STUDY OF THE STRESS AND DEFORMATION IN ACME NUT THREADS

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Background

Developing an understanding of the mechanisms responsible for the critical wear exhibited by the Alaska Airlines jackscrew requires an understanding of the loading and thread meshing under operating conditions. In the current study, we utilize finite element analysis methods to study the stress and contact pattern in the jackscrew assembly as a function of wear and operating temperature. The issues to be addressed in the current study are:

1. Is the load transfer from the screw to the nut uniform across all threads?
2. Does the temperature drop at altitude cause additional freeplay between threads of the screw and nut?
3. Is the wear pattern of an individual thread uniform?
4. As thread wear progresses, is there a point at which the process becomes unstable and switches from a wear process to a structural failure?

The investigation will utilize a combination of two dimensional, axisymmetric and three dimensional analyses to address the issues raised in the four questions posed. The objective is not to perform an exact simulation of the wear-out process for the acme nut, but to define the mechanisms of load transfer from the screw to the nut at various stages of nut thread wear. Questions 1 through 3 address issues critical to interpreting experimental data on the wear rate of the acme nut material. Questions 3 and 4 provide information needed in interpreting the behavior of the jackscrew just prior to failure and the formation of the thread remnant found at the accident scene.

Modeling and Analysis Procedure

The jackscrew mechanism consists of a nut eight inches long fabricated from C95500 aluminum bronze and a screw made from a hardened 4140 steel. The nut has 32 threads divided into two separate alternating helices so that the pitch for an individual thread is 0.50 inches. The baseline dimensions of the screw and nut are shown in Figure 1. The details of the torque tube/screw geometry have been neglected since they do not influence the contact condition between the nut and screw. This geometry was used to define the meshes and contact surfaces for the finite element analyses (FEA). For the axisymmetric models, the basic geometry was duplicated thirty two times to replicate the actual length of the nut and the portion of the screw that was in contact with the nut. The analyses were performed using the geometry as shown, and also with the clearance between the crest of the screw and the root of the nut reduced to zero.

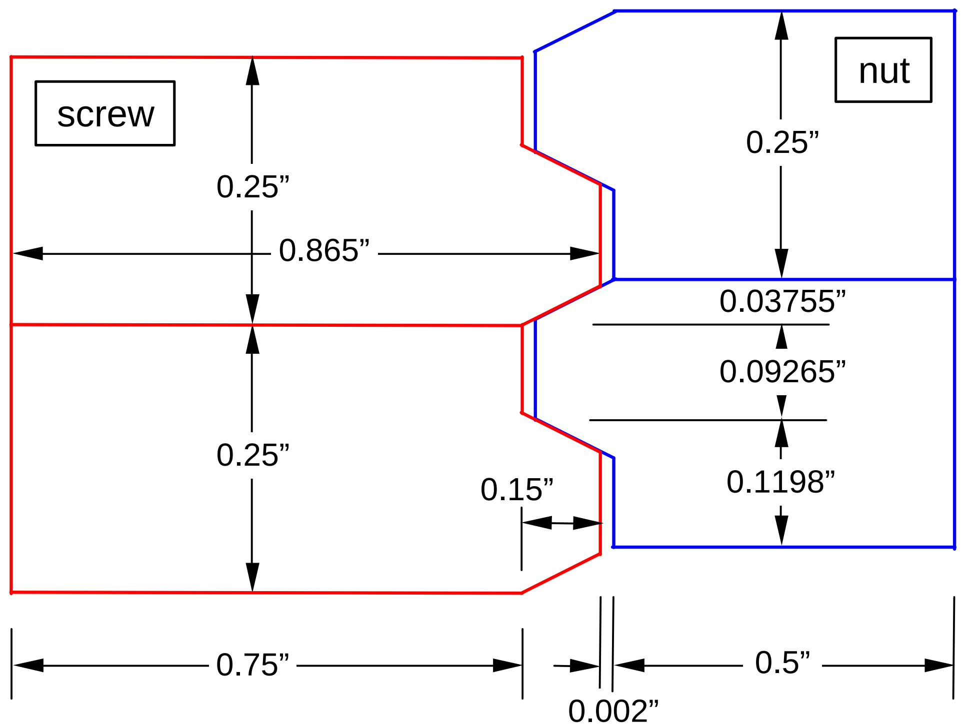

Figure 1. Basic Geometry of Screw and Nut Treads

For the axisymmetric analyses, which were performed primarily to study the mechanism of axial load transfer between the screw and the nut and the mesh required to accurately predict the contact pressure distribution at the screw-nut contact surfaces, the results were not sensitive to this spacing. The details of the meshes used will be discussed in the sections dealing with each stage of the study. In addition to geometry, the material properties used are common to all analyses. Table 1 lists the basic thermo-mechanical properties used for all analyses. Because detailed data was not available for the behavior in the plastic regime, the analyses assumed a small amount of strain hardening for the both materials. As will be discussed later, this proved to not influence the results because all analyses stayed in the elastic range. The analyses were all run using the ABAQUS[1] code utilizing finite deformation theory in order to accurately model the contact behavior.

Material	**Elastic Modulus, E,** GPa (msi)	**Yield Strength,** s_y, MPa (ksi)	**Poisson’s Ratio,** n	**Coeff. of Thermal Exp.,**/C° (/F°)
Alum. Bronze	110 (15.9)	413 (60)	0.34	9×10^{-6} (5×10^{-6})
4140 Steel	200 (29)	1,100 (160)	0.29	7×10^{-6} (3.9×10^{-6})

Table 1. Thermo-mechanical Properties Used in Finite Element Analyses

[1] Hibbitt, Karlsson & Sorensen, Inc., *ABAQUS/Standard User’s Manual, Version 6.2*, Pawtucket, RI; 2001

Load Transfer Between Screw and Nut

In order to define the level of the contact pressure between the screw and nut, an analysis of the entire length of the nut must be performed. It was estimated that an accurate three dimensional model of all thirty two teeth would require more than one million degrees of freedom (DOF) in the FEA. An analysis of this size is outside of the scope of the present study. It was decided that a two dimensional analysis would properly account for distribution of the axial load transfer between the screw and nut. To justify this, one can consider the simplified design equations available for jackscrews. The relationship between the axial load, *P*, transmitted per thread and contact force, *N*, normal to the thread surface is given by[2],

$$N = \frac{P}{\left(\cos l - m \sin l\right)} \tag{1}$$

where l is the pitch angle and m is the coefficient of friction. The term $m \sin l$ in the denominator represents the contribution of the frictional shear stress to vertical equilibrium. In an axisymmetric analysis, the pitch angle is zero and the torque and thrust are therefore uncoupled. However, ignoring the frictional contribution maximizes the contact pressure and thus provides a good upper bound on the load transfer to the teeth as a function of axial position.

An additional consideration in modeling the acme nut is how to best approximate the boundary conditions restraining the nut in the gimbal supports. Figure 2 shows a picture of the nut in its gimbal support attached to the airframe of the accident aircraft. It is apparent from the figure that both the geometry and support arrangement are geometrically irregular and very rigid. The rigidity of the attachments should allow the major influences of the attachment to be captured without the necessity of

Figure 2. Acme Nut Installed in Accident Aircraft

[2] Hamrock, Jacobson & Schmid, *Fundamentals of Machine Elements*, McGraw-Hill, New York, NY; 1999

modeling the local attachment geometry in detail. For the first series of axisymmetric models which are designed to study the nature of the load transfer from the screw to the nut, a series of boundary conditions constraining motion of the nut were used to determine if there was strong sensitivity of the loading pattern to the nut boundary conditions. The mesh used for these analyses is shown in Figure 3. The mesh has a total of 19,220 four-noded isoparametric elements, 20,334 nodes and 41,890 DOF. As can be seen from the figure, the majority of the elements are concentrated in the teeth in order to obtain an accurate modeling of the contact. A 35,586 N (8,000 lbs) axial load was applied to the top of the screw at the centerline to maximize the ability of the load to diffuse down the threads as far as possible. No displacement boundary conditions were applied to the screw so that contact pressure and friction are the only means for equilibrating it under the applied loads.

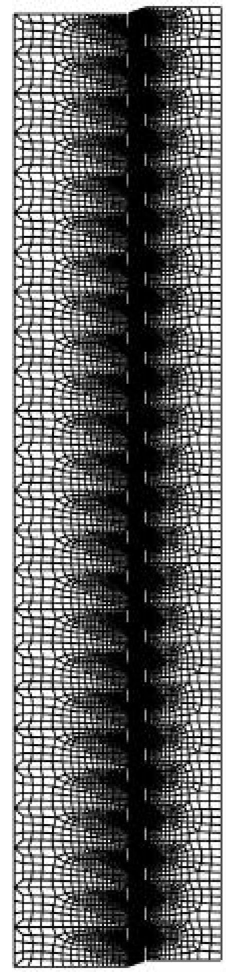

Figure 3. Finite Element Mesh Used for Axisymmetric Analyses

Figure 4 shows the distribution of axial stress, σ_{22} in the screw and nut for three different boundary conditions. In (I) the entire outside surface of the nut is constrained against axial movement. In (II) axial displacement is restrained at pairs of nodes located one third and two thirds along the length and in (III) the bottom surface of the nut is restrained from axial movement. Case (I) leads to the stiffest configuration for the nut and case (III) the most flexible. The actual condition, ignoring the circumferential effect, is most likely somewhere in between cases (I) and (II). At any axial station along the screw, the net axial load being transferred is the integral of the axial stress over the screw cross-section. Thus, if the stress is constant across a cross section, the net axial force being transmitted through the screw at that station is given by the axial stress multiplied by the cross-sectional area. In case (I), the axial stress in the screw is down to almost zero (between 3.6 MPa (520 psi) and 11 MPa (1,595 psi)) at the 5th thread. This corresponds to a total load transfer below the fifth thread of approximately one eighth of the original load. From this we can conclude that the first four to five threads of the acme nut are carrying almost all of the thrust transmitted through the jackscrew. Case (II)

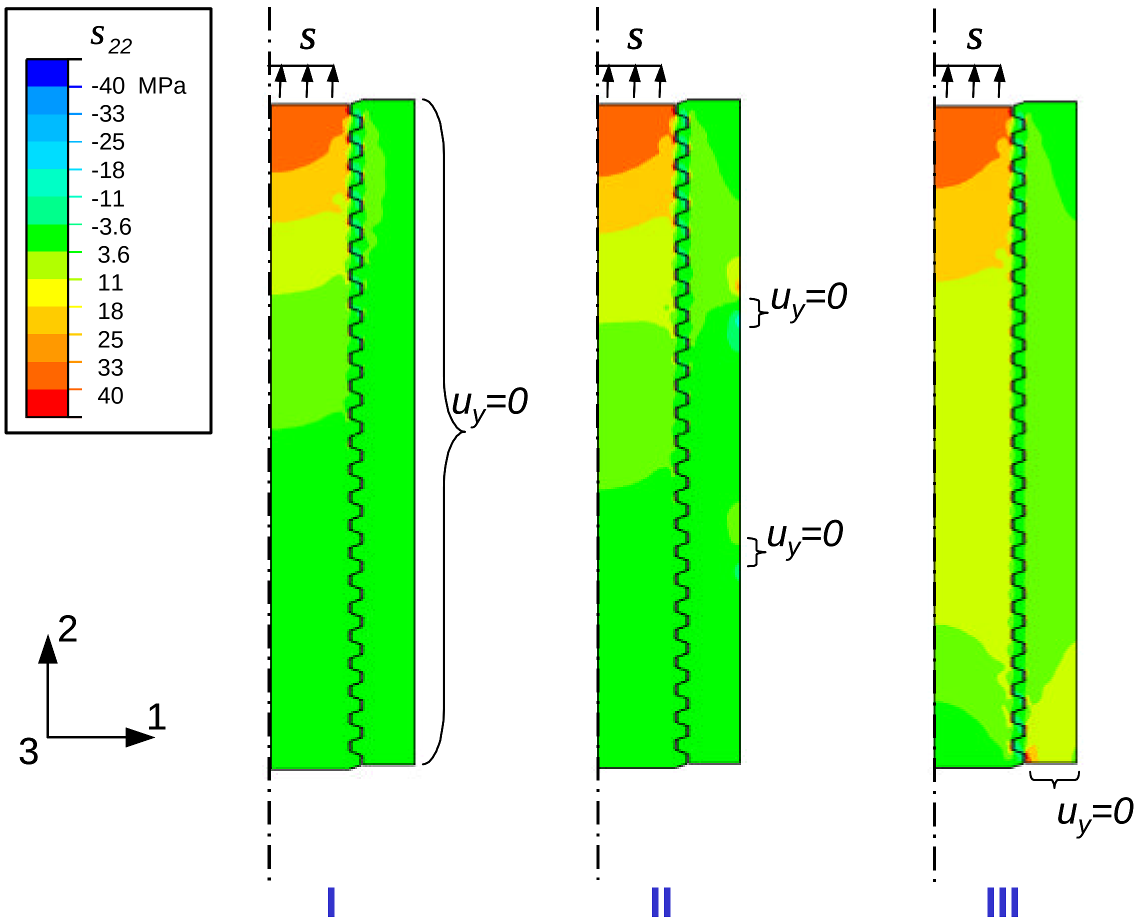

Figure 4. Axial Stress Distribution in Axisymmetric Model with Three Different Axial Constraints

shows an almost identical behavior to case (I). For the most flexible possible arrangement, case (III), we only see small changes from the other two cases. The stress increase seen at the bottom of the screw in case (III) is a local effect associated with the boundary condition and does not represent load transfer from the screw to the nut. For all analyses aimed at looking at the stress state caused by sliding of the screw across the nut threads, we have therefore assumed that the entire applied axial load is transmitted through four thread revolutions of the acme nut.

Stresses in Screw and Nut Due to Temperature Drop

For the second issue, which was to be addressed using an axisymmetric, the full-length model was utilized to study the influence of a temperature drop at altitude. Because of possible variations in pitch due to fabrication tolerances, it was desired to demonstrate whether a temperature drop would worsen or improve the tooth engagement. Because the coefficient of thermal expansion of the aluminum bronze is approximately 28% larger than the steel, it was thought that the space between the teeth on the nut would shrink around the screw thread and increase the contact. As would be assumed, the central part of the screw is in compression and the central part of the nut is in tension of an equal magnitude. Because the stresses in both the screw and nut are in the elastic range and the deformations in both bodies are small, the results of the axial force loading combined with the with a temperature drop of –100C° (–180 F°) can be obtained by superposition of the two solutions. For a static condition with no sliding between the surfaces, the axial load transfer would be unaffected by the temperature change. Under sliding conditions, the increased contact pressure caused by the temperature drop would lead to an increase in the frictional contact pressure which would increase the load transfer in lightly loaded threads according to equation (1). Because the total load transferred from the screw to the nut is fixed, this effect would reduce the magnitude of the maximum axial force carried by any individual thread revolution. Thus, the temperature effect will not be considered in defining the loading on threads for a detailed wear/deformation analysis.

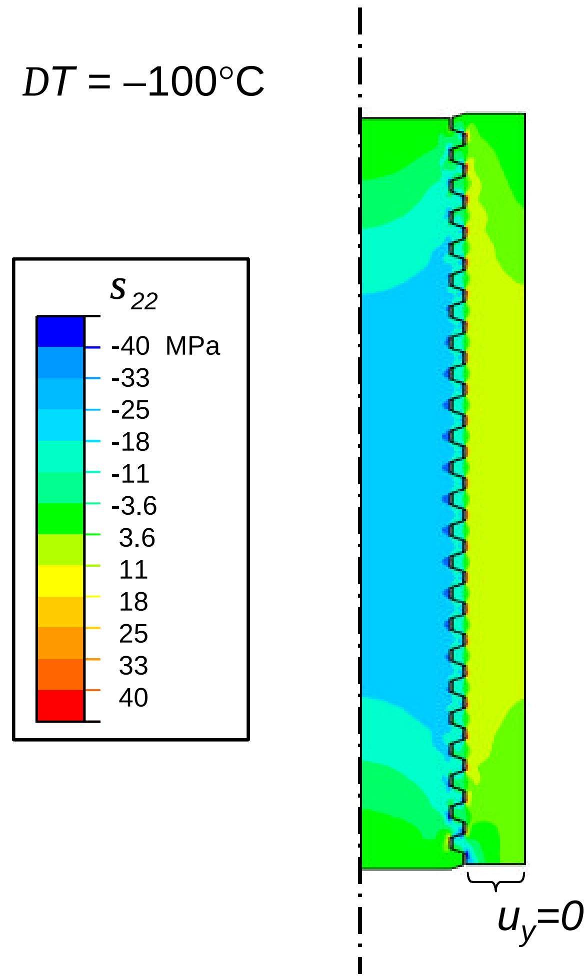

Figure 5. Axial Stress Distribution in Axisymmetric Model Due to –100C° Temperature Change.

Finite Element Model and Analysis Development

Because of the *helical nature* of the threads in a screw-nut assembly, a two dimensional analysis cannot properly compute the local stresses due to contact under load with friction. This was explained by equation (1) which illustrates how a finite pitch angle l together with the law of friction leads to a coupling between the normal and tangential forces. However, a three dimensional analysis of an entire jackscrew/acme nut assembly would involve prohibitive computing resources and time. As was previously discussed, the objective of this study is not to perform an exact simulation of the wear-out process for the entire acme nut, but to define the mechanisms of load transfer from the screw to the nut at various stages of nut thread wear. For this, it is only necessary to capture the nature and mechanisms of the load transfer from the screw to the nut. To reduce the problem to one of manageable size, several assumptions were made. In an axisymmetric problem, a three dimensional problem is reduced to a two dimensional one by the assumption of no displacement in the circumferential direction. Helical threads do not exhibit an axisymmetric geometry, however, away from the ends of the thread, the local pattern of deformation (exhibited by the strain field) should be independent of position along the helical generator unless as with the acme nut being studied the boundary conditions are not axisymmetric. Referring to Figure 2, it can be seen that the outside perimeter of the acme nut is constrained from displacing in all three directions by thick sections located 180° apart. These sections fit into a groove in the heavy gimbal mechanism. As a first assumption, it will be assumed that the entire circumferential periphery is restrained. This provides a mechanism for developing a modeling strategy which can greatly reduce the size of the three dimensional model required. A model for one 360° segment of the nut and screw is shown in Figure 6. The finite element mesh was generated by taking the two dimensional mesh section from the axisymmetric model and repeating it at fixed circumferential stations with the axial (x_3) coordinate increased at a fixed multiplier given by the angle multiplied by the pitch divided by $2p$.

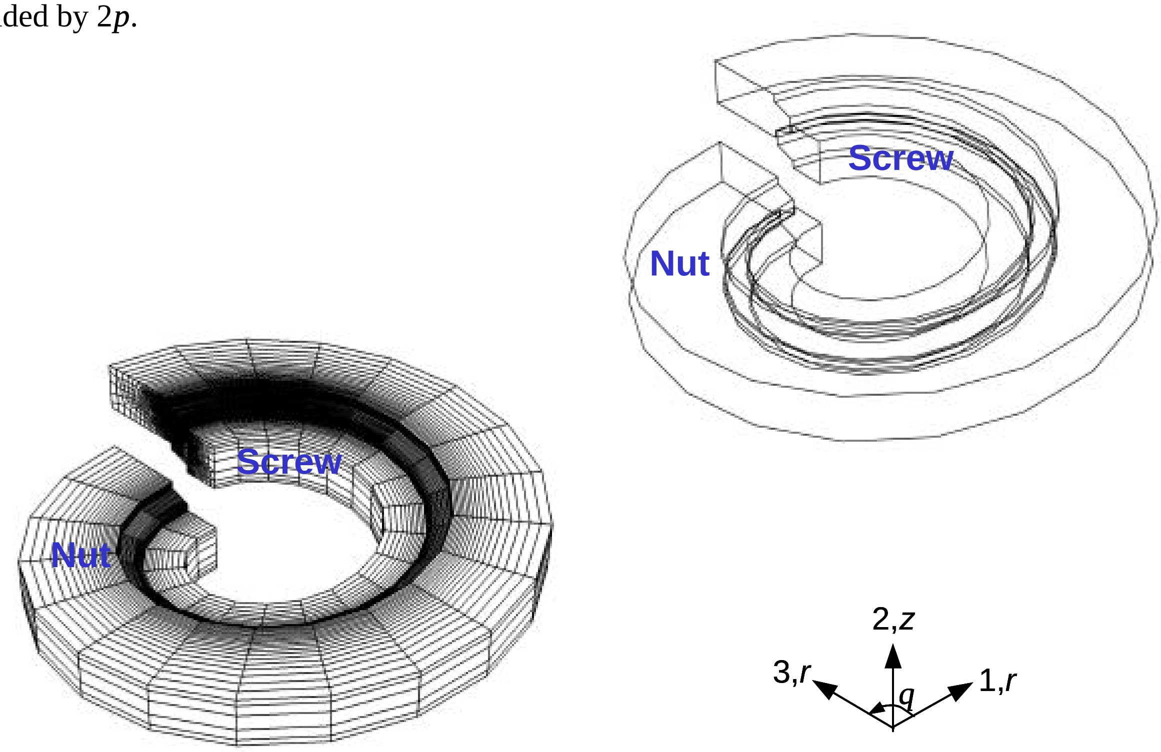

Figure 6. Three Dimensional Finite Element Mesh for 360° Section of Jackscrew/Acme Nut Assembly

The mesh for this model had a total of 34,880 eight-noded 3D elements with 39,674 nodes and 120,366 degrees of freedom (DOF). The large number of contact elements in these computations makes the computational effort for each analysis very time consuming. This model was analyzed with a series of boundary conditions on the nut in order to evaluate the influence of the localized constraints along the outer periphery.

Figure 7 shows contours of the von Mises stress when the vertical displacement component is constrained to zero along the outer circumference of the model. Additionally, the radial and circumferential displacements a constrained to the same value for all nodes in the nut on a plane of constant circumferential angle *q*. The axisymmetric analyses indicated that four to five thread revolutions carried about 90% of the jackscrew load. Under normal operating conditions, the maximum

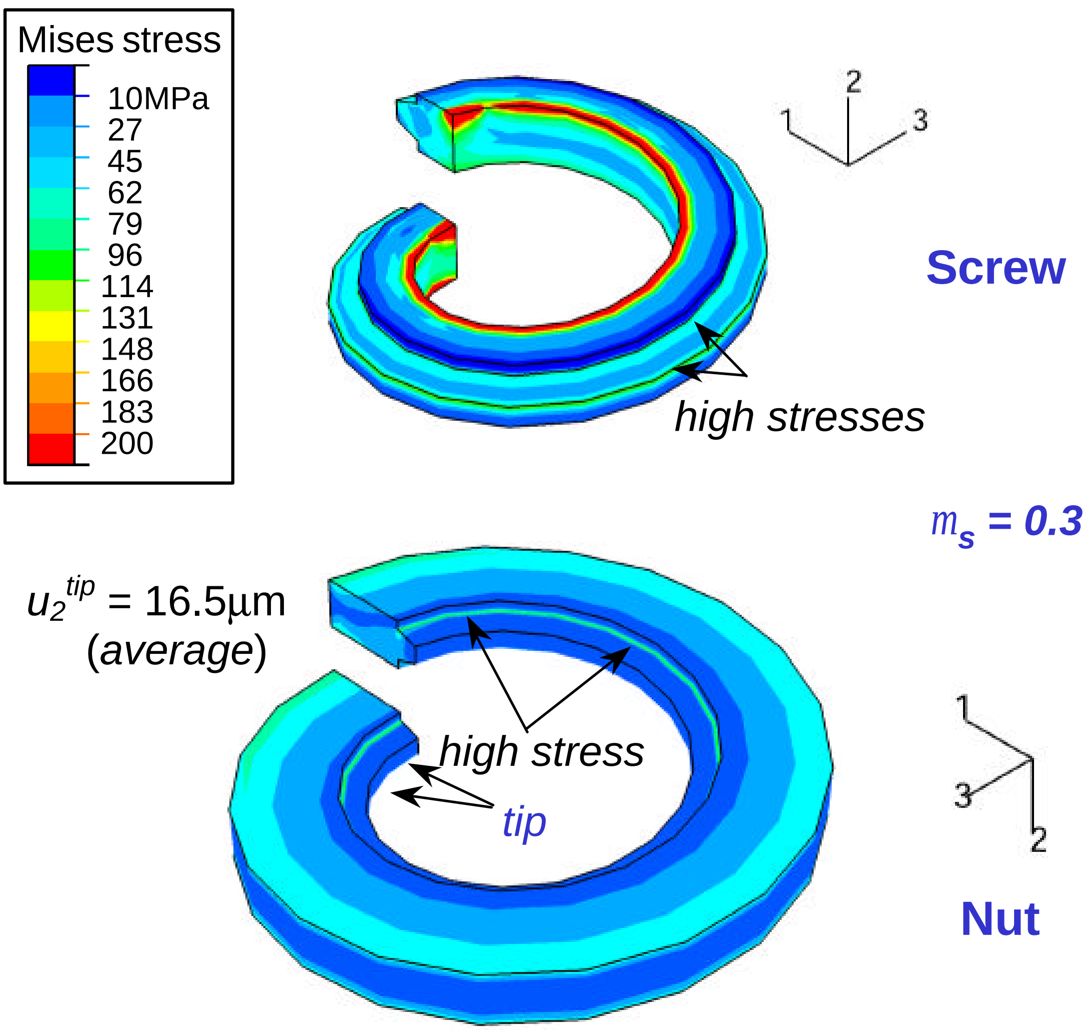

Figure 7. Stress Distribution in Single Thread Model with 360° Boundary Constraints

jackscrew load is approximately 35,584 N (8,000 pounds). Based on this, the loading in all cases is set as 8,896 N (2,000 pounds) per thread revolution. The stresses in the acme nut are very low. Peak stresses in the acme nut are below 40% of the yield stress of the aluminum brass material. The majority of the contact region between the screw and nut are at very low stress levels. The stress distribution is seen to be independent of the circumferential position q except for small regions near the 0° and 360° edges of the model. This validates that the boundary conditions and constraints validate the quasi-axisymmetric, generalized plane strain type of stress state hypothesized.

The results just discussed ignored the non-axisymmetric nature of the constraints on the acme nut. To evaluate the influence of the localized fixation on the contact stresses in the acme nut, a series of 360° analyses were run with boundary conditions which were intended to bracket the actual conditions. To evaluate the influence of boundary conditions on the level of stress, the 360° model was analyzed using a selection of boundary conditions which should bracket the actual conditions. Figure 8 shows the results for five different boundary conditions. Here cases were examined with the entire outer boundary constrained and with nodes at 180° restrained and the remainder free. In addition, the upper and lower surfaces of the nut were modeled both as stress free and with the plane-strain like conditions imposed by constraining points along radial lines to have the same displacement. Figure 8 shows a comparison of the von Mises stress in the nut for these combinations. The free surface case is seen to give a higher stress. This is due to its increased flexibility. This boundary condition is more representative of threads at the ends of the nut and will be used for the remaining analyses. Referring to Figure 2, it is seen that the reinforcement and constraint on the outer periphery of the nut is complex and covers a significant portion of the surface. For this reason, the model with complete restraint is considered to be more representative of the actual conditions. In the thread region of the nut, the overall stress state is quite similar for both cases. The local constraint does show a higher peak stress at the acme nut thread root, but this occurs over a very small region and the stresses have very close magnitudes for the majority of the thread surface. As was discussed earlier, the boundary conditions at 0° and 360° give a periodic like solution for repeating threads.

For the case of complete constraint on the outer circumference that was shown to give a solution that was independent of q except in small regions near the edges at 0° and 360°. This result implies that the q independent solution should be obtained from a reduced model covering a small angular sector with a single element row and the boundary constraints used in the 360° model. Figure 9 compares the von Mises stress for the 360° model and a 1° slice of elements. The stresses are seen to be essentially

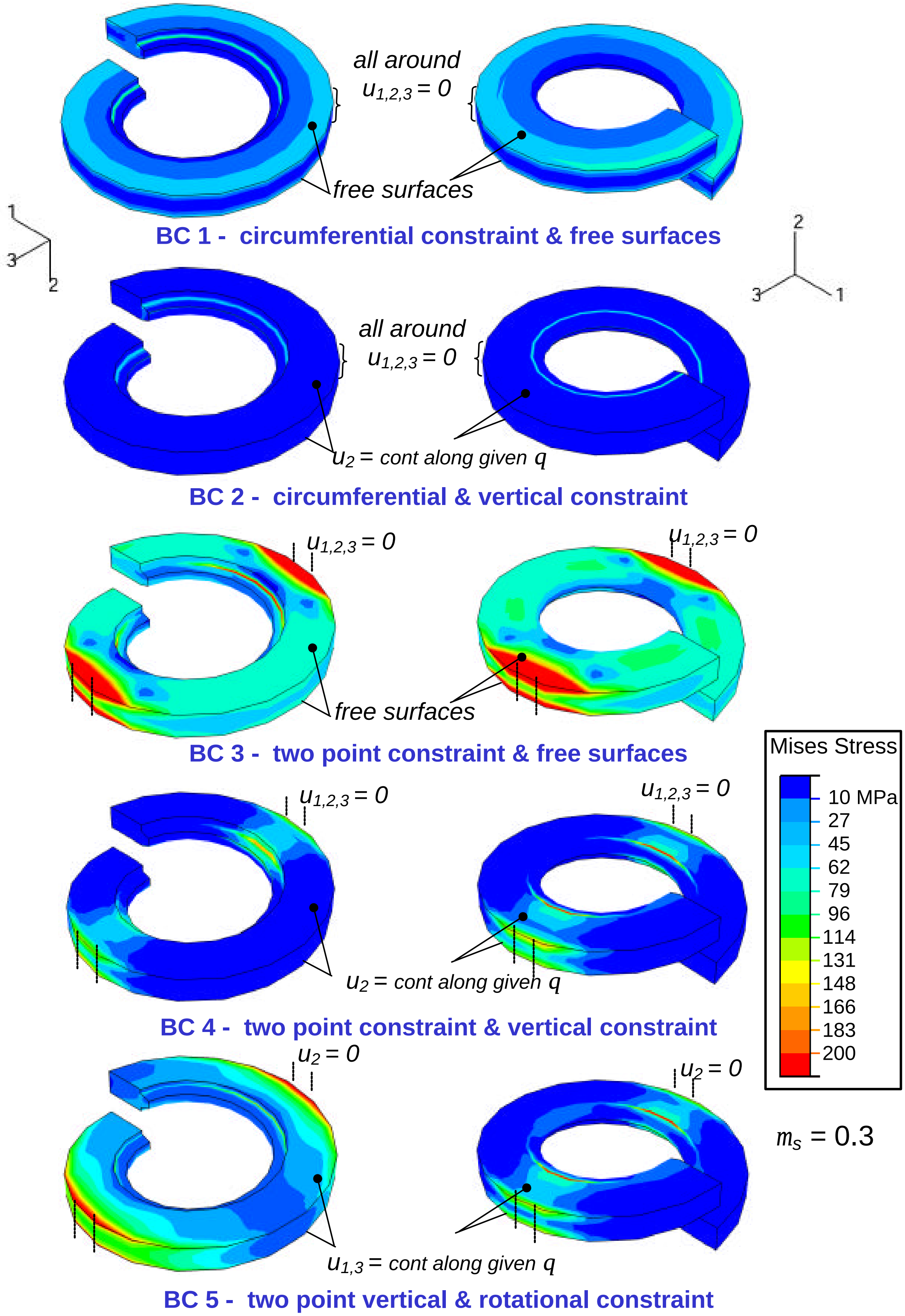

Figure 8. Influence of Boundary Conditions on Stress Distribution in Acme Nut.

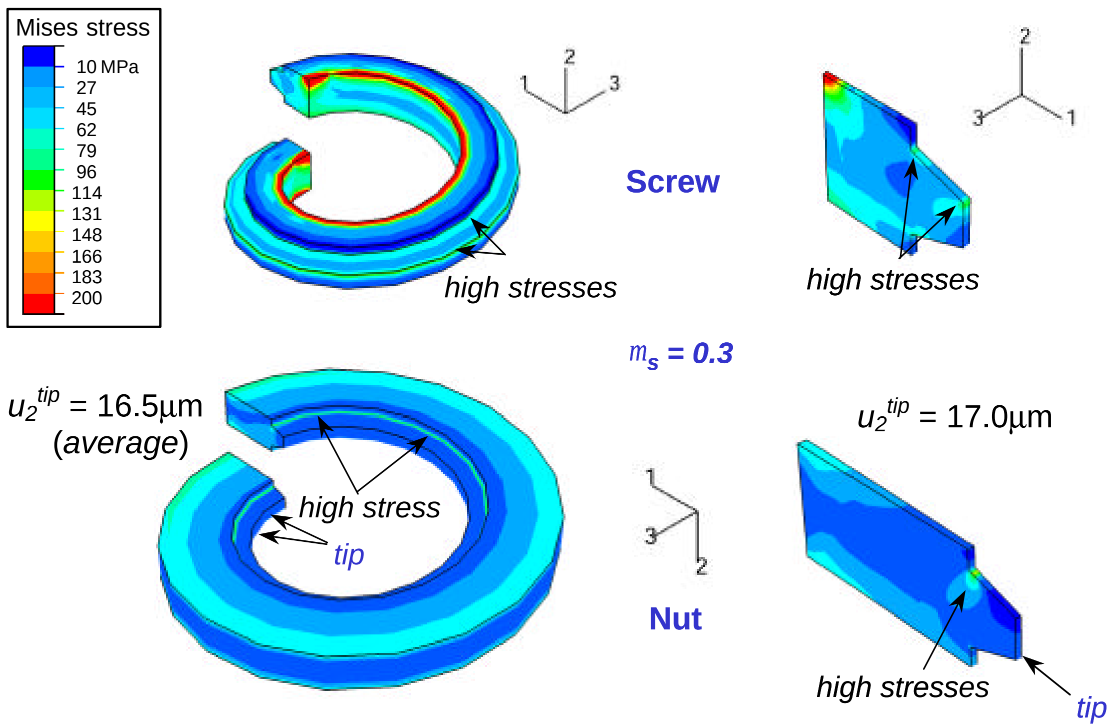

Figure 9. Comparison of 360° Model and 1° Model

identical except near the edges of the 360° model. In addition, the average tip vertical deflections of 16.5 µm (0.00065 in) for the 360° model and 17 µm (0.00067 in) for the 1° model agree to within 2%.

Three Dimensional Analysis of Stress and Deformation

Based on the results of the previous section, the 1° model was chosen to use to study the details of the contact stress pattern and the influence of acme nut thread wear on the level and pattern of stress and deformation in the acme nut. Figure 10 shows the von Mises stresses for the base case of original geometry and the loading previously defined. Looking at the contact surface of the acme nut, it is observed that the majority of the nut has a stress less than 50 MPa (7,250 psi) except for a very small region near the thread root. Since the wear rate for material removal is proportional to the contact pressure and shear stress, these are shown in Figure 11. The axial stress s_{zz} and the shear stress s_{rz} are seen to be small in comparison to the von Mises stress. This indicates that the dominant deformation mode in the thread is a beam bending type behavior which predominantly develops a s_{rr} stress in the threads. The magnitude of all the stress components is below 30 MPa, (4,350 psi) less than 10% of the yield stress of the material. At these stress levels the wear rate for the acme nut/jackscrew material combination is well within normal design ranges for properly lubricated materials. Significantly, the

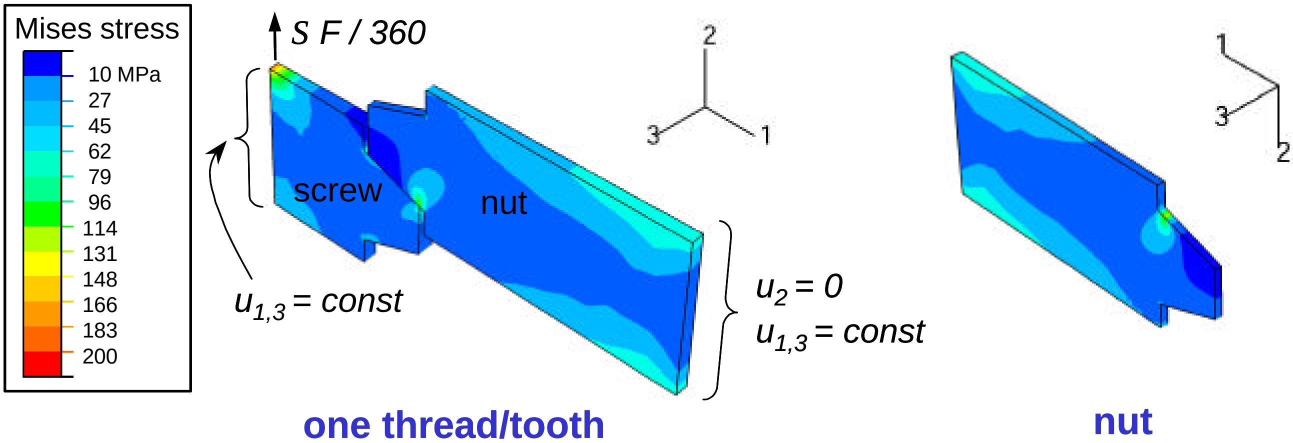

Figure 10. Von Mises Stress for Acme Nut with Zero Wear

majority of the surface is at a negligible stress level. Only a small region near the thread root sees normal stresses near 30 MPa (4,350 psi) and shear stresses in the 10 MPa (1,450 psi) range.

The stress state on the acme nut surface, due to contact with the jackscrew, implies that the acme nut surface would not wear in a uniform manner. The material with the highest contact stresses would wear at a more rapid rate than the remainder of the surface. This implies a complex wear behavior for the acme nut thread surfaces. At any instant, the region of the thread surface with the highest stress would wear more rapidly than adjacent material, creating an instantaneous contact pressure relief. The adjacent material would then see an increase in contact pressure and wear away, causing the process to continue across the thread surface. Thus the wear can be simply envisioned as a continuous process of removing material from the thread surface beginning at the root and moving to the tip and then beginning again at the root. The actual process is complicated by the local deformation of the acme nut surface due to contact pressure and the overall thread deformation pattern due to bending of the threads under load. To examine the influence of this wear mode on the stress in the jackscrew, the element located at the highest stress point was removed and the stress state in the nut was reexamined. While no attempt was made to model the actual wear process, because a fully nonlinear deformation analysis was performed, this analysis provides an accurate discrete material removal evaluation of the proposed continuous process. The maximum axial stress for this case is less than 60 MPa (8,700 psi) and the maximum shear stress was approximately 10 MPa (1,450 psi). The elements in the wear region of the acme nut thread were all of a uniform size, 0.15mm (0.0059 in) wide by 0.14mm (0.0055 in) deep. The contact stresses were found to stay very low as elements were successively removed.

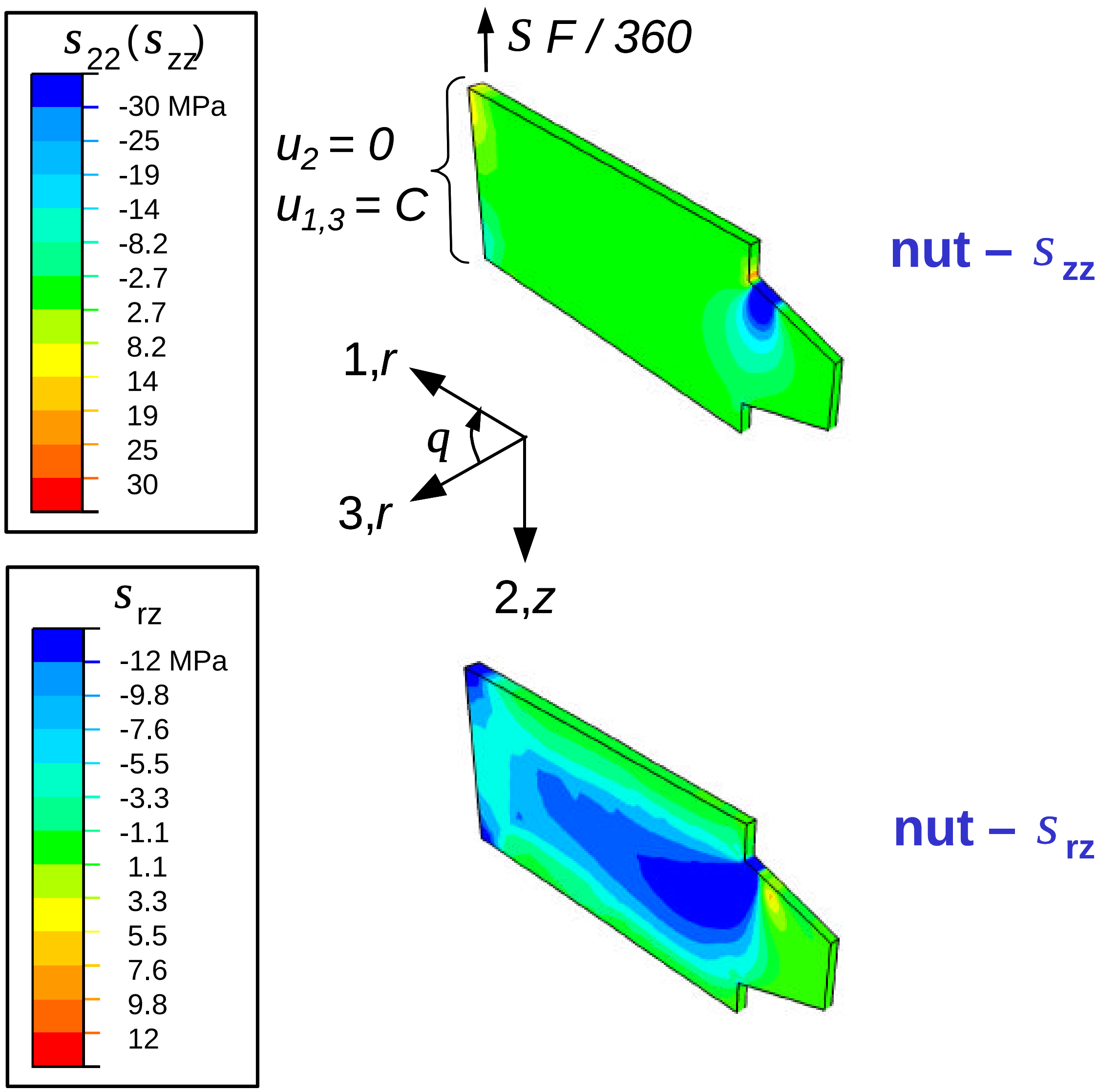

Figure 11. Axial and Shear Stress on Acme Nut Contact Surface

Figure 12 shows the radial direction shear stress in the acme nut as wear is simulated by removal of elements. The first figure is the original profile. An extremely small region near the edge of the screw-nut contact shows a shear stress concentration with a peak stress of –70 MPa (10,150 psi). These numbers are larger than what would be predicted for a continuous removal process where the step change in surface profile would be much less than 0.14mm (0.0055 in). The second figure shows the shear stress when half of the top layer of elements, 0.14 mm (0.0055 in) was removed. The magnitude of the peak shear stress has not changed but the size of the zone with stresses near the peak has grown. The last two figures are for the case when 1.68 mm (0.066 in) has been removed and when half of the surface is down to 1.82 mm (0.072 in) removal. Again we see no change in the stress levels and

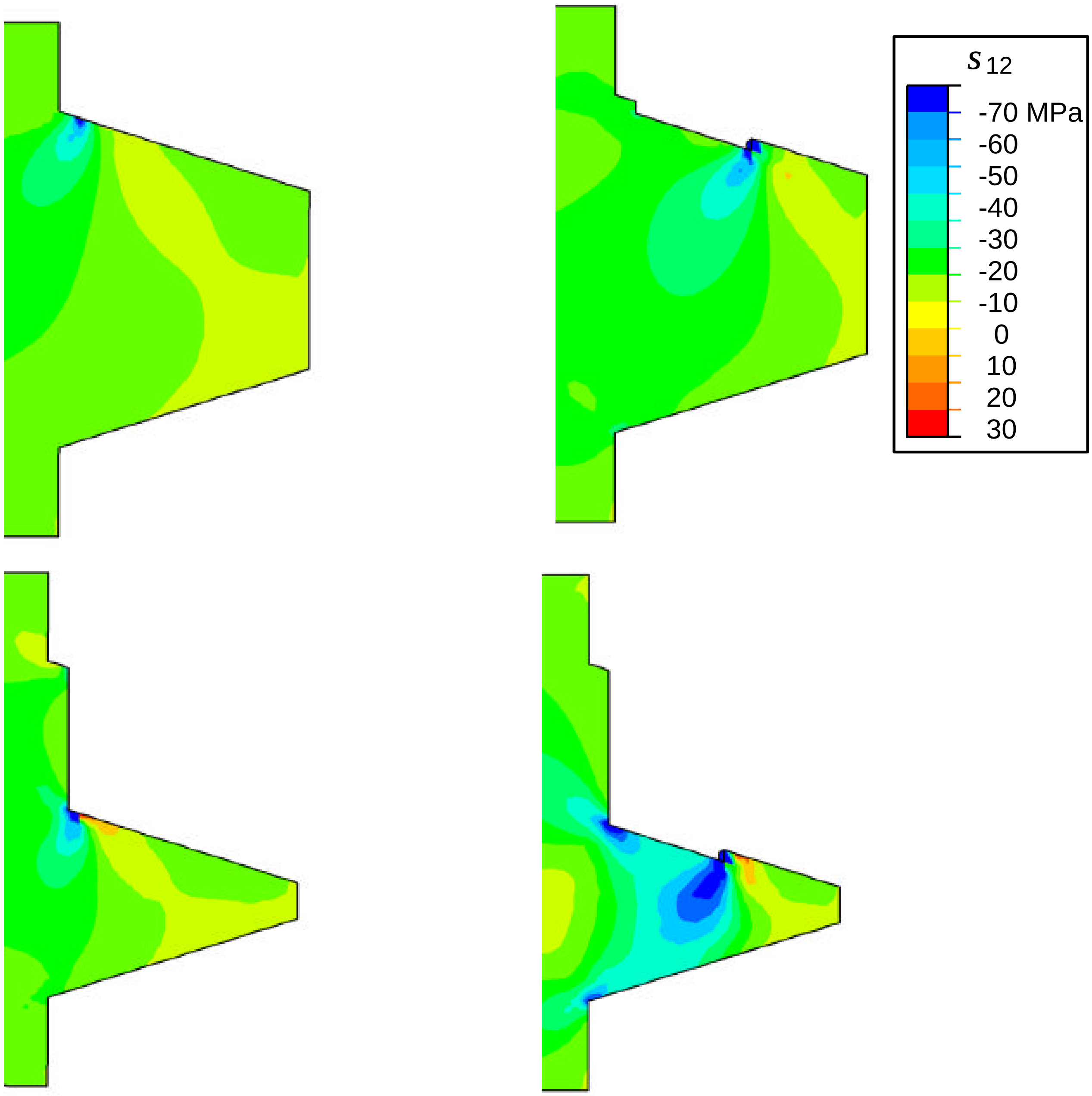

Figure 12. Shear Stress Contours in Nut as Material Wear in Simulated by Element Removal

patterns, just an increase in the size of the zones of highest stress. An exception to this last conclusion is in the shear stress pattern for the 1.68 mm (0.066 in) wear cases. While the peak magnitude of the shear stress remains at approximately –70 MPa (10,150 psi), a small region of opposite direction shear stress of with an approximate magnitude of +30 MPa (4,350 psi) magnitude is observed on the contact surface adjacent to the peak area. This will be discussed further in the next section. Figure 13 shows the results for the axial stress for the same cases of wear as in Figure 12. The stresses are again seen to

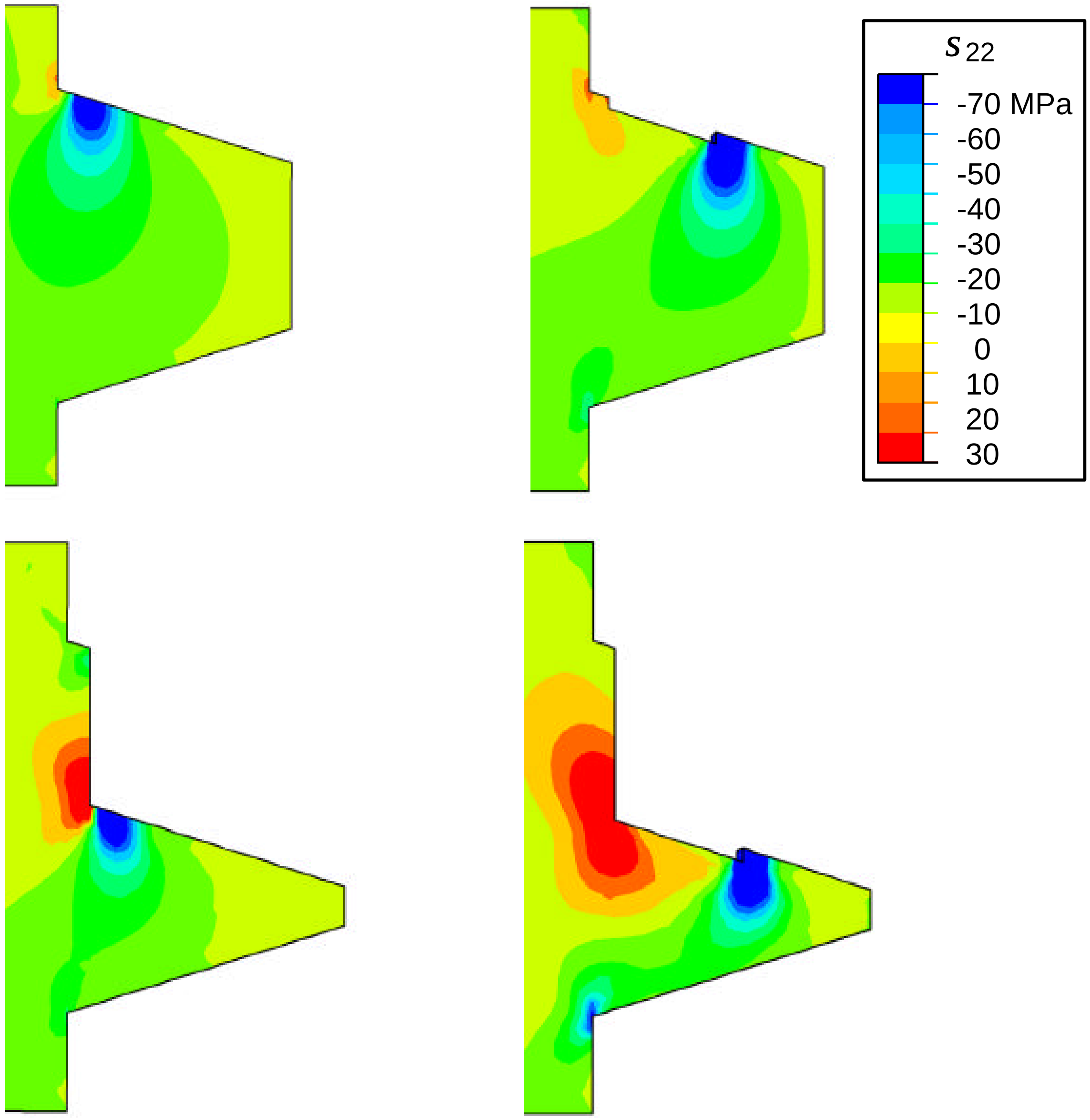

Figure 13. Axial Stress Contours in Nut as Material Wear in Simulated by Element Removal

remain quite low and only the size of the higher stress region changes not the magnitude, except near the root where the increased bending causes higher tensile stress. These cases show that as the acme nut wears to levels way in excess of its design, the stresses under load remain quite low so that one would not expect any significant change in the nature of the wear process.

Acme Nut Thread Deformation As Wear Progresses

In the previous section it was shown that the acme nut was capable of sustaining operating loads without developing stresses above 50% of its yield stress at wear levels exceeding the maximum

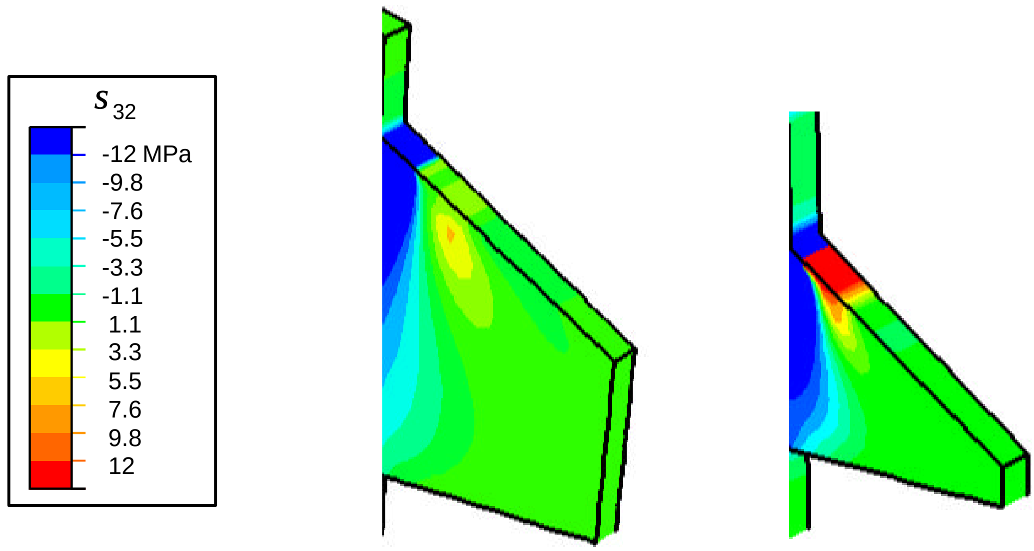

Figure 14. Circumferential Shear Stress Distribution for Case of Zero Wear and 2.10 mm (0.0825 inches) Wear.

allowed in its design. One mechanism for developing a nonstandard wear pattern and accelerating the instantaneous wear rate on the acme nut threads is the incompatible screw-nut deformation implied by the change in sign for the shear stress discussed in the previous section for the worn nut thread. This characteristic of the shear stress distribution is not seen at lower levels of thread wear. In Figure 14, an expanded view of the circumferential shear stress distribution on the acme nut thread contact surface is shown of the case of zero thread wear and 2.10 mm (0.0825 inches) thread wear. A region of reversed shear stress direction means that the surface is attempting to slide opposite to the direction implied by the axial loading of the screw. The nature of this deformation can be understood from an examination of the thread deformation modes implied from the stress field. The acme nut threads are undergoing a bending deformation as indicated by the sharp gradient in the transverse shear stress, (s_{rz}), in the vicinity of the high contact pressure region. The net vertical load acts in a direction which is not aligned with the natural helical axes of the thread and therefore the bending is of an unsymmetrical nature[3], leading to a cross sectional rotation proportional to the local gradient in transverse shear stress. This leads to a rotation which acts counter to the natural sliding motion of the screw relative to the nut thread surface. Under frictional sliding, this deformation would lead to accelerated wear in the region of opposing shear stress. Another possible failure mechanism could be excessive deformation of the thread due to wear. As the thread wears, large bending of the thread could cause interference with the sliding of the screw across the nut surface.

[3] Benham, Crawford and Armstrong; *Mechanics of Engineering Materials;* Longman Group, Essex, England; 1996

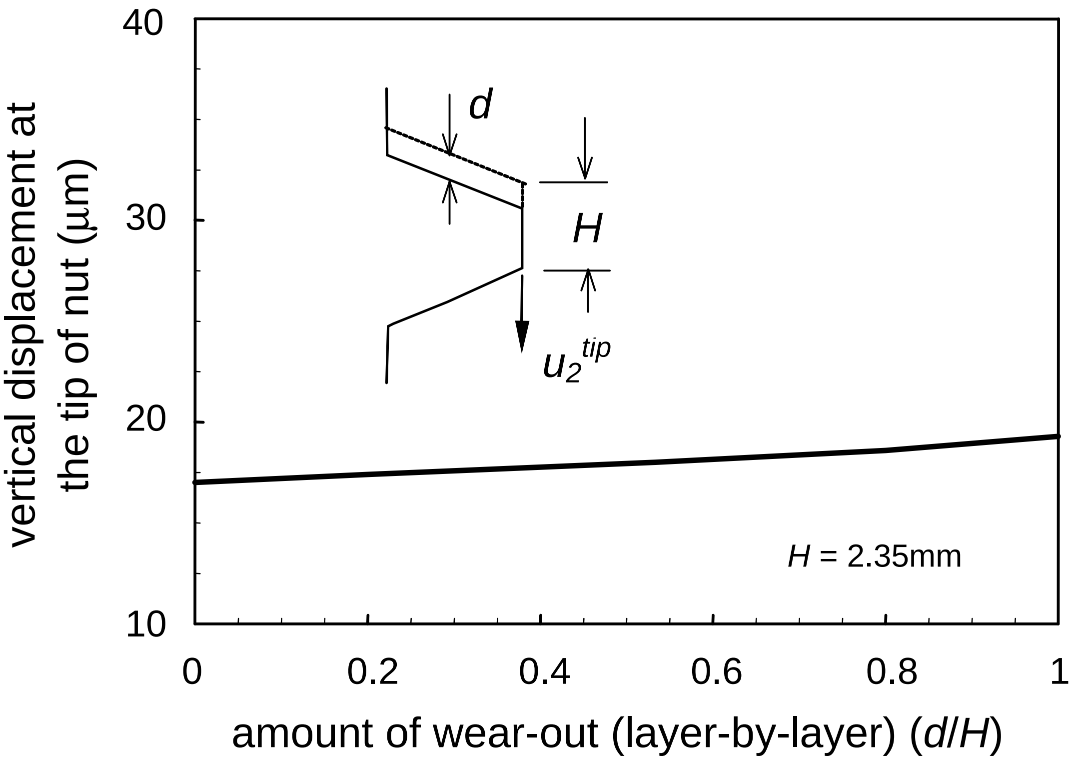

Figure 15. Change in Vertical Displacement of Acme Nut Thread as Layers are Removed.
In All Cases, Contact Occurs Near the Root.

In Figure 15, the maximum axial displacement of the acme nut thread is plotted versus the fraction of the thread face removed. It is seen that there is very little change in displacement, even when the thread is worn down to a point. This is easily explained by the results from the contact stresses on the nut surface. Most of the surface sees a contact stress near zero, so that the loading is predominantly near the root of the thread. Such loading produces almost zero bending moment and therefore does not deflect the thread. This conclusion is verified in Figure 16 which shows the change in tip displacement for two levels of layer wear as wear progresses along the layer surface. Level *b* in the graph refers to a removal of 1.68 mm (0.066 inches) of surface and level *a* refers to the complete removal (2.35 mm or 0.0925 inches) of the thread. For the level *b*, as the thread continues to wear from the root to the tip, the tip deflection increases by approximately 100%. For this case the tip deflection remains small, never going above 40 μm (0.00015 in). A very different behavior is seen for the behavior as wear from the root to the tip progresses for level *a*. When approximately one half of the surface has worn to the next layer, the rate of increase in deflection with wear increases significantly. When approximately 75% of the layer is removed, no equilibrium solution is possible in the finite element model. At this point the

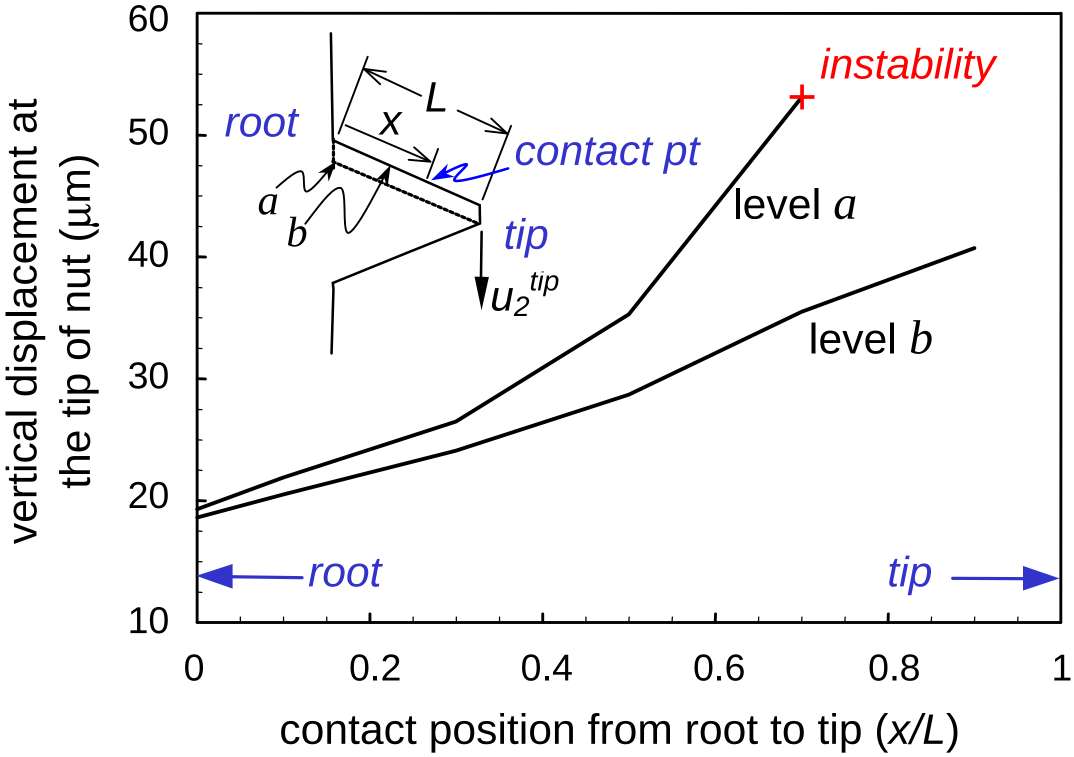

Figure 16. Change in Vertical Displacement of Acme Nut Thread as Wear Progresses Along a Layer.

deflected shape of the thread does not allow for enough contact area with the screw surface to prevent the screw from sliding unstably across the thread surface (at '+' in the figure).

Conclusions

A study has been conducted using finite element analysis methods to address the stress and contact pattern in the jackscrew assembly as a function of wear and operating temperature. The investigation utilized a combination of two dimensional, axisymmetric and three dimensional analyses. It was found that four of the thirty-two threads of the acme nut carried most of the load transmitted through the jackscrew. The stress levels in the acme nut stay very low in comparison to the yield stress of the aluminum bronze material even for wear levels greatly exceeding the maximum amount allowed for the jackscrew in service. The wear mechanism for the acme nut threads is found to be a layer by layer removal with wear across the layer progressing from the root outward. This wear mechanism is consistent with physical evidence from the inspection of worn jackscrew assemblies. Figure 17[4] shows

[4] NTSB, Materials Laboratory Factual Report No. 01-082, September 24, 2001

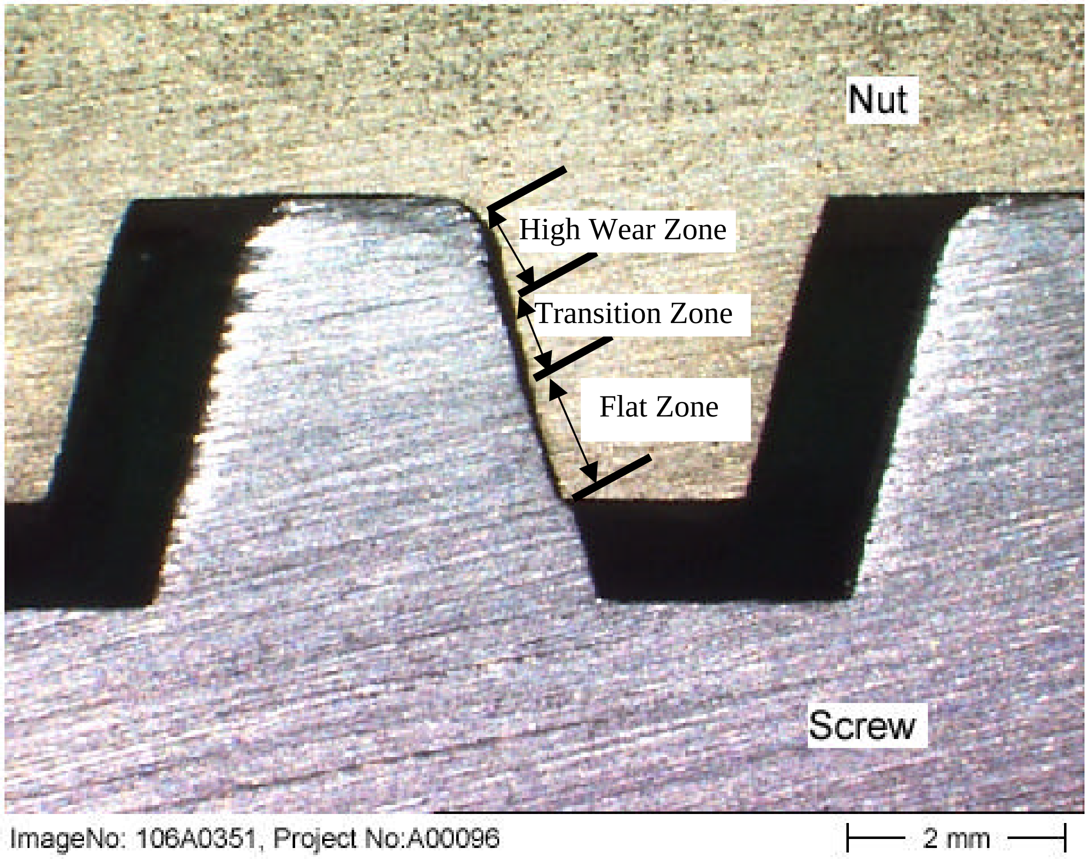

Figure 17. Cross Section of Highly Worn Acme Nut Removed from Service.

a cross section of a highly worn acme nut removed from service. If one studies the tooth flank profile, three regions can be identified. They are: near the root, a region which is worn down below the main flank profile; near the crown of the thread a flat region is observed which is essentially parallel to the screw thread flank; and an angled transition zone connects these regions. This flank profile is consistent with the wear mechanism identified by the finite element calculations. This thread has been worn down to a level associated with the flat zone thickness. The next stage of wear has begun and extends to near the boundary between the high wear zone and the transition zone.

Based on the results of the axisymmetric analyses of the complete nut length, it was determined that the four thread revolutions nearest the end of the nut would carry approximately 90% of the total axial load. As the threads on these heavily loaded acme nut thread revolutions wear down, the screw would attempt to move to maintain contact with the heavily loaded thread revolutions. This movement would be inhibited by thread revolutions which had experienced less wear. This process would result in a reduction of loaded on the worn thread revolutions and a shifting of load to other regions of the thread. The exact pattern of this process would most likely be a continuous movement of the high wear rate process zone from the end of the acme nut to adjacent threads. This process would continue until all of the threads had achieved a nearly uniform level of wear (to within the initial machining tolerances of the threads) and would then repeat itself. During this process, the load distribution through the acme nut threads could distribute itself in a manner in which more than four threads were heavily loaded relative to the remaining threads. The assumptions used in this study can thus be seen to represent an upper bound on the stresses seen locally in the jackscrew threads as a function of wear. For a given load, the contact pressure and stresses predicted are representative of the maximum that would exist in a jackscrew for a given amount of maximum wear.

In addition to the changes in the global behavior of the jackscrew assembly at large levels of acme nut thread wear, local deformation mechanisms which alter the nature of the sliding contact and implied wear patterns are observed at large wear levels. This manifests itself in two distinct mechanisms, each of which would alter the wear and failure process of the acme nut threads. At wear levels which exceed the design limit, the acme nut thread contact surface has a region near the root which deforms in a manner which would make it shear against the normal sliding motion of the screw. This would cause an area of accelerated wear near the thread root. Furthermore, under the load level used in this study, structural failure occurred not due to material failure but from an inability of the nut to sustain the screw in equilibrium at a very large level of wear. This leads to a sliding instability where the screw thread would slide along the surface of the nut thread in a shaving type of motion. This implies that at a stage when wear has exceeded the design limits for the acme nut, the nature of the contact and sliding motion between the screw and nut will change drastically. Rather than a smooth contact and sliding during jackscrew rotation, the screw will try to penetrate into the surface of the acme nut. This would lead to a different wear pattern on the acme nut threads. While the exact shape of the resulting worn acme nut thread cannot be determined from the results of this study, it is apparent that wear will not occur in a manner consistent with the undeformed acme nut surface remaining parallel to the contacting screw surface. While an exact simulation of the wear process is not technically feasible with the current level of technology, a wear pattern and bending deformation similar to that observed in the thread remnants recovered from the accident aircraft[5] is consistent with the wear mechanisms implied by the two deformation modes found to develop at high levels of wear.

[5] NTSB, Materials Laboratory Factual Report No. 00-145, November 12, 2000