MODEL OF THE FRICTIONAL HEATING OF INCONEL 718 AND TITANIUM (Ti-6Al-4V) IN HELIUM

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ABSTRACT

The frictional heating of metals in sliding contact in an oxygen-rich environment can result in the catastrophic ignition and combustion of metallic parts. In order to investigate the susceptibility of specific metals to this phenomenon the NASA White Sands Test Facility (WSTF) has conducted numerous experimental investigations of this phenomenon over the past decade using a unique apparatus developed at WSTF. A number of theoretical investigations of the frictional heating and subsequent ignition have also been pursued at WSTF. But, due to the complexity of the different phenomena involved in the frictional heating of metals in an oxygen-rich environment (e.g., the kinetics of the oxidation reactions or the effect of mechanical, physical and chemical changes at the interface on the actual amount of heat produced by friction) it is not possible at present to accurately predict whether the ignition phenomena will occur. The heat introduced to the interface of the samples during the experiment via friction will be balanced by the heat loss from the interface due to conduction through the samples away from the interface. This conduction of heat away from the interface will be enhanced by heat losses from the samples to the surroundings via convection, radiation and conduction. In order to accurately predict the possibility of ignition it is therefore important to understand the effect these phenomena have on the resulting temperature at the interface. It is also important to understand the actual heat produced by the frictional rubbing of the metallic samples. At present no quantitative theory exists for predicting the actual conversion of mechanical to thermal energy via friction (i.e., there is no method for predicting friction coefficients).

A computer model of the frictional heating of metals in an inert environment has been developed which incorporates the effects of the heat loss from the samples due to conduction, radiation and convection to the surroundings. This model allows the measured temperatures to be used to determine the amount of heat produced at the interface during the experiment by the sliding contact of two different metallic samples. The results of the simulation for an experiment run at WSTF show that for the same heat production at the interface the heat losses have a significant effect on the temperatures in the samples. But, the heat losses do not significantly affect the different calculated heat flows (or friction coefficients) at the interface necessary to correlate the measured temperatures.

INTRODUCTION

Over the past decade a number of theoretical and experimental reports have been published regarding the frictional heating of metals in gaseous oxygen,[1-5] the majority of which was conducted at the NASA White Sands Test Facility (WSTF). Jenny and Wyssmann[1] developed a theoretical model of frictional heating which incorporated conductive heat transfer along the sample and heat loss to the surroundings via radiation. They also assumed that the friction coefficient was dependent on the interface temperature, though they were unable to relate this temperature dependence to any physical property and instead relied on an empirical relationship.

At WSTF Yuen and co-workers[2-4] developed a number of models for the frictional heating apparatus. The model by Zhu et al.[3] incorporated conduction along the samples, convective and radiative heat loss and heat production due to oxidation reactions at the interface. Although the model used a number of empirical parameters it was able to qualitatively represent the experimental results.

The purpose of this modeling effort is to test a model of the frictional heating of cylinders under relatively simplistic conditions (i.e., no reaction). By removing the contribution to the heat input due to the oxidation reactions (which, at present, are not well understood mathematically) it should be possible to determine with higher accuracy the contributions of the other phenomena (friction, conduction, convection, etc.) on the observed temperatures in the samples.

EXPERIMENTAL RESULTS

The experimental results used in this study are from test number 830-411 performed on October 11, 1990 at WSTF. The experimental apparatus for the frictional heating tests has been described in detail elsewhere[5] and need not be repeated here. From a modeling standpoint we need only consider those aspects of the design and operation of the experiment which may have an effect on the observed results.

The test samples consist of two annular rods, one stationary (made of Inconel 718) and one rotating (made of the titanium alloy Ti-6Al-4V). Both samples are approximately 0.850 in. long with an outer diameter of approximately 1 in. and an inner diameter of 0.8 in. The stationary sample is placed in a holder which has a diameter approximately 0.0015 in. wider than the outer diameter of the sample and is 0.4 in. long. For the rotating sample the holder has a diameter approximately 0.001 in. less than the inner diameter of the sample, is approximately 0.5 in. long and is connected to the inner rotating shaft. After 0.5 in. the inner shaft diameter is reduced to 0.57 in. for a length of 0.8 in. where it then reaches the minimum diameter of 0.37 in. Both samples are enclosed in a cylindrical chamber with a gas volume of 3 in^3 (total volume approximately 4.5 in^3) and surrounded by a replaceable copper sleeve with a diameter of 1.5 in. The length of the chamber is 2.5 in.

The measurements of interest in this study were the transient temperature changes measured by two thermocouples placed in the stationary sample. The two thermocouples were located 0.05 in. (TC-702) and 0.20 in. (TC-703) from the junction with the rotating rod. The thermocouple closest to the junction shows a very interesting temperature change. There are four distinct time intervals during the course of the experiment denoted by changes in the temperature of TC-702 (fast, slow, fast, slow).

MODEL DEVELOPMENT

The model to be used in this study will be a simple transient one-dimensional model for heat conduction. Due to the high temperatures involved, though, the temperature dependence of the physical properties will be included. Therefore, for each annular rod (i=1,2) we will have the following energy balance[6]

$$\rho_i C_{p,i} \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} k_i \frac{\partial T}{\partial x} - Q_v - Q_c - Q_r$$

where ρ_i = the density of rod i, $C_{p,i}$ = the heat capacity of rod i, T = temperature in the rod, t = time, x = axial position in rod, k_i = the thermal conductivity of rod i, Q_v = the heat loss per unit rod volume due to convection, Q_c = the heat loss per unit rod volume due to conduction and Q_r = the heat loss per unit rod volume due to radiation.

The initial condition will be:

$$T = T_o \quad \text{for } -\infty < x < +\infty$$

The boundary conditions will be:

At x = 0:

$$-k_1 \frac{\partial T}{\partial x} + k_2 \frac{\partial T}{\partial x} = Q_F \qquad \text{and} \qquad T_{0+} = T_{0-}$$

where Q_F is the frictional energy flux at the interface.

At $|x| \to \infty$:

$$T = T_w$$

Frictional Energy:

The frictional energy flux Q_F is given by the equation

$$Q_F(t) = P(t)v(t)\mu(t) \qquad \text{in Btu/in}^2\text{ sec.}$$

Although the applied pressure (P) and velocity (v) terms are measured quantities, the coefficient of friction (μ) is not measured. Due to the substantial frictional forces introduced in this experiment it is likely that this parameter will change over the course of the experiment. At present there is very little suitable theory describing how the coefficient of friction will change over the course of the experiment. A number of methods were attempted in order to develop a suitable fit of the data. The following correlations gave the best results.

Correlation A: The first model simply states that the frictional heat flux Q_F will be held constant over a specified time interval Δt_i, i.e.

$$Q_F = Q_{F,i} \qquad \text{for } t_i < t < t_i+\Delta t_i$$

The values for $Q_{F,i}$ will be determined by the best fit of the experimental data. From these values the average values for the friction coefficient can be calculated using the following analysis:

Define the "average" flux over a time interval t_i to $t_i+\Delta t_i$ as

$$Q_{F,i} = \frac{\int_{t_i}^{t_i+\Delta t_i} Q_F dt}{\int_{t_i}^{t_i+\Delta t_i} dt} = \frac{\int_{t_i}^{t_i+\Delta t_i} P(t)\,v(t)\,\mu(t)\,dt}{\int_{t_i}^{t_i+\Delta t_i} dt}$$

For this experiment P(t) varies linearly with time and v is constant. Letting P(t) = kt gives the average friction coefficient over the time interval as

$$\mu_i = \frac{2Q_{F,i}\Delta t_i}{kv[(t_i+\Delta t_i)^2 - t_i^2]}$$

Correlation B: The second model will hold the friction coefficient μ constant over the time interval Δt_i, i.e.

$$\mu = \mu_i \qquad \text{for } t_i < t < t_i + \Delta t_i$$

The frictional heat flux Q_F will then be calculated using this value for μ. The values for μ_i will be determined by the least-squares fit.

Effect of Gas on Frictional Heating:

For the gas phase (helium) we can develop the following energy balance (assuming the gas phase to be well-mixed):

$$m_g C_{p,g} \frac{dT_g}{dt} = 2\pi R_o \int h_v (T - T_g)\, dx - A_g h_v (T_g - T_w)$$

where m_g = the mass of gas in the system, $C_{p,g}$ = the heat capacity of the gas, h_v = convective heat transfer coefficient, T = the rod temperature (varies with position), T_g = the temperature of the gas, T_w = the chamber wall temperature, R_o = the outer rod radius and A_g = the area for heat transfer from the gas to the surrounding chamber.

External Heat Transfer From Rods:

Convection:

The heat loss from the rods due to convection can be described by the equation

$$Q_v = S_v h_{vi} (T - T_g)$$

where S_v = the surface area per unit rod volume available for convective heat transfer ($= 2R_o/(R_o^2 - R_i^2)$), R_i = the inner radius of the annulus and h_{vi} = the convective heat transfer coefficient for rod i.

For the stationary Inconel rod it will be assumed that, due to the high temperatures involved in the experiment, free convection will be the major mode of heat transfer by convection for the part of the rod exposed to the chamber. The heat transfer coefficient can be given by the correlation from Chapman[7].

For the rotating titanium rod forced convection was assumed to be the main mode of convective heat transfer. A correlation for heat transfer for an annulus with the inner cylinder rotating is given by Maron and Cohen.[8]

Conduction:

For the heat transfer from the cylinders to the holders over the small distances (0.0015 and 0.001 in., respectively, for the Inconel and titanium samples) the heat loss due to conduction can be approximated by assuming a "pseudo" heat transfer coefficient h_c

$$Q_c = S_{c,i}h_c(T - T_w) = \frac{S_{c,i}k_g}{\Delta x}(T - T_w)$$

where $S_{c,i}$ = the surface area per unit volume of rod i available for conductive heat transfer, k_g = the thermal conductivity of the gas (helium) evaluated at the average temperature of T and T_w and Δx = the distance separating the cylinder and holder.

Radiation:

The heat loss due to radiative heat transfer can be calculated via

$$Q_r = S_{r,i}\sigma F_{i-j}(T^4 - T_w^4)$$

where all temperatures are absolute (°R), σ = Stefan's constant (= 3.3063×10^{-15} Btu/s in^2 °R^4), $S_{r,i}$ = the surface area per unit volume of rod i available for radiative heat transfer and F_{i-j} = the shape factor for radiative heat transfer from body i to body j.

Due to the geometry of the apparatus, the shape factors are dependent on the axial position x. It is assumed that radiation will occur in both directions from the rods (i.e., from the outer surface of the rods to the chamber and from the inner surface of the rods to the shaft). The equations for radiative heat transfer between two gray bodies is given by Chapman.[7]

NUMERICAL SOLUTION

The numerical solution of the differential equations describing the temperature profiles in the annular rods is accomplished using a finite-difference approximation for the spatial derivatives and integrating in time using the ODEPACK subroutine LSODE (allowing integration via Adam's or the implicit Gear's methods). The nonlinear least-squares program for determining the frictional heating parameters was the MINPACK subroutine LMDIF1 which utilizes a finite-difference Jacobian.

For the finite-difference approximation the following equation was used for the conduction terms:

At $x = x_i$:

$$\frac{\partial}{\partial x} k_j \frac{\partial T}{\partial x} = \frac{k_{j,i+}(T_{j,i+1} - T_{j,i}) - k_{j,i-}(T_{j,i} - T_{j,i-1})}{h^2}$$

where h is the grid spacing. The subscript j refers to the rod (Inconel, j = 1, or titanium, j = 2). The subscript i refers to the axial position (the value of i increases with increasing distance from the junction of the rods for both values of j). The axial position x = 0 refers to the junction of the two rods and x > 0 refers to the Inconel rod while x < 0 refers to the titanium rod. The thermal conductivities will be evaluated at the temperatures at the mid-points between x_{i+1} and x_i (for $k_{1,i+}$) and x_i and x_{i-1} (for $k_{1,i-}$). These temperatures will be approximated using a second-order difference formula.

The finite-difference approximation for the boundary condition at x = 0 was handled in the following manner:

Using $O(h^2)$ approximations for the derivatives gives

$$-\frac{k_{1,0}(T_{1,1} - T_1^*)}{2h} + \frac{k_{2,0}(T_2^* - T_{2,1})}{2h} = Q_F$$

where T^*_1 is the temperature in the Inconel rod at an imaginary point located a distance h into the titanium rod. T^*_2 is a similar point for the titanium rod.

Since we assume that the temperature at x = 0 is the same for both the Inconel and titanium rods for all times, this implies that at x = 0 $\partial T/\partial t$ for both rods are equal. For these terms to be equal the conduction terms must also be equal. We can therefore set the finite-difference approximations at x = 0 for both rods equal to each other, i.e.

$$\frac{k_{1,0+}(T_{1,1} - T_0) - k_{1,0-}(T_0 - T_1^*)}{\rho_1 C_{p,1,0}} = \frac{k_{2,0-}(T_2^* - T_0) - k_{2,0+}(T_0 - T_{2,1})}{\rho_2 C_{p,2,0}}$$

where the subscript 0 refers to values at x = 0.

Solving the boundary condition for T^*_1 in terms of T^*_2 gives

$$T_1^* = T_{1,1} + \frac{2hQ_F + k_{2,0}(T_{2,1} - T_2^*)}{k_{1,0}}$$

Substituting this into the previous equation gives T^*_2 solely in terms of the other temperatures (which are all known at the previous time step). This value for T^*_2 can then be used in the transient equation for the titanium rod at x = 0. Unfortunately, since T^*_2 and T^*_1 are necessary in order to calculate the thermal conductivities, an iterative procedure is necessary to first approximate these temperatures, calculate the thermal conductivities and then check the values obtained using the approximate values (in practice only a few iterations were necessary for convergence). The initial guesses for T^*_2 and T^*_1 were obtained using a second-order finite difference extrapolation.

The integration for the energy balance on the gas was accomplished using the trapezoidal rule.

RESULTS AND DISCUSSION

Figure 1 shows the results of the simulation with the frictional heat at the interface approximated using correlation A with 16 equally spaced values for $Q_{F,i}$. The agreement with the experimental results is within a relative error of 6%. Note that the calculated temperature at the interface approaches the phase transition temperature for Ti-6Al-4V (which is between 1750 and 1850°F). But, for the short period that the interface temperature enters this range the value is due primarily to difficulties in fitting the experimental data with the transient model. Except for this short interval the calculated interface temperature never enters this range. The effect of the phase transition on the energy balance was therefore neglected.

The average friction coefficient using 16 time intervals (parameters) in correlation A or B is shown in Figure 2. The initially high value for the friction coefficient suggests a Coulomb-type relationship (i.e., as $P \rightarrow 0$ the frictional force approaches a finite non-zero value). The results compare favorably with the friction coefficients determined from torque measurements as reported by Zhu et al.[3] After the initial time period the qualitative nature of the curve also agrees with the conceptual model of Suh[9] regarding the contributions to the friction coefficient from adhesion, plowing and polishing.

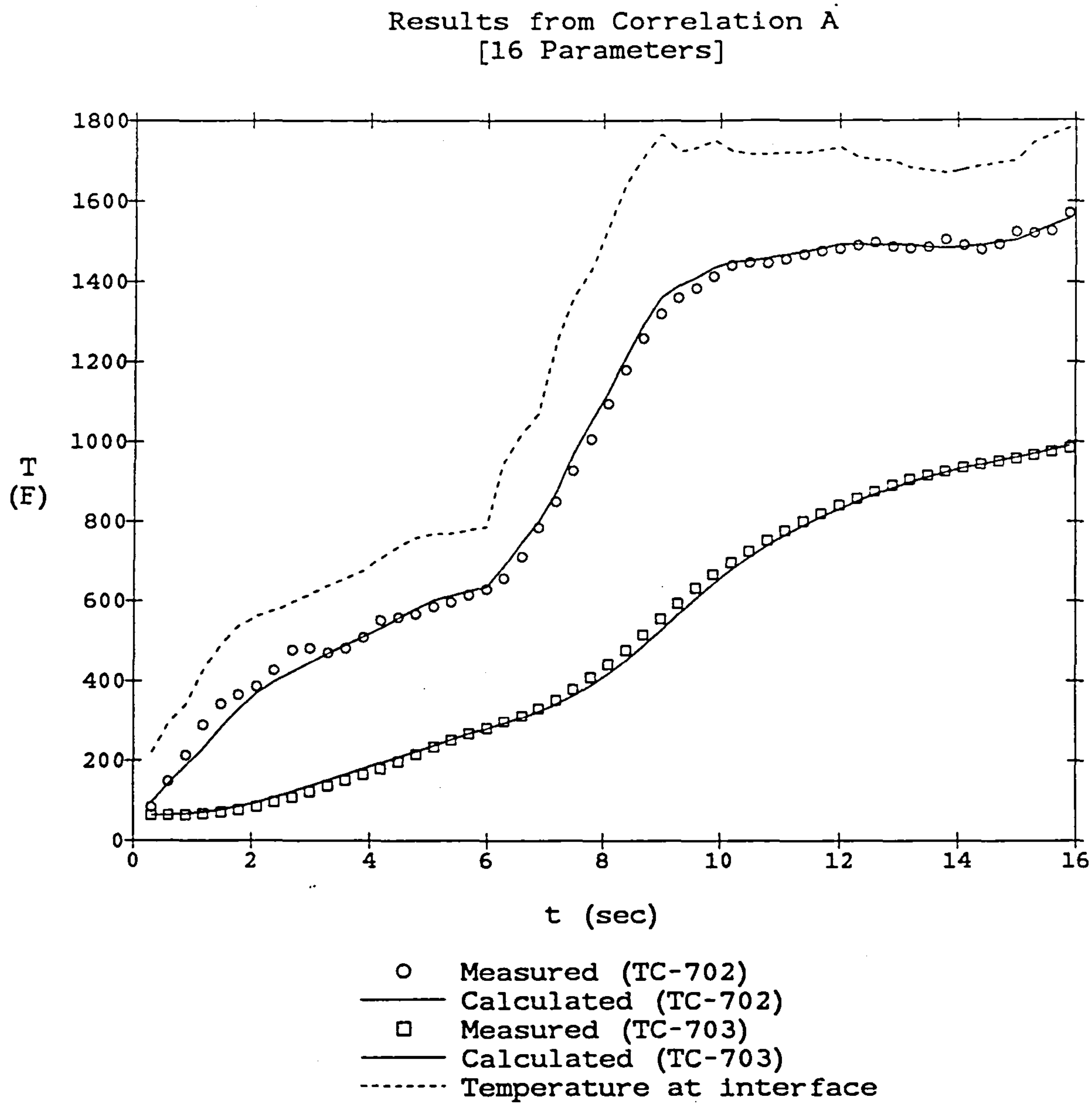

Figure 1.- Comparison of measured and calculated temperatures in Inconel 718 rod using correlation A with 16 time intervals.

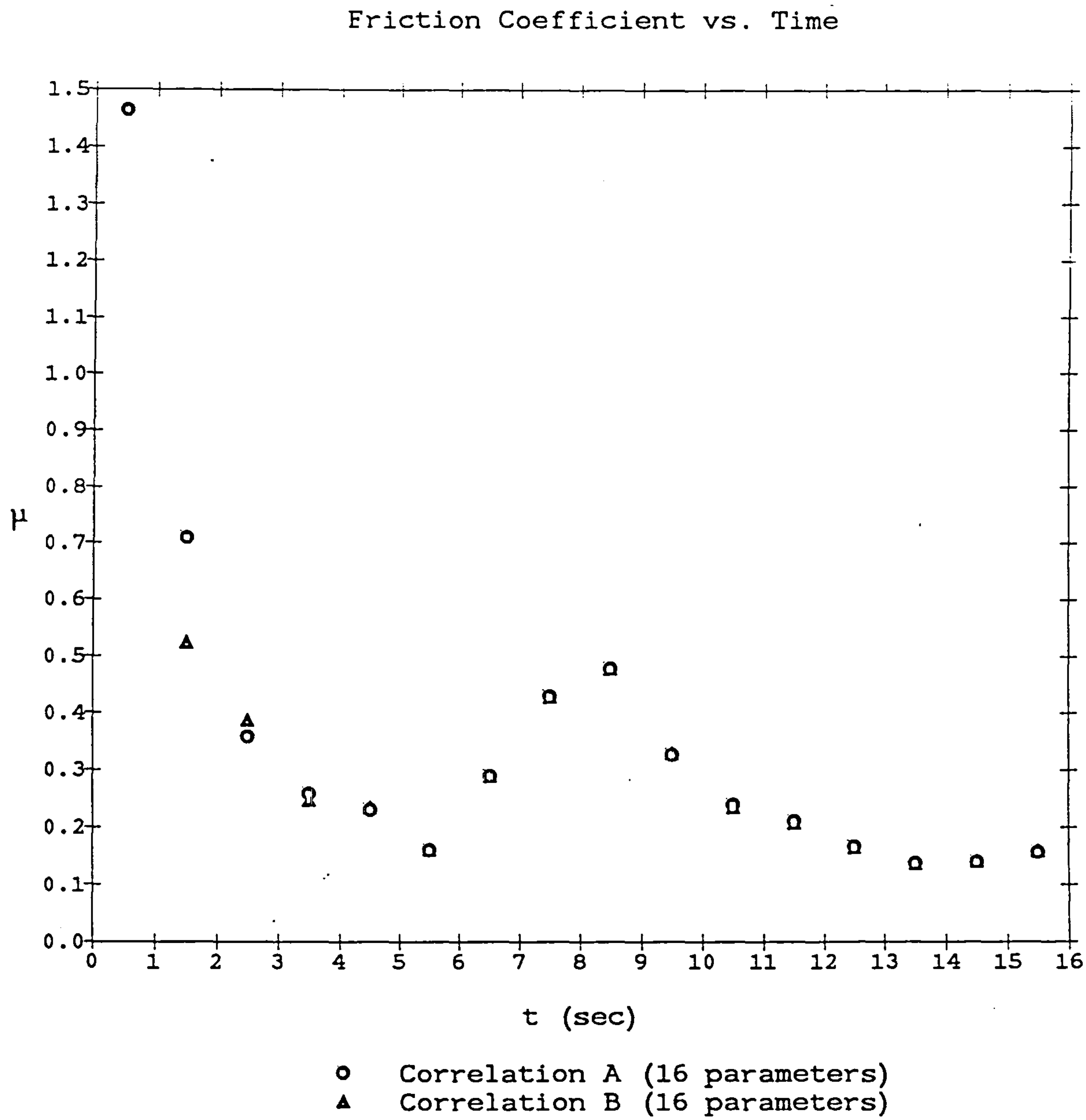

Figure 2.- Comparison of the average friction coefficients calculated using correlation A or B.

Figure 3 shows the contributions of conductive, radiative and convective heat loss on the calculated temperature profiles in the Inconel sample at the end of the simulation (for each curve the same values for the heat flux at the interface were used). For the stationary Inconel sample free convection and radiation make the largest contribution to the temperatures at the thermocouple positions (x = 0.05 in. and x = 0.2 in.).

For the rotating titanium sample forced convection contributes the most with conduction and radiation about equal. The large effect of conduction is due to the holder which extends from x = -0.85 to x = -0.35 in. This holder (which was assumed to remain at a constant temperature of 64°F) provides a significant sink for conductive heat transfer.

Figure 4 shows a comparison of the friction coefficients calculated using correlation A for two cases: 1) with the conductive, radiative and convective heat losses included and 2) with no heat losses included. These results, in conjunction with Figure 3, show the extreme sensitivity of the sample temperatures to the heat flux at the interface (i.e., the friction coefficients). In Figure 3 the values of the heat production corresponding to those used for case 1 (above) were used for all of the simulations, even those for which no heat losses were included (i.e., case 2). For the results in Figure 4 the values of the heat production required to match the measured temperatures were used for each simulation. From these results we can see that the required friction coefficients did not vary significantly between the two extreme cases. But, from Figure 3 we see that when the same values for the friction coefficient are used the temperatures at the thermocouple points (x = 0.05 in. and x = 0.20 in) vary by over one hundred degrees. This can be a significant difference when trying to predict the possibility of ignition for a given set of test conditions.

For the energy balance of the gas the final calculated temperature was 170°F which compares favorably with the measured value of 160°F.

The results of this research effort have shown that it is possible to accurately model the measured temperature in the metallic samples resulting from frictional heating in an inert environment. Future work will concentrate on attempting to extend the applicability of the model to reactive environments (i.e., gaseous oxygen) and the ability to separate the different contributions to the heat flux at the interface due to friction and reaction.

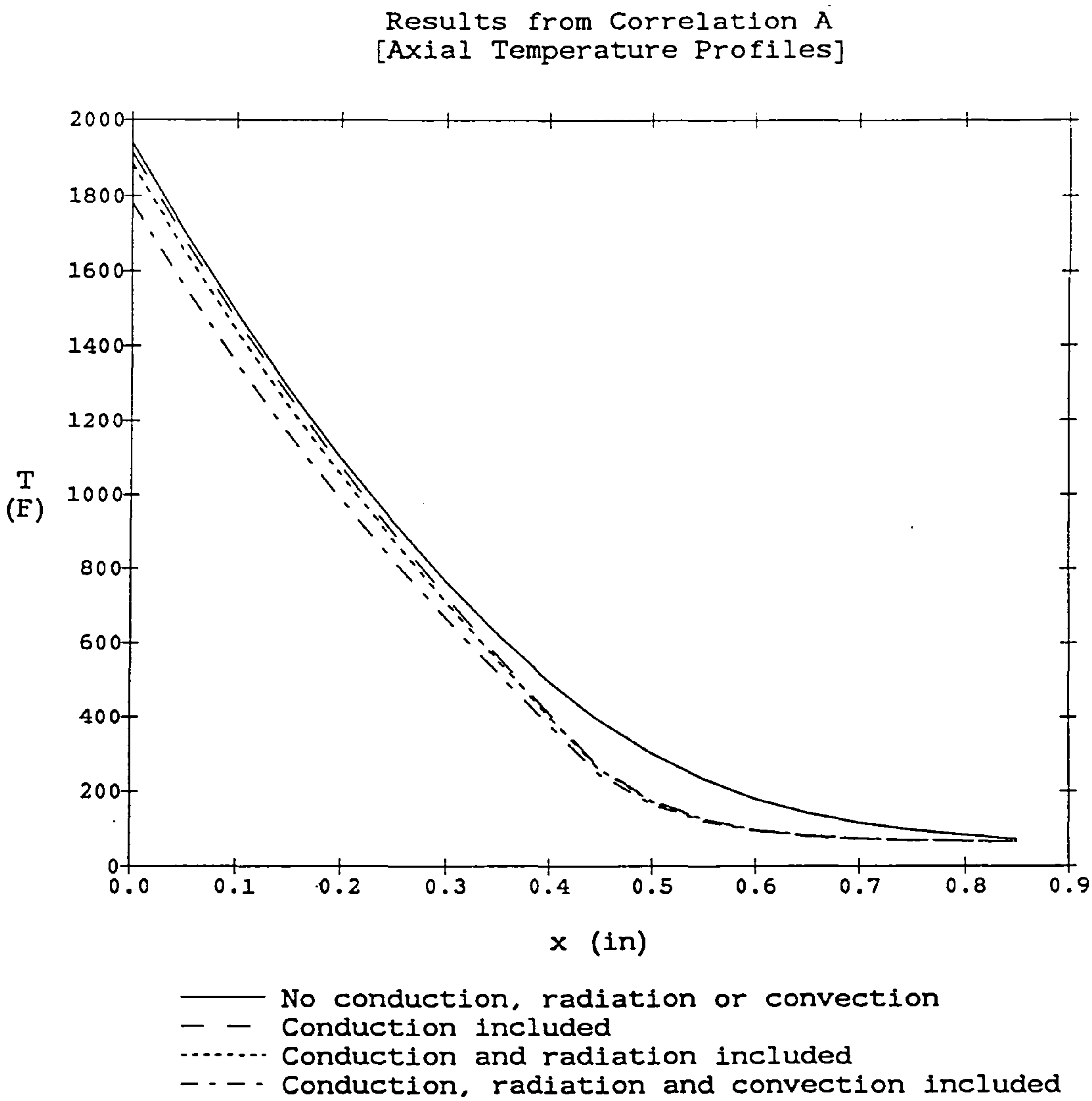

Figure 3.- Comparison of the effects of different modes of heat loss on the axial temperature profile in the Inconel 718 rod.

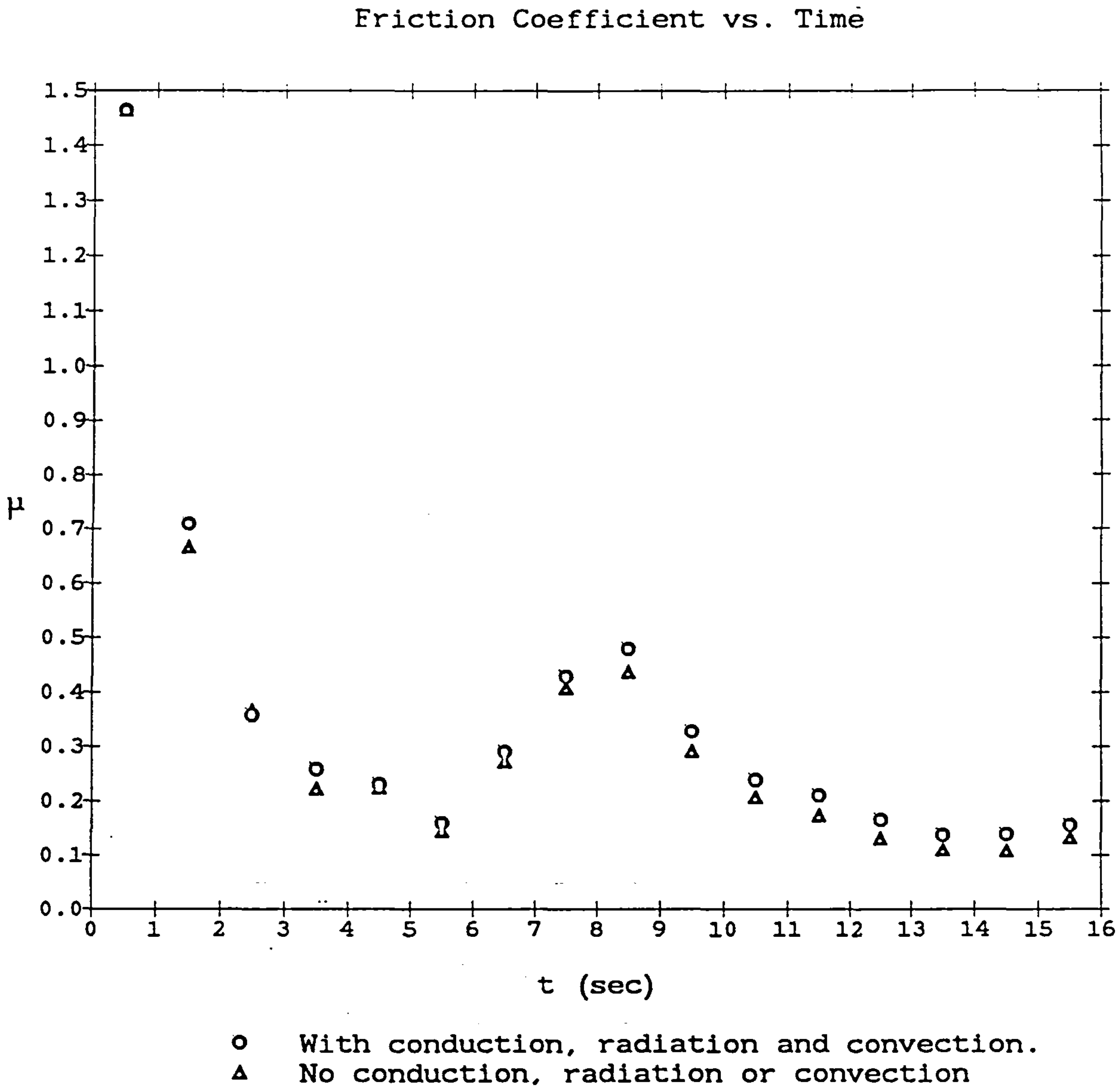

Figure 4.- Comparison of the average friction coefficients calculated using correlation A for two cases: 1) with all modes of heat loss from the sample included and 2) with no heat loss included.

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