

WCAP-17942-NP (APR1400-A-M-NR-14001-NP)  
Revision 0

November 2014

# **KHNP APR1400 Flywheel Integrity Report**

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## **KHNP APR1400 Flywheel Integrity Report**

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**November 2014**

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## ABSTRACT

This report demonstrates that the APR1400 reactor coolant pump (RCP) flywheel satisfies all of the RCP flywheel integrity criteria of the design specification and U.S. NRC Regulatory Guide 1.14. The combination of shrink-fit and rotational stresses at normal operation and at overspeed are calculated and shown to be acceptable with respect to the prescribed criteria. The speed at which separation of the main flywheel from the hub and the separation of the hub from the shaft is computed to be significantly above the separation speed of 150% of normal operation speed required in the design specification.

The critical speeds for ductile and non-ductile fracture are computed and shown to be significantly greater than two times normal speed as required by the Regulatory Guide. It is demonstrated that excessive deformation will not result from an overspeed condition.

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## 1 BACKGROUND AND PURPOSE

In support of the Design Certification Application (DCA) of the APR1400 Advanced Light Water Reactor (ALWR) design, Korea Hydro & Nuclear Power (KHNP) requested Westinghouse to evaluate the integrity of the APR1400 RCP motor flywheel design. The subject matter of this report is the flywheel integrity analyses performed by Westinghouse to quantify the structural integrity of the flywheel as well as the critical speeds of the flywheel under normal and off-normal operating conditions. Specifically, the objective of the analyses is to compute the stresses and critical speeds and to compare the results to criteria in the design specification [1] and U.S. NRC Regulatory Guide 1.14 [2].

Portions of this report contain proprietary information. Proprietary information is identified and bracketed. For each of the bracketed sections, the reasons for the proprietary classification are provided using letters superscripted letters “a”, “c”, and “e”. These letters stand for:

- a. The information reveals the distinguishing aspects of a process or component, structure, tool, method, etc. The prevention of its use by Westinghouse’s competitors, without license from Westinghouse, gives Westinghouse a competitive economic advantage.
- c. The information, if used by a competitor, would reduce the competitor’s expenditure of resources or improve the competitor’s advantage in the design, manufacture, shipment, installation, assurance of quality, or licensing of a similar product.
- e. The information reveals aspects of past, present, or future Westinghouse- or customer-funded development plans and programs of potential commercial value to Westinghouse.

## **2 REQUIREMENTS**

### **2.1 LIMITS OF APPLICABILITY**

This report is applicable to the KHNP APR1400 flywheel only.

### **2.2 OPEN ITEMS**

This WCAP contains no open items.

### **2.3 DISCUSSION OF SIGNIFICANT ASSUMPTIONS**

This calculation note contains no significant assumptions.

### **2.4 ACCEPTANCE CRITERIA**

The acceptance criteria in the specification [1] include:

1. The total stress in the flywheel at standstill and normal operating speed does not exceed one-third of the ultimate tensile strength.
2. The total stress at design overspeed does not exceed two-thirds of the minimum yield strength, where design overspeed is 125% of normal operating speed.
3. The total stress at joint release speed shall not exceed the lower of one-half of the minimum specified ultimate strength or two-thirds of the minimum specified yield strength. The joint release speed shall be equal to or greater than 150% of normal operation speed.

The Regulatory Guide acceptance criteria are established in [2] as:

4. Flywheel assembly should be designed to withstand normal conditions, anticipated transients, the design basis LOCA, and the safe shutdown earthquake without loss of structural integrity.
5. The critical speed for ductile fracture should be predicted.
6. The critical speed for non-ductile fracture should be predicted.
7. The normal speed should be less than one-half of the lowest critical speeds.
8. The predicted LOCA overspeed should be less than the lowest critical speeds.
9. The critical speed for excessive deformation of the flywheel should be predicted.

### 3 METHODOLOGY

This analysis calculates the stresses in the flywheel resulting from the shrink fit of the sections of the flywheel and the rotation of the flywheel at various speeds. Using these stresses, the integrity of the flywheel is evaluated by assuming a crack in the highest stressed location of the flywheel to assess if the criteria of [1] and [2] are satisfied.

#### 3.1.1 Stresses and Radial Displacement due to Shrink Fit of Sections of Flywheel

The stresses due to shrink fit are determined by the formulae [3, page 683].

The flywheel is constructed by shrink fitting three parts. First, the outer wheel is shrink fitted to the hub. Next, the assembly of the outer wheel and hub is shrink fitted to the shaft. Then, the amount of shrink fit required to maintain contact between the outer wheel and the shaft is determined by the relative radial displacement due to rotation at the maximum rotational speed.

The following definitions apply:

a = outside radius of outer wheel

b = inner radius of outer wheel

c = outer radius of the hub, nominally equal to b

d = averaged inner radius of hub

e = outer radius of the shaft, nominally equal to d

The relationship between radial displacement of the inner radius and contact pressure for the outer wheel is expressed as:

$$b = (Pb/E)[(a^2 + b^2)/(a^2 - b^2) + \nu] \quad \text{Equation 1 [3]}$$

In Equation 1:

a = outside radius of wheel

b = inside radius of wheel

P = contact pressure between outer wheel and hub

E = Young's modulus

$\nu$  = Poisson ratio

radial stress:

$$S_r = -Pb^2(a^2 - r^2)/[r^2(a^2 - b^2)] \quad \text{Equation 2 [3]}$$

In Equation 2:

r = distance from center of wheel

P = pressure on inner radius of wheel

tangential stress:

$$St = Pb^2(a^2 + r^2)/[r^2(a^2 - b^2)] \quad \text{Equation 3 [3]}$$

The relationship between the radial displacement of the outer radius and the contact pressure for the hub is expressed as:

$$c = -(Pc/E)[(c^2 + d^2)/(c^2 - d^2) - \nu] \quad \text{Equation 4 [3]}$$

In Equation 4:

c = outside radius of hub

d = averaged inside radius of hub

radial stress:

$$Sr = -Pc^2(r^2 - d^2)/[r^2(c^2 - d^2)] \quad \text{Equation 5 [3]}$$

In Equation 5:

r = distance from center of wheel

P = pressure on outer radius of hub

tangential stress:

$$St = -Pc^2(d^2 + r^2)/[r^2(c^2 - d^2)] \quad \text{Equation 6 [3]}$$

### 3.1.2 Stresses and Radial Displacement due to Rotation of Flywheel

The stresses and displacements due to rotation of a disk of uniform thickness are taken from [3, page 746].

The change in the inner radius of the outer wheel due to rotation of “W” in radians/sec is expressed as:

$$b = (1/4)(\text{density/gravity}) * W^2 * (b/E) * [(3 + \nu)a^2 + (1 - \nu)b^2] \quad \text{Equation 7 [3]}$$

The change in the outer radius of the hub due to rotation of “W” in radians/sec is expressed as:

$$c = (1/4)(\text{density/gravity}) * W^2 * (c/E) * [(1 - \nu)c^2 + (3 + \nu)d^2] \quad \text{Equation 8 [3]}$$

tangential stress due to rotation:

$$St = (1/8)(\text{density/gravity}) * W^2 * [(3 + \nu) * (a^2 + d^2 + a^2 d^2 / r^2) - (1 + 3\nu)r^2] \quad \text{Equation 9 [3]}$$

In Equation 9, “r” is the radial location of the stress

Radial stress due to rotation:

$$S_r = (1/8) * (3 + \nu) * (\text{density/gravity}) * W^2 * [a^2 + d^2 - (a^2 d^2 / r^2) - r^2] \quad \text{Equation 10 [3]}$$

These equations apply to the outer wheel, the hub, and the shaft when the appropriate inner and outer radii are used. For the shaft, the inner radius is zero.

### 3.1.3 Stress Intensity Factor of Assumed Crack in Flywheel

The stress intensity factor for an axial crack in a cylinder will be used to compute the effect of a crack in the outer wheel. The highest stress location is the inner radius of the outer wheel. The stress intensity factor formula is taken from [4]. This formula computes the stress intensity factor for a fourth order polynomial stress distribution. The tabular coefficients in [4] are listed up to a thickness to inside radius ratio ( $t/R_i$ ) of 1.0.

$$\left( \begin{array}{c} \text{Coefficients for } S_t \text{ and } KI \end{array} \right) \quad \text{a, c, e}$$

$$S_t = S_0 + S_1(x/t) + S_2(x/t)^2 + S_3(x/t)^3 + S_4(x/t)^4 \quad \text{Equation 11}$$

$$KI = [G_0 S_0 + G_1 S_1(a/t) + G_2 S_2(a/t)^2 + G_3 S_3(a/t)^3 + G_4 S_4(a/t)^4] \text{SQRT}(\dot{\lambda} a) \quad \text{Equation 12}$$

In Equations 11 and 12:

$S_t$  = tangential stress

$a$  = radial dimension of assumed crack

$t$  = distance between inner and outer radii of wheel

$x$  = distance from cracked edge

## 4 INPUT

The units of dimensions and material properties are given in metric units. The calculations are performed in customary U.S. units. Some metric units will be stated as reference values for comparison with other flywheel documentation.

The stress analysis in [6] is performed in metric units. The results of the analysis in [6], when converted to customary U.S. units, are slightly different than the results calculated herein (where the input values are converted from metric units).

### 4.1 MATERIALS

The material selected for both parts of the flywheel is 26NiCrMoV14-5 [6]. This report uses the stated properties for evaluation at normal operation of 120°F, so it is recognized that the properties are appropriate for use at 120°F.

The modulus of elasticity is 204,000N/mm<sup>2</sup> or 29,587,956 lbs/in<sup>2</sup>.

The minimum yield strength is 640 N/mm<sup>2</sup> or 92,825 lbs/in<sup>2</sup>.

The minimum tensile strength is 800 N/mm<sup>2</sup> or 116,000 lbs/in<sup>2</sup>.

The normal operating temperature is 49°C or 120°F [1].

The minimum fracture toughness,  $K_{IC}$ , at normal operating temperature is at least 165 MPa m<sup>1/2</sup> or 150 ksi in<sup>1/2</sup> [1].

The density of the flywheel material is 7,850 kg/M<sup>3</sup> or 0.283 lbs/in<sup>3</sup>.

### 4.2 GEOMETRY



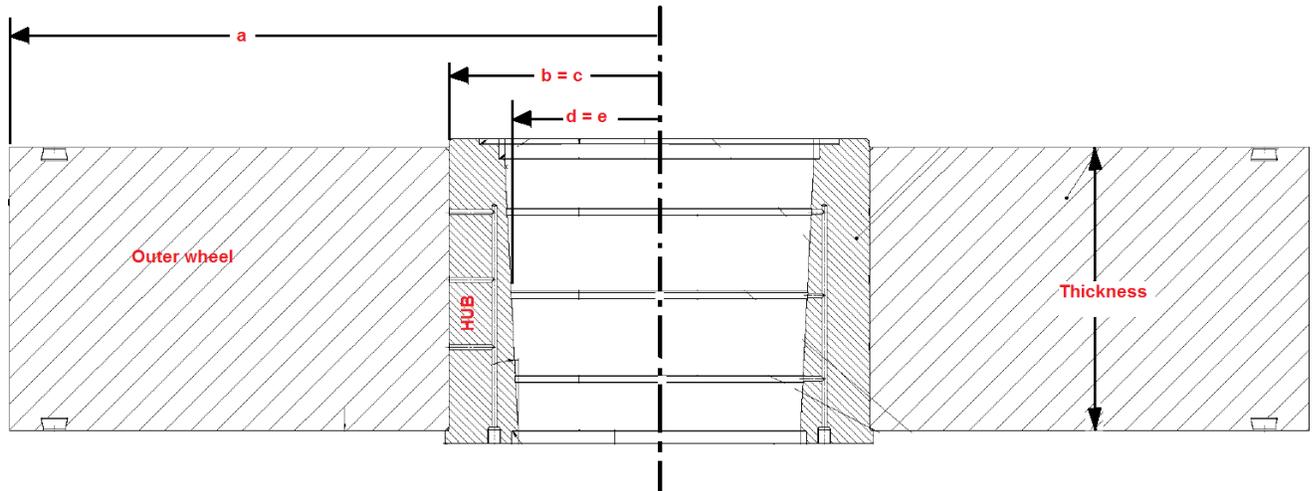


Figure 4-1: APR1400 Flywheel Sketch [6]

### 4.3 ROTATIONAL SPEEDS PER THE DESIGN SPECIFICATION

The speeds that are to be addressed include:

- standstill, only shrink fit stress
- normal operation at 1,200 rpm
- 125% of normal operation = 1,500 rpm
- 150% of normal operation = 1,800 rpm
- loss of shrink fit
- critical speed for ductile fracture
- critical speed for non-ductile fracture
- critical speed for excessive deformation

## 5 EVALUATION AND ANALYSIS

### 5.1 SHRINK FIT REQUIREMENTS

The amount of shrink fit that is required to achieve the criterion that the joint release speed is at least 150% of normal speed is determined by computing the radial deformation of the flywheel segments.

The radial displacement of the inside of the outer wheel due to 1,800 rpm is computed from Equation 7 to be 0.0152 inches.

The radial displacement of the outside of the hub due to 1,800 rpm is computed from Equation 8 to be 0.0011 inches.

Therefore, the required amount of shrink fit to preserve the connection between the hub and the outer wheel at 1800 rpm is a radial shrink of  $0.0152 - 0.0011 = 0.0141$  inches.

After the outer wheel and hub are joined, the assembly is shrink fitted onto the shaft.

The radial displacement of the inside of the assembly (the inside radius of the hub) due to 1,800 rpm is computed from Equation 7 with the dimensions of the wheel assembly to be 0.0106 inches.

The radial displacement of the outside of the shaft due to 1,800 rpm is computed from Equation 8 with the dimensions of the shaft to be 0.00015 inches.

Therefore, the required amount of shrink fit to preserve the connection between the shaft and the wheel assembly at 1,800 rpm is a radial shrink of  $0.0106 - 0.00015 = 0.0105$  inches.

Due to manufacturing tolerances, the applied shrink fit will be greater than the minimum computed.

The manufacturing tolerance between the hub and the shaft is [ ]<sup>a, c, e</sup>. The value for smoothing of the interface is [ ]<sup>a, c, e</sup> [6]. The manufacturing tolerance between the hub and the outer wheel is [ ]<sup>a, c, e</sup>. The value for smoothing of the interface is [ ]<sup>a, c, e</sup> [6]. Therefore, the nominal shrink fit between the outer wheel and the hub will be increased to [ ]<sup>a, c, e</sup> [6], and the shrink fit between the wheel assembly and the shaft will be increased to [ ]<sup>a, c, e</sup> [6]. The minimum values will also be considered for seismic loading.

## 5.2 WHEEL ASSEMBLY SHRINK FIT CONTACT PRESSURE

The contact pressure between the hub and the outer wheel is computed by Combining Equation 1 and Equation 4.

$$b - c = 0.0166 \text{ inches}$$

This results in a contact pressure,  $P = 9,095$  psi.

## 5.3 WHEEL ASSEMBLY SHRINK FIT STRESSES

The shrink fit stresses in the wheel assembly before attaching to the shaft are computed using Equations 2, 3, 5, and 6, and the contact pressure,  $P = 9,095$  psi.

The stresses as a function of radial position are shown in Figure 5-1.

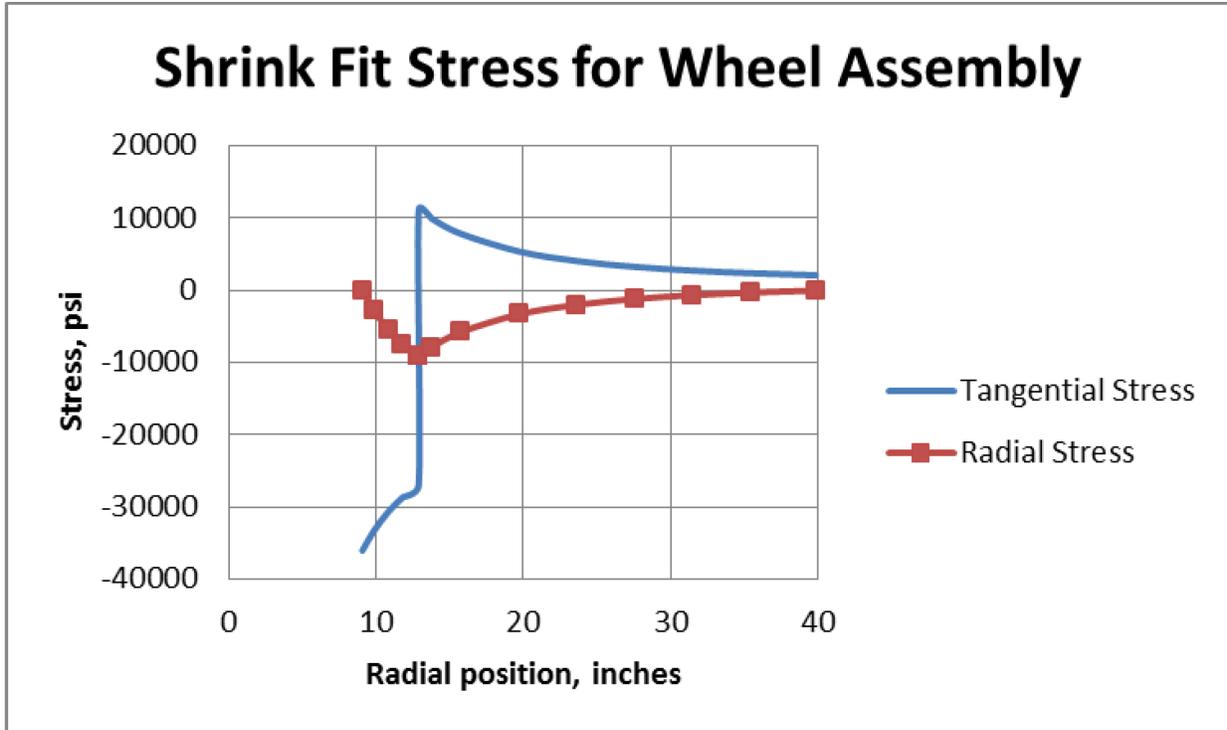


Figure 5-1: Stress due to Wheel Assembly

#### 5.4 ATTACH TO SHAFT SHRINK FIT CONTACT PRESSURE

The contact pressure due to the shrink of the wheel assembly onto the shaft is computed by combining Equation 1 for the wheel assembly and Equation 4 for the shaft:

$$d - e = 0.0138 \text{ inches}$$

This results in a contact pressure,  $P = 22,894$  psi

For this relationship the radial flexibility of the shaft is modified by a factor of  $[ \quad ]^{a, c, e}$  to estimate the decreased flexibility of the shaft due to the axial spread of the radial stress due to the contact pressure.

The result of using this factor was found to correlate well with the stress results in [6].

## 5.5 ATTACH TO SHAFT SHRINK FIT STRESSES

The stresses due to the shrink fit onto the shaft computed by Equation 2, Equation 3, and the contact pressure  $P = 22,894$  psi for the wheel assembly are shown in Figure 5-2. These stresses are added to the shrink fit stresses due to the assembly of the outer wheel and hub to be the total stresses at standstill, shown in Figure 5-3.

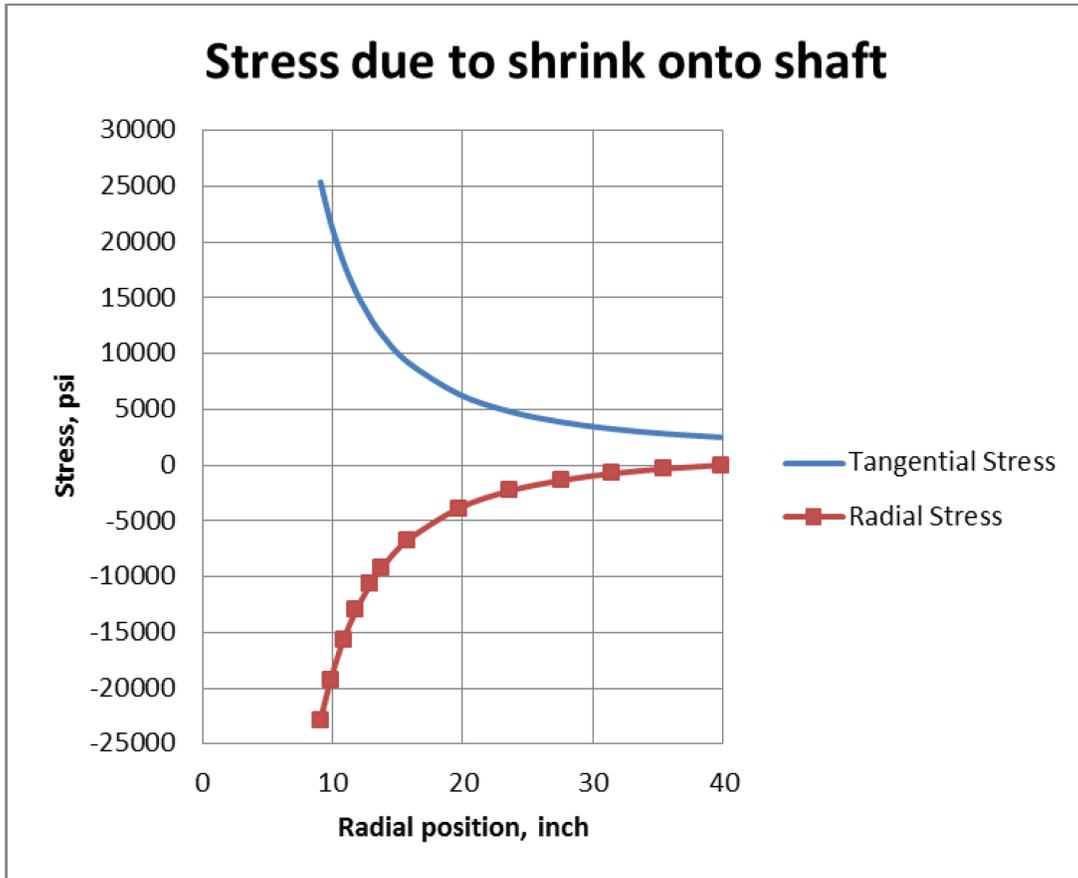
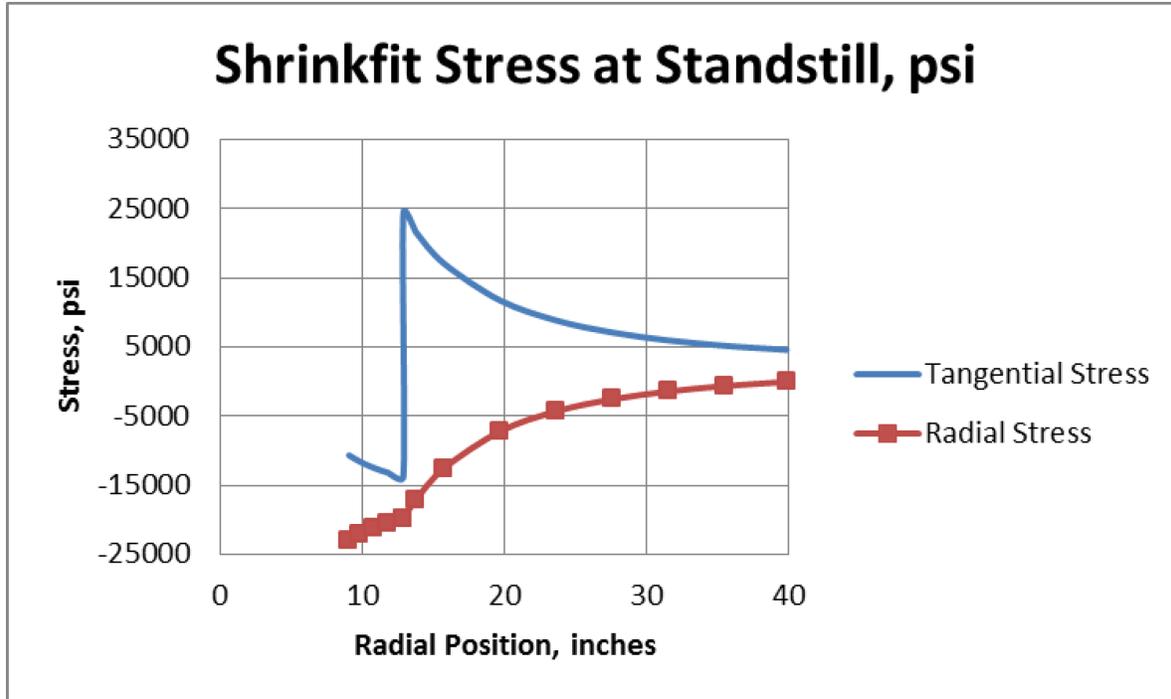


Figure 5-2: Stress due to Shrink Fit onto Shaft



**Figure 5-3: Total Stress at Standstill**

The radial contact stress between the hub and the outer wheel is -19,789 psi.  
The peak stress at the inside radius of the outer wheel is 24,418 psi.

## 5.6 STRESSES DUE TO ROTATION AT 1,200 RPM

The outer wheel and hub assembly will rotate as a unit as long as there is a compressive radial stress at the boundary between the parts. The stress in the assembly due to rotation will be combined with the shrink fit stress of the assembly. The rotation of the wheel assembly will reduce the contact with the shaft; the shrink fit stresses due to the contact with the shaft will be reduced due to the rotation of the shaft and the wheel assembly.

Using Equation 7 for the wheel assembly and Equation 8 for the shaft, the change in shrink fit at the shaft is computed for 1,200 rpm.

From Equation 7, the radial displacement of the inside of the wheel assembly at 1,200 rpm is 0.00469 inches.

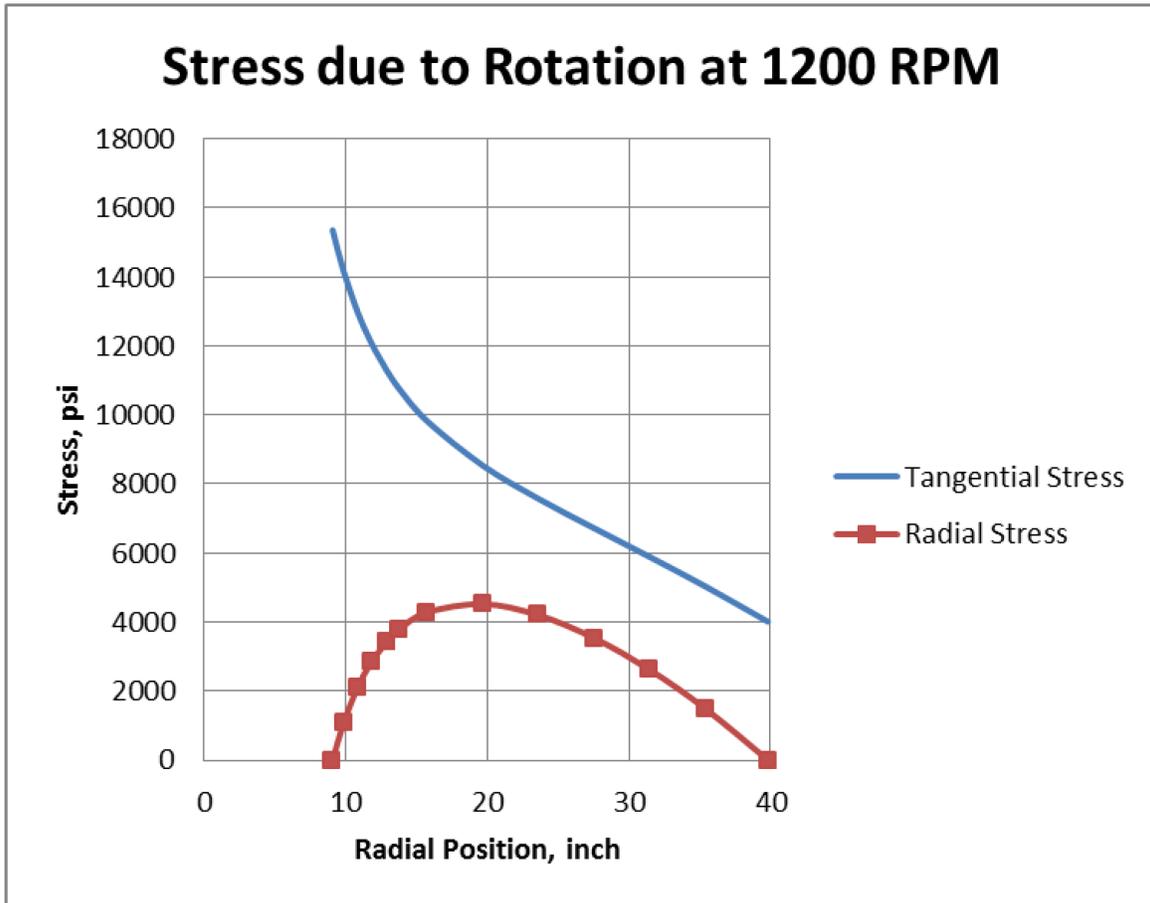
From Equation 8, the radial displacement of the outside of the shaft due to 1,200 rpm is 0.00005 inches.

Based on these displacements, the new contact pressure is obtained by combining Equation 1 and Equation 4:

$$\text{original shrink} - (d - e) = 0.0138 - (0.00469 - 0.00005) = 0.00916 \text{ inches}$$

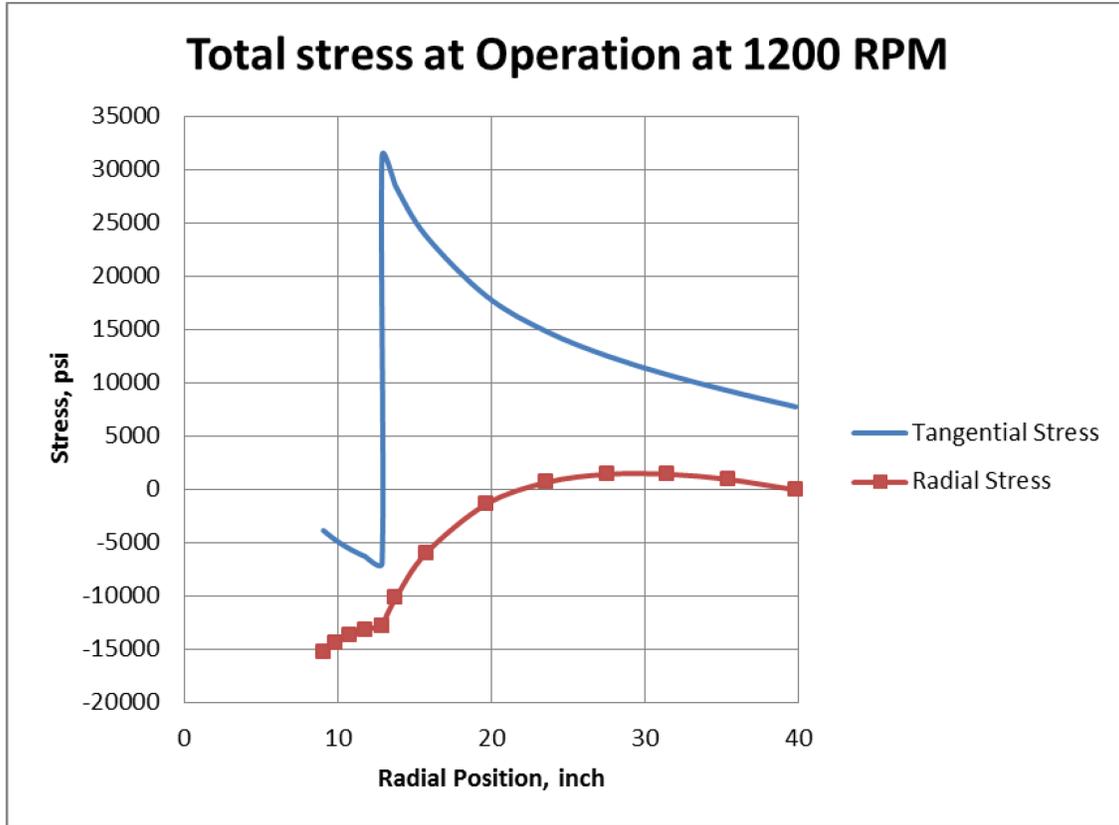
This results in a contact pressure of  $P = 15,203$  psi.

The tangential and radial stresses due to rotation are determined from Equations 9 and 10, respectively. These stresses are plotted for a rotation of 1200 rpm in Figure 5-4.



**Figure 5-4: Stress due to Rotation at 1,200 rpm**

The total stress at operation at 1,200 rpm is the combination of the stress due to rotation, the stress due to shrink fit of the assembly, and the reduced stress due to the reduced shrink fit at the shaft. The resulting stress is shown in Figure 5-5.



**Figure 5-5: Total Stress at 1,200 rpm**

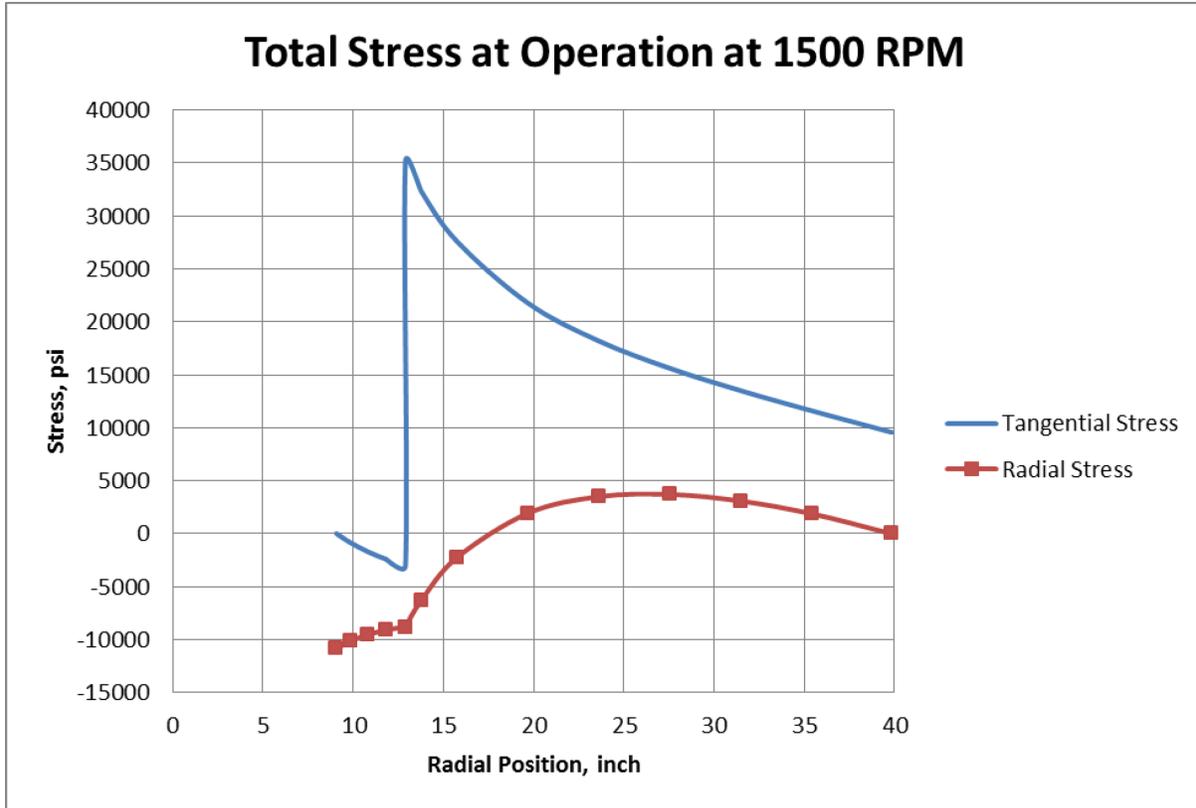
The radial contact stress between the hub and the outer wheel is -12,770 psi.

The peak stress at the inside radius of the outer wheel is 31,240 psi.

The stresses computed in [6] and converted to U.S. customary units are -12,707 psi and 30,722 psi. These results are very similar, and confirm the analysis in U.S. customary units.

## 5.7 STRESSES DUE TO ROTATION AT 1,500 RPM

Using the same procedure and equations as used in the previous section, the stresses at operation at 1,500 rpm are computed. These stresses are shown in Figure 5-6.



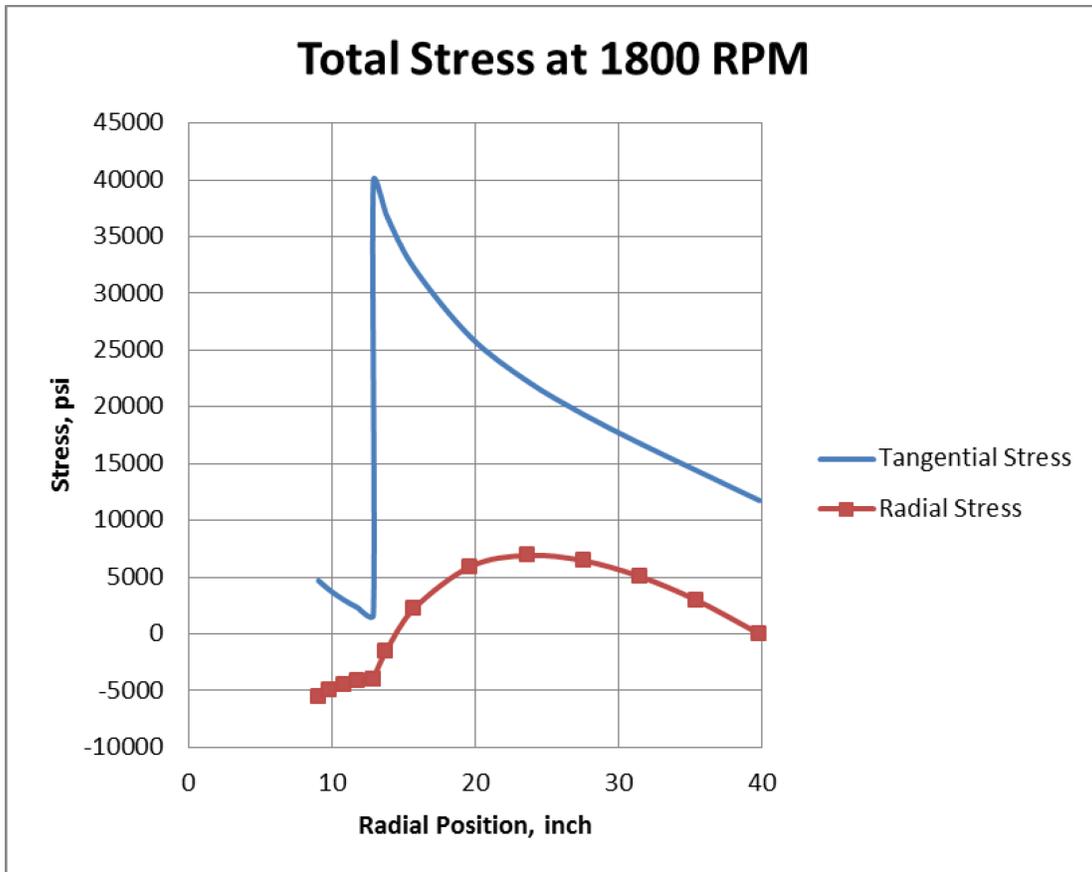
**Figure 5-6: Total Stress at Operation at 1,500 rpm**

The radial contact stress between the hub and the outer wheel is -8,722 psi.

The peak stress at the inside radius of the outer wheel is 35,078 psi.

## 5.8 STRESS DUE TO ROTATION AT 1,800 RPM

Using the same procedure and equations as used in the previous section, the stresses at operation at 1,800 rpm are computed. These stresses are shown in Figure 5-7.



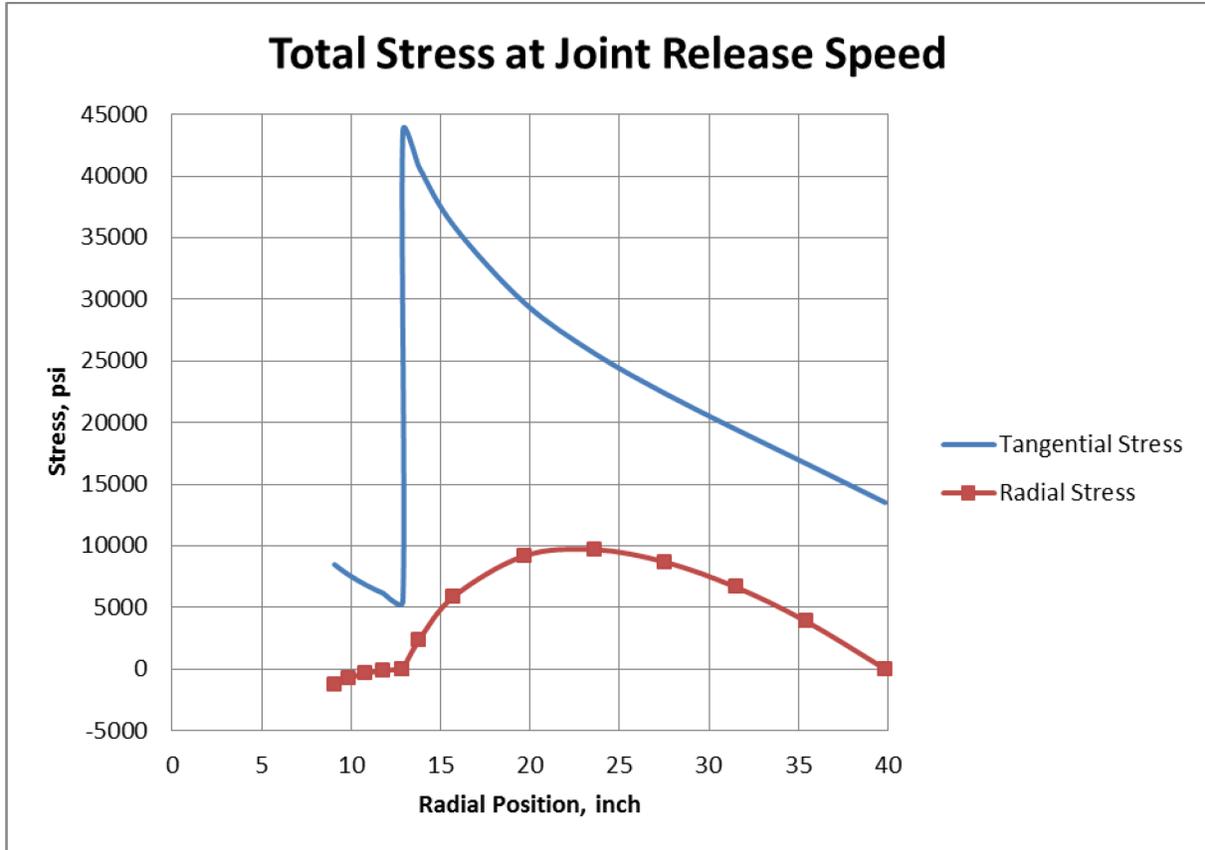
**Figure 5-7: Total Stress at 1,800 rpm**

The radial contact stress between the hub and the outer wheel is -3,997 psi.

The peak stress at the inside radius of the outer wheel is 39,768 psi.

## 5.9 JOINT RELEASE SPEED FOR PRESCRIBED SHRINK VALUES

The joint release speed is obtained by increasing the speed until either the contact pressure between the wheel assembly and the shaft is negligible, or until the contact pressure between the hub and the outer wheel is negligible. The joint between the hub and outer wheel becomes loose at 2,010 rpm for the nominal shrink fit values. The stresses are shown in Figure 5-8.



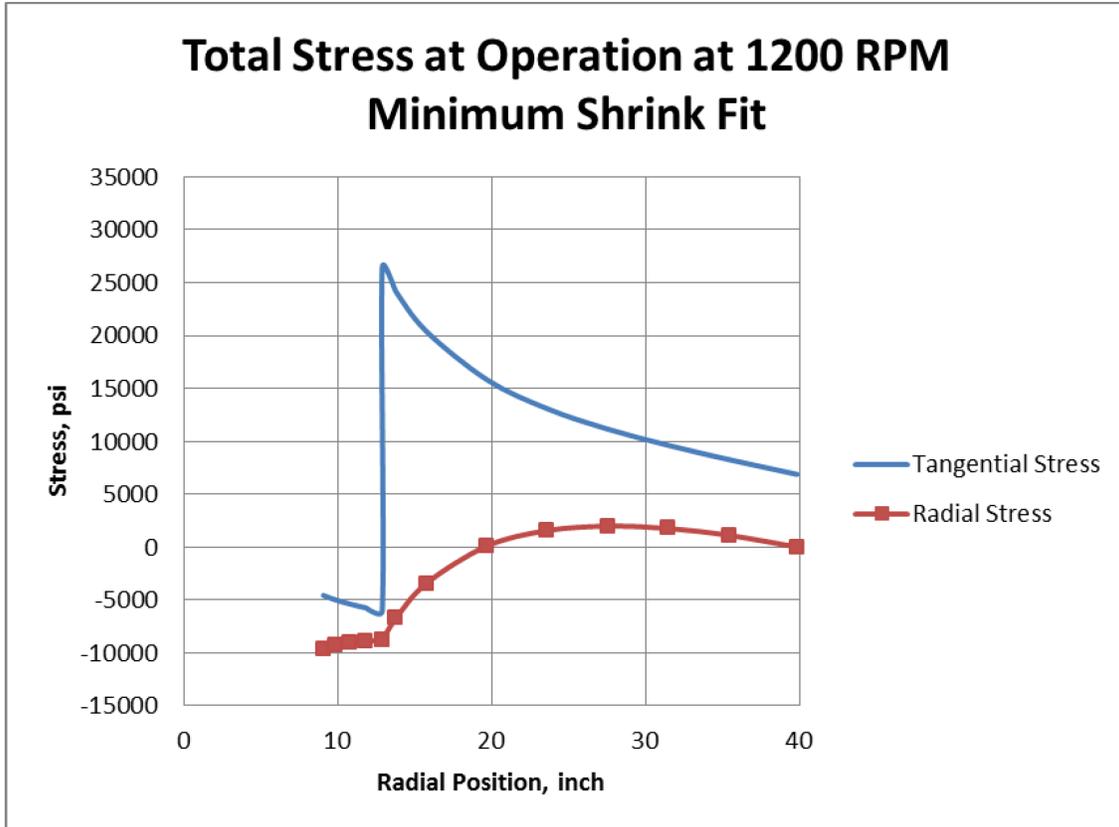
**Figure 5-8: Stresses at Joint Release Speed of 2,010 rpm**

The radial contact stress between the hub and the outer wheel is zero.

The peak stress at the inside radius of the outer wheel is 43,559 psi.

## 5.10 CONSIDERATION OF MINIMUM SHRINK FIT

Using the minimum values of shrink fit computed in Section 5.1, as well as the formulae and procedures of Section 5.6, the stresses at normal operation considering the minimum shrink fit of 0.0152 inches between the outer wheel and the hub, and 0.0105 inches between the shaft and the wheel assembly, are shown in Figure 5-9.



**Figure 5-9: Stress at 1,200 rpm with Minimum Shrink Fit**

The radial contact stress between the hub and the outer wheel is -8,823 psi.

The radial contact stress between the shaft and the hub is -9,659 psi.

The peak stress at the inside radius of the outer wheel is 26,370 psi.

## 5.11 EVALUATION OF STRESSES

Criterion 1: Total stresses in the flywheel at standstill and normal operating speed shall not exceed one-third of the minimum specified yield strength. The total stress for this application is defined as the von Mises stress =  $[1/\text{SQRT}(2)] * [(St - Sr)^2 + (St^2 + Sr^2)]^{1/2}$ .

The total von Mises stress at 1,200 rpm calculated in [6] is 266.65 N/mm<sup>2</sup> or 38,674 psi, which is less than one-third of the ultimate strength of 800/3 (N/mm<sup>2</sup>) = 38,677 psi.

The maximum total stress at standstill is 38,354 psi, which is less than the stress at 1,200 rpm.

Criterion 2: Total stress at design overspeed shall not exceed two-thirds of the minimum yield strength, where design overspeed is 125% of normal operating speed.

The maximum total stress at 125% of normal operation, 1,500 rpm = 40,156 psi, is less than two-thirds of the yield strength = 92,825 \* 2/3 = 61,883 psi.

Criterion 3: The joint release speed shall be equal to or greater than 150% of the normal operation speed. The total stress at the joint release speed shall not exceed the lower of one-half of the minimum specified ultimate strength or two-thirds of the minimum specified yield strength.

The maximum total stress at the joint release speed is 43,599 psi. This is less than one-half of ultimate stress of 116,000 psi = 58,000 psi, and less than two-thirds of the yield strength = 61,883 psi.

## 5.12 CONSIDERATION OF SEISMIC EVENT

The potential concern for seismic loads is a vertical (axial) force, which could exceed the friction resistance. (The friction resistance maintains the connection of the flywheel components.) The minimum friction forces at normal operation, the assumed starting point for a seismic event, would occur for the case of the minimum original shrink fit. Horizontal seismic acceleration has a negligible effect on the shrink fit of the flywheel assembly.

The radial contact stress between the hub and the outer wheel is -8,823 psi.

The radial contact stress between the shaft and the hub is -9,659 psi.

Assuming a coefficient of friction of 0.2, the total axial resistance to vertical movement of the flywheel assembly relative to the shaft is  $0.2 \times 9,659 \text{ psi} \times \text{thickness of the assembly} \times \text{circumference of the shaft}$  is 1,906,765 lbs.

The weight of the flywheel assembly is 23,174 lbs.

The ratio of the force to the weight is the acceleration required to slip,  $A_{\text{slip}} = 1,906,765 / 23,174 = 82.3$  times gravity. This is significantly above the seismic requirement of three times gravity specified in [1].

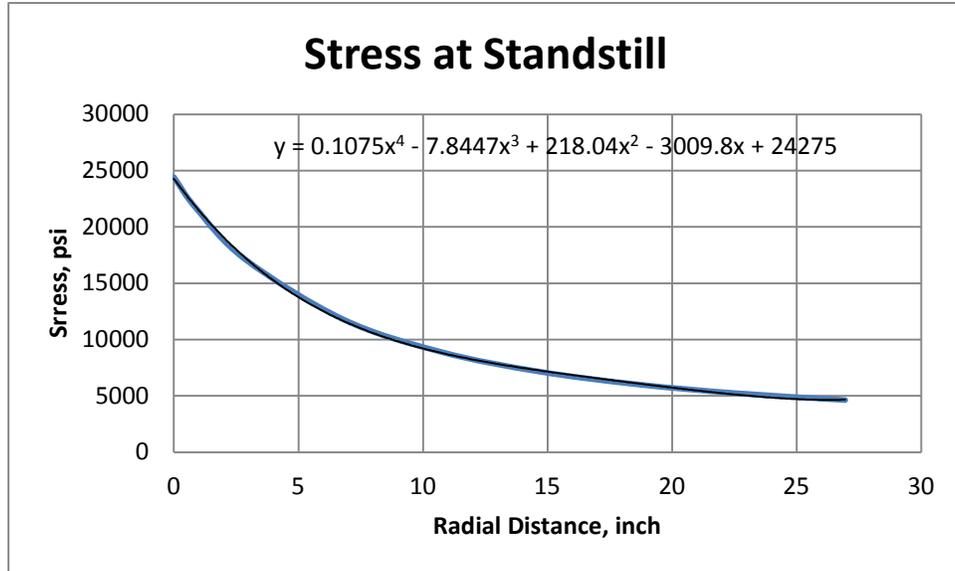
Similarly, since the connection between the hub and the outer wheel has a larger contact area, the acceleration required to slip is 107 times gravity. This is significantly above the seismic design requirement. Therefore, Criterion 4 is satisfied.

## 5.13 STRESS INTENSITY FACTOR OF CRACK AT HIGHEST STRESSED LOCATION

The stress intensity factor as a function of crack depth is computed from the stresses computed in the previous sections of this report. In all cases analyzed, the highest tensile stress is located on the inner radius of the outer wheel. The stress distribution from this point toward the outer edge of the wheel can be represented by the fourth order Equation 11, and the stress intensity factor due to this stress is computed using Equation 12.

### $K_I$ at Standstill

The tangential stress is taken from Figure 5-3. The stress is fit to a fourth order polynomial in Figure 5-10.



**Figure 5-10: Polynomial Tangential Stress at Standstill**

The radial extent of the outer wheel from the inner to outer radius is [ ]<sup>a, c, e</sup>. The stress, in Equation 11 form, is:

$$S_t = S_0 + S_1(x/t) + S_2(x/t)^2 + S_3(x/t)^3 + S_4(x/t)^4$$

$$[S = 24,275 - 3,009.8(26.95)(x/t) + 218.04(26.95^2)(x/t)^2 - 78,447(26.95^3)(x/t)^3 + 0.1075(26.95^4)(x/t)^4]$$

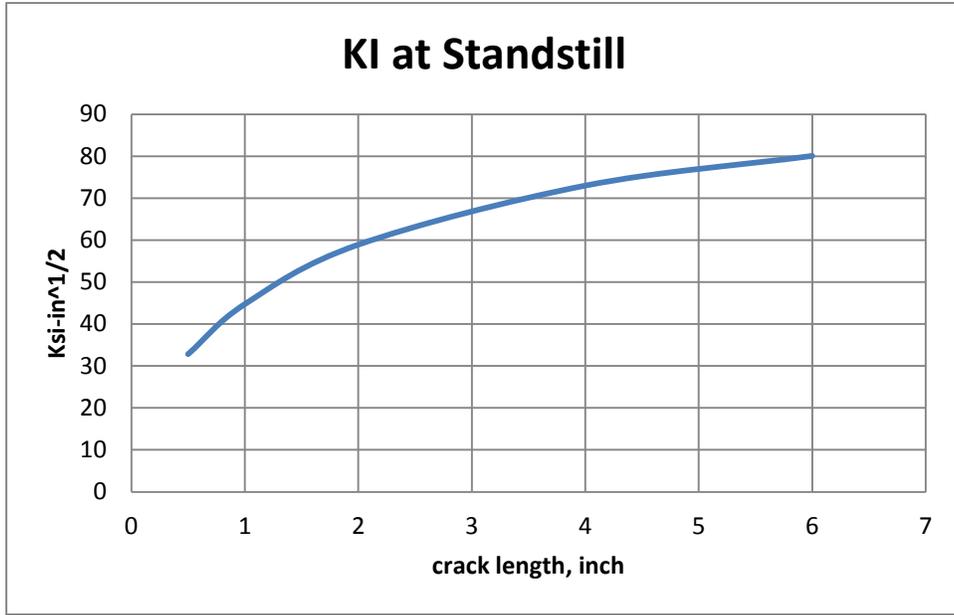
$K_I$  from Equation 12 is shown in Figure 5-11:

$$K_I = [G_0S_0 + G_1S_1(a/t) + G_2S_2(a/t)^2 + G_3S_3(a/t)^3 + G_4S_4(a/t)^4] \text{SQRT}(\dot{A}a)$$

In the previous equation:

a = crack length in radial direction

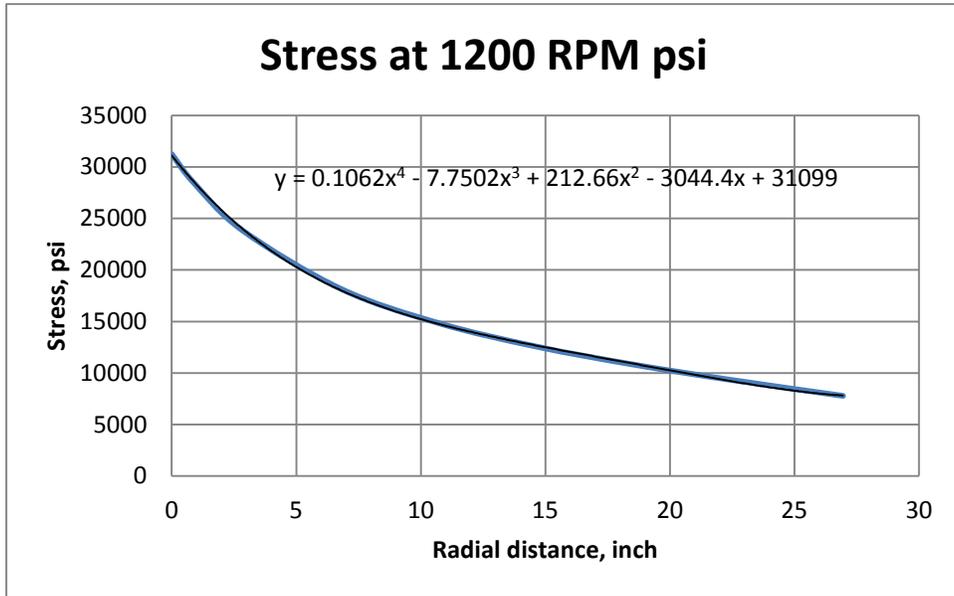
t = radial thickness of outer wheel



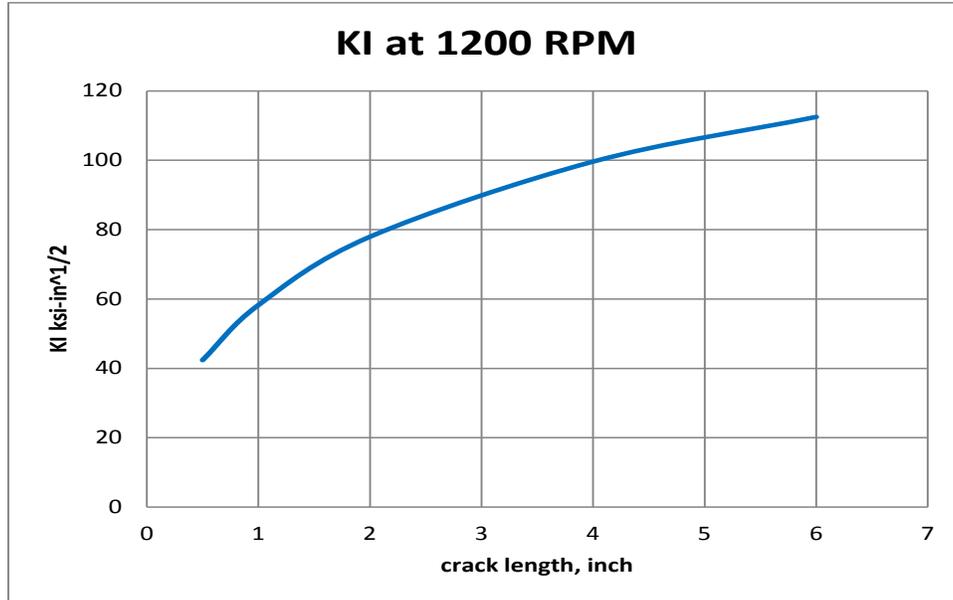
**Figure 5-11: Stress Intensity Factor versus Crack Length at Standstill**

K<sub>t</sub> at 1,200 rpm

The tangential stress is taken from Figure 5-5. The stress is fit to a fourth order polynomial in Figure 5-12.



**Figure 5-12: Polynomial Tangential Stress at 1,200 rpm**

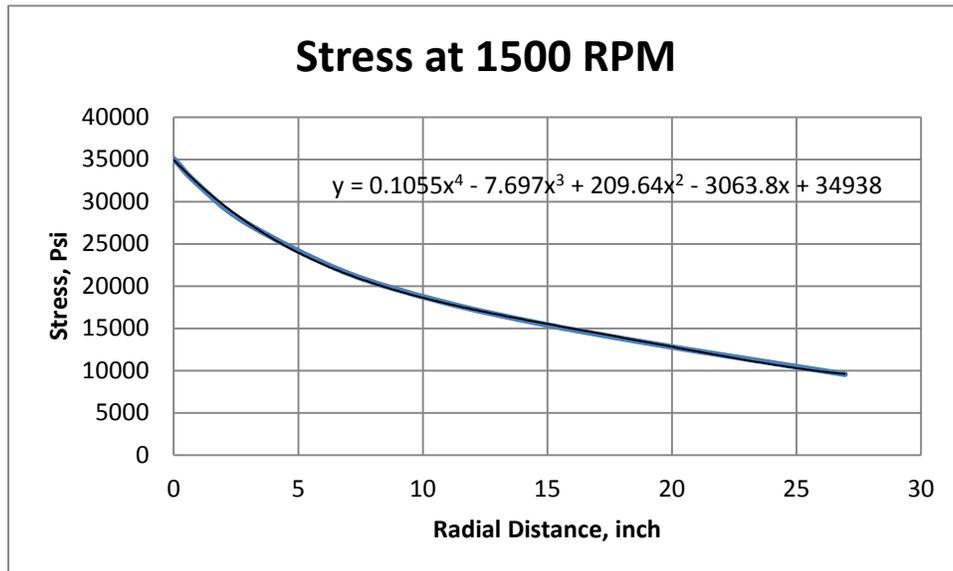


**Figure 5-13: Stress Intensity Factor versus Crack Length at 1,200 rpm**

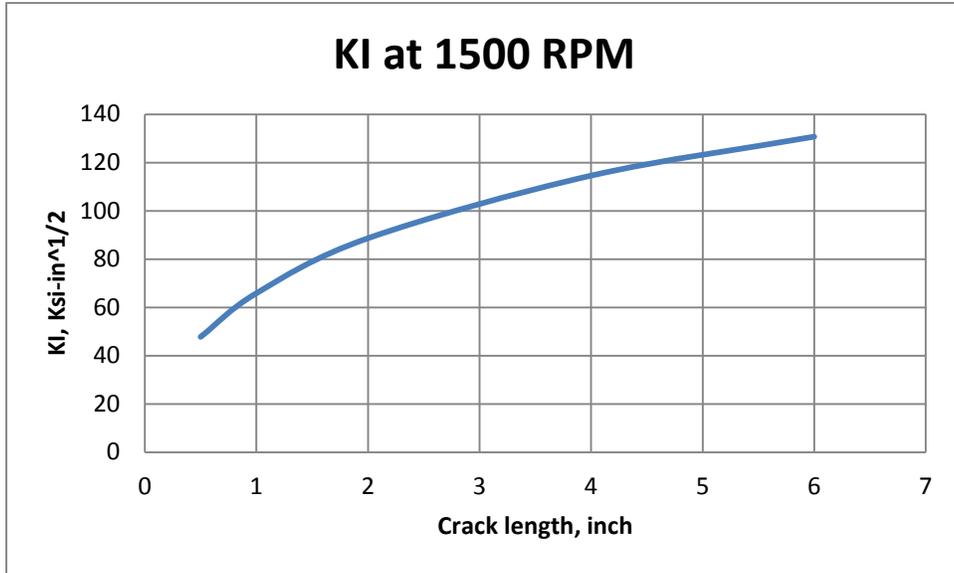
For a 0.5-inch crack,  $K_I = 42.4 \text{ ksi-in}^{1/2}$ . Therefore,  $K_{IC} / K_I = 3.54$ .

#### $K_I$ at 1,500 rpm

The tangential stress is taken from Figure 5-6. The stress is fit to a fourth order polynomial in Figure 5-14.



**Figure 5-14: Polynomial Tangential Stress at 1,500 rpm**

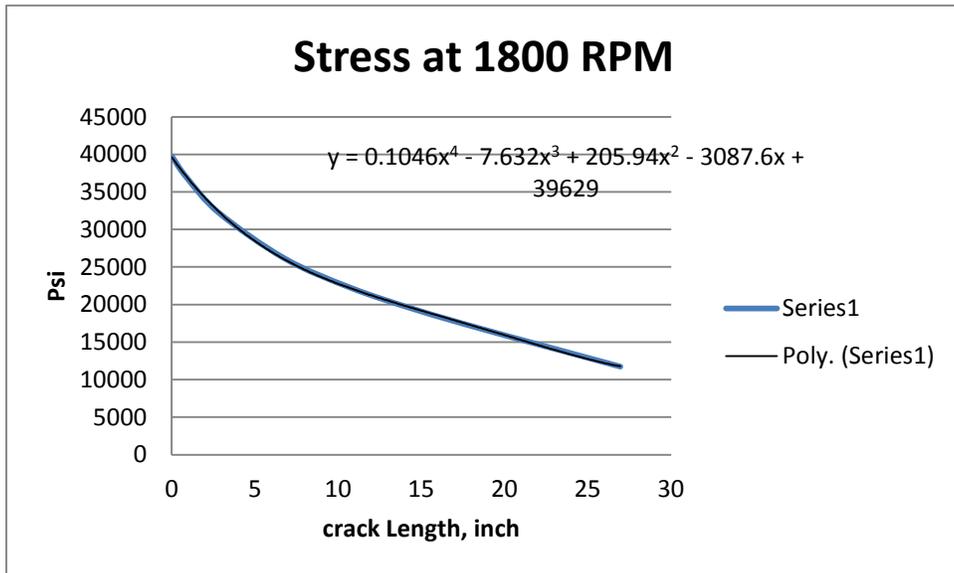


**Figure 5-15: Stress Intensity Factor versus Crack Length at 1,500 rpm**

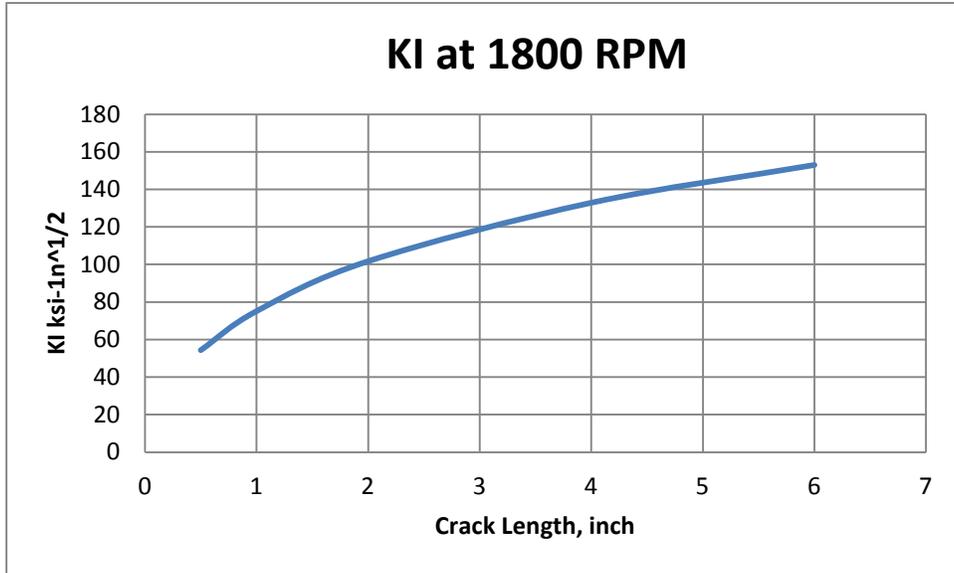
For a 0.5-inch crack,  $K_I = 47.8 \text{ ksi-in}^{1/2}$ . Therefore,  $K_{IC} / K_I = 3.14$ .

$K_I$  at 1,800 rpm

The tangential stress is taken from Figure 5-7. The stress is fit to a fourth order polynomial in Figure 5-16.



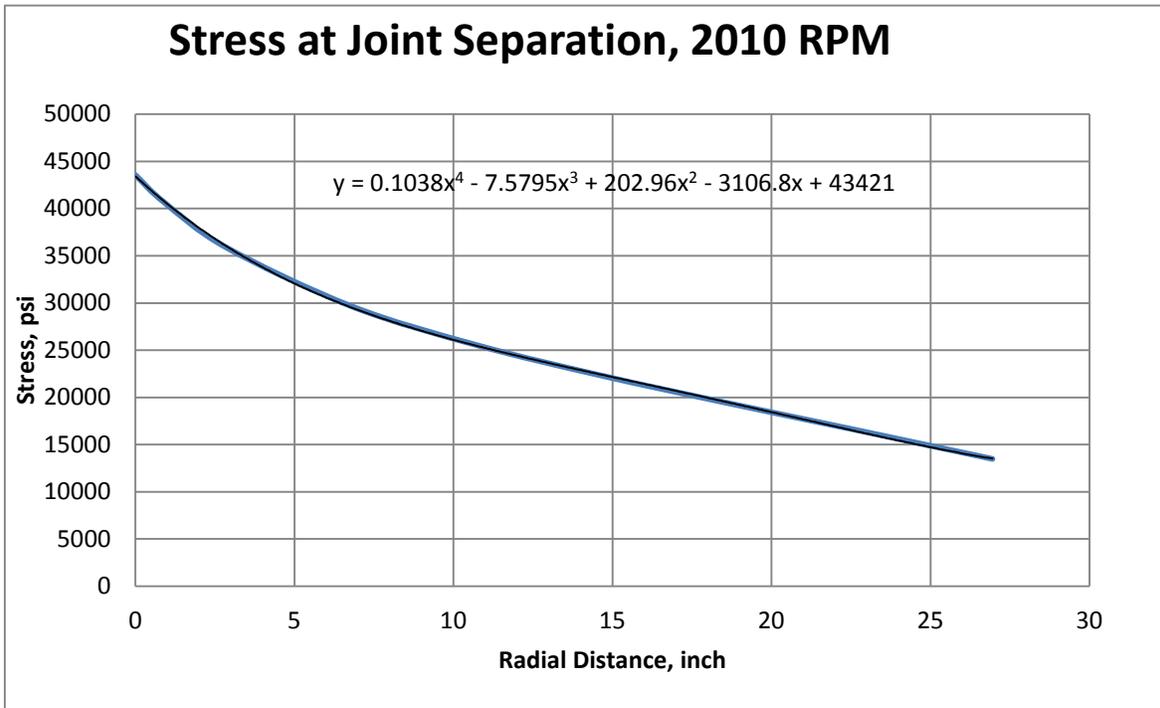
**Figure 5-16: Polynomial Tangential Stress at 1,800 rpm**



**Figure 5-17: Stress Intensity Factor versus Crack Length at 1,800 rpm**

K<sub>t</sub> at Joint Separation at 2,010 rpm

The tangential stress is taken from Figure 5-8. The stress is fit to a fourth order polynomial in Figure 5-18.



**Figure 5-18: Polynomial Tangential Stress at Joint Separation at 2,010 rpm**

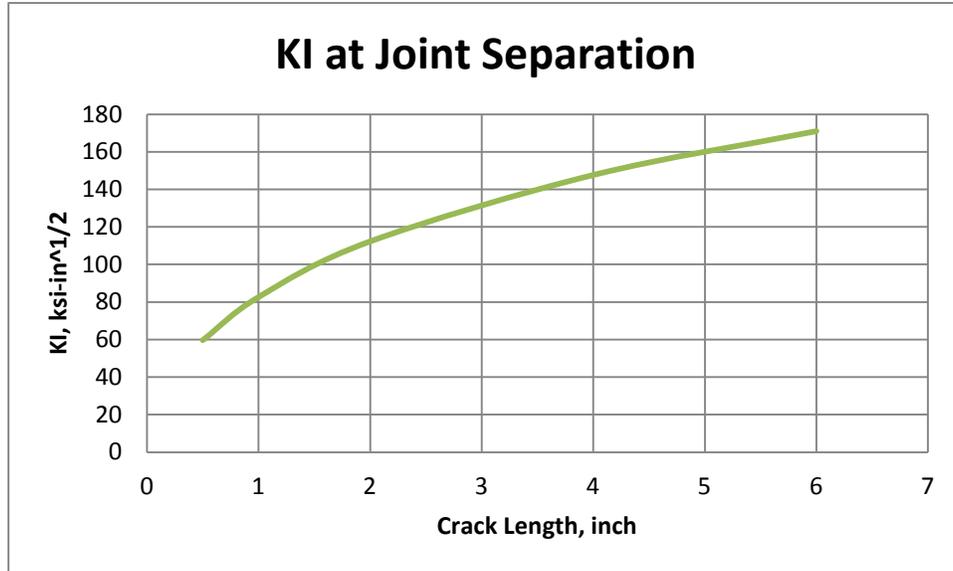


Figure 5-19: Stress Intensity Factor versus Crack Length at Joint Separation at 2,010 rpm

#### 5.14 EVALUATION OF INTEGRITY

Criterion 5: The critical speed for ductile fracture should be predicted.

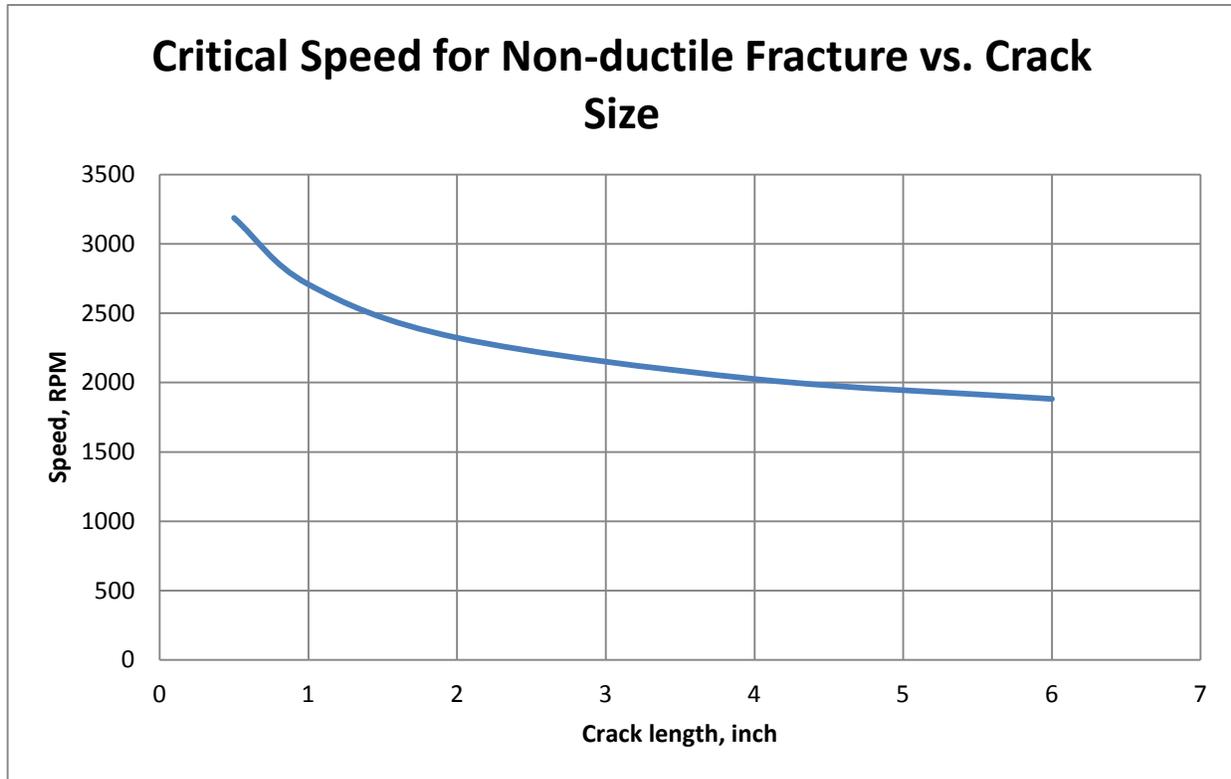
The critical speed for ductile fracture can be conservatively estimated by determining the speed at which the maximum stress reached 0.7 times the ultimate stress. This criterion is consistent with ASME Code Section III, Article F-1330 [5].

Figure 5-8 shows that, at separation speed (2,010 rpm), the maximum stress is 43,559 psi. Since the shrink fit stress is no longer present, the stress scales as the square of the rotation speed. Therefore, the speed at which the stress reaches 0.7 times the ultimate stress, (81,200 psi) is 2,744 rpm.

$$(2,010 \text{ rpm})^2: 43,559 \text{ psi} = (\text{critical rpm})^2: 81,200 \text{ psi}$$

$$\text{critical rpm} = 2,744$$

Criterion 6: The critical speed for non-ductile fracture should be predicted. To predict the critical speed for non-ductile fracture, a crack size must be hypothesized. For conservatism, a crack length of 0.50 inches is evaluated. The stress intensity factor at separation speed is shown in Figure 5-19. The rotation speed to reach the minimum  $K_I$  of 150 ksi-in<sup>1/2</sup> is plotted in Figure 5-20 as a function of crack size.



**Figure 5-20: Rotation Speed to Reach a  $K_I$  of 150 ksi-in<sup>1/2</sup>**

The critical speed for a crack length of 0.5 inches is 3,187 rpm.

Criterion 7: The normal speed should be less than one-half of the lowest critical speeds.

**Table 5-1: Normal and Critical Speed Comparison**

Critical Speed	Speed (rpm)	Normal Operation Speed/Critical Speed Ratio	Critical Speed/Joint Separation Speed Ratio
Ductile Fracture	2,744	0.44 < 0.5 Per Criterion 7	1.37
Non-ductile Fracture	3,187	0.38 < 0.5 Per Criterion 7	1.59
Excessive Deformation	2,934	0.41 < 0.5 Per Criterion 7	1.46

Note: Since critical speeds are greater than the separation speed of 2,010 rpm, these critical speeds are only hypothetical, and could not occur in real life.

Criterion 8: The predicted LOCA overspeed should be less than the lowest critical speed.

The design overspeed is 125% of normal operation = 1,500 rpm. This is less than all critical speeds.

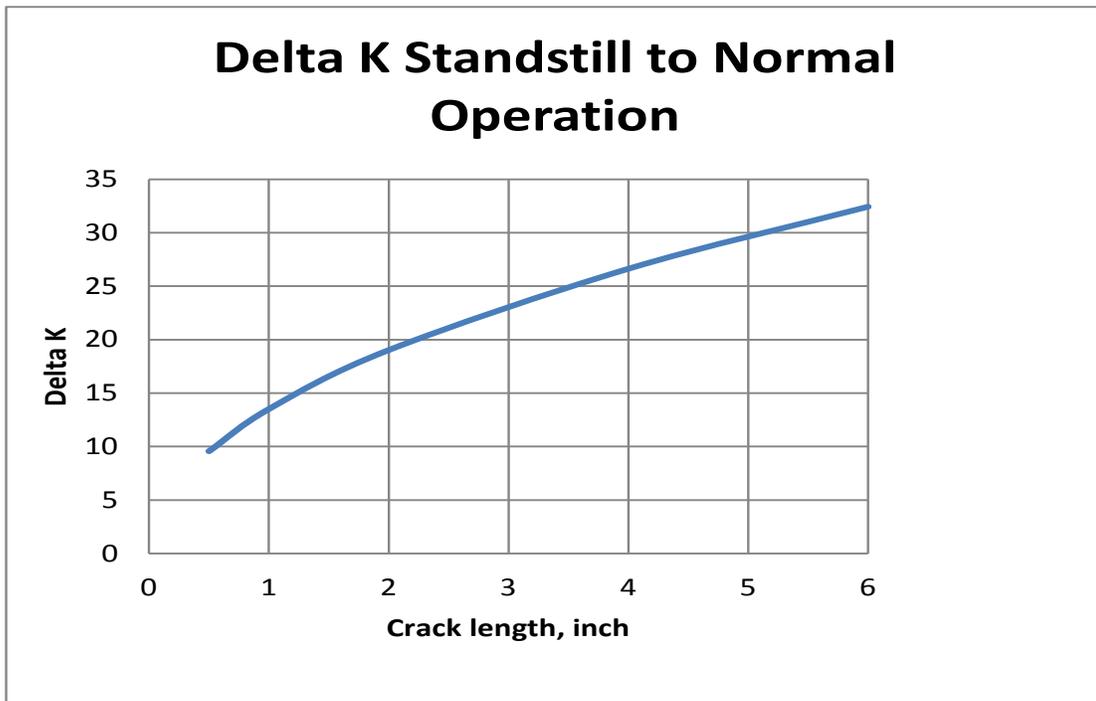
Criterion 9: The critical speed for excessive deformation of the flywheel should be predicted.

The critical speed for excessive deformation can be conservatively estimated by determining the speed at which the maximum stress reached the yield stress.

Figure 5-8 shows that, at separation speed (2,010 rpm), the maximum stress is 43,559 psi. Since the shrink fit stress is no longer present, the stress scales as the square of the rotation speed. Therefore, the speed at which the stress reaches the yield stress (92,825 psi) is 2,934 rpm. Since this is greater than the joint separation speed, it is a hypothetical value that is impossible to be reached. Therefore, excessive deformation at overspeed conditions (i.e., 1,500 rpm) will not occur.

### 5.15 FATIGUE CRACK GROWTH

The fatigue crack growth due to 6,000 cycles from standstill to normal operation can be predicted by the fatigue crack growth rates available in [5]. The delta  $K_I$  from standstill to normal operation is determined by comparing the  $K_I$  in Figure 5-11 and Figure 5-13. The delta  $K_I$  is shown in Figure 5-21.



**Figure 5-21: Change in  $K_I$  from Standstill to Normal Operation**

Using the formulation in [5]:  $da/dn = C_0 (K_I)^n$

In the previous equation:

$$C_0 = 1.99 \times 10^{-10} S$$

$$S = 25.72 \times (2.88 - R)^{-3.07}$$

$$n = -3.07$$

$$R = K_{Imin}/K_{Imax}$$

A crack grows from 0.5 inches to 0.5032 inches in 6,000 cycles from standstill to normal operation. This is negligible growth.

## 6 RESULTS AND CONCLUSIONS

This report demonstrates that the APR1400 reactor coolant pump (RCP) flywheel satisfies all of the RCP flywheel integrity criteria of the design specification [1] and U.S. NRC Regulatory Guide 1.14 [2]:

The acceptance criteria in the specification [1] include:

1. The total stress in the flywheel at standstill and normal operating speed does not exceed one-third of the specified ultimate strength. This criterion is met.
2. The total stress at design overspeed does not exceed two-thirds of the minimum yield strength, where design overspeed is 125% of normal operating speed. This criterion is met.
3. The total stress at joint release speed does not exceed the lower of one-half of the minimum specified ultimate strength or two-thirds of the minimum specified yield strength. The joint release speed is 2,010 rpm, which is greater than 150% of normal operation speed. This criterion is met.

The Regulatory Guide acceptance criteria are established in [2] as:

4. Flywheel assembly is designed to withstand normal conditions, anticipated transients, the design basis loss of coolant accident (LOCA), and the safe shutdown earthquake without loss of structural integrity.
5. The critical speed for ductile fracture is 2,877 rpm.
6. The critical speed for non-ductile fracture is 3,187 rpm.
7. The normal speed is less than one-half of the lowest critical speeds for fracture. This criterion is met.
8. The LOCA overspeed of 1,500 rpm is less than the lowest critical speeds. This criterion is met.
9. Excessive deformation can be conservatively defined as total stress reaching material yield strength,  $S_y$ . The shrink fit release speed is 2,010 rpm, with a maximum stress of 43,559 psi. This is much less than the  $S_y$  of 92,825 psi. The hypothetical speed at  $S_y$  is 2,934 rpm, which will not occur because the shrink fit will separate at 2,010 rpm. Additionally, the separation speed of 2,010 rpm is significantly greater than the LOCA overspeed of 1,500 rpm. Consistent with past practice [7], the excessive deformation criterion is considered satisfied.

## 7 REFERENCES

1. KEPCO Design Specification, 11A60-FS-DS485, Rev. 02, "Design Specification for Reactor Coolant Pump Motors," November 14, 2014.
2. US Nuclear Regulatory Commission Regulatory Guide 1.14, Rev. 1, "Reactor Coolant Pump Flywheel Integrity," August 1975.
3. Young, Warren C. and Richard G. Budynas, "Roark's Formulas for Stress and Strain," Seventh Edition, McGraw-Hill Companies, Inc., New York, NY, 2002.
4. API 579-1/ASME FFS-1, "Fitness-For-Service," Annex C, "Compendium of Stress Intensity Factor Solutions," June 5, 2007.
5. ASME Boiler and Pressure Vessel Code, Section III, Division 1, Rules for Construction of Nuclear Facility Components, 2007 Edition with 2008 Addenda, Section XI, Appendix A 4300.
6. Siemens Document, 4D5.0170.83-575711F, Rev. F, "Flywheel Calculation," May 30, 2011.
7. Westinghouse Report, WCAP-15666-A, Rev. 1, "Extension of Reactor Coolant Pump Motor Flywheel Examination," October 2003.