The Study of a Neoprene Elevator Roller Guide

Robert Michael and David Roth
Penn State Erie, Erie, PA, USA
Richard Ruskey
Elsco, Elevator Safety Co, Owings Mills, MD, USA
Ernie Ferro
President, Corry Rubber Corp. (CRC), Corry, PA, USA

Abstract

This study uses ANSYS to improve the manufacturing and design of a neoprene elevator roller guide. The study uses 2d and 3d hyperelastic and contact elements to model neoprene material tests and also highly deformed roller guide proof tests. The ANSYS analysis succeeded in modeling the neoprene material performance. Geometry changes in the shape of the neoprene were studied to reduce adhesion stresses between the neoprene and aluminum center hub, and yet maintain the spring stiffness of the current roller guide design.

Introduction

An elevator car navigates between rails that are attached to the structure of the building. Roller guides position the elevator by surrounding each rail on three sides, helping to control side-to-side and front-to-back movement. Roller guides play an important role in the vibration and noise control of elevators and are carefully manufactured to precise specifications for added safety and stability. More information about roller guides can be found on the Elsco website, www.elscoguides.com.

The elevator roller guide is currently made of an aluminum hub casting with a ½ inch thick trapezoidal shaped neoprene elastomer pad adhered to the aluminum circumference. A typical application is a three-wheel assembly that is also spring mounted. See Figure 1. The neoprene and the springs allow the guide to provide the smoothest possible ride, even when the elevator rails are rough or slightly misaligned.

Figure 1. Moderate Speed Roller Guide

The roller guide contacts the guide rails through the roller wheel. The roller wheel is generally preloaded 25 to 50 lbs to maintain contact with the guide rails as the elevator traverses the elevator shaft. The roller wheels also see side-to-side loading which puts the neoprene elastomer in shear. Several factors contribute to the shear loading of the wheel including misalignments between the guide rails and elevator; misalignment of the guide rails themselves, and non-uniform weight distribution within the elevator. The wheels also see cyclic radial loading. For example, a typical elevator speed of 2,000 ft/min will result in a dynamic radial load of 1,270 cycles/minute on a 6" wheel. This cyclic radial load may add or subtract from the radial preload depending on rail alignment. See Figure 2.

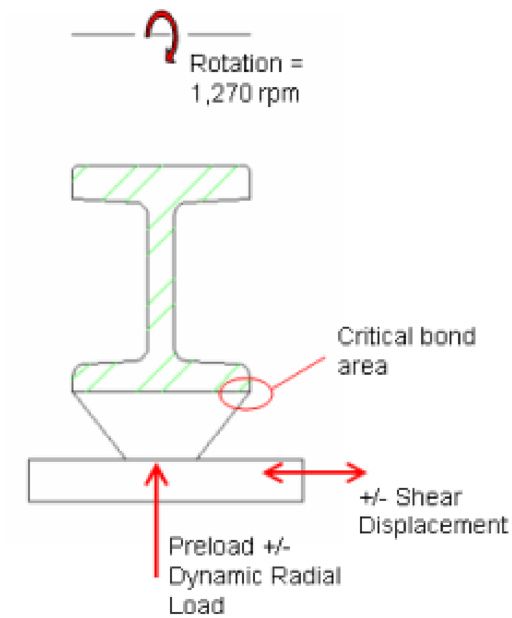

Figure 2. Loads and Motions for 6" Roller Wheel

Due to the aggressive nature of the application, roller wheels are 100% proof tested to examine the adhesive bond between the neoprene elastomer and aluminum hub. See Figures 3 and 4. The proof test displaces the neoprene to levels much beyond any actual elevator application displacements. Since the test does not weaken the roller, it guarantees the integrity of the bond and material of each wheel sold. Occasional bond failures during the proof testing are likely to occur at the edge of the part. It is believed there are several factors, which lead to these edge bond failures. These include the high stress concentrations at the edge of the neoprene, premature heating of the aluminum metal component during the bonding process and thin sections of elastomer, which tend to cure before the adhesion process can take place. The focus of this paper is to describe how the stress concentration plus the thin edge at the neoprene-aluminum interface were eliminated, thereby lessening the number of bond failures during the proof tests.

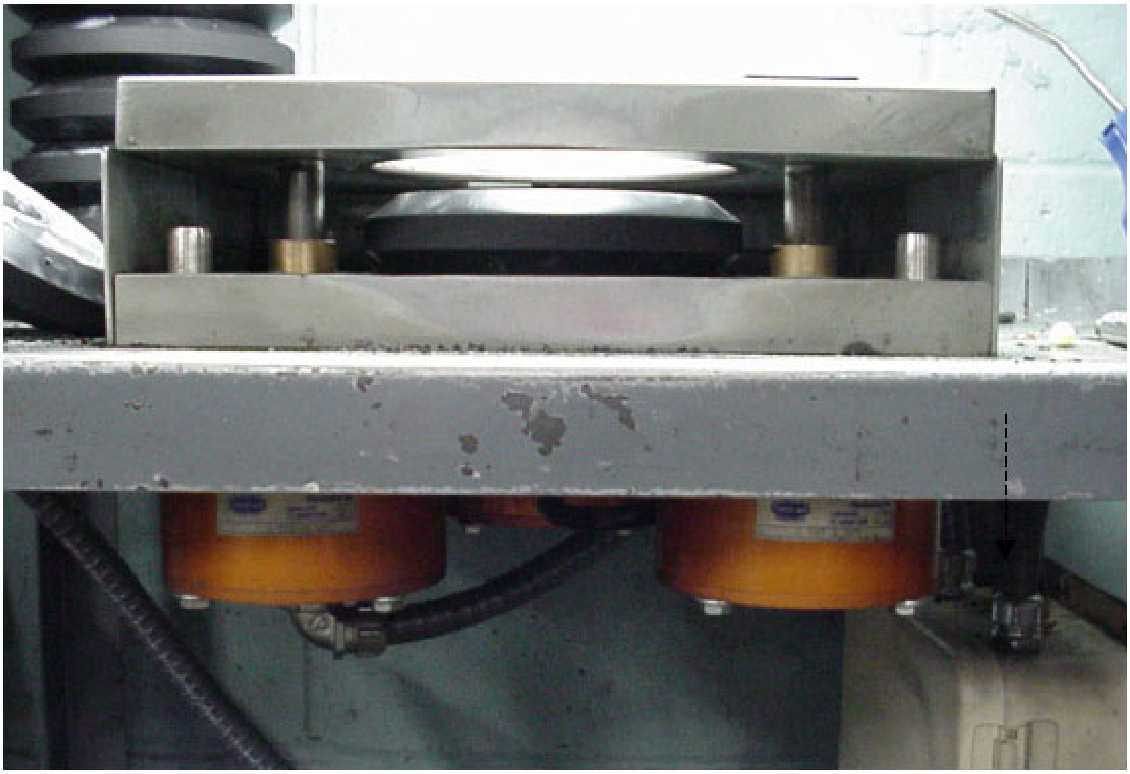

Figure 3. Transverse "Proof Test" Load

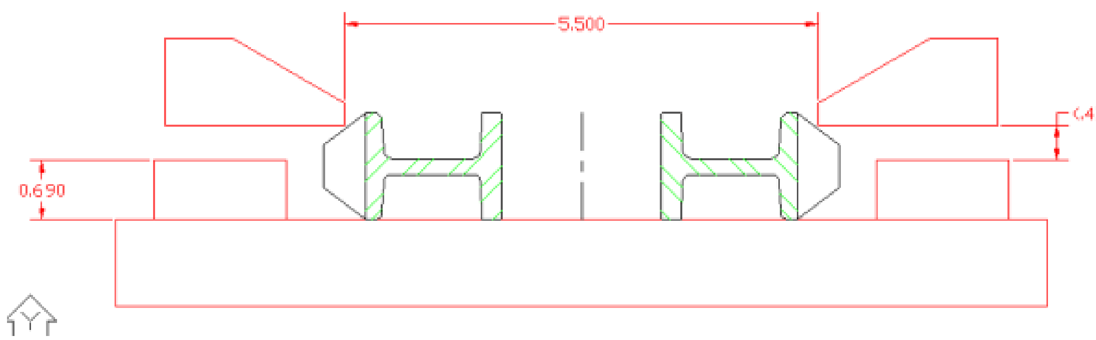

Figure 4. Schematic Showing Transverse Load "Proof Test"

Procedure

It is important that any design change to the neoprene elastomer shape or volume does not affect the total spring stiffness of the roller guide. This is because the current spring stiffness has an excellent history of providing a safe and comfortable ride. To ensure that the spring stiffness of the neoprene is maintained, experiments needed to be performed to record the current load-deflection response of the guide. In addition to providing a basis for any change in the design of the neoprene, the data also was important to choosing a material model for the neoprene. In this manner, constants for the hyperelastic material model could be determined for the ensuing ANSYS models. See Figures 5 and 6.

Figure 5. Front View - Radial Load Versus Deflection for Neoprene Stiffness

Figure 6. Side View - Radial Load Versus Deflection for Neoprene Stiffness

According to Alan Gent, [Ref 1], "Since poisons ratio for elastomers is between 0.499 and 0.5, most finite element programs will not properly analyze the elastomer." This was written in 1992, and since then much has been done to improve hyperelastic elements. [Ref 2] A 3-D ANSYS model using nonlinear elastic hyperelastic elements was used to match the response of the model to the test data for the radial loading. By using this model hyperelastic material constants were obtained for use for the rest of the study using ANSYS axisymmetric models. The goals for studying the axisymmetric models were two-fold: One, to increase the amount of neoprene at the outer edges of the contact of the neoprene and the aluminum for better curing of the adhesive; and two, if possible, to reduce the high stress concentration in this area.

Analysis

Radial Testing and ANSYS Modeling for Radial Loads

The radial stiffness of the neoprene wheel was determined from testing of actual wheels loaded to 500 pounds. See Figures 5 and 6. The radial load-deflection curve was recorded up to approximately 1/8-inch elastomer compression. The data is shown in Figure 7.

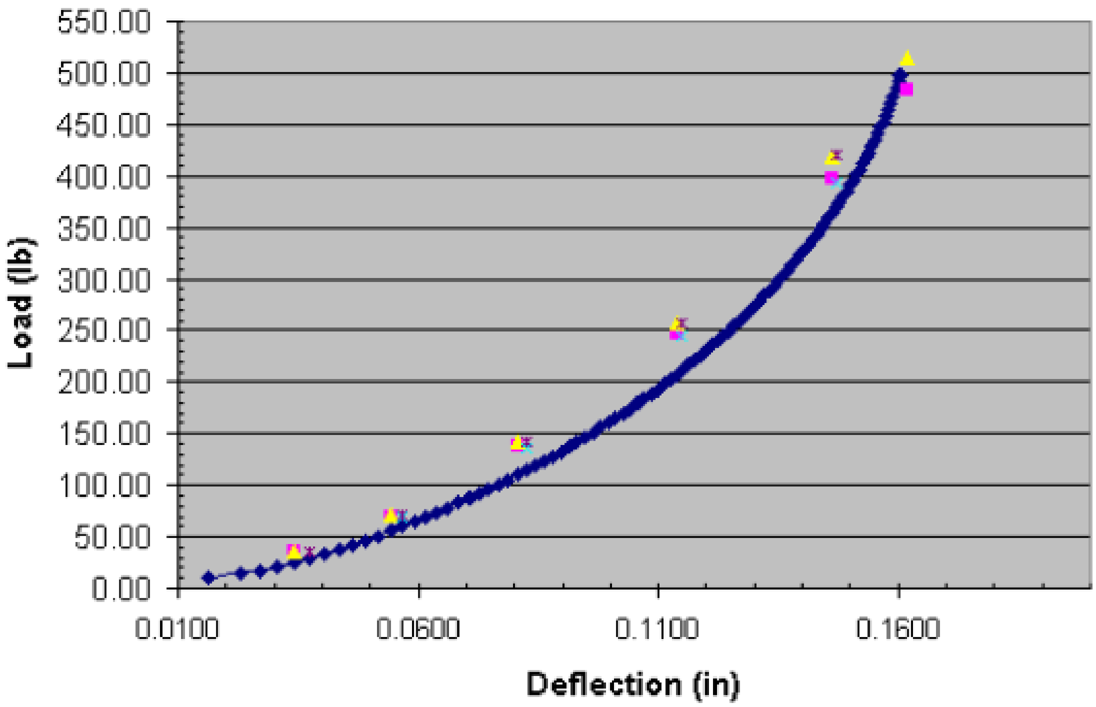

Figure 7. Static Radial Load-Deflection Curve – 500 Pound Test

A quarter-section 3-D model was created using Solid 45 elements for the aluminum hub and Hyper 86 elements for the neoprene. Surface contact elements were used at the interface of the neoprene pad and the aluminum hub. The aluminum hub was oriented in the model such that a rib within the hub directly aligned with the contact point of the neoprene in order to give the stiffest possible path from the axel to the neoprene. This stiff path would concentrate a maximum force on the neoprene, allowing the model to represent the most severe stresses. See Figure 8.

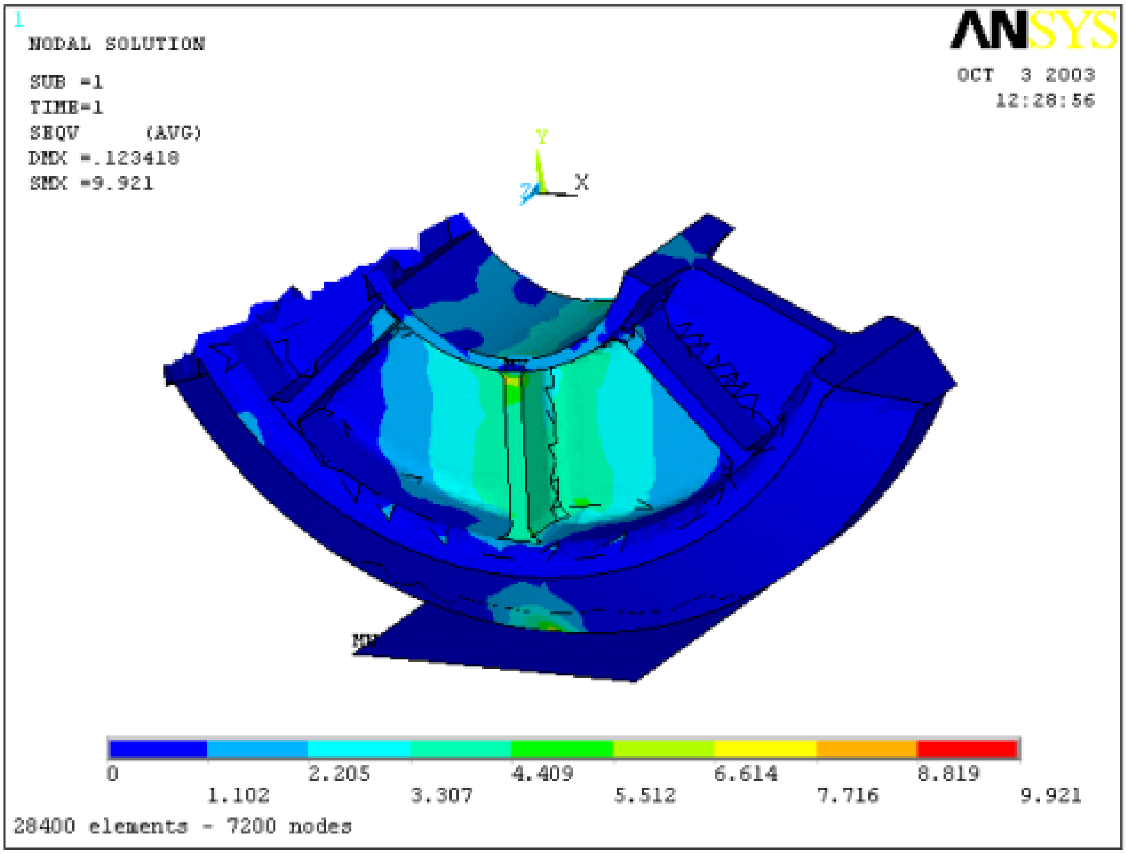

Figure 8. ANSYS Results for Static Radial Load-Deflection

ANSYS Modeling for Transverse Loads

An axisymmetric ANSYS model was created to replicate the transverse deflection of the neoprene during a proof test. ANSYS Plane 42 elements were used for the aluminum hub, and hyperelastic plane 182 elements were used for the neoprene. ANSYS Target 169 and Contact 172 elements were used for the interface between the neoprene and the aluminum. The contact elements were used for two reasons: One; the nodes along the interface would be representing only one material rather than one common node averaging elements of aluminum and neoprene properties. Two, stresses for normal and sliding were readily available and easily reviewed for the contact elements in the ANSYS POSTprocessor. This eliminated the need for element tables for the Plane 182 element results. This particular axisymmetric model was useful in providing an important "bench-mark" for which any design changes could be compared. See Figure 9.

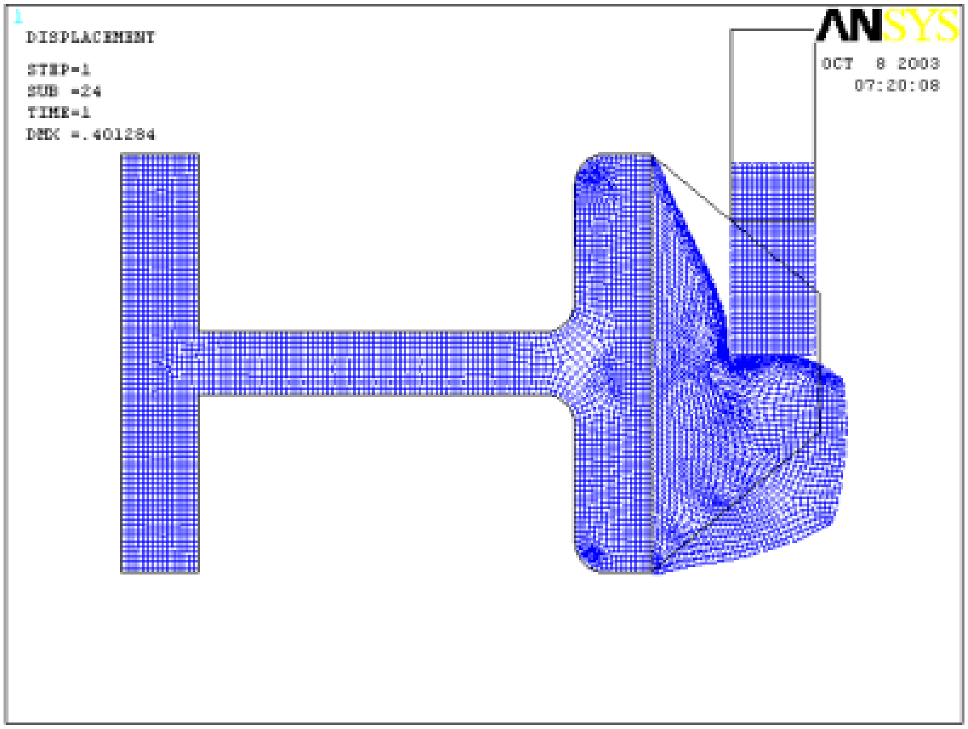

Figure 9. Neoprene Deformation from Transverse Proof Loading

Various geometries were modeled and studied. The geometries were labeled as Case A through Case E; whereas Case A is the original and current design. For the Case variations, See Figure 10. In each model at least 0.02 inches of neoprene elastomer was added to the edge of the interface between the aluminum and the neoprene. It was originally felt that the more neoprene elastomer located at the edge, the less the stress riser there would be due to eliminating any abrupt change in geometry. It should be noted that the current design has the neoprene approaching the aluminum as a sharp edge and no radius.

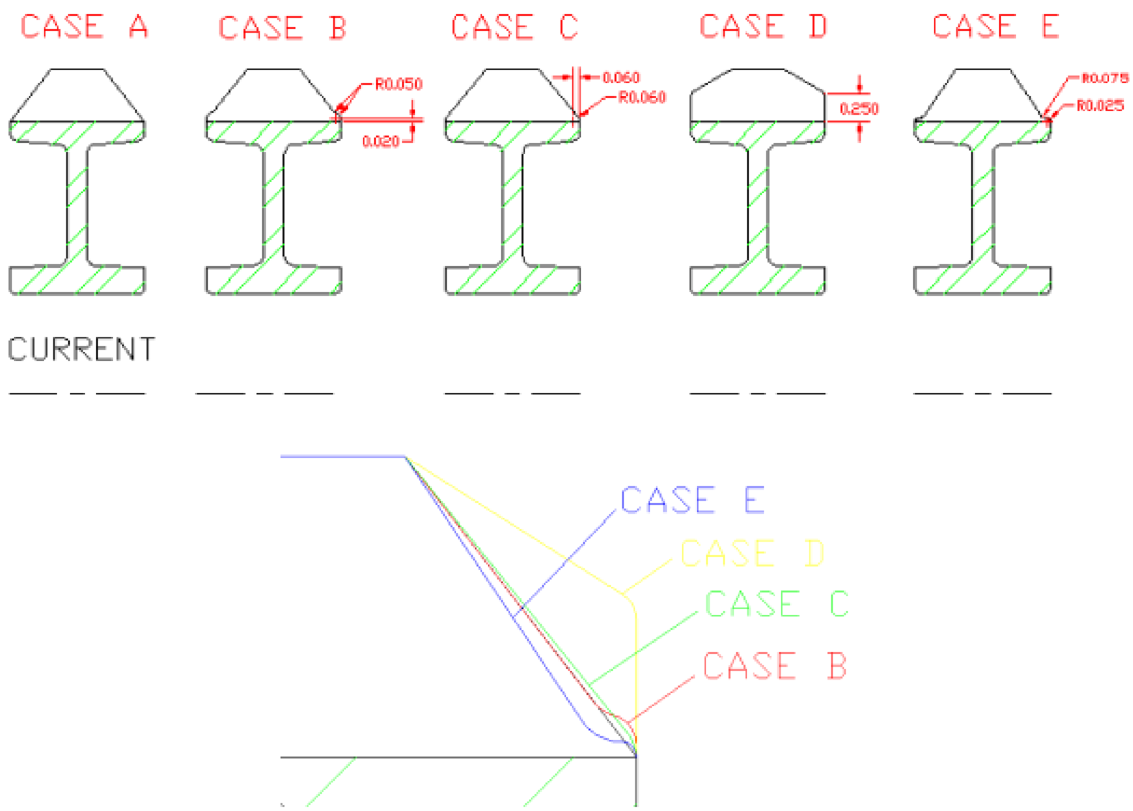

Figure 10. Geometry Variations – Case Studies

Analysis Results & Discussion

The quarter-section 3D ANSYS analysis resulted in radial deformations up to those deformations used in the radial load test data. The radial load versus deformation data is shown as a solid-line on the graph. See Figure 7. Minor adjustments were made to the Mooney Rivlin Hyperelastic material constants, $C_{01}+C_{10}$, in an effort to improve correlation with experimental load-deflection data. However, changing each material constant, while holding their sum constant, did little to better match the analysis to the experimental data. However, changing the sum of $C_{01}+C_{10}$ resulted in the neoprene elastomer being either too stiff or too soft. These two constants combined represent the shear modulus of the neoprene = $2(C_{01}+C_{10})$ [Ref 3]. The various trials of these two constants are shown as discrete data next to the solid line on the Figure 7. The total shear modulus used for the ANSYS modeling was 300 psi, which is very reasonable for neoprene. [Ref 2]. It is the authors' opinion that the Mooney-Rivlin material model is sufficient to represent the load-test data since the model closely followed the test data. Other non-linear elastic models, such as the Ogden or Neo-Hookean models [Ref 2] could be tried if a closer correlation was to be attempted. Once the material constants were determined, the ANSYS models representing the proof test loading were analyzed.

Since the location of the tearing and separation of the neoprene elastomer from the aluminum during the proof testing occurred on the interface edge located on the tension side of the elastomer, only that small area was studied in the ANSYS results. A typical mesh of the axisymmetric model showing one of the geometries studied is seen in the Figure 11. This model shows a metal indenter pushing down on the hyperelastic elastomer elements representing the transverse load of the proof test. See Figure 4.

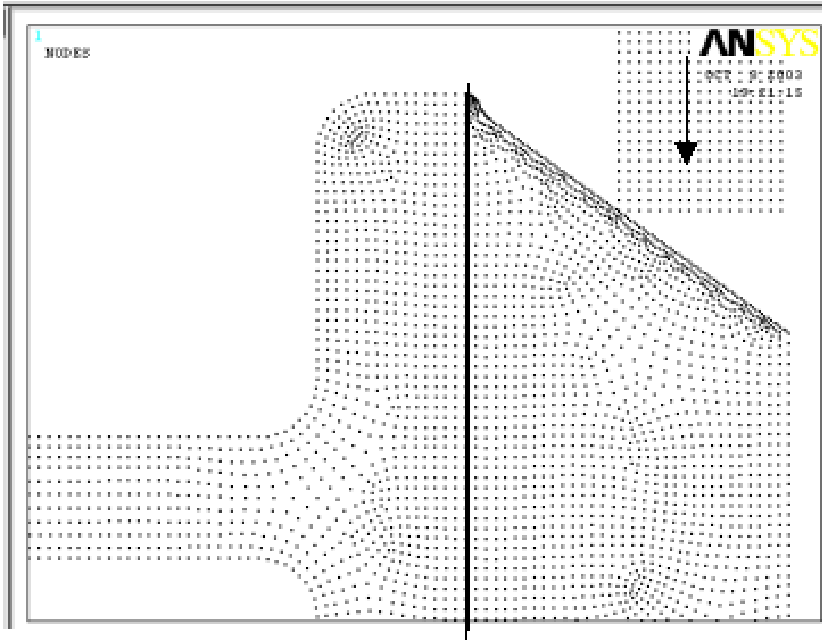

Figure 11. Meshed Model for Case B

A few key interface elements were selected from the entire model. The shear stress, the normal stress, and the total stress in the interface elements were recorded for the different geometries. Each Case was carefully studied. A typical total stress plot of the interface contact elements is seen in Figure 12. The values of the stress varied slightly from the edge to 0.035 inches from the edge. Two elements were within this range and their nodal stresses were averaged to represent the total stress in that area. The stresses shown are the average of the total stress (combined normal and sliding) for the three nodes within the 0.035 inches from the edge. The following table shows the results of each Geometry case.

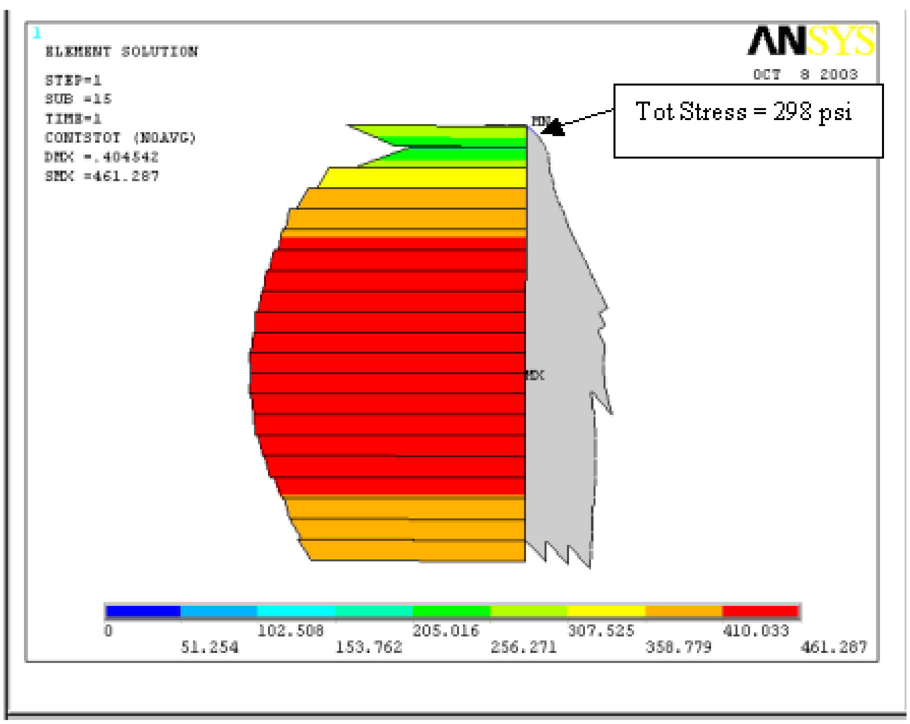

Figure 12. ANSYS Results for Static Radial Load-Deflection

Table 1: Summary Of Total Stress At The End of the Interface

	Geometry A	Geometry B	Geometry C	Geometry D	Geometry E
Avg. Stress(psi)	303.3psi	202.7psi	580.0psi	640.0psi	142.0ps

Conclusion

The original premise of adding 0.02 inch radius of elastomer at the edge of the roller guide was expected to reduce adhesion concerns in manufacturing and reduce the stress concentration at the end. Since the proposed design (Case B) resulted in a smaller stress than the original design (Case A), this premise was proven to be valid.

A second conclusion surprised the authors, but after some thought, seemed quite reasonable. Studying the analyses of various geometries helped the authors understand what was actually happening at the interface when elastomer was added or subtracted from that particular edge. Although the added 0.02 inch of elastomer was desired on the edge because of better adhesion, it turned out that if too much elastomer was added, the new section actually became stiffer, and resulted in more stress being transferred to the aluminum at the edge. Proof of this is seen from Case D, which represents a gross amount of neoprene added to the edge. The total interface stress of the current design (Case A) was more than doubled that stress shown for Case D. So a designer should be careful not to add too much stiffness to the edge where the adhesion is suspected to be weakest.

An additional conclusion is to not subtract away too much neoprene from the edge or the volume of the neoprene pad would be reduced. This reduction of volume would affect the spring stiffness of the roller guide, which is not acceptable. ANSYS proved to be a valuable resource in monitoring how much neoprene could be added or subtracted such that stresses were reduced in the critical area.

Of the several geometries analyzed, Case E showed the lowest stress in the critical area of the interface. A 36% reduction in stress is found by sufficiently rounding the corner with elastomer, yet not removing enough to affect the spring-stiffness of the roller guide. This geometry adds the right amount of elastomer to aid in the adhesion to the aluminum, and it lowers the adhesion stress. The authors are currently modeling an ANSYS analysis to verify that the change indeed does not affect the spring-stiffness of the roller guide.

Additional research beyond this study would include exploring other mechanisms that affect part performance. An understanding of the heat generation due to hysteretic damping during cyclic loading could lead to further refinement of the elastomer contour for improved durability. Also, a thorough understanding of how the aluminum metal components heat up during the bonding process could lead to refinement of the mold for improved bonding.

References

[1] Alan A.Gent, Engineering with Rubber, Hanser Publishing, Oxford University Press, NY, 1992, p. 240

[2] ANSYS Theory Reference Manual, Structures with Material Nonlinearities 4.5, Hyperelastcity, ANSYS 7.1 Documentation, 2003.

[3] ANSYS Element Reference, General Element Features, 2.5 Implicit Analysis, Section 2.5.2.2, Mooney-Rivlin Hyperelastic Material Constants, ANSYS 7.1 Documentation, 2003.