TABLE 5-5

Maximum Stress Summary
Type 5 - Shoe Rib and Wall
Uniform Temperature Case

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**TABLE 5-6**  
Maximum Stress Summary  
Type 6 - Shoe Web  
Uniform Temperature Case

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## TABLE 5-7
### Maximum Stress Summary
**Type 7 - Shoe Fillets**
**Uniform Temperature Case**

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5.2 Pure Tensile Load

This section presents the 3-D results for the pure tensile load - Case 1. A free body diagram for this load case is illustrated in Figure 3-1. Five separate load cases were actually analyzed to investigate the effect of several parameters and to calibrate the model with the Goodyear test results.

Table 5-8 summarizes the significant results of the five tensile load cases. Case 1.5 is the final tensile case and is considered the best model simulation of the track shoe. Detailed stress and displacement plots for Case 1.5 are presented later in this section. But first, the significant findings of Table 5-8 will be discussed.

A load of 36,000 lbs. per shoe (same as Goodyear test) was assumed for all tensile cases. The first approximation rubber properties (E = 20 x 10^6 psi and ν = .49) as developed in Appendix B were assumed for Cases 1.1 and 1.2. The only difference between the first two cases is the type of pin support at the outside pin connector. Figures 5-1 and 5-2 show displacement plots for the simple support and clamped support, respectively. There is a significant difference in both displacements and stresses between these two cases. Case 1.2, the clamped support, is considered to be the more realistic representation of the pin connector.

Reviewing the stress results for Cases 1.1 and 1.2, one important and sig-
significant observation regarding the rubber stresses was made. Although the maximum stress intensity in the rubber is only 4218 psi for Case 1.1, the hydrostatic stress component is very high. For example, the three principal stresses at node 506 in the rubber are SIG1 = 13,907 psi, SIG2 = 11,321 psi, and SIG3 = 9689 psi. This hydrostatic stress state, where the three normal stresses are nearly equal, could cause a problem in a material with a very large Poisson's ratio, such as, rubber.

From the theory of elasticity, the relation between volume expansion and the sum of the three normal stresses can be derived from Hooke's law and is:

\[ e = \frac{(1-2v) \theta}{E} \]

where: \[ e = \epsilon_x + \epsilon_y + \epsilon_z \]
\[ \theta = \sigma_x + \sigma_y + \sigma_z \]

For a uniform hydrostatic stress state:

\[ \sigma_x = \sigma_y = \sigma_z = \sigma_0, \text{ and} \]
\[ e = \frac{3(1-2v) \sigma_0}{E} \]

For an incompressible material, \( v \) is 1/2 and thus the unit volume expansion \( e \) is zero. Therefore, the element will not distort for any value of \( E \) and the material will act like a rigid cube.

In our case, there is a large hydrostatic stress component and a very small
rubber displacement. This hydrostatic stress in the rubber is modeling induced, however. Since the rubber is only one element in width, it cannot distort because the nodal displacements are essentially fixed. One side of the rubber is connected to relatively stiff steel and the other side is connected to the relatively stiff aluminum. One way to eliminate this problem is to put more rows of rubber elements between the steel and aluminum and this will permit the rubber to distort. This action was considered too costly because of the large number of additional elements needed and therefore was not taken. Additionally, actual rubber stresses are not significant for this class of problems.

The manner selected to eliminate this modeling induced rigidity problem is to set the Poisson's ratio of rubber equal to zero. This essentially makes a series of radial (uniaxial) springs out of the rubber elements. Since the primary purpose of the rubber is to transfer load from the shaft to the shoe, this assumption is considered entirely satisfactory for that purpose.

Case 1.3 of Table 5-8 is exactly the same as Case 1.2 except that Poisson's ratio for the rubber was set to zero. There is a significant increase in shaft deflection due to the more flexible rubber model, and the stresses either remain the same or increase.

Cases 1.3, 1.4, and 1.5 are exactly the same except for the rubber modulus. As can be seen, the shaft deflection increases as the rubber modulus decreases. Case 1.4 is an intermediate case and the detailed stress results were not pro-
cessed. The shaft deflection for Case 1.5 is approximately .088" which is nearly equal to the .098" deflection obtained from the Goodyear test. As a result of these tensile load calibration runs, the model as defined by Case 1.5 is considered to be qualified and will be used to make the demonstration load case runs described in Section 3.0.

Displacement and stress contour plots are presented at six cross-sections in the 3-D shoe model. These six cutting planes are illustrated in Figure 5-3. Figures 5-4 and 5-5 show selected nodal points on Planes 1 and 2, respectively. These sketches are used to locate the nodes when the displacements are summarized.

Tables 5-9 to 5-15 show the maximum stress summaries in the seven element types for the pure tensile load case. This data is in the same format as that presented for the uniform temperature case in Section 5.1. Displacements of the shaft and rubber in Plane 1 for this case are summarized in Table 5-16. Note that the minus sign for the relative displacement column means compression and the plus sign means tension. Since the rubber preload is approximately 0.1" and the maximum rubber stretch is only .079", the rubber preload is still maintained for this loading.

Figures 5-6 to 5-12 show displacement plots at the six cutting planes as shown in Figure 5-3. It should be noted that some of the displacement plots are greatly exaggerated and some of the components appear to overlap. This is only
caused by the large scale factor. Stress contour plots are shown at the same six planes in Figures 5-13 to 5-22. Most of the plots are stress intensity plots; however, on two planes the principal stress plots are also shown.

It should be emphasized that the maximum loads assumed for this load case and the two succeeding load cases are somewhat arbitrary, although the load selected for this case corresponds to the maximum load used in the Goodyear test (Figure B-5). The analytical model is linear and the results can be scaled as long as the rubber preload is maintained in the binocular section.

The stresses presented in the stress summary tables represent peak surface stresses, for the most part, and are not necessarily the controlling parameter for material failure. Generally, membrane and bending stresses on a given section are more related to ductile failures than the peak surface stress. The actual stress evaluation is beyond the scope of work of this current contract. Additionally, the applied loads must be known in order to make a judgment on the structural adequacy of the component.

The stresses calculated in the steel shaft are relatively high (Table 5-9) and are primarily bending stresses at the symmetry plane and at the connector end. The large forces and moments causing these large shaft stresses are being reacted by the connectors which may also be experiencing high stresses. Additional studies should be performed on the shaft and connector to assess their load-carrying capability.
## TABLE 5-8
Summary of the 3-D Model
Tensile Load Studies

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### Results

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*Element Types - See Figure 2-9 for identification

**Node Numbers - See Figures 2-11 to 2-17 for locations
## TABLE 5-9
Maximum Stress Summary
Type 1 - Steel Shaft
Pure Tensile Load - Case 1.5

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Type 2 - Rubber Bushing
Pure Tensile Load - Case 1.5

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### Table 5-11
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Pure Tensile Load - Case 1.5

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Type 4 - Shoe End Plates
Pure Tensile Load - Case 1.5

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### TABLE 5-13
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Pure Tensile Load - Case 1.5

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#### POSTI MODAL STRESS LISTING
### TABLE 5-14
Maximum Stress Summary
Type 6 - Shoe Web
Pure Tensile Load - Case 1.5

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### TABLE 5-16
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### TABLE 5-15
Maximum Stress Summary
Type 7 - Shoe Fillets
Pure Tensile Load - Case 1.5

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TABLE 5-16  
Displacement Summary for Tensile Load Case 1.5

SHAFT DISPLACEMENTS

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RUBBER DISPLACEMENTS

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NOTES:  
(1) Refer to Figure 5-4 for node locations.  
(2) Minus sign on relative displacements (Δ) means compression.
Figure 5.2 - Displacement Plot, Tensile Case 1.2

CLAMPED SUPPORT CONDITION
AT PIN CONNECTOR

NOTE: DISPLACEMENTS ARE EXAGGERATED

DISP. - HOR. PLANE THRU CL OF SHAFT - TENSILE LOADING CASE 2
Figure 5-3 - Cutting Planes Used to View Finite Element Results
Figure 5-4 - Selected Nodal Points on Plane 1
Figure 5-5 - Selected Nodal Points on Plane 2
Figure 5-6: Displacements, Plane 1, Tensile Load Case 1.5

NOTE: Displacements are exaggerated.

Disp. - hor. plane thru cL of shaft - tensile load case 5
Figure 5-7 - Displacements, Plane 1, Tensile Load Case 1.5
Displacements to Scale, Scale = 1.0
Figure 5-8 - Displacements, Plane 2, Tensile Load Case 1.5
Figure 5-9 - Displacements, Plane 3, Tensile Load Case 1.5
Figure 5-10: Displacements, Plane 4, Tensile Load Case 1.5
Figure 5-11 - Displacements, Plane 5, Tensile Load Case 1.5
Figure 5-12 - Displacements, Plane 6, Tensile Load Case 1.5
Figure 5-14 - Stress Intensity, Plane 2, Tensile Load Case 1.5.
Figure 5-17 - Stress Intensity, Plane 3, Tensile Load Case 1.5
Figure 5-18 - Stress Intensity, Plane 4, Tensile Load Case 1.5
Figure 5-19 - SIG1 Principal Stress, Plane 5, Tensile Load Case 1.5
Figure 5-21 - Stress Intensity, Plane 5, Tensile Load Case 1.5
Figure 5-22 - Stress Intensity, Plane 6, Tensile Load Case 1.5
5.3 Out-of-Plane Load

The displacement and stress results for the out-of-plane loading (Case 2) are presented in this section. A free body diagram for this load case is illustrated in Figure 3-2. The load is applied at a $30^\circ$ angle with the horizontal plane of the shoe. The same set of tables and plots that were presented in Section 5.2 for the pure tensile load case are presented here.

Tables 5-17 to 5-23 summarize the maximum stresses that were calculated for the out-of-plane load in the various element types. Figures 5-23 to 5-28 show displacement plots at the six cutting planes which are illustrated in Figure 5-3. Stress contour plots for the same six planes are shown in Figures 5-29 to 5-38.
### TABLE 5-17
Maximum Stress Summary
Type 1 - Steel Shaft
Out-of-Plane Load, Case 2

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### TABLE 5-18

Maximum Stress Summary

Type 2 - Rubber Bushing

Out-of-Plane Load, Case 2

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## TABLE 5-19
Maximum Stress Summary
Type 3 - Shoe Binocular
Out-of-Plane Load, Case 2

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DEAC ENGINEERING ANALYSIS CORPORATION

REPORT NO.  DEAC-TR-120
REV. NO. ALC-85-003
PROJECT NO.  BY
DATE  CHECKD. BY  DATE  PAGE  76
## TABLE 5-20
Maximum Stress Summary
Type 4 - Shoe End Plates
Out-of-Plane Load, Case 2

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Maximum Stress Summary
Type 6 - Shoe Web
Out-of-Plane Load, Case 2

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Post-Element Stiffness Matrix

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### TABLE 5-23
**Maximum Stress Summary**

**Type 7 - Shoe Fillets**

**Out-of-Plane Load, Case 2**

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#### ERASE FOR LABEL TYPE FROM 7 TO 7 BY 1

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Figure 5-23 - Displacements, Plane 1, Out-of-Plane Load, Case 2

DISP. - HOR. PLANE THRU CL OF SHAFT - BENDING LOAD
Figure 5-24 - Displacements, Plane 2, Out-of-Plane Load, Case 2
Figure 5-25 - Displacements, Plane 3, Out-of-Plane Load, Case 2
Figure 5-26 - Displacements, Plane 4, Out-of-Plane Load, Case 2
Figure 5-27 - Displacements, Plane 5, Out-of-Plane Load, Case 2
Figure 5-28 - Displacements, Plane 6, Out-of-Plane Load, Case 2
Figure 5-29 - Stress Intensity, Plane 1, Out-of-Plane Load, Case 2
Figure 5-30 - Stress Intensity, Plane 2, Out-of-Plane Load, Case 2
Figure 5-31 - SIG1 Principal Stress, Plane 3, Out-of-Plane Load, Case 2
Figure 5-32 - SIG3, Principal Stress, Plane 3, Out-of-Plane Load, Case 2

SIG3 - LAT. PLANE THRU THK. (OUTER) PLATE - BENDING LOAD
Figure 5-33 - Stress Intensity, Plane 3, Out-of-Plane Load, Case 2
Figure 5-34 - Stress Intensity, Plane 4, Out-of-Plane Load, Case 2

SI - LAT. PLANE THRU CENTERLINE OF SHOE - BENDING LOAD
Figure 5-36 - SIG3 Principal Stress, Plane 5, Out-of-Plane Load, Case 2
Figure 5.37 - Stress Intensity, Plane 5, Out-of-Plane Load, Case 2

SI - LAT. PLANE THRU THIN (BTUN. SHOES) PLATE - BENDING LOAD
Figure 5-38 - Stress Intensity, Plane 6, Out-of-Plane Load, Case 2
5.4 Twisting Load

This section presents the results for Case 3, a twisting load on the shoe, as illustrated in Figure 3-3. The format used to present these results is identical to that of the two previous sections and is summarized below:

- Tables 5-24 to 5-30: Maximum stress summaries
- Table 5-31: Shaft and rubber displacements
- Figures 5-39 to 5-44: Displacement plots
- Figures 5-45 to 5-54: Stress contour plots
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Twisting Load, Case 3

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## TABLE 5-25

Maximum Stress Summary
Type 2 - Rubber Bushing
Twisting Load, Case 3

### BEAM FOR LABEL: TYPE FROM 2 TO E BY 1

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Maximum Stress Summary
Type 4 - Shoe End Plates
Twisting Load, Case 3

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**Maximum Stress Summary**

Type 5 - Shoe Rib and Wall

Twisting Load, Case 3

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### TABLE 5-29

Maximum Stress Summary
Type 6 - Shoe Web
Twisting Load, Case 3

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### TABLE 5-30
Maximum Stress Summary
Type 7 - Shoe Fillets
Twisting Load, Case 3

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TABLE 5-31
Displacement Summary for Twisting Load, Case 3

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**RUBBER DISPLACEMENTS**

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<th>UX&lt;sub&gt;j&lt;/sub&gt;</th>
<th>RELATIVE DISPLACEMENT (Δ = UX&lt;sub&gt;i&lt;/sub&gt; - UX&lt;sub&gt;j&lt;/sub&gt;)</th>
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**NOTES:**
(1) Refer to Figure 5-5 for node locations.
(2) Minus sign on relative displacements (Δ) means compression.
Figure 5-39 - Displacements, Plane 1, Twisting Load, Case 3
Displacements to Scale, Scale = 0
Figure 5-40 - Displacements, Plane 2, Twisting Load, Case 3
Figure 5-43 - Displacements, Plane 5, Twisting Load, Case 3

DISP. - LAT. PLANE THRU THIN PLATE - TWIST LOAD
Figure 5-44 - Displacements, Plane 6, Twisting Load, Case 3
Figure 5.45 - Stress Intensity, Plane 1, Twisting Load, Case 3

SI - HOR. PLANE THRU CL OF SHRT - TWIST LOAD
Figure 5-47 - SIG1 Principal Stress, Plane 3, Twisting Load, Case 3
Figure 5-48 - SIG3 Principal Stress, Plane 3, Twisting Load, Case 3
Figure 5-49 - Stress Intensity, Plane 3, Twisting Load, Case 3
Figure 5-52 - SIG3 Principal Stress, Plane 5, Twisting Load, Case 3
Figure 5-53 - Stress Intensity, Plane 5, Twisting Load, Case 3
Figure 5-54 - Stress Intensity, Plane 6, Twisting Load, Case 3
6.0 LOG OF COMPUTER FILES ON ALCOA'S DEC VAX 11/785

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<td>[KAHRS2.DEAC]</td>
<td>ANSYS PREP7 file that contains B.C.'s and loadings for the pure tensile load case 1.5.</td>
</tr>
<tr>
<td>SHTENS512.OUT</td>
<td>[KAHRS2]</td>
<td>ANSYS binary FILE12 which contains all data for post-processing of the pure tensile load case 1.5.</td>
</tr>
<tr>
<td>SHOEBEND.DAT</td>
<td>[KAHRS2.DEAC]</td>
<td>ANSYS PREP7 file that contains B.C.'s and loadings for the out-of-plane load.</td>
</tr>
<tr>
<td>SHBEND12.OUT</td>
<td>[KAHRS2]</td>
<td>ANSYS binary FILE12 which contains all data for post-processing of the out-of-plane load.</td>
</tr>
<tr>
<td>SHOETWIST.DAT</td>
<td>[KAHRS2.DEAC]</td>
<td>ANSYS PREP7 file that contains B.C.'s and loadings for the twisting-couple load.</td>
</tr>
<tr>
<td>SHTWIST12.OUT</td>
<td>[KAHRS2]</td>
<td>ANSYS binary FILE12 which contains all data for post-processing of the twisting-couple load.</td>
</tr>
<tr>
<td>POST.COM</td>
<td>[KAHRS2]</td>
<td>ANSYS POST1 file that generates the post-processed stress tables and plots.</td>
</tr>
<tr>
<td>AEXEC.COM</td>
<td>[KAHRS2]</td>
<td>VAX command file to execute a job on the FPS.</td>
</tr>
<tr>
<td>ANS27.COM</td>
<td>[KAHRS2]</td>
<td>VAX command file to generate a FILE27 for use on the FPS.</td>
</tr>
<tr>
<td>INT.COM</td>
<td>[KAHRS2]</td>
<td>VAX command file to &quot;wake up&quot; ANSYS interactively.</td>
</tr>
</tbody>
</table>
APPENDIX A

Scope of Work
Defined for Project
Stress analysis of tank track shoe by Finite Element Method

Create a 3 dimensional finite element model of the track shoe represented by the attached drawing and pictures. An actual part will be shipped to Design Engineering Analysis Corporation, DEAC, to aid in the creation of the model.

DEAC shall develop a ¼ symmetry model of the track shoe using ANSYS STIFF 45 isoparametric solid elements. The model shall include the steel pin and rubber bushing in order to develop the proper loading on the shoe, particularly in the binocular section of the shoe. There shall be approximately 3000 elements in the quadrant, with 1600 elements in the shoe forging and 1200 elements in the pin and bushing.

DEAC shall then use the 3-D model to analyze various loading conditions on the shoe. The tensile and side loads will be evaluated with the quadrant model by applying the proper boundary conditions to the planes of symmetry. The specific set of load cases to be analyzed are to be as follows:

1. Pure tension
2. Out-of-plane bending
3. Twisting

These load cases shall demonstrate part performance at the given loads. The results of the three load cases shall be reviewed, plotted separately and combined within ANSYS POST1. Other ANSYS POST1 processing shall include displacements, stress contour plots and a summary of maximum stresses.

The model shall be created using ANSYS PREP7 on a DEC VAX 11/785 located at the Alcoa Technical Center near Pittsburgh, Pa. DEAC is to supply their own terminal devices which must be compatible with 1200 baud asynchronous dial up modem devices. A Floating Point Systems FPS-164 is networked to the VAX and is available for the analysis run.
DEAC is responsible for successfully completing an analysis run for one set of load cases as described above. In the event of a numerical instability caused by modeling the rubber bushing with solids, DEAC shall replace the bushing elements with STIFF 40 combination element (2 parallel springs with a gap element) and obtain a converged solution. This work if necessary, shall be done at NO additional charge to ALCOA.

DEAC shall provide three copies of the final report. The report shall include applicable command file listings and plots. Prior to completion of the final report, a rough draft shall be submitted for review. Weekly verbal updates to ALCOA engineers will be expected.

**TIMING:** The project shall be complete by October 4, 1985.
<table>
<thead>
<tr>
<th>REPORT NO.</th>
<th>REV. NO.</th>
<th>PROJECT NO.</th>
<th>PAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>DEAC-TR-120</td>
<td></td>
<td>ALC-85-003</td>
<td>126</td>
</tr>
</tbody>
</table>

APPENDIX B

Development of Rubber Properties for Bushing Material
The bushing material between the steel pin and aluminum forging is natural rubber. The following material specifications and fabrication procedures were provided by Dan Carbaugh of Alcoa by telephone on September 3, 1984:

**Bushing Material:**  Natural Rubber (NR)

- Ultimate tensile strength = 3000 psi
- Shore A durometer = 65-70

**Fabrication:**  Rubber rings or doughnuts are molded and bonded to steel pins at intervals. Rubber doughnuts are compressed 35% from free state when pressed into track shoe binocular. Rubber expands axially to close up gaps between the doughnuts.

Rubber is an elastomeric or viscoelastic material which exhibits high elongation and high speed of retraction. The idealized behavior most nearly approximating that of rubberlike materials is known as linear viscoelastic behavior. Rubber does not follow Hooke's law and can be characterized by a nonlinear elastic behavior which becomes stiffer with increasing strain, i.e., the tangent modulus increases with increasing load. Rubber also exhibits some time-dependent permanent viscous or creep deformation.

Probably the most important property of rubber for design purposes is the modulus of elasticity which is difficult to specify since the material is nonlinear. Additionally, the stress-strain curves for rubber in tension, compression, and shear are all different. For these reasons, it is common to specify the durometer hardness of rubber materials. The rubber hardness is not important
of itself, but it is merely an approximate and convenient measurement which is related to the modulus of elasticity and is independent of a specimen shape factor.

Since a compressive stress-strain curve for the natural rubber is not available, a compressive stress-strain curve for a similar rubber was used to determine an effective modulus of elasticity for our study. Figure B-1 shows a compressive stress-strain curve for Nitrile Butyl Rubber (NBR) with a 90 durometer reading. Since the durometer hardness readings of both rubbers are similar (70 vs. 90), the NBR stress-strain curve was used for analysis purposes. The rubber comparisons shown in Tables B-1 and B-2 also suggest similarities between NR and NBR rubber.

Since rubber is basically a nonlinear elastic material, a plastic-type iterative solution is required to follow the rubber stress-strain curve. A nonlinear analysis is not economically feasible for large 3-D models as in our case. Additionally, the rubber is only incidental to our problem in that it is required for proper load transfer from the pin to the forging. Our approach will be to select an effective Young's modulus from the rubber stress-strain curve that will approximate the actual rubber stiffness of the assembly.

Since the rubber is compressed 35% when press-fit into the binocular, determine the free height of the doughnut.
Let \( a = \frac{1.375}{2} = .6875 \) = pin outside radius

\( b = \frac{1.781}{2} = .8905 \) = binocular inside radius

\( c = \) free radius of rubber doughnut

\[
\Delta \omega = \frac{c-b}{c-a} = .35
\]

\[
c = \frac{b - .35a}{.65} = 1.000''
\]

Therefore, the rubber is compressed or preloaded to \((c-b) = 1.000'' - .8905'' = .1095''\) on a radius. The final annular compressed thickness of the rubber is \((b-a) = .8905'' - .6875'' = .203''\) which corresponds to a 35% preload.

Using the stress-strain curve in Figure B-1, the tangent modulus at 35% strain is equal to \(E_1\). Therefore, assume an elastic modulus of \(E = 20,000\) psi as a first approximation for our analysis. Rubber is essentially an incompressible substance that deflects by changing shape rather than changing volume.
Therefore, Poisson's ratio approaches 1/2, and for analysis purposes we will use a value of .49.

This assumption will be checked out using a simplified ANSYS STIF42 plane model of the pin and rubber. The plane model will have a unit depth. Therefore, the model will use an equivalent pin height to maintain the proper bending stiffness.

The moment of inertia of the steel pin is:

\[ I = \frac{\pi}{64} (D_0^4 - D_1^4) = \frac{\pi}{64} (1.375^4 - .7375^4) = .1609 \text{ in}^4 \]

For an equivalent rectangular beam of unit depth:

\[ d^3 = \frac{12I}{b} = \frac{12(.1609)}{1} = 1.9308 \]

\[ d = 1.2452'' \]

The ANSYS 2-D model of the pin and rubber is shown in Figure B-2. The model node numbers are shown in Figure B-3, and the boundary conditions are shown in Figure B-4. This model assumes the aluminum is infinitely rigid for these studies. A load of 18,000 lbs. was applied to this half-symmetry model which corresponds to the load deflection test performed by Goodyear.

Figure B-5 shows the load-deflection curve provided by Goodyear. A maximum load of 36,000 lbs. was applied to the shoe and the resulting deflection was
.098". These values will be used as a basis to select an effective elastic modulus and qualify the model.

Two limiting cases (plane strain and plane stress) were run with the 2-D model described in Figure B-2. The plane strain case tends to over-estimate the rubber stiffness because the strain in the Z-direction (into the plane of page) is $\varepsilon_Z = 0$. Since rubber is incompressible, the only deformation it can take is out the ends. The plane stress case tends to under-estimate the rubber stiffness because the stress in the Z-direction is $\sigma_Z = 0$. This allows deformation in the Z-direction and out the ends. The actual 3-D case is probably between these limiting cases.

The plane strain deformation plots are shown in Figures B-6 and B-7. The displacement scale (DSCALE = 4.14) is exaggerated in Figure B-6 and is equal to 1.0 in Figure B-7. Figures B-8 and B-9 show the displacement plots for the plane stress case. Table B-3 shows the ANSYS PREP7 input listing for the plane stress case. Tables B-4 and B-5 list the complete displacement solution for both cases.

The maximum displacements for the two limiting cases are:

Plane strain, Displacement = .04590"
Plane stress, Displacement = .11019"

The test results show that the measured displacement for 36,000 lbs. is .098"
which lies between the calculated values. The average displacement of the two calculated values is .078" which is in the same ballpark as the test results. Noting that the 2-D model did not include the aluminum flexibility, the assumed rubber properties of $E = 20,000$ psi and $v = .49$ are considered satisfactory for initial use in the 3-D model. These properties will be modified as the 3-D model is qualified in the calibration runs in Section 5.2.
### TABLE B-1
Relative Properties of Various Rubbers

#### RUBBER SPRINGS

<table>
<thead>
<tr>
<th>Polymer designation</th>
<th>Common name</th>
<th>Shore A hardness range</th>
<th>Max. tensile strength (psi)</th>
<th>Com-pression set</th>
<th>Tear resistance</th>
<th>Resilience</th>
<th>Heat resistance</th>
<th>Outdoor aging resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural NR...</td>
<td>Natural</td>
<td>30-100</td>
<td>4,000</td>
<td>Good</td>
<td>Good</td>
<td>High</td>
<td>High</td>
<td>Fair</td>
</tr>
<tr>
<td>SBR...</td>
<td>SBR</td>
<td>40-100</td>
<td>2,000</td>
<td>Good</td>
<td>Fair</td>
<td>Fair</td>
<td>Poor</td>
<td>Fair</td>
</tr>
<tr>
<td>Neoprene CR...</td>
<td>Neoprene</td>
<td>40-85</td>
<td>2,000</td>
<td>Poor (GH)</td>
<td>Good (W)</td>
<td>Fair</td>
<td>Excellent</td>
<td>Good</td>
</tr>
<tr>
<td>Butyl HIR...</td>
<td>Butyl</td>
<td>60-75</td>
<td>2,000</td>
<td>Fair</td>
<td>Good</td>
<td>Fair</td>
<td>Excellent</td>
<td>Good</td>
</tr>
<tr>
<td>EPDM...</td>
<td>EPDM</td>
<td>45-100</td>
<td>2,000</td>
<td>Fair</td>
<td>Fair</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Nitrile NBR...</td>
<td>Nitrile</td>
<td>50-100</td>
<td>2,500</td>
<td>Good</td>
<td>Fair</td>
<td>Medium</td>
<td>Medium</td>
<td>Poor</td>
</tr>
<tr>
<td>Propylene PO...</td>
<td>Propylene</td>
<td>45-80</td>
<td>2,000</td>
<td>Fair</td>
<td>Fair</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Urethane --</td>
<td>Urethane</td>
<td>60-80</td>
<td>1,300</td>
<td>Poor</td>
<td>Good</td>
<td>Medium</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Silicon Si...</td>
<td>Silicon</td>
<td>50-90</td>
<td>1,000</td>
<td>Excellent</td>
<td>Fair</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Hyplon CSM...</td>
<td>Hyplon</td>
<td>45-95</td>
<td>2,000</td>
<td>Fair</td>
<td>Good</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Polychloroprene ACM</td>
<td>Polychloro-</td>
<td>45-90</td>
<td>1,000</td>
<td>Good</td>
<td>Fair</td>
<td>Medium</td>
<td>Medium</td>
<td>Excellent</td>
</tr>
<tr>
<td>Fluoro-rubber FPM</td>
<td>Fluoro-rubber</td>
<td>60-90</td>
<td>2,000</td>
<td>Excellent</td>
<td>Fair</td>
<td>Medium</td>
<td>Medium</td>
<td>Excellent</td>
</tr>
</tbody>
</table>

* The relative ease of obtaining good adhesion to metal without employing costly metal treatments and elements.

# Table B-2
Comparison of Various Rubber Properties

## Rubber — Molded, Extruded

<table>
<thead>
<tr>
<th>PHYSICAL PROPERTIES</th>
<th>NR Natural Rubber (Ch-2)</th>
<th>Butadiene- Styrene (BR-2)</th>
<th>Chloroprene (CR)</th>
<th>Butyl (IIR)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific Gravity</td>
<td>0.92</td>
<td>0.94</td>
<td>1.15</td>
<td>0.90</td>
</tr>
<tr>
<td>Elongation, %</td>
<td>0.052</td>
<td>0.143</td>
<td>0.112</td>
<td>0.053</td>
</tr>
<tr>
<td>Refractive Index</td>
<td>1.45</td>
<td>1.43</td>
<td>1.42</td>
<td>1.41</td>
</tr>
</tbody>
</table>

## MECHANICAL PROPERTIES

<table>
<thead>
<tr>
<th>Property</th>
<th>NR Natural Rubber</th>
<th>Butadiene-Styrene</th>
<th>Chloroprene</th>
<th>Butyl</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tear, psi</td>
<td>3500</td>
<td>3000</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td>Abrasion Resistance</td>
<td>Good</td>
<td>Excellent</td>
<td>Good</td>
<td>Excellent</td>
</tr>
</tbody>
</table>

## CHEMICAL RESISTANCE

<table>
<thead>
<tr>
<th>Substance</th>
<th>NR Natural Rubber</th>
<th>Butadiene-Styrene</th>
<th>Chloroprene</th>
<th>Butyl</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sunlight Aging</td>
<td>Fair</td>
<td>Poor</td>
<td>Fair</td>
<td>Poor</td>
</tr>
<tr>
<td>Oxidation</td>
<td>Good</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Solvents</td>
<td>Good</td>
<td>Good</td>
<td>Good</td>
<td>Excellent</td>
</tr>
<tr>
<td>Aliphatic Hydrocarbons</td>
<td>Fair</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Aromatic Hydrocarbons</td>
<td>Fair</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Oxygenated Alcohols</td>
<td>Fair</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Oil, Grease</td>
<td>Poor</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Animal, Vegetable Oils</td>
<td>Poor</td>
<td>Poor</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Acids</td>
<td>Fair</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Dyes</td>
<td>Fair</td>
<td>Fair</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td>Permeability to Gases</td>
<td>Fair</td>
<td>Excellent</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
</tbody>
</table>

## USES

- Pneumatic tires and hoses: power transmission belts and conveyor belts; gaskets; mounting; hose; chemical tank linings; printing press rollers; mould or shell阿里巴巴; wards against air, moisture, sand and dirt.
- Same as natural rubber: carbon-black, dephosphorus, self-sealing fast tanks, aircraft hose, gas dams, gasoline and oil hoses, cables, machinery mounting, printing rolls.
- Wire and cable: wire, hose, electrical insulations, electrical cables, electrical conduits, electrical connectors, electrical switches, electrical fuses, electrical fuses, electrical switches.
- Truck and automobile tire inner tubes, covering bags for tire충전기, and molding, lining, cushion bags.
- Chemical tanks lining, printing press rollers: wards against air, moisture, sand and dirt.

**TABLE B-3**

ANSYS Input Listing for 2-D Model Plane Stress Case

```
PREP7
TITLE, SAMPLE PROBLEM TO TEST RUBBER PROP, PLANE STRESS
ET,1,463.3
ET,2,463.3
R,1
H,1.0,0.0,0.0
H,2.0,0.0,0.0
H,4.0,0.0,0.0
H,5.0,0.0,0.0
HGEM,1,1.6,-5
HGEM,2,56.15,4.73
HGEM,3,56.2,4.73
HGEM,3,56.3,4.73
RAT,E
TYPE,2
REAL,1
E,6.06,2.7
MAT,1
TYPE,1
E,7.53,8
RPE,1,1,1.1
RAT,E
TYPE,2
E,5.10,9.4
EGEN,10.5,-4
EGEN,2,50,2.3
EGEN,3,5,-2
EPL0T
HL0T,1
EX,1,30E6
ALFX,1,0
NUXY,1,3
EX,2,30E6
ALFX,2,0
NUXY,2,49
CP,1,UV,0.67,68.69
TIME,0
ITER,1,1,1
ERF,2
ETEMP,0
D,1.UV,0.05,1
D,1.UV,0.05,1.5
D,2.UV,0.05,1.5
F,FF,FF,10000
C,LINE,1
DBLC,1
FBC,1
EPL0T
LIMIT
AFRACE
FINISH
FTR,27
FINISH
POST1
SET,1,1
MODERASE
PPL0ST
VJERASE
PPL0ST
ASET,1,1,1
PPL0ST
FINISH
```
### TABLE B-4
Displacement Solution for
2-D Model Plane Strain Case

<table>
<thead>
<tr>
<th>NODE</th>
<th>UX</th>
<th>UV</th>
<th>LOAD STEP</th>
<th>ITERATION</th>
<th>CUM. ITER.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00000E+00</td>
<td>0.00000E+00</td>
<td>47</td>
<td>0.02851E-02</td>
<td>103627E-01</td>
</tr>
<tr>
<td>2</td>
<td>0.00000E+00</td>
<td>-0.29915E-02</td>
<td>48</td>
<td>0.05409E-01</td>
<td>103290E-01</td>
</tr>
<tr>
<td>3</td>
<td>0.00000E+00</td>
<td>-0.36562E-02</td>
<td>49</td>
<td>0.02815E-02</td>
<td>103527E-01</td>
</tr>
<tr>
<td>4</td>
<td>0.00000E+00</td>
<td>-0.29915E-02</td>
<td>50</td>
<td>0.03628E-01</td>
<td>103627E-01</td>
</tr>
<tr>
<td>5</td>
<td>0.00000E+00</td>
<td>0.00000E+00</td>
<td>51</td>
<td>0.06715E-01</td>
<td>103050E-01</td>
</tr>
<tr>
<td>6</td>
<td>-0.25954E-02</td>
<td>0.00000E+00</td>
<td>52</td>
<td>0.00574E-01</td>
<td>103290E-01</td>
</tr>
<tr>
<td>7</td>
<td>-0.15878E-02</td>
<td>-0.29765E-02</td>
<td>53</td>
<td>0.00746E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>8</td>
<td>-0.14834E-17</td>
<td>-0.36144E-02</td>
<td>54</td>
<td>0.00574E-01</td>
<td>103290E-01</td>
</tr>
<tr>
<td>9</td>
<td>-0.18367E-02</td>
<td>-0.29765E-02</td>
<td>55</td>
<td>0.07386E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>10</td>
<td>-0.29155E-02</td>
<td>0.00000E+00</td>
<td>56</td>
<td>0.72765E-01</td>
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<tr>
<td>11</td>
<td>-0.47351E-02</td>
<td>0.00000E+00</td>
<td>57</td>
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<td>103527E-01</td>
</tr>
<tr>
<td>12</td>
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<td>0.00000E+00</td>
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<td>0.11294E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>13</td>
<td>-0.26829E-17</td>
<td>0.39649E-02</td>
<td>59</td>
<td>0.16499E-01</td>
<td>103527E-01</td>
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<tr>
<td>14</td>
<td>-0.48236E-03</td>
<td>0.20330E-02</td>
<td>60</td>
<td>0.26094E-01</td>
<td>103527E-01</td>
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<tr>
<td>15</td>
<td>-0.77991E-02</td>
<td>0.00000E+00</td>
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<td>103527E-01</td>
</tr>
<tr>
<td>16</td>
<td>-0.89435E-02</td>
<td>0.00000E+00</td>
<td>62</td>
<td>0.45623E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>17</td>
<td>-0.91965E-03</td>
<td>-0.27378E-02</td>
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<td>0.11294E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>18</td>
<td>-0.39573E-17</td>
<td>-0.29256E-02</td>
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<td>0.16499E-01</td>
<td>103527E-01</td>
</tr>
<tr>
<td>19</td>
<td>-0.61865E-02</td>
<td>-0.27378E-02</td>
<td>65</td>
<td>0.26094E-01</td>
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*Note: The table represents the displacement solution for a 2-D model plane strain case, showing the node numbers, UX and UV coordinates, load steps, and iteration numbers.*
TABLE B-5
Displacement Solution for 2-D Model Plane Stress Case

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<th>ITERATION</th>
<th>CUR. ITER.</th>
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MAXIMUMS

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Figure B-1 - Compressive Stress-Strain Curve for NBR Rubber with 90 Durometer
Figure B-2 - Simplified ANSYS 2-D Plane Model of Pin and Rubber
**Figure B-3 - 2-D Model Nodal Point Locations**

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</table>

**SAMPLE PROBLEM TO TEST RUBBER PROP**
P = APPLIED LOAD FOR ½ SYMMETRY MODEL = 18,000 LB.

DISPLACEMENT CONSTRAINTS (△) SHOWN ABOVE.

2-D MODEL ASSUMES ALUMINUM IS INFINITELY RIGID.

SAMPLE PROBLEM TO TEST RUBBER PROP
REFERENCE: Test data obtained by Goodyear - curve supplied by Alcoa.

In letter from D.P. Corbridge, 4/2/85.

Figure B-5 - Load-Deflection Curve of M-1 Bushing Assembly for One Pin/Bushing Installed in One Binocular.
Figure B-6 - Displacement Plot for Plane Strain Case,
Displacement Scale = 4.14
NOTE: MODEL DISPLACEMENTS ARE TO SCALE WITH
MODEL DIMENSIONS DUE TO SCALE FACTOR = 1.0

19,000 lb

.02213" .004590""

DISPLACEMENT

USER SCALING
ZV=1
DIST=3.3
XF=3
DMAX=.0798
DSCA=1

SAMPLE PROBLEM TO TEST RUBBER PROP. PLANE STRAIN
Figure B-8 - Displacement Plot for Plane Stress Case. Displacement Scale = 2.04

NOTE: Model displacements are exaggerated.

P = 19,000 lb

ORIGINAL POSITION

Sample problem to test rubber prop, plane stress
Figure B-9 - Displacement Plot for Plane Stress Case, Displacement Scale = 1.0
APPENDIX C

Parametric Studies of a 2-D Interaction Model of the Shaft, Rubber, and Shoe Endplate
The purpose of this study was to investigate the interaction of the shaft, rubber, and endplate due to a tensile pull load using an economical model. The sensitivity of the rubber modulus and the effect of rubber preload were investigated in this study. These results were used to guide the 3-D model analysis presented in Section 5.0 and to gain some insight into the load paths of the assembly. These studies were run on the Data General MV-8000 computer at Swanson Analysis Systems, Inc. for the sake of expediency.

The 2-D ANSYS plane model of the shoe endplate is shown in Figure C-1. This model is identical to a slice taken through the 3-D model endplate section as described in Section 2.0 of the report. The model is constructed of the ANSYS STIF42 solid elements using the plane stress option. The nodal point locations are shown in Figure C-2.

The model was loaded by a 10,000 lb. load applied to the shaft. This load was considered to be representative of the load being carried by the endplate in the 3-D model. The actual magnitude of the load is not significant, although it should be representative, and it was held constant throughout the study. The material properties for the steel and aluminum are the same as in the 3-D model. The rubber modulus was varied from 20,000 psi to 7,000 psi and Poisson's ratio for rubber was set to zero. As discussed in Section 5.2, Poisson's ratio for rubber was set to zero for the 3-D model and this model in order to eliminate the rigid cube effect due to the incompressible nature of rubber and the modeling approach used for the rubber.
Five parametric load cases were investigated with this model and the significant results are summarized in Table C-1. Case 4 is very similar to Case 5 and is not included in the summary. The only difference between Cases 4 and 5 is that the rubber preload was set to .10 inches for Case 4. The parameters that were varied in the studies are listed in the top four rows of the table. Selected stress and displacement results are listed in the remainder of the table. Figure C-3 shows the location of the three key stress points in the shoe model. The ANSYS input listing for Case 5 is shown in Table C-2.

Cases 1 and 2 compare the effect of the rubber modulus, 20,000 psi vs. 7,000 psi. Although the rubber modulus was reduced by almost a factor of three, the aluminum shoe stresses only increased by 10% or less. The shaft deflection increased by approximately a factor of three which is a direct result of the modulus change.

Displacement and stress contour plots for Cases 1 and 2 are shown in Figures C-4 to C-11. One displacement plot and three stress contour plots are shown for each case. Figure C-4 shows an exaggerated distortion plot of the model. Note that the maximum displacement is only .0292" and the scale factor is 7.11 which exaggerates the motions. A scale factor of 1.0 would show actual displacements to scale with the model dimensions. Figures C-5 and C-6 show the first principal stress contours (SIG1) and the third principal stress contours (SIG3), respectively. Figure C-7 shows the stress intensity contour plot. The distorted shape of the shoe (dashed lines) on the stress plots show that the binocular tends to ovalize under load. The maximum tensile stresses occur at points A and C which is consistent with the stress distributions in a circular ring under
diametral loading. Figures C-8 through C-11 show the same set of plots for Case 2. All stress values listed in Table C-1 were obtained directly from the contour plots. All extrapolated values are indicated with the approximate sign (~).

The 2-D model was modified to include a ring of gaps (ANSYS STIF12) between the rubber and aluminum shoe elements. This model was used to analyze Cases 3, 4, and 5 for the purpose of evaluating rubber separation and preload. Case 3 is the same as Case 2 except that the rubber was allowed to pull away from the aluminum as shown in Figure C-12. As indicated in Table C-1, the maximum tensile stresses for Case 3 are approximately twice those of Case 2. This is due to the fact that the preload was overcome (preload was set to zero for Case 3) and the rubber was allowed to separate. This separation effectively cuts the rubber stiffness in half and allows more deflection and higher stresses in the aluminum. See the UY deflection at Node 590 for Cases 2 and 3 in Table C-1. Figures C-13 to C-15 show the stress contour plots for Case 3.

Case 5 extends Case 3 by preloading the rubber before the external load is applied. This case demonstrates the importance of the rubber preload on the shoe stresses. Case 5 was run in two load steps: Load step 1 is the preload and load step 2 is the preload plus external load. Figure C-16 shows the displacement plot for the preload step. Note that the preload is shown as a gap in the model for illustration purposes only. ANSYS graphics handles gaps in this manner. The actual hardware obviously does not have a gap. Figures C-17 to C-19 show the
stress contour plots for the preload case. The maximum preload principal stress is approximately 13,000 psi (Figure C-17).

Figures C-20 to C-23 show the set of plots for load step 2 of Case 5. The stresses for Case 5 are also summarized in Table C-1. Note that the third column of Case 5 is the difference between the first two columns and represents the effect of the applied load without preload. This result is very significant in that it is approximately equal to the Case 2 results. Remember that Case 2 is only a one load step problem without explicitly modeling the gap interface and preload interference step. This means that, as long as the preload is maintained, the problem can be modeled as a linear system without gaps and preload and that the rubber elements can take both compression and tension. Preload stresses can be superposed on Case 2 stresses if desired. The slight difference in stresses between the Case 2 and Case 5 subtracted results can be attributed to the friction free interface between the rubber and aluminum for Case 5. Since the Case 2 model is continuous across the three materials, shear forces can be transmitted across the boundary and thus have some effect on the stress pattern.

Conclusions

Based on the results of these parametric studies, the following observations can be made:

1. The value of Young's modulus assumed for the rubber has a very small effect on the resulting aluminum shoe stresses. The modulus of rubber,
however, has a significant effect on the shaft displacement within the 
binocular. Compare Cases 1 and 2.

2. The rubber preload has a significant effect on the aluminum shoe 
stresses. When the rubber preload is exceeded or separation occurs, 
the shoe stresses are higher than they would be if adequate preload were 
maintained. Compare Cases 3 and 5.

3. As long as rubber preload is maintained, the track shoe system can be 
modeled as a linear system with the rubber capable of supporting both 
tensile and compressive loads. The tensile loads are only reducing 
the compressive preload in the rubber. Compare Cases 2 and 5. This 
conclusion is significant in that the detailed 3-D model of the shoe 
can be analyzed as a linear system without gaps and costly iterations 
to achieve convergence.
### TABLE C-1
2-D MODEL PARAMETRIC STUDIES
SUMMARY OF RESULTS

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Notes:
1. Case 4 is very similar to Case 5 and is not shown in this table. The Case 4 rubber preload was set to .10".

2. SIGPR is the principal stress and can be either SIG1 or SIG3 depending whether the largest stress is positive or negative.

3. See Figure C-3 for locations.

* This column was calculated by subtracting the preload column from the total column.
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**TABLE C-2**

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Figure C-1 - 2-D ANSYS Interaction Model of the Shaft, Rubber, and Shoe Endplate
Figure C-2 - Node Point Description for 2-D Plane Model
Figure C-3 - Sketch of Shoe Model Showing High Stress Locations
Figure C-4 - Plane Model, Case 1, Displacement Plot
Figure C-5 - Plane Model, Case 1, SIG1 Principal Stress

ALCOA TANK SHOE PLANE MODEL - TENSILE LOADING CASE 1
Figure C-6 - Plane Model, Case 1, SIG3 Principal Stress
Figure C-8 - Plane Model, Case 2, Displacement Plot
Figure C-9 - Plane Model, Case 2, SIG1 Principal Stress
Figure C-10 - Plane Model, Case 2, SIG3 Principal Stress
Figure C-11 - Plane Model, Case 2, Stress Intensity
Figure C-12 - Plane Model, Case 3, Displacement Plot
Figure C-13 - Plane Model, Case 3, SIG1 Principal Stress
Figure C-14 - Plane Model, Case 3, SIG3 Principal Stress
Figure C-16 - Plane Model, Case 5, Preload Displacement Plot
Figure C-17 - Plane Model, Case 5, Preload
SIGI Principal Stress
Figure C-18 - Plane Model, Case 5, Preload
SIG3 Principal Stress
Figure C-19 - Plane Model, Case 5, Preload
Stress Intensity
Figure C-20 - Plane Model, Case S, Preload plus Applied Load Displacement Plot
Figure C-22 - Plane Model, Case 5, Preload plus Applied Load
SIG3 Principal Stress
Figure C-23 - Plane Model, Case 5, Preload plus Applied Load Stress Intensity
APPENDIX D

ANSYS 3-D Model

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<p>| CSIZX                  |
| BOTTOM RIB NODES &amp; CYLINDER INTERSECTION NODES |</p>
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<th>Instruction</th>
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<td>RP2, -1180, 1200, 1200</td>
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<td>Move cylinder to match web bottom</td>
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<td>LOCAL, 22.1, 74, 3.769, 3.852, 90, 190</td>
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<td>MGEM, 3, 13, 6001, 6013, 1, .315</td>
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<td>C5, 20, 1, 1, 8901, 8902</td>
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<td>C5, 21, 0, 6001, 6001, 6012</td>
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<td>C262</td>
<td>Move cylinder tangent to meet web</td>
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<td>C263</td>
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<td>Move back plates</td>
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| RPE | 928,2,1,159,159,4,1,159,159 |
| E   | 1489,6447,6449,6447,1538,6249,6449,6449 |
| E   | 6249,6449,1630,1630,6251,6451,1780,1780 |
| E   | 6447,6449,1630,1480,6647,6449,1629,1479 |
| E   | 4397,1179,6644,6444,4398,1030,6444,6444 |
| E   | 6244,4398,6444,6444,1180,1180,1180,1180 |
| E   | 6262,6451,6251,6251,1780,1780,1780,1780 |
| CIII | FILLET ELEMS |
| TYPE | 7 |
| MAT | 3 |
| RP8 | 928,1,159,159,1,6,150,150,1 |
| E   | 7001,6262,6263,6263,7002,6268,6269,6269 |
| RP8 | 1,6,6,1,6,6,6 |
| E   | 7016,4676,4677,6312,7009,4663,4664,6311 |
| E   | 6312,4677,4676,7016,6313,4690,4690,7017 |
| RP3 | 1,1,13,13,1,1,13,13 |
| E   | 6306,7016,6312,6312,6307,7017,6313,6313 |
| RP3 | 1,1,1,1,1,1,1,1 |
| E   | 6264,6262,7001,7001,6270,6269,7002,7002 |
| RP7 | 5,6,1,1,6,6,1 |
| E   | 6306,6305,7008,7008,7016,6311,7009,7009 |
| E   | 1780,1929,1929,1779,6252,6262,7001,6451 |
| E   | 1780,6451,1779,1779,6651,6651,6651,6651 |
| E   | 6263,6253,7001,7001,6254,6254,6254,6254 |
| E   | 6451,6451,6254,6254,7001,7001,7001,7001 |
| E   | 3270,3130,3129,3129,7009,7009,7009,7009 |
| E   | 3279,3280,4663,4663,7009,7009,7009,7009 |
| E   | 4663,3280,7009,7009,4663,4664,4664,4664 |
| CIII | WAVE RE-ORDERING |
| MAT | 2,49,47 |
| USTART | 902,973,1 |
| Wmore | 902,1,451,1 |
| Wmore | 5155,5157,1 |
| Wmore | 5255,5257,1 |
| Wmore | 4352,4352,1 |
| Wmore | 4362,4362,1 |
| Wmore | 4378,4381,1 |
| Wmore | 4383,4383,1 |
| Wmore | 4391,4391,1 |
| Wmore | 4404,4412,1 |
| Wmore | 4417,4425,1 |
| Wmore | 4430,4438,1 |
| Wmore | 4443,4451,1 |
| USTART | 1952,2033,1 |
| Wmore | 2048,2065,1 |
| Wmore | 3231,5531,100 |
| Wmore | 3228,5228,1 |
| Wmore | 5135,5139,1 |
| Wmore | 5028,5039,1 |
TABLE D-2
ANSYS PREP7 INPUT LISTING FOR
3-D MODEL LOAD CASE 1 - PURE TENSILE LOAD

/PREP7
/TITLE   ALCOA TANK SHOE   -  TENSILE LOADING CASE 5
ET,1,45,1   SHAFT
ET,2,45,1   RUBBER BUSHING
ET,3,45,1   CYLINDER CASTING
ET,4,45,1   THICK & THIN BACK PLATES
ET,5,45,1   WEB & INTERSECTING CYLINDER WALL
ET,6,45,1   WEB ONLY
ET,7,45,1   FILLETS
CIII MATE 1 STEEL
EX,1,1.30E6
ALPX,1,0
MUXY,1,0
DENM,1,283
CIII MATE 2 RUBBER
EX,2,2.40E3
ALPX,2,0
MUXY,2,0
DENM,2,4336
CIII MATE 3 ALUMINUM
EX,3,3.10E6
ALPX,3,0
MUXY,3,0
DENM,3,1088
CIII
CIII
CIII
CPSIZE,50
PRSTR,-1.1,1.2,3,4,5,6,7
TIME,0
ITER,1,1,1
LPRINT,1
KRF,2
KTEMP,0
TREF,0
TUNIF,70
SYMBC,0,2,3,769
SYMDC,0,3,11,870
D,4709,UX,0.
CPIGEN,1,UV,2.49,1
CPIGEN,2,UV,3982,3949,1
CPIGEN,3,UV,2.49,1
F,2,FY,-18000.
F,3982,FY,-18000.
LURITE
AFURITE
FINISH
TABLE D-3
ANSYS PREP7 Input Listing for
3-D Model Load Case 2 - Out-of-Plane Load

PREP7
PREP7
T: TITLE, ALCOA TANK SHOE - BENDING LOAD
ET. 1, 45, SHAFT
ET. 2, 45, RUBBER BUSHING
ET. 3, 45, CYLINDER CASTING
ET. 4, 45, THICK BACK PLATE
ET. 5, 45, WEB & INTERSECTING CYLINDER WALL
ET. 6, 45, WEB ONLY
ET. 7, 45, FILLETS
MAT. 1, STEEL
EX. 1, 30.0E6
ALP. 1.0
NMY. 1.0
DENS. 8.003
MAT. 2, RUBBER
EX. 2, 4.0E3
ALP. 2.0
NMY. 2.0
DENS. 2.0303
MAT. 3, ALUMINUM
EX. 3, 10.0E6
ALP. 3.0
NMY. 3.0
DENS. 3.008
MAT. 5, INSERT GEOMETRY HERE
CP. 50
PRE. -1, 1, 2, 3, 4, 5, 6, 7
TIME, 0
ITER, 1, 1
LPRINT, 1
KRF, 2
CTERP, 0
TREF, 0
TURF, 0
SYRBC, 0, 2, 2, 3, 769
SYRBC, 0, 3, 11, 870
D, 6265, UX, 0, 0, 6267.1
RPS, 0, 0, 0
D, 2659, UX, 0, 0, 6261.1
D, 6481, UX, 0, 0, 6416.1
D, 6424, UX, 0, 0, 6429.1
D, 6437, UX, 0, 0, 6439.1
D, 6687, UX, 0, 0, 6629.1
D, 6537, UX, 0, 0, 6642.1
CPQEM, 1, UV, 2, 49.1
CPQEM, 2, UV, 3962, 3949.1
CPQEM, 3, UV, 2, 49.1
CPQEM, 4, UV, 2, 49.1
CPQEM, 5, UV, 3962, 3949.1
F, 2, FX, 9000
F, 3982, FX, 9000
F, 2, FY, -15500
F, 3982, FY, -15500
LURITE
AFURITE
FINISH
TABLE D-4
ANSYS PREP7 INPUT LISTING FOR
3-D MODEL LOAD CASE 3 - TWISTING LOAD

/PREP7
/STL
/THK
/COF
/COFF
/GEN
/LOAD
/END
TABLE D-5
ANSYS POST1 INPUT LISTING FOR POSTPROCESSING
3-D MODEL RESULTS (MAXIMUM STRESS SUMMARIES,
DISPLACEMENT AND STRESS CONTOUR PLOTS)

```plaintext
SET DEF [KAMHS23]
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STRESS, SIG1, 45, 101
SIGE, 45, 102
SIG3, 45, 103
SINT, 45, 104
SIGE, 45, 105
SET, 1, 1
SSET, NTYP, 1, 1
BEGIN, MAC
ERSEL, TYPE, NTYP
MELEM
ESORT, SINT, 15
PRESTR
PRELEM
EUSORT
NSORT, SI, 15
PRMSTR, PRIM
MALL
EALL
SEND
BDO, MAC, 1, 6
ERSEL, TYPE, 1
MELEM
PRMSTR, ALL
MALL
EALL
ERSEL, TYPE, 2
MELEM
PRMSTR, ALL
MALL
EALL
ERSEL, TYPE, 3, 7, 1
MELEM
PRMSTR, ALL
MALL
EALL
GLOBAL, 1
PRFOR
/TITLE, DISP. - VERT. PLANE THRU CL OF SHAFT - TWIST LOAD
/DSCALE, 1, 1
/VIEU, 1, 1
/FOCUS, 1, 2
/ANG, 90
MALL
EALL
PLDISP
/ANG, 100
/TITLE, DISP. - HOR. PLANE THRU CL OF SHAFT - TWIST LOAD
/VIEU, 1, 1
/FOCUS, 1, 2
MALL
EALL
PLDISP
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TABLE D-5
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PLMSTR.SI
/FOCUS,...805
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EALL
ERSEL,TYPE,3.7.1
PLMSTR,SI
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OAD

PLMSTR,SI
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OAD

PLMSTR,SI
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OAD

PLMSTR,SI
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OAD

ST LOAD

PLMSTR,SI
/TITLE, SI - LAT. PLANE THRU THK (BTUN. SHOES) PLATE - TU

OAD

UIST LOAD

PLMSTR,SI
/TITLE, SIG1 - LAT. PLANE THRU THK (BTUN. SHOES) PLATE - T

OAD

UIST LOAD

PLMSTR,SI
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OAD

IST LOAD

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/PLDISP
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OAD

LOAD

/TITLE, HIDDEN LINE SI PLOT OF HALF-SYMM SHOE MODEL - TUIST

OAD