

The Effect of Wear Groove on Vibration and Noise of Aircraft Brakes: Theoretical and Experimental Evidence

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ABSTRACT

The goal of this paper is to delineate recent experimental evidence that the presence of conforming surface wear groove tends to stabilize the vibration and noise response of aircraft brakes. This finding is consistent with an earlier theoretical study in which the contact between Carbon-Carbon (C/C) composite brake disks was assumed to be visco-elastic and through this assumption it was found that the existence of conforming grooves results in increasing dynamic stability of brake disk interaction. Therefore, the presumption of visco-elastic contact for C/C brakes seems to agree with the experimental observation in a subscale dynamometer.

The present paper summarizes both theoretical analysis and the test results. In the tests C/C composites were heat treated for one hour at temperatures 1800°C and 2400°C, respectively. They were then subjected to frictional tests in a subscale aircraft brake dynamometer at 50 % relative humidity (RH) level. Coefficient of friction (μ), vibration and noise were recorded during simulated braking. The surface topography was determined after each sequence of friction tests. The nanohardness of fiber and matrix in bulk composite were measured using a nanoindenter. The hardness of fiber was found to be two times that of the matrix for the samples heat treated at 2400°C as compared to the samples heat treated at 1800°C. The surface roughness (sRa) was measured as 1.96 μm for the sample heat treated at 2400°C after the 100% simulated normal landing energy conditions. The sample heat treated at 1800°C exhibited smoother friction surface (sRa=0.86 μm). During braking, μ varied by a factor of 2 or more which led to undesirable vibration.

Both the theoretical model and the tests results point to the same conclusion; existence of conforming grooves enhances dynamic stability of a disk pair, resulting in significant reduction in vibration and noise in braking.

INTRODUCTION

The interested range of dimensionless asperity (micron bumps) radius of curvature, β , is from 100 to 140, corresponding to elastic contact. A dimensionless length parameter is the ratio of the length in question to the standard deviation of roughness asperity height distribution of the disk surface. In the steady state studies it is assumed that a rotor comes into contact with a stator disk in which the stator is considered to have normal and angular freedom due to flexibility of the connections. Assuming constant, the normalized separation of two disks, h , is varied from 3 to 4.5 and β from 100 to 140. The governing equations of the dynamic model were used to study the steady state relation between angular speed of upper disk and friction torque for the range of h and β .

Within the range of h and β , the critical parameters obtained from steady state analysis were then applied in simulink model to investigate the dynamic response of the system for both models.

As shown in Fig. 1, a mass-spring-damper system comprising an upper annular disk with a fixed angular velocity ($\dot{\phi}_1$) is supported by a lower initially stationary disk that is permitted to rotate about its central axis and translate along its axis (normal direction). The structural stiffness and damping in the normal direction are denoted, respectively, K_n and C_n while those for torsional motion are represented by K_{tor} and C_{tor} . The h_0 indicates the initial mean plane separation of two contacting rough surfaces due to an applied normal force, F_{na} . To simplify the study, the upper disk, m_1 , is assumed to be fixed in normal position after the application of a pre-load. Referring to Fig. 2a, the governing equation of motion for normal can be expressed as

$$F_{kn} - F_{nc} - F_c = m_2 \ddot{x} \quad (1)$$

or

$$F_p - F_{nc} - K_n x - C_n \dot{x} = m_2 \ddot{x} \quad (2)$$

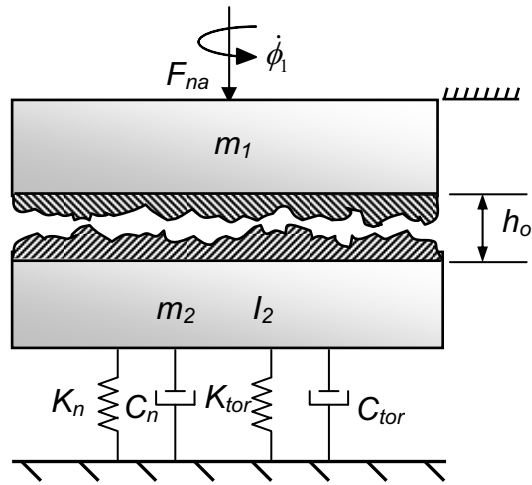
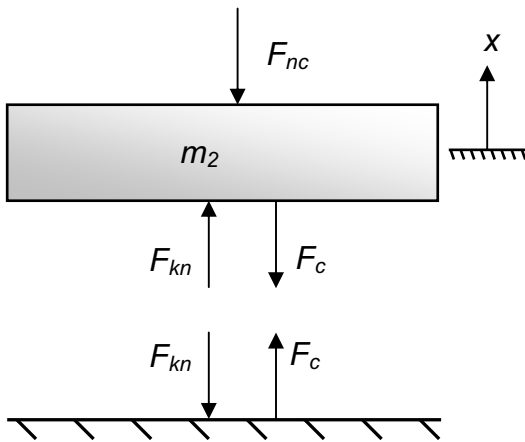
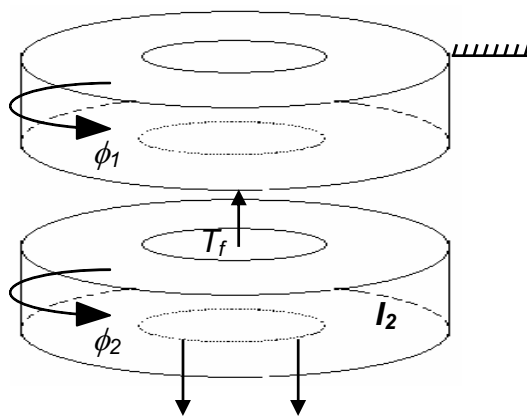


Figure 1 Schematic of the two-disk model



(a) Normal motion



(b) Torsional motion

Figure 2 Free body diagram in (a) normal and (b) torsional motions

Where, F_p is the elastic force due to the pre-load, K_n and C_n are the structural stiffness and damping coefficient respectively. Function F_{nc} is the normal contact force between the two contacting rough surfaces

contributed by additive visco-elastic contact components as

$$F_{nc} = F_{ne}^+ + F_{ne}^- + F_{nv}^+ + F_{nv}^- \quad (3)$$

The terms F_{ne}^+ and F_{ne}^- denote, respectively, the elastic normal contact forces at both positive and negative asperity interference slopes. F_{nv}^+ and F_{nv}^- are those contributed by rate dependent forces at both asperity shoulders.

STEADY STATE ANALYSIS

CONTACT OF NOMINALLY FLAT ROUGH SURFACES

The steady state response of the two-disk system is investigated to study the relation between friction torque, T_f , angular speed of upper disk, $\dot{\phi}_1$, and the dimensionless average radius of summit curvature, β . The analysis involves the cases for different applied pre-load, F_p , and the dynamic factor, f_s , that dictates elastic contact contribution to friction torque.

In steady state condition, the angular speed of the stator, $\dot{\phi}_2$, and the normal velocity, $\dot{h}$, vanish. The governing equation of normal motion is

$$m_2 \ddot{x} = F_p - K_n x - C_n \dot{x} - 2 C_{ne} A f_{ne} - 2 C_{nv} A_{nv} f_{nv} \dot{h} - 2 C_{nvr} A_{nvr} f_{nvr} (\dot{\phi}_1 - \dot{\phi}_2) \quad (4)$$

At steady state, $\ddot{x} = 0$, $\dot{x} = 0$, $\dot{h} = 0$, reducing equation (1) to

$$F_p - K_n x - 2 C_{ne} A f_{ne} - 2 C_{nvr} A_{nvr} f_{nvr} (\dot{\phi}_1) = 0 \quad (5)$$

To find the angular speed of upper disk, $\dot{\phi}_1$, equation (5) can be recast as follows

$$\dot{\phi}_1 = \frac{F_p - K_n \sigma (h_0 - h) - 2 C_{ne} A f_{ne3}}{2 C_{nvr} A_{nvr} f_{nvr2}} \quad (6)$$

or, after substituting the appropriate approximate functions

$$\dot{\phi}_1 = \frac{F_p - K_n \sigma (h_0 - h) - 2 C_{ne} A (319.956 \beta - 115.254) h e^{-4.9h}}{2 C_{nvr} A_{nvr} (-0.039 \beta^2 + 13.976 \beta + 285.238) h e^{-4.6h}} \quad (7)$$

The functions f_{ne3} and f_{nvr2} were used since the interested range of h is from 3 to 4.5. The pre-load is basically the applied elastic normal force and it can be obtained as follows

$$F_p = 2 C_{ne} A f_{ne3}(h_0), \quad 3 \leq h \leq 4.5 \quad (8)$$

Or,

$$F_p = 2 \left(\frac{8}{3} \frac{\pi}{\sqrt{2\pi}} E' \eta^2 \sigma^4 \sqrt{\beta} \right) \pi (R_0^2 - R_i^2) (319.956 \beta - 115.254) h_0 e^{-4.9h_0} \quad (9)$$

Where, h_0 is obtained from the equations governing the rotation motion of the stator by setting $\dot{\phi}_2$ and $\dot{h}$ to zero. The friction torque is found to be given by the following equation.

$$T_f = f_s C_a \left(\frac{8}{3} \frac{\pi}{\sqrt{2\pi}} E' \eta^2 \sqrt{\beta} \sigma^4 \right) \pi (R_o^3 - R_i^3) f_{te2} + \quad (10)$$

$$C_b \left(\frac{4\pi}{\sqrt{2\pi}} \eta^2 \sqrt{\beta} E \eta_v \sigma^3 \right) \left(\frac{\pi}{2} \right) (R_o^4 - R_i^4) f_{nv2} (\dot{\phi}_1)$$

A simple program was setup using Matlab software to solve the equations in steady state. First, the pre-load was chosen, and the initial values of normalized separation, h_0 , were calculated using equation (9). β was varied from 100 to 140 with step size of 0.5. The next step is to solve for $\dot{\phi}_1$ using equation (7). For a fixed pre-load, F_p , the corresponding h_0 is used in the equations while varying β from 100 to 140 and h from 3 to 4.5. Finally, the calculated $\dot{\phi}_1$ values are substituted into equation (10) to find the friction torque at the corresponding h and β .

Results of Steady State Analysis in Contact of Plain Rough Surfaces

First, cases resulting in stable and unstable dynamic response are presented. It is shown that it is possible for certain choice of surface parameters, axial load and material properties that the steady state response results in negative friction-sliding speed characteristics. Such negative slopes have been known to create instability in the dynamic response of two disks, manifested through noisy or vibration prone braking experience. The negative friction-torque is known as the Stribek effect.

Negative Rate-Dependent Friction (Stribek Effect)

The results of the steady state are plotted in graphs of friction torque versus angular speed of upper disk for various h and β . Figures 3 and 4 illustrate the results for potentially unstable or noisy case of braking since beyond a critical speed, the friction torque decreases with sliding speed. The various plots correspond to different values of dimensionless asperity radius of curvature.

The steady state investigations involved β from 100 to 140 with step size of 0.5, and h from 3 to 4.5 with step size of 0.02. Generally, as shown in the figures, the friction torque increases with sliding speed $\dot{\phi}_1$. This observation can be seen in all the figures at lower angular speed region. Throughout the entire range of $\dot{\phi}_1$, the increases in β will consistently lower the friction torque in all cases. The results suggest that higher speed and rougher surface (lower β) will generate more friction torque. Nevertheless, there is a critical point where friction torque will reach a maximum value beyond which it will decrease as the angular speed (sliding speed) is increased further as shown in higher speed region in Figures 3 and 4. This is a significant behavior as negative slope in friction/speed curves can lead to dynamic instability.

The friction torque/speed curves shown in Figures 3 and 4 are reminiscent of friction speed curve in wet

friction systems in which the curve shows a negative slope in the boundary and mixed (partial) lubrication regimes as shown in Figure 5. This phenomenon is known as Stribek effect [1]. The friction torque also reaches a maximum value and shows negative slope region in the figures, this indicates that the material rate-dependent effect is a dominant factor that causes the negative slope curve. It is important to realize that the negative slopes are equivalent to negative damping coefficients that can affect dynamic instability of the system.

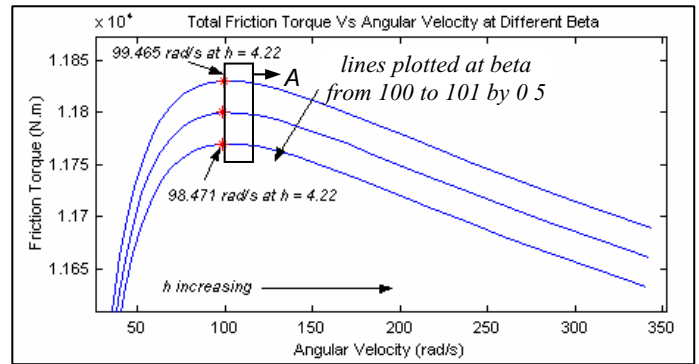


Figure 3 Friction torque vs. angular velocity at $\beta = 100$ to 101

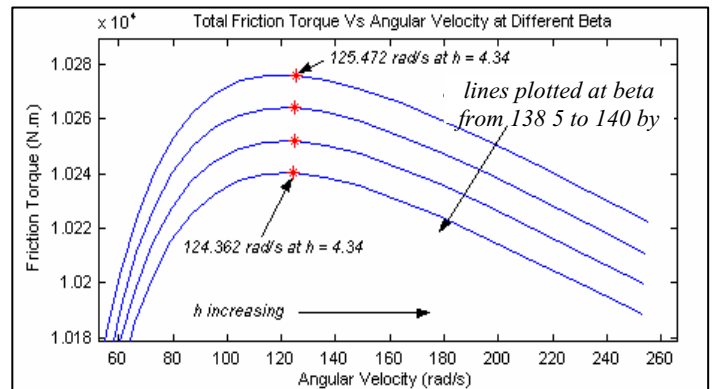


Figure 4 Friction torque vs. angular velocity at $\beta = 138.5$ to 140

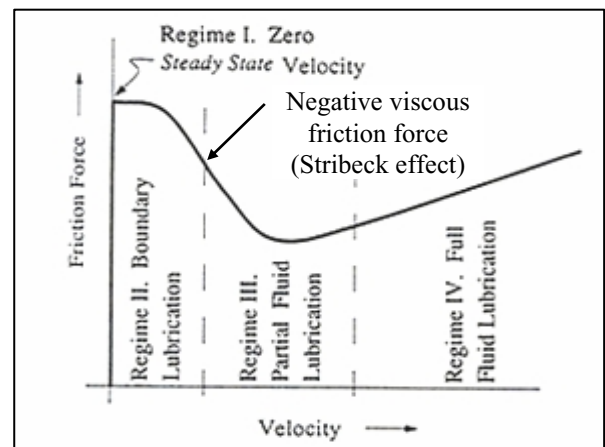


Figure 5 The generalized Stribek curve (Armstrong, 1991)