THERMAL STRESS IN HIGH TEMPERATURE CYLINDRICAL FASTENERS

Max L. Blosser

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Abstract

Uninsulated structures that are fabricated from carbon or silicon-based structural materials and that are allowed to become hot during flight are attractive for the design of some components of hypersonic vehicles. They have the potential to reduce weight and increase vehicle efficiency. Because of manufacturing constraints, these structures will consist of parts which must be fastened together. The thermal expansion mismatch between conventional metal fasteners and carbon or silicon-based structural materials may make it difficult to design a structural joint which is tight over the operational temperature range without exceeding allowable stress limits. In this study, algebraic, closed-form solutions for calculating the thermal stresses resulting from radial thermal expansion mismatch around a cylindrical fastener are developed. These solutions enable a designer to quickly evaluate many combinations of materials for the fastener and structure. Using the algebraic equations developed in this study, material properties and joint geometry were varied to determine their effect on thermal stresses. Finite element analyses were used to verify that the closed-form solutions derived in this study give the correct thermal stress distribution around a cylindrical fastener and to investigate the effect of some of the simplifying assumptions made in developing the closed-form solutions for thermal stresses.

Introduction

The National Aerospace Plane (NASP) program has generated a renewed interest in research relating to hypersonic vehicles. The severe aerodynamic heating encountered by reusable hypersonic vehicles makes the structural design of such vehicles a challenge. Uninsulated structures, which are allowed to become hot during flight, are attractive for the design of some components of hypersonic vehicles because they have the potential to reduce weight and increase vehicle efficiency. Some hot structures being considered may reach temperatures approaching 3000 °F. Carbon or silicon-based materials will likely be useful for such structures because they are light and retain strength at elevated temperature.

An example of such a hot structure is a carbon-carbon body flap which was studied as a replacement for the insulated aluminum flap on the Shuttle (ref. 1). In addition, NASA Langley Research Center has a current contract for the development of oxidation protected carbon-carbon lightly loaded structures. This contract includes the design and fabrication of a carbon-carbon control surface test article for NASP, such as that shown in Figure 1. Because of manufacturing constraints, these structures and similar hot structures will consist of smaller pieces which must be fastened together. Ductile metal fasteners are desirable for assembling these multiple pieces. However, the thermal expansion mismatch between the metal fastener and carbon or silicon-based structural materials makes it difficult to design a structural joint which is tight over the operational temperature range without exceeding allowable stress limits.

Fig. 1: Hot structures for reusable hypersonic vehicles

The problems encountered using a standard cylindrical fastener in a high temperature joint are illustrated in Figure 2. In the figure the fastener is assumed to have a much larger coefficient of thermal expansion (CTE) than the structure. If the fastener is snug in the radial direction at room temperature, high thermal stresses will result. If the fastener is snug in the axial direction at room temperature, high thermal stresses will result at elevated temperature. If the fastener is insufficiently prestressed at room temperature to remain snug at elevated temperature, the radial thermal stress or axial prestress required to maintain an acceptably tight joint over the operational temperature range may cause premature failure of the joint. Therefore, a joint...
consisting of a standard cylindrical fastener that joins pieces of the selected high temperature material should be analyzed to determine if there are satisfactory combinations of radial clearance and axial prestress at room temperature which will produce acceptable stresses at elevated temperature.

![Diagram of cylindrical fastener joints](image)

**Fig. 2: Problem areas for cylindrical fastener joints in high temperature applications**

<table>
<thead>
<tr>
<th>Joint condition</th>
<th>Radial direction</th>
<th>Axial direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Room temperature</td>
<td>Snug, Loose</td>
<td>Snug, Pre-stressed</td>
</tr>
<tr>
<td>High temperature</td>
<td>High thermal stresses</td>
<td>Loose, Snug</td>
</tr>
</tbody>
</table>

In this paper, simple solutions for calculating the thermal stresses resulting from radial thermal expansion mismatch around a cylindrical fastener are developed to enable a designer to evaluate quickly many combinations of materials. The solutions developed do not address the axial prestress.

Although similar solutions were found in the literature, no explicit solution for the thermal stresses around a cylindrical fastener was found. References 2-4 contain axisymmetric plane-stress solutions, but these solutions do not involve temperature or thermal expansion. References 5-9 contain similar solutions involving thermal expansion, but do not address the same problem as this paper. (A more detailed discussion of references 2-9 is given in Appendix A.) Consequently, algebraic equations for the thermal stresses in a finite ring of material around a cylindrical fastener of a different material are developed in this study. Similarly, algebraic equations for the thermal stresses around a hollow cylindrical fastener are also developed. Equations are also presented that account for the effect of an initial clearance between the fastener and surrounding material. The equations developed in this study are used to study the effect of varying joint geometry and material properties on the thermal stresses in a cylindrical fastener joint. Three-dimensional finite element analyses are used to verify these solutions and to evaluate the effect of some of the simplifying assumptions on the magnitude and distribution of the thermal stress.

<table>
<thead>
<tr>
<th>Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$ radius of fastener</td>
</tr>
<tr>
<td>$a_i$ inside radius of hollow fastener</td>
</tr>
<tr>
<td>$A, B, C, D$ constants defined by boundary conditions</td>
</tr>
<tr>
<td>$b$ radius of structure around fastener</td>
</tr>
<tr>
<td>$E$ modulus of elasticity</td>
</tr>
<tr>
<td>$L$ length</td>
</tr>
<tr>
<td>$L_0$ length at initial temperature</td>
</tr>
<tr>
<td>$n$ integer</td>
</tr>
<tr>
<td>$P$ pressure at interface between fastener and structure</td>
</tr>
<tr>
<td>$r, \theta$ cylindrical coordinates</td>
</tr>
<tr>
<td>$T$ temperature</td>
</tr>
<tr>
<td>$T_0$ initial temperature</td>
</tr>
<tr>
<td>$u$ displacement</td>
</tr>
<tr>
<td>$\alpha$ coefficient of linear thermal expansion (CTE)</td>
</tr>
<tr>
<td>$\delta$ initial clearance around a cylindrical fastener</td>
</tr>
<tr>
<td>$\phi$ Airy stress function</td>
</tr>
<tr>
<td>$c$ strain</td>
</tr>
<tr>
<td>$\nu$ Poisson's ratio</td>
</tr>
<tr>
<td>$\sigma$ stress</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Subscripts</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f$ fastener</td>
</tr>
<tr>
<td>$g$ gap</td>
</tr>
<tr>
<td>$\text{max}$ maximum</td>
</tr>
<tr>
<td>$r$ r direction</td>
</tr>
<tr>
<td>$s$ structure around the fastener</td>
</tr>
<tr>
<td>$z$ z direction</td>
</tr>
<tr>
<td>$\theta$ $\theta$ direction</td>
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</table>

**Closed-Form Solutions**

The objective of the following analysis is to develop a simple equation to predict the thermal stresses around a cylindrical fastener at elevated temperature. The thermal stresses predicted by this analysis occur because the fastener has greater thermal expansion in the radial direction than the surrounding structure. The fastener is also assumed to have greater thermal expansion in the axial direction than the surrounding structure and, with increasing temperature, will tend to loosen axially, relieving any axial prestress. If appreciable normal stresses develop at the interface between the fastener and structure, friction will inhibit the axial slippage of the fastener and thereby induce out-of-plane stresses in the joint. However, in the present analysis the assumption is made that there is no friction between the fastener and surrounding structure. Thus, no significant stresses are introduced in the thickness direction of the material and a condition of plane stress results. The fastener and surrounding structure are assumed to undergo a uniform temperature change and to expand radially as isotropic materials although the CTE's of the two materials differ. For simplicity, the fastener is assumed to be surrounded by a disk of material with the fastener at the center. In most joints the shape of the structure will be much more complicated, and the thermal stresses will vary somewhat around the fastener. Also, for simplicity the ends of the bolt, including the head, are neglected (see fig. 3). With these assumptions the problem becomes an axisymmetric, plane-stress problem.
For the structure surrounding the fastener, the boundary conditions are:

at $r = a$, $\sigma_{rs}(a) = -P$ and at $r = b$, $\sigma_{rs}(b) = 0$

Applying these boundary conditions to equations (B12), the constants $A$ and $C$ are found to be:

$$A = \frac{P b^2}{1 - (\frac{b}{a})^2} \quad \text{and} \quad C = -\frac{P}{2} \left( \frac{1}{1 - (\frac{b}{a})^2} \right)$$  (5)

The expression for radial displacement in the structure surrounding the fastener therefore becomes:

$$u_{rs} = \frac{P}{E} \left( \frac{1}{\frac{b}{a}} \right) \left( \frac{b^2}{1 + \nu_s} \right) + (1 - \nu_s) r a s \Delta T$$  (6)

Displacement compatibility at the interface between the fastener and structure, $r = a$, requires that the radial displacements be equal at that point. Equations (3) and (6), with $r = a$, can be combined and solved for the interface pressure, $P$.

$$P = \frac{E_s \left( \frac{b}{a} \right)^2 (1 + \nu_s) + (1 - \nu_s) r a \Delta T}{1 - (\frac{b}{a})^2}$$  (7)

The stress distributions in the structure surrounding the fastener can be found by combining equations (B12) and (5) to produce:

$$\sigma_{rs} = -P \left( \frac{\frac{b}{a}}{\left( \frac{b}{a} \right)^2 - 1} \right)$$  (8)

$$\sigma_{rs} = -P \left( \frac{\frac{b}{a}}{\left( \frac{b}{a} \right)^2 - 1} \right)$$  (9)

where $P$ is given by equation (7).

Several limiting cases of the solution for thermal stresses around a cylindrical fastener are now considered. These limiting cases give insight into the behavior of the joint and provide simpler expressions for thermal stresses in the joint. If the structure around the fastener is shrunk to an infinitely thin ring, the stresses in the fastener approach zero. The only nonzero stress is the hoop stress in the ring of structure which is given by the simple expression:

$$\sigma_{rs} = E_s \left( \alpha_f - \alpha_s \right) \Delta T$$  (10)

If the fastener is surrounded by an infinite sheet ($b/a = \infty$), equation (7) reduces to the following form:

$$P = \frac{E_s \left( \alpha_f - \alpha_s \right) \Delta T}{1 + \nu_s}$$  (11)

Therefore, the radial displacement for the fastener is given by:

$$u_{rs} = -\frac{P}{E} \left( 1 - \nu_a \right) + r a \Delta T$$  (12)

Substituting equations (2) into (B12) shows that the fastener is in hydrostatic compression,

$$\sigma_{rs} = \sigma_{rs} = -P$$  (13)
and the stresses in the structure around the fastener become:

\[ \sigma_\theta = - \sigma_r = \left( \frac{3}{2} \right) \frac{p}{r} \]  
(12)

If the fastener material is much stiffer than the surrounding structure (\( E_f \gg E_s \)), then equation (11) reduces to:

\[ p = \frac{E_s (\alpha_f - \alpha_s) \Delta T}{1 + \nu_s} \]  
(13)

and the stresses may be found by substituting equation (13) into equations (8) and (9). Similarly, if the surrounding structure is much stiffer than the fastener material (\( E_s \gg E_f \)), then equation (11) reduces to:

\[ p = \frac{E_f (\alpha_f - \alpha_s) \Delta T}{1 - \nu_f} \]  
(14)

Hollow Fastener Solution

The solution for thermal stresses of a hollow cylindrical fastener can be found by following a similar procedure. The equations for stresses and displacements in the structure surrounding the fastener will have the same form for the hollow fastener as for the solid fastener, however, the equation for the interface pressure between the fastener and surrounding structure will be different.

The stresses and displacement for the idealized hollow fastener (fig. 3) can be found by applying the following boundary conditions:

at \( r = a_1 \), \( \sigma_r (a_1) = 0 \)

and at \( r = a \), \( \sigma_r (a) = -p \)

Applying these boundary conditions to equations (B12), the constants A and C are found to be:

\[ A = \frac{p a_2^2}{a_2^2 - a_1^2}, \quad C = -\frac{p a_2^2}{2 (a_2^2 - a_1^2)} \]  
(15)

The expression for radial displacement in the fastener, therefore, becomes:

\[ u_r = -\frac{p (-1 + \frac{1}{\nu_f} + (1 - \nu_f)) r}{1 - \left( \frac{a_1}{a} \right)^2} + r \sigma_r \Delta T \]  
(16)

As in the solid fastener solution, displacement compatibility at the interface between the fastener and surrounding structure, \( r = a \), requires that the radial displacements be equal at that point. Equations (6) and (16), with \( r = a \), can be combined and solved for the interface pressure \( P \) to give:

\[ \frac{E_s}{a} \left( \frac{5}{2} - 1 \right) (\alpha_f - \alpha_s) \Delta T \]

\[ \left( \frac{5}{2} \right) \left( 1 + \nu_f \right) + \left( 1 - \nu_f \right) \left( \frac{E_s}{E_f} \right) \left( \frac{5}{2} - \frac{a_2^2}{a_1^2} \right) \left( 1 - \nu_f \right) \]

\( \sigma_f = -P \left( \frac{a_1}{a} \right)^2 \left( \frac{1}{\nu_f} - 1 \right) \)

\( \sigma_s = p \left( \frac{a_1}{a} \right)^2 \left( \frac{1}{\nu_f} - 1 \right) \)

Fastener With Initial Clearance

Fasteners may have an initial clearance at room temperature. Equation (7) may be easily modified to account for an initial clearance. The temperature change which closes the gap due to the initial clearance must be calculated. This temperature change can be found by equating the radial position of the expanded fastener and hole. As discussed in reference 10, if the CTE's are assumed to be independent of temperature, the exact expression for thermal expansion can be found by integrating the equation:

\[ dL = L a \Delta T \]  
(20)

to produce

\[ L = L e^{a(T - T_s)} \]  
(21)

Equating the radial position of the expanded fastener and hole therefore produces:

\[ a e \Delta T_g = (a + \delta) e \Delta T \]  
(22)

Solving equation (22) for the temperature change produces:

\[ \Delta T_g = \frac{\ln (1 + \delta)}{\alpha_f - \alpha_s} \]  
(23)

where \( \delta \) is the initial clearance. The temperature change calculated using equation (23) can be subtracted from the total temperature change, and the net temperature change can be substituted into equation (7) to find the corresponding thermal stresses. Note, however, that the term \( (\Delta T - \Delta T_g) \) must be positive.

Finite Element Analyses

The finite element method is commonly used to numerically analyze the stresses in joints. Therefore, finite element solutions were obtained
for comparison with the mathematically derived solutions presented in this study. Finite element analyses were used to verify that the equations derived in this study give the correct thermal stress distribution around a cylindrical fastener. In addition the finite element analyses were used to investigate the effect of friction and the protruding end of the fastener on the thermal stress distribution. An existing, general-purpose finite element program, Engineering Analysis Language (EAL) (ref. 11), was used for the analyses.

A three-dimensional finite element model (see figure 4) was used to represent a solid cylindrical fastener joint configuration. A five-degree wedge-shaped model of the fastener and structure was used as indicated on the figure. Similar models were used to model both a hollow fastener joint and a joint with a fastener which protruded from the structure.

Fig. 4: Finite element model of solid cylindrical fastener and surrounding structure

Particular, but arbitrary, dimensions were chosen for each joint modeled. The radius of the cylindrical fastener was 0.1 inch and the radius of the surrounding structure was 1.0 inch. The thickness of the cylindrical fastener model was 0.15 inch, but the symmetry constraints applied on the lower surface made the model respond as though it were 0.3 inch thick. The inside radius of the hollow fastener was 0.05 inch. For the model of the protruding fastener, the fastener extended 0.1 inch above the structure. An arbitrary set of isotropic stiffness properties was used (see figure 4).

Three-dimensional, six-node and eight-node, linear elements were used to model the fastener and eight-node linear elements were used to model the surrounding structure. The radial dimensions of the elements were varied parabolically to achieve a finer mesh near the interface between the fastener and surrounding structure. The finer mesh near the interface was required for efficient prediction of the expected stress gradients in that region. The resulting finite element mesh can be seen in figure 4. Nodes on the sides and bottom of the model were constrained to remain in their respective planes but were free to move within those planes.

The most challenging portion of the finite element analysis was modeling the contact between the fastener and surrounding material along the bearing surface. Adjacent nodes along this boundary were connected by zero-length elements which had high stiffness perpendicular to the bearing surface and no stiffness tangent to the surface. The elements allowed relative motion along the boundary between the adjacent nodes, but not perpendicular to it. The model was set up so that if one of these elements was found to be in tension, the element stiffness could be reduced to an insignificant value. For the cases analyzed, however, none of the zero length elements was in tension. For the solid fastener, one case was analyzed with the zero length elements having large (essentially infinite) stiffnesses in all degrees of freedom. This case simulated the highest possible stresses due to friction.

Results and Discussion

Comparison of Closed-Form and Finite Element Solutions

The thermal stresses calculated from the closed-form solutions developed in this paper are compared to those calculated from the finite element models previously described. Figure 5 shows the thermal stress comparison for a solid fastener subjected to a 1000 °F temperature increase. The radial and hoop stresses are shown as a function of radial position for both the fastener and structure. The shaded region of the figure represents the fastener, and the unshaded region represents the surrounding structure. Stresses calculated using the three-dimensional finite element model are indicated by symbols, and the plane-stress solution stresses are indicated by solid lines. The results show excellent agreement for both the magnitude and distribution of thermal stresses. The peak stress of approximately 80,000 psi is indicative of the large thermal stresses which may occur in hot structures with cylindrical fasteners. The peak stresses in the structure occur at the interface between the fastener and structure. The magnitude of the compressive radial stress is slightly less than the magnitude of the tensile hoop stress.

Fig. 5: Thermal stresses for solid cylindrical fastener
The fastener is in hydrostatic compression with stress equal to the peak radial stress in the structure, as shown by the constant hoop and radial stresses in the fastener. Because the structural hoop and radial stresses are tensile and compressive respectively, the von Mises stress in the structure will be considerably larger than in the fastener. The geometry and loading are axisymmetric, so the in-plane shear stress is inherently zero. The plane-stress solution neglects the z stress and out-of-plane shear stresses. In the absence of friction between the fastener and structure, the three-dimensional finite element analysis also predicts negligible out-of-plane stresses for the relatively thick model analyzed.

Figure 6 shows a similar comparison of thermal stresses for a hollow fastener. The geometry, material properties, and loading are the same as for figure 5, except that the fastener is hollow. The shaded band represents the wall of the hollow fastener. Again, the only significant stresses are the radial and hoop stresses in the fastener and structure. The figure shows the excellent agreement between the three-dimensional finite-element and plane-stress solutions. The stress distribution in the structure is similar to that for the solid fastener, however, the magnitudes of the stresses are lower. The stress distribution in the hollow fastener is completely different than in the solid fastener. The radial stress goes to zero on the inner surface of the fastener, as expected. The hoop stress in the hollow fastener is much larger than that in the solid fastener. The peak hoop stress occurs on the inner surface of the hollow fastener, and for this example, is more than double the stress in a solid fastener. Again, the out-of-plane stresses predicted by the three-dimensional finite element solution were negligible.

![Fig. 6: Thermal stresses for hollow fastener](image)

**Effects of Geometry and Material Properties on Thermal Stress**

The equations for thermal stress around a solid cylindrical fastener were used to determine the effect of varying material properties and geometry. Figure 7 shows the maximum thermal stresses in the structure around a solid fastener as a function of ratio of structure-to-fastener radius.

![Fig. 7: Maximum structural thermal stresses for solid fastener as a function of ratio of structure-to-fastener radius](image)

- **Effects of Geometry and Material Properties on Thermal Stress**

  The Poisson's ratios of the fastener and structure were also varied to determine their effect on thermal stress. In figure 8 the maximum thermal stress (hoop stress at r=a) in a solid fastener in an infinite sheet of structure is shown as a function of Poisson's ratio of the fastener. The Poisson's ratio of the structure is also varied to produce a set of curves. The values of Poisson's ratio are varied from zero to 0.5, the range possible with isotropic materials. Increasing the fastener Poisson's ratio effectively stiffens the fastener, which is in two-dimensional hydrostatic compression, and thereby increases the peak thermal stress. Increasing the structure Poisson's ratio, however, decreases peak thermal stresses.

![Fig. 8: Maximum structural thermal stresses for solid fastener as a function of Poisson's ratio](image)
Figure 8 shows a comparison of the maximum thermal stresses of a hollow fastener to those of a solid fastener with similar geometry, material properties, and loading. The ratio of hollow-to-solid fastener stresses is shown as a function of the ratio of the hollow fastener inner-to-outer radius. The figure shows that even a small hole in the center of a fastener increases the stress in the fastener by a factor of 2. The stress increase occurs because the stress distribution is changed by the introduction of the hole. On the inner surface of the fastener, the normal stress must be zero. The hoop stress increases significantly, compared to a solid fastener. As the wall of the hollow fastener becomes thinner, the stress in the structure is reduced, but the fastener stress remains high. When the fastener modulus is greater than that of the structure, the fastener stress ratio increases as the fastener wall is made thinner. When the moduli are equal, the fastener stress ratio remains constant with wall thickness. When the structural modulus is greater, the fastener stress ratio decreases as the fastener wall is made thinner. For a specified wall thickness, a stiffer fastener modulus produces a lower stress ratio.

Figure 9 shows a comparison of the maximum thermal stresses of a hollow fastener to those of a solid fastener with similar geometry, material properties, and loading. The ratio of hollow-to-solid fastener stresses is shown as a function of the ratio of the hollow fastener inner-to-outer radius. The figure shows that even a small hole in the center of a fastener increases the stress in the fastener by a factor of 2. The stress increase occurs because the stress distribution is changed by the introduction of the hole. On the inner surface of the fastener, the normal stress must be zero. The hoop stress increases significantly, compared to a solid fastener. As the wall of the hollow fastener becomes thinner, the stress in the structure is reduced, but the fastener stress remains high. When the fastener modulus is greater than that of the structure, the fastener stress ratio increases as the fastener wall is made thinner. When the moduli are equal, the fastener stress ratio remains constant with wall thickness. When the structural modulus is greater, the fastener stress ratio decreases as the fastener wall is made thinner. For a specified wall thickness, a stiffer fastener modulus produces a lower stress ratio.

Fastener Protrusion Beyond Surrounding Structure

Most fasteners extend through the structure being fastened. However, to develop the algebraic equations in this study, the fastener was assumed not to protrude beyond the surrounding structure. A finite element model was used to investigate the effects of this assumption on the thermal stresses for a particular fastener which protrudes beyond the structure. Figure 10 shows the distribution of radial stress in such a joint. Away from the fastener the stresses are uniform through the thickness, which is consistent with the plane-stress solution. Near the fastener the stresses in the structure increase slightly toward the protruding end of the fastener, compared to the stresses in a joint in which the fastener does not protrude. The portion of the fastener protruding beyond the structure is free to thermally expand radially, but the portion of the fastener within the structure is constrained by the surrounding structure. Thus, the fastener expands more near the protruding end of the fastener than towards the center of the structure. The presence of a fastener head would therefore be expected to increase the local, peak thermal stresses in the structure even more than a protruding fastener shank. The peak stresses for a joint with a protruding fastener are shown in Table 1, along with the stress on the using the plane-stress solution developed in this study. The peak radial stress in the structure is about 15 percent higher for the protruding fastener. However, these higher stresses are localized at the fastener/structure interface at the free surface and for many materials may be alleviated by local yielding. Significant out-of-plane stresses, not present in the plane-stress solution, were calculated for the joint with a protruding fastener. However, these out-of-plane stresses are low compared to the in-plane stresses. If brittle materials or materials with low out-of-plane strengths are used in the joint, a more detailed analysis of the joint may be required.
Table 1: Maximum calculated thermal stresses (psi)

<table>
<thead>
<tr>
<th>Type of Stress</th>
<th>Maximum Stress (psi)</th>
</tr>
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<tbody>
<tr>
<td>Friction Between Fastener and Structure</td>
<td></td>
</tr>
<tr>
<td>Radial</td>
<td>-51,000</td>
</tr>
<tr>
<td>Hoop</td>
<td>-51,000</td>
</tr>
<tr>
<td>Radial</td>
<td>-51,000</td>
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<tr>
<td>Hoop</td>
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</tr>
<tr>
<td>Shear</td>
<td>0</td>
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<tr>
<td>Shear</td>
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</table>

Table 2: Limits of clearance for running and sliding fits from ref. 12

<table>
<thead>
<tr>
<th>Diameter Range, in.</th>
<th>Close Fit</th>
<th>Medium Fit</th>
<th>Loose Fit</th>
</tr>
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<tbody>
<tr>
<td>0.25</td>
<td>0.18</td>
<td>0.65</td>
<td>0.05</td>
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<tr>
<td>0.5</td>
<td>0.83</td>
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<td>0.75</td>
<td>0.95</td>
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<tr>
<td>2.75</td>
<td>1.9</td>
<td>2.3</td>
<td>1.05</td>
</tr>
</tbody>
</table>

Fig. 11: Temperature change required to close initial clearance

Concluding Remarks

Algebraic equations derived from closed-form solutions of elasticity equations are presented for thermal stresses in solid and hollow
cylindrical fasteners. The fastener was idealized as a pin surrounded by an annulus of structural material so that an axisymmetric plane-stress solution could be developed. Friction between the fastener and structure was neglected. For these assumptions, excellent agreement was obtained between the thermal stresses predicted by the algebraic equations and by a three-dimensional finite element analysis.

An equation was also developed to predict the temperature required to close an initial clearance around a fastener. This temperature can be subtracted from the total temperature change to calculate the thermal stress of a fastener with an initial clearance at room temperature. For a sample joint the initial fastener fit is shown to have a significant effect on the thermal stresses.

Using the algebraic equations developed in this study, material properties and joint geometry were varied to determine their effect on thermal stresses. The effect of varying the ratio of structure to fastener radius exceeds five or six the peak stresses become insensitive to further increases in the ratio. The effect of varying the modulus ratio was also shown. Increasing fastener modulus increases thermal stresses in the joint. Thermal stresses of similar hollow and solid fasteners were compared. Hollowing out a fastener increases fastener stresses and decreases structural stresses. The effects of varying the Poisson's ratio of both the fastener and structure are also shown. Increasing the fastener Poisson's ratio effectively stiffens the fastener, which is in two-dimensional hydrostatic compression, and thereby increases the peak thermal stress. Increasing the structure Poisson's ratio, however, decreases peak thermal stresses.

Finite element analysis was used to bound the effects of friction on thermal stresses and to show the effects of a fastener protruding beyond the structure surrounding the fastener was determined. When the ratio of structure to fastener radius exceeds five or six the peak stresses become insensitive to further increases in the ratio. The effect of varying the structure-to-fastener modulus ratio was also shown. Increasing fastener modulus increases thermal stresses in the joint. Thermal stresses of similar hollow and solid fasteners were compared. Hollowing out a fastener increases fastener stresses and decreases structural stresses. The effects of varying the Poisson's ratio of both the fastener and structure are also shown. Increasing the fastener Poisson's ratio effectively stiffens the fastener, which is in two-dimensional hydrostatic compression, and thereby increases the peak thermal stress. Increasing the structure Poisson's ratio, however, decreases peak thermal stresses.

Several solutions for thermal stresses in the vicinity of cylindrical inclusions were found in the literature. References 5 and 6 present thermal stress solutions for a cylinder with an infinite radius of material surrounding the cylinder. Similarly, the solution in reference 6, when simplified for a constant temperature change and converted to plane stress, also agrees with the current solution and that of reference 5 for a cylinder with an infinite radius of material surrounding the cylinder. The solutions in references 5 and 6, however, do not account for a finite radius of material surrounding the cylindrical inclusion. In references 7 and 8 a plane-stress solution for thermal stresses in a disk with a central shaft is given. The central shaft, however, is rigid and does not expand with temperature. A solution for thermal stresses in a circular plate with a central hot spot is also presented in reference 8. Reference 9 a more complicated problem with a pair of circular inclusions and a nonuniform temperature distribution.

Although the general solution and similar solutions were found in the literature, no explicit solution for the thermal stresses around a cylindrical fastener was found. Consequently, algebraic equations for the stresses in a finite ring of material around a cylindrical fastener of a different material were developed in this study.

Appendix B

General Solution For Axisymmetric Plane Stress

The general plane-stress solution for axisymmetric problems, as outlined in references 2 and 3, is shown in the following mathematical development. The biharmonic equation which defines the Airy stress function for axisymmetric plane stress is:

$$\frac{d^2 \phi}{dr^2} + \frac{1}{r} \frac{d \phi}{dr} = \frac{1}{r^2} \frac{d}{dr} \left( \frac{d \phi}{dr} \right) = 0$$  \hspace{1cm} (B1)

Equation (B1) has a solution in terms of integer powers of \(r\):

$$\phi = r^n$$  \hspace{1cm} (B2)

Substituting equation (B2) into equation (B1) produces a characteristic equation which leads to the general solution.
\[ \phi = A \ln(r) + B r^2 \ln(r) + C r^2 + D \quad (B3) \]

In polar coordinates, the stresses in terms of the Airy stress function are

\[ \sigma_r = \frac{1}{r} \frac{\partial \phi}{\partial r} + \frac{A}{r} \frac{\partial^2 \phi}{\partial \theta^2} \]
\[ \sigma_\theta = \frac{A}{r} \frac{\partial^2 \phi}{\partial r^2} \quad (B4) \]
\[ \sigma_r = -\frac{B}{2} \frac{\partial^2 \phi}{\partial r \partial \theta} \]

Therefore, by substituting the expression for \( \phi \) into equations (B4), the stresses for an axisymmetric problem are found to be

\[ \sigma_r = \frac{A}{r} + 2B \ln(r) + B + 2C \]
\[ \sigma_\theta = \frac{A}{r} + 2B \ln(r) + 3B + 2C \quad (B5) \]
\[ \sigma_r = 0 \]

In polar coordinates, the definitions of strain are given by

\[ \epsilon_r = \frac{1}{E} \left( \frac{\partial u_r}{\partial r} - \nu \frac{u_\theta}{r} \right) \]
\[ \epsilon_\theta = \frac{1}{E} \left( \frac{\partial u_\theta}{\partial r} + \nu \frac{u_r}{r} \right) \]
\[ \epsilon_r = \frac{1}{2} \left( \frac{\partial^2 u_r}{\partial r^2} + \frac{1}{r} \frac{\partial u_r}{\partial \theta} \right) \quad (B6) \]

Hooke's stress-strain relations for an isotropic material in plane stress are

\[ \epsilon_r = \frac{1}{E} (\sigma_r - \nu \sigma_\theta) \]
\[ \epsilon_\theta = \frac{1}{E} (\sigma_\theta - \nu \sigma_r) \quad (B7) \]
\[ \epsilon_r = \frac{1}{E} \sigma_r \]

The displacements may be calculated from the stresses by combining equations (B5), (B6), and (B7) and solving for radial displacement. First solve for the strains to obtain:

\[ \epsilon_r = \frac{1}{E} \left( \frac{A (1+\nu)}{r^2} + 2C (1-\nu) \right. \]
\[ + B (1-\nu)[2 \ln(r) + 1] - 2 \nu B \] \quad (B8)
\[ \epsilon_\theta = \frac{1}{E} \left( \frac{A (1+\nu)}{r^2} + 2C (1-\nu) \right. \]
\[ + B (1-\nu)[2 \ln(r) + 1] - 2 \nu B \] \quad (B9)

Solution of equations (B8) and (B9) produces two expressions for radial displacement which must be consistent. They are

\[ u_r = \frac{1}{E} \left( -\frac{A (1+\nu)}{r} + 2C (1-\nu) \right) \]
\[ + B (1-\nu)[2 \ln(r) - r] - 2 \nu B r + \text{const} \quad (B10) \]
\[ u_r = \frac{1}{E} \left( -\frac{A (1+\nu)}{r} + 2C (1-\nu) \right) \]
\[ + B (1-\nu)[2 \ln(r) + r] + 2B r \quad (B11) \]

For equations (B10) and (B11) to be consistent the constant B and the constant in equation (B10) must be zero. The general expressions for plane stress are therefore:

\[ \sigma_r = \frac{A}{r} + 2C \]
\[ \sigma_\theta = -\frac{A}{r} + 2C \quad (B12) \]

and the radial displacement is given by:

\[ u_r = \frac{1}{E} \left( -\frac{A (1+\nu)}{r} + 2C (1-\nu) \right) \quad (B13) \]

The constants A and C can be obtained by substitution of boundary conditions into equations (B12) and (B13).

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References


Thermal Stress in High Temperature Cylindrical Fasteners

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Uninsulated structures that are fabricated from carbon or silicon-based structural materials and that are allowed to become hot during flight are attractive for the design of some components of hypersonic vehicles. They have the potential to reduce weight and increase vehicle efficiency. Because of manufacturing constraints, these structures will consist of parts which must be fastened together. The thermal expansion mismatch between conventional metal fasteners and carbon or silicon-based structural materials may make it difficult to design a structural joint which is tight over the operational temperature range without exceeding allowable stress limits. In this study, algebraic, closed-form solutions for calculating the thermal stresses resulting from radial thermal expansion mismatch around a cylindrical fastener are developed. These solutions enable a designer to quickly evaluate many combinations of materials for the fastener and structure. Using the algebraic equations developed in this study, material properties and joint geometry were varied to determine their effect on thermal stresses. Finite element analyses were used to verify that the closed-form solutions derived in this study give the correct thermal stress distribution around a cylindrical fastener and to investigate the effect of some of the simplifying assumptions made in developing the closed-form solutions for thermal stresses.