Influence of Elastic and Plastic Contact Models on the Overall Thermal Resistance of Bolted Joints

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ABSTRACT — A bolted joint, which operates in vacuum is analyzed in this work. A resistance network, composed of material, contact and constriction resistances is presented for the overall thermal resistance of bolted joints. Thermal stress theory is used to determine contact stress variations with temperature level. Dimensional and non-dimensional forms of the thermal resistances and of the thermal stress models are given. Two contact resistance models, one based on the elastic and the other on the plastic deformation of contacting asperities, are used and their impact on the overall thermal resistance is studied.

INTRODUCTION

With the improvements in the performance level of the electronic devices, which lead to higher component and package densities, thermal engineers are constantly faced with problems of heat dissipation. For the thermal design of modern electronic devices, several parameters are necessary, including the physical properties of the materials, the thermal resistances between contacting surfaces and the heat flux constriction resistances. A large amount of research has been performed to investigate these parameters.

In many applications, the electronic devices are installed inside boxes, which are connected to panels through bolted joints. The heat dissipated by these devices flows through the bolted joints before being lost to the environment. Therefore the overall thermal resistance of bolted joints is important. A general procedure for modelling this overall resistance has been developed by Mantelli [1], and was applied for junctions operating in a vacuum. The overall thermal resistance of bolted joints consists of several material and contact resistances, combined in series and in parallel. Several models are available for the determination of the contact resistance between conforming surfaces, which can be classified according to the type of deformation: plastic or elastic. In this work the influence of the use of the two types of models, in the overall thermal resistance of bolted joints, are studied.

LITERATURE REVIEW

The determination of the thermal resistance of bolted joints is not a simple task, since it includes the understanding of the thermal behavior of the joint components, the mechanical characteristics of the contacts and the metrological aspects of the contacting surfaces. Several studies attempted to determine the pressure distribution between plates joined by a bolt [2,3,4,5,6] and others concentrated on the thermal aspects, as several experimental works [2,7,8]. Some correlations were developed [9,10], some analytical results for constriction resistance of joint components [10,11] but few analytical studies of the overall thermal resistance have been done in this area. Recently, Maddren [12] compared his experimental data for the contact resistance between two flat surfaces with the predictions of models that employed elastic and plastic theory. He concluded that the elastic theory was more accurate. In this work, the influence of the type of model in the overall thermal resistance of bolted joints is investigated.

ANALYTICAL MODELLING

Problem Definition

The bolted joint analyzed in this work is composed of two thin aluminum plates of similar thicknesses connected by a stainless steel bolt, as shown in the schematic in Fig. 1. Three washers are positioned between the plates, one washer between the bolt head and the upper plate and another one between the nut and lower plate. All washers are identical and made of stainless steel. A vacuum environment is assumed, therefore convection heat transfer is considered negligible. This mounting is typical for satellite applications and the washers between plates are usually a requirement of the structural design.

Overall Thermal Resistance Model

The analogy between thermal and electrical circuits is used to model the overall thermal resistance of bolted joints. Heat is supplied to the upper plate and dissipated from the lower plate

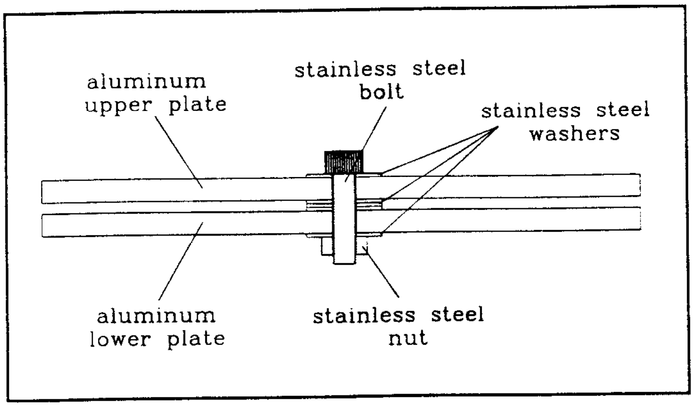

Fig. 1. Typical Satellite Bolted Joint.

(see Fig 2). After the heat is constrained to the washer area, it can follow one of the two paths: either through the washers between the plates or through the bolt. Previous studies by Mantelli [1] showed that the bolt path and all the radiation resistances can be neglected with an error of around 5%. Therefore, the resulting thermal circuit is composed of constriction, material and contact resistances in series, as shown in Fig. 2. The description of the thermal resistances which appear in this figure is shown in Table 1.

The material resistance is determined by means of the well known one-dimensional heat flux thermal resistance expression:

Table 1. Description of the Resistances of the Network

Thermal Resistance	Description
R_{ct1}	constriction resistance of plate 1
R_{mp1}	material resistance of the upper plate
$R_{cp1,1}$	contact resistance between the upper plate and washer
R_{mwi}	material resistances of the washers between plates (i varies from 1 to n, where n is the number of washers)
$R_{cwi,i+1}$	contact resistance between washers
$R_{cwn,p2}$	contact resistance between the washer and lower plate
R_{mp2}	material resistance of plate 2
R_{ct2}	constriction resistance of plate 2

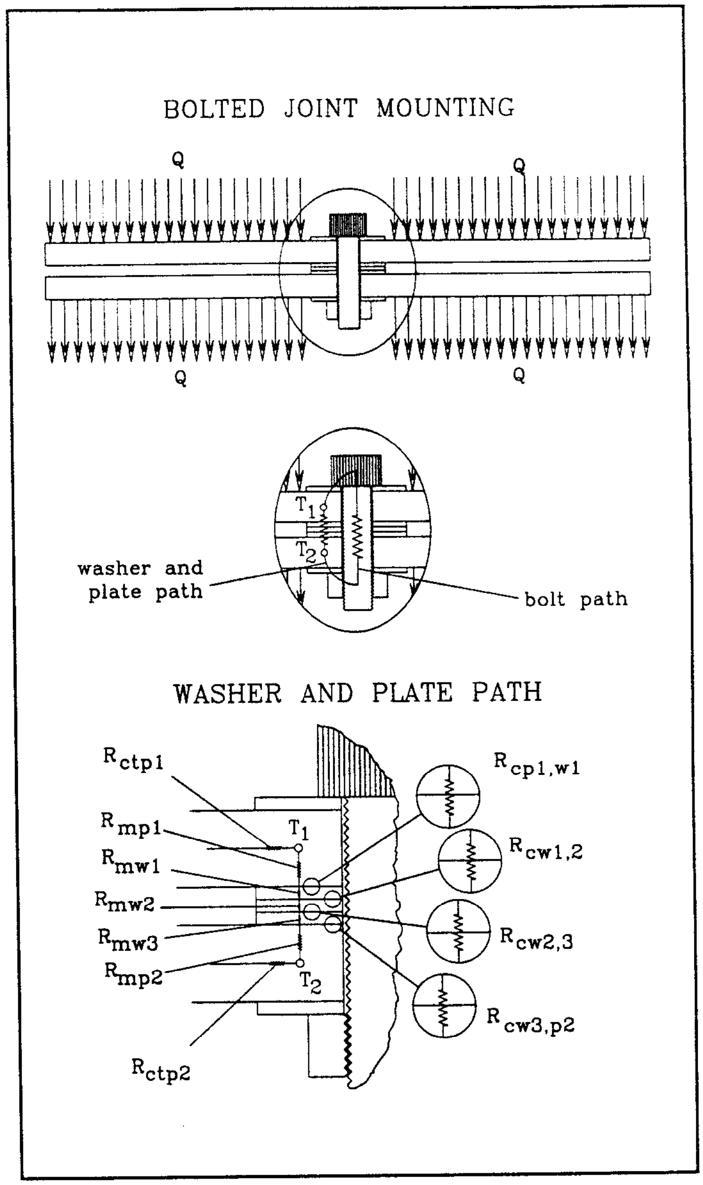

Fig. 2. Resistance Network.

$$R_m = \frac{L}{k\,A} \tag{1}$$

where L is the heat path length in the bolted joint axial direction, k is the conductivity of the material and A is the heat transfer area.

Figure 3 shows a schematic of the upper plate and its boundary conditions. The mathematical model was developed by Mantelli [1] and the final equation is:

$$R_{ct} = \frac{1}{8 \pi k L} \frac{(c^2 + b^2)}{(c^2 - b^2)} + \frac{C_1}{q \pi (c^2 - b^2)^2} \left[b^2 \left(\ln b - \frac{1}{2} \right) - c^2 \left(\ln c - \frac{1}{2} \right) \right] + \frac{C_2}{q \pi (c^2 - b^2)} \tag{2}$$

where a and b are the inside and outside radii of the washer, c is the outside radius of the plate, q is the heat flux, and C_1 and C_2 are the integration constants, given by:

$$C_1 = \frac{q c^2}{2 k L} \tag{3}$$

and:

$$C_2 = \frac{\frac{q}{2 k L} \left(\frac{c^2}{b} - b \right)}{m \left(\frac{K_1 (ma) I_1 (mb)}{I_1 (ma)} - K_1 (mb) \right)} \left(K_0 (mb) - \frac{K_1 (ma) I_0 (mb)}{I_1 (ma)} \right) + \frac{q}{2 k L} \left(\frac{b^2}{2} - c^2 \ln b \right) \tag{4}$$

K and I are the modified Bessel functions of first and second kind of order 0 or 1, respectively.

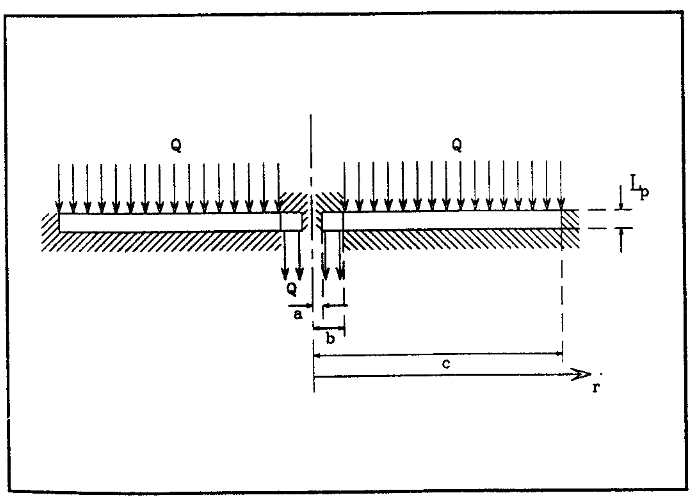

Fig. 3. Physical Model of the Plate Constriction Resistance.

Thermal Contact Resistance Models

Over the past 30 years, several thermal contact resistance models were developed for isotropic conforming rough surfaces in contact under vacuum. They can be divided into models that considered elastic and those that considered plastic deformation of the contacting asperities. The elastic model assumes that the contacting asperities, after unloading, return to their original shape and the plastic model assumes that the asperities are permanently deformed.

A thermal contact resistance model is composed of three different models: thermal, deformation and surface geometry. In 1966, Greenwood and Williamson [13] developed a model for the deformation of contacting asperities, which they considered elastic. In 1970, Whitehouse and Archard [14] developed a model similar to the Greenwood and Williamson's model for the elastic deformation. In 1969, Cooper et. al.[15] proposed a complete model for the conductance of contacting asperities undergoing plastic deformation but differing in surface geometry aspects. Later, in 1974, Mikic [16] adapted this model for elastic deformation. In 1982, Yovanovich [17] developed a correlation of the Cooper et. al. [15] model, in which a non-dimensional conductance was defined as a function of the pressure and of the softer contacting material bulk hardness. In 1982 and 1983, Yovanovich et. al. [18,19] and Yovanovich and Hegazy [20] realized that the correlation precision increases significantly if one uses the appropriated microhardness instead of bulk hardness. In 1988, Song and Yovanovich [21] developed an explicit expression to calculate the appropriate value of the microhardness and their expression will be used in the present work, for the plastic contact model. In their paper, Sridhar and Yovanovich [22], (1993) present a comparison of all these models. In 1994, Sridhar and Yovanovich [23] developed an elastoplastic model, that covers the entire range of contact deformation behavior, from elastic to plastic. Since an explicit expression to determine dimensionless contact pressure is unavailable, their model should be used only in the case where the surface deformation is partially elastic and partially plastic, and where the use of one of the plastic or elastic models lead to large errors. In this work, the Song and Yovanovich's [21] expression is used for the plastic and Mikic's [18] expression for the elastic deformations, because they involve less number of surface parameters, and therefore are easier to work with.

According to Mikic [16], the resistance between contacting surfaces, based on the elastic deformation of the asperities, can be calculated through the expression:

$$\frac{\sigma}{m} \frac{h_c}{k_s} = 1.54 \left(\frac{P}{H_e} \right)^{0.94} \tag{5}$$

where σ is the RMS surface roughness heights of the contacting surfaces 1 and 2 ($\sigma^2 = \sigma_1^2 + \sigma_2^2$), m is the effective mean surface slope ($m^2 = m_1^2 + m_2^2$), h_c is the contact conductance, k_s is the harmonic mean thermal conductivity $(k_s=2k_1k_2/(k_1+k_2))$ and P is the nominal contact pressure. H_e is given by:

$$H_e = \left[\frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2}\right]^{-1} \frac{m}{\sqrt{2}} \quad (6)$$

where E_1 and E_2 are the modulus of elasticity of the materials 1 and 2, and ν_1 and ν_2 are the Poissons ratio of materials 1 and 2.

According to Song and Yovanovich [21], the resistance between contacting surfaces whose asperities are deformed plastically can be obtained from the expression:

$$\frac{\sigma}{m}\frac{h_c}{k_s} = 1.25\left(\frac{P}{H_p}\right)^{0.95} \quad (7)$$

where P/H_p is given by:

$$\frac{P}{H_p} = \left[\frac{P}{c_1\left(1.62\,\frac{\sigma}{m}\right)^{c_2}}\right]^{\frac{1}{1+0.071\,c_2}} \quad (8)$$

and where c_1 and c_2 are the Vickers correlation coefficients of the softer material.

The contact resistance is determined from:

$$R_c = \frac{1}{h_c\,A} \quad (9)$$

According to Fig. 2, the overall thermal resistance R_t of the bolted joint under investigation is calculated from the expression:

$$R_t = R_{ctp1} + R_{mp1} + R_{cp1,w1} + R_{mw1} + R_{cw12} + R_{mw2} + R_{cw23} + R_{mw3} + R_{cw3p2} + R_{mp2} + R_{ctp2} \quad (10)$$

where R_m (material resistances) are determined from Eq. 1, R_{ct} (constriction resistances) from Eq. 2 and the R_c (contact resistances) from Eq. 9, where h_c is obtained from Eq. 5 or 7.

Thermal Stress Model

It has been observed by Mantelli and Basto [24] and Mantelli and Yovanovich [25] that if a bolted joint is assembled at room temperature T_0 and submitted to a different operating temperature T_m, the initial stress applied to the mounting can vary significantly, according to the material properties and initial torque. The decrease of the pressure of a bolted joint can be a critical situation for many applications, as for satellites. Thermal stress theory was applied to model the influence of the temperature on the contact stress between elements of a bolted joint. Fig. 4 presents the physical model, where the following simplifications have been made:

1- The bolt and nut are considered a single body, since they are made of the same material and are firmly fastened.

2- The outside diameter of the plates, washers, bolt head and nut are considered equal to the washer outside diameter.

3- The contact pressure is assumed to be uniformly distributed over the contacting surfaces.

4- No radial forces are considered.

5- The bolt traction forces are assumed to act in the upper and lower ends of the bolt shaft.

6- The variation of the properties with temperature is considered linear, for the bolted joint operating temperature range. Therefore, as the studied bolted mounting (see Fig. 1) presents an axial symmetry with relation to the middle washer, the same mean temperature T_m is used for the determination of the properties of the different types of materials of the mounting.

The final expression of the stress P (see reference [1]), for the mounting under investigation is:

$$P = \frac{2\,L_p\,(\alpha_b - \alpha_p)\,(T_m - T_0)}{\frac{1}{E_b}\left[4\,L_w\left(1 - \frac{(b^2-a^2)}{r_b^2}\right) - 2\,\frac{(b^2-a^2)}{r_b^2}\,L_p + 2\,L_p\,\frac{E_b}{E_p}\right]} + P_i \quad (11)$$

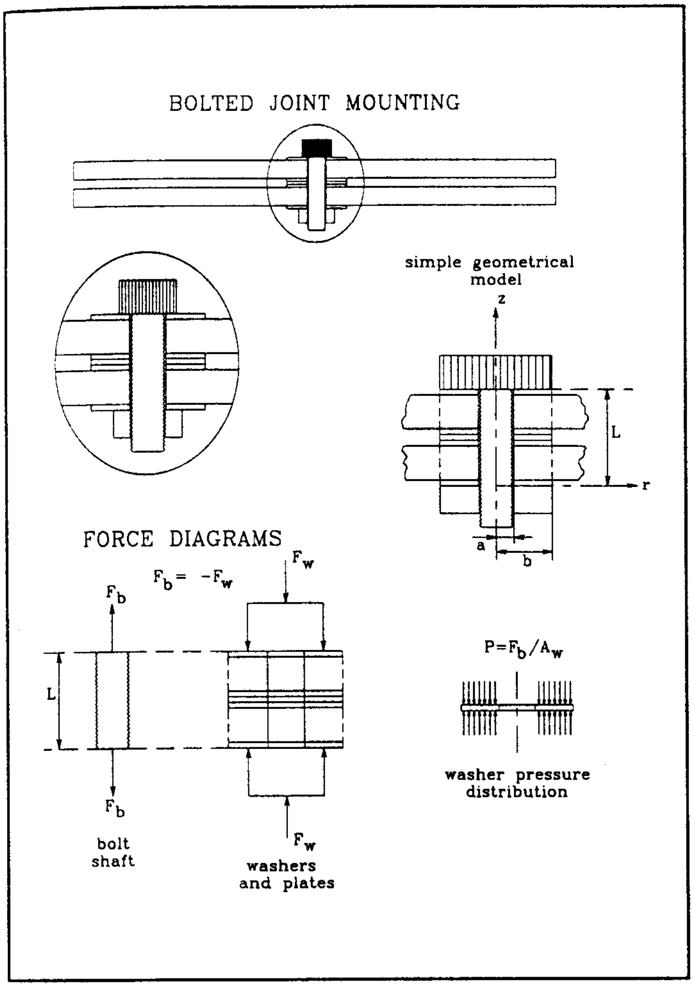

Fig. 4. Physical Model for Thermal Stress Analysis.

where P_i is the initial stress (at room temperature), α is the coefficient of thermal expansion, E is the elasticity modulus, r_b is the bolt shaft radius, and the subscripts p and b refer to the plate and bolt material respectively.

Non-dimensional Form of Equations

The use of the non-dimensional form of the equations is very convenient, because the results are more general. In this section the non-dimensional form of the resistance and stress equations are obtained. To simplify the analysis, the material resistances of two surfaces (1 and 2) in contact are joined in a single resistance. The following non-dimensional parameters are used to non-dimensionalize the thermal resistance expressions:

$$a^* = \frac{a}{b}, \quad b^* = \frac{b}{L_s}, \quad c^* = \frac{c}{b},$$

$$h_m^* = \frac{b\left(k_1 + k_2\right)}{L_1 k_2 + L_2 k_1}, \quad k^* = \frac{k_s L_s}{k_p L_p},$$

$$L_s^* = \frac{L_s}{b}, \quad R^* = R\, k_s\, L_s, \quad \xi = \sqrt{h_c^*\, k^*\, b^*}, \tag{12}$$

$$\overline{\psi}^* = -\frac{1}{8} k^* \left(c^{*2} + 1\right) + \frac{1}{2} k^* \frac{c^*}{c^{*2} - 1} \cdot \left[\left(\ln b - \frac{1}{2}\right) - c^{*2}\left(\ln\left(b\, c^*\right) - \frac{1}{2}\right)\right] + \frac{1}{2} k^* \left[\frac{\left(c^{*2} + 1\right)}{\xi} \frac{\phi_2}{\phi_1} + \left(\frac{1}{2} - c^{*2} \ln b\right)\right]$$

where ϕ_1 and ϕ_2 are given by:

$$\phi_1 = \frac{K_1\left(\xi a^*\right) I_1\left(\xi\right)}{I_1\left(\xi a^*\right)} - K_1\left(\xi\right)$$
$$\phi_2 = K_0\left(\xi\right) + \frac{K_1\left(\xi a^*\right) I_0\left(\xi\right)}{I_1\left(\xi a^*\right)} \tag{13}$$

The parameter L_s is the harmonic mean of the thicknesses of the contacting plates $L_s = 2L_1L_2/(L_1+L_2)$, where 1 and 2 refer to the two contacting surfaces.

There are two definitions of the dimensionless contact conductance parameter, since two models for the deformation of asperities (plastic and elastic) are used in this work. One should note that k_s is not present in these definitions. For the elastic contact, the h_c^* parameter is:

$$h_c^* = \frac{1.54\left(\frac{P}{H_e}\right)^{0.94} b}{\frac{\sigma}{m}} \tag{14}$$

For the plastic model, the h_c^* parameter is defined as:

$$h_c^* = \frac{1.25 \left(\frac{P}{H_p} \right)^{0.95} b}{\frac{\sigma}{m}} \quad (15)$$

The non-dimensional form of the material resistance is:

$$R_m^* = \frac{L_s^*}{h_m^* \, \pi \left(1 - a^* \right)} \quad (16)$$

Similarly, the non-dimensional form of the contact resistance is:

$$R_c^* = \frac{L_s^*}{h_c^* \, \pi \left(1 - a^* \right)} \quad (17)$$

Finally, the constriction resistance in non-dimensional form is:

$$R_{ct}^* = \frac{\overline{\psi}^*}{\pi \left(c^{*2} - 1 \right)} \quad (18)$$

The following dimensionless parameters are used to non-dimensionalize the thermal stress model:

$$E^* = \frac{E_b}{E_p}, \quad L^* = \frac{L_w}{L_p}$$
$$\alpha^* = \frac{\left(\alpha_b - \alpha_p \right) q \, b^2}{k_s \, L_s} \quad (19)$$

where L_w is the thickness of the washer.

The temperatures and the stresses are non-dimensionalized by the following expressions, respectively:

$$T^* = \frac{T \, k_s \, L_s}{q \, b^2}, \quad P^* = \frac{P}{E_{ss}} \quad (20)$$

The thermal stress non-dimensional expression is:

$$P^* = \frac{\alpha^* \left(T_m^* - T_0^* \right)}{2 \, L^* \left[1 - \frac{\left(1 - a^{*2} \right)}{r_b^{*2}} \right] - \frac{\left(1 - a^{*2} \right)}{r_b^{*2}} + E^*} \quad (21)$$

These expressions are used in the comparative study of the next section.

Comparison of Elastic and Plastic Contact Resistance Models

The overall non-dimensional thermal resistance, which is the combination of the individual non-dimensional thermal resistances according to Eq. 10, is dependent on the contact stress. The contact stress is dependent on the temperature levels of the components of the mountings, which in turn are dependent on the boundary conditions (heat input and one known fixed temperature) and on the overall thermal resistance. Therefore, the calculation of the overall thermal resistance is an iterative process, which was implemented in Maple V5† computer algebra

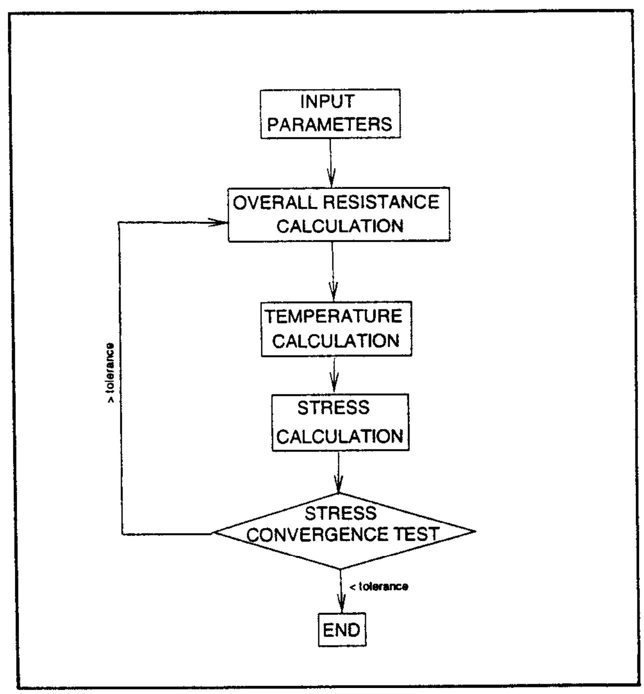

Fig. 5. Flow Chart of the Iterative Calculation Routine

† Maple V5, release 3, Maple Software, Waterloo, Canada

software. A flow chart of the iterative calculation routine is presented in Fig. 5. To study the influence of the temperature level, different heat inputs were tested (convergence was achieved for each heat input), and variable temperature levels and contact stress were obtained. In this study, it is considered that the heat is removed from the lower plate to a cold plate with known temperature, at the cryogenic level. Therefore, the cold plate is the fixed temperature boundary condition.

The physical dimensions, physical properties and the initial values settings in this study, for the mounting described in Fig. 1, are presented in Tables 2 and 3 respectively.

Table 2. Physical Dimensions of the Components of the Mounting

a	0.0037 m
b	0.0083 m
c	0.0889 m
r_b	0.0020 m
L_p	0.0032 m
L_w	0.0064 m
σ_{wp}/m_{wp} †	3.557 10^{-6} m
σ_{ww}/m_{ww} ‡	7.639 10^{-6} m
m_{wp} †	0.0915
m_{ww} ‡	0.0550

† wp refers to the washer-plate contact
‡ww refers to the washer-washer contact

Table 3. Physical Properties of the Components of the Mounting and Initial Settings.

k_p [26]	210.0 W/m°C
k_b [27]	14.8 W/m°C
α_p [27]	22.0 10^{-6} m/m
α_b [28]	14.5 10^{-6} m/m
E_p [27]	6.895 10^{10} Pa
E_b [27]	1.931 10^{11} Pa
ν_p [29]	0.33
ν_b [29]	0.30
c_{1p} [30]	1103.9 MPa
c_{1b} [30]	-0.00487 10^{-6}
c_{2p} [31]	6271.0 MPa
c_{2b} [31]	-0.229 10^{-6}
P_i	10 MPa
T_0	300 K

RESULTS

Figure 6 presents a comparison between two plots of the non-dimensional resistance R_t^* as a function of the non-dimensional pressure P^* for the model which considers elastic deformation and for one which considers plastic deformation. One sees that the difference between these curves is negligible. Figure 7 shows a similar plot, where the non-dimensional overall resistance R_t^* (left y axis) and the non-dimensional pressure P^* (right y axis) are plotted against the non-dimensional temperature, T_m^*. One should keep in mind that the temperature is divided by q when it is non-dimensionalized (see Eq. 21). The several temperature levels studied are obtained from the variation of q. The rate of temperature increase with q increase is lower than the rate of increase of q itself, therefore , as q increases, the non-dimensional temperature decreases. So, from Fig. 7 one observes that, for low temperature levels, T_m^* and R_t^* present high values, but P^* presents a low value. In other words, the resistance increases as the pressure decreases, as expected. In both figures, the difference between the values obtained from the plastic and elastic models are never larger then 4%.

COMMENTS AND CONCLUSIONS

In this work, the influence of the use of plastic and elastic models in the determination of the overall thermal resistance of bolted joints is studied. The bolted joint studied is composed the two plates, which are fastened by a bolt. Three washers are installed between the plates, one washer under the bolt head and another one above the nut. As the system operates in vacuum, only material, constriction and contact resistances were found. For this configuration, the use of plastic or elastic contact model had a small influence in the overall thermal resistance of bolted joints. This occurs because the magnitude of the contact resistances found in the junction are small when compared with the other resistances (constriction and material). Therefore, it is recommended that the choice of the deformation model should be based on the availability of the physical properties and the geometric parameters. A study of the relative importance of the thermal resistances in systems such as bolted joints should always be performed. Even if the results of this work are not general, it points out that in some cases, the contact resistance may not be the dominant factor. Most of the studies deal with bolted joints with no washers between the plates, where the efforts were concentrated on the theoretical and experimental determination of the contact resistance between the plates. Therefore, it is recommended that the various resistances found in a mounting configuration should be compared to determine the importance of the contact resistance in the overall system.

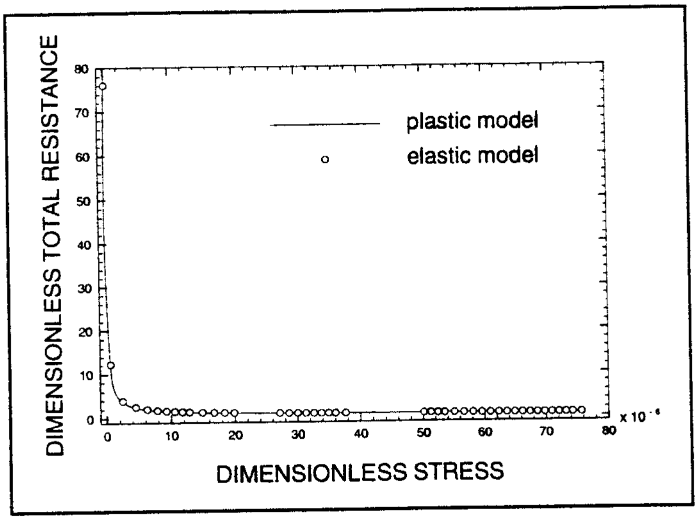

Fig. 6. R_t^* as a Function of P^* for the Plastic and Elastic Model.

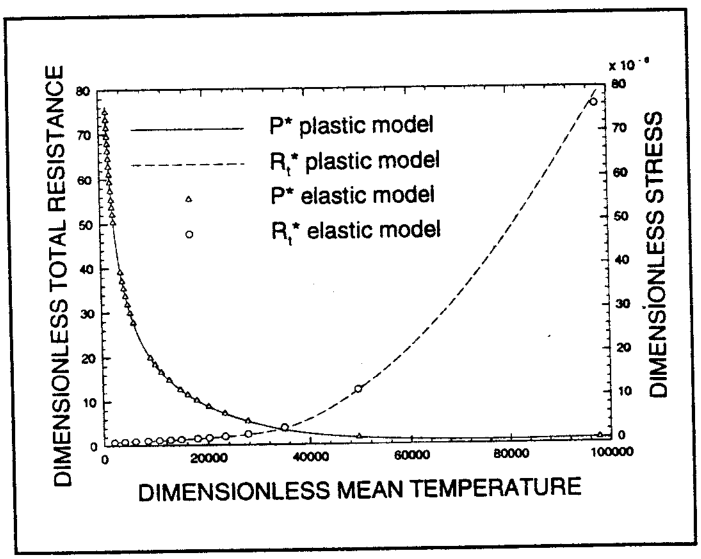

Fig. 7. R_t^* and P^* as a Function of T_m^* for the Plastic and Elastic Model.

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