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Analysis of Disc Brake Noise at High and Low Frequency with the Effect of the Friction

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Abstract: This paper is a study on the stick-slip oscillation of a discrete system with contact interaction as a friction curve. The stick-slip oscillation with a single degree-of-freedom was examined by means of numerical time integration ODE45, while that with two degrees was by using FEM method. Beam on rotating disc was used to investigate the effect of friction at low velocity. The response indicated that the friction ratio was responsible for the separation amplitude value. The plate on disc was molded and connected by using matrix27 to investigate the effect of friction on a high frequency system. The results showed that friction causes damping at low frequency while at high frequency, it may generate the squeal.

Key words:

INTRODUCTION

The noise has been classified into two different phenomena depending on the frequency range; brake groan (0-500) Hz and brake squeal (1-16) kHz, Mikio 1993. The groan is attributed to the stick-slip motion while the squeal is attributed to modal coupling. Stick–slip vibrations show up in many kinds of engineering systems and in everyday life, e.g. as sounds from squeaking chalks and shoes, creaking doors, chattering machine tools, and grating brakes—most of which are attributed to surfaces sliding with friction. The brake system is a self-excited and nonlinear dynamic system wherein exists friction induced dynamic instability caused by negative damping, Stick-Slip, Sprag-Slip, structure modal coupling, parametric excitation, or internal resonance. The friction forces in disc brakes are important in decelerating or stopping road vehicles. The disc and the pad motion may also generate surface waves between the interfaces, which may lead to stick-slip motion that causes the dynamic instability (Chen, F., 2000). The stick-slip is responsible for the squeal at the final stage of braking. The stick-slip phenomenon is responsible for squeal noises, Pilipchuk and Tan (2004). A new cause for the noise was found in the longitudinal vibration of the disc rotor at high frequency. When the disc brake generates noise, it indicates that the disc is vibrating in one diametral mode. It was found that the squeal frequency is more dependent on the natural frequency of the disc rotor. The dynamic stability of the system studied, depends on the diametral mode shape of the disc. The squeal frequency usually matches well with the modal frequency of recorded vibrations of the disc brake (Alex 1992). Gregory (1999) used a stationary brake system consisting of a pad, calipers, and rotor. In his experimental work, he found that the natural frequencies and mode shape of the operating system were identical to those of the stationary system under the assumption that the systems were linear and that the rotation frequency of the rotor was much smaller than the natural frequency of either system. The aim of this paper is to understand the effect of friction on the stability of the system with different frequency range. This paper includes three new ideas. The first is to use a simple friction equation to study the stick-slip phenomenon. The second is to use a rigid disc with beam in an attempt to split the coupled mode in order to observe the effect of the friction on the system response only. The third is to use a plate on disc instead of beam on disc which gives a better representation of the brake system compared to Ripin’s (1995) thesis result.

2. Stick-Slip with One-Degree of Freedom:

A stick-slip system was conducted by using mass on a moving belt. The system was one degree of freedom. Figure 1 shows a system: a mass M on a belt that moves at constant speed (). The mass is a rigid body having position X (t) at time t. It is subjected to gravity loading Mg, linear spring-loading KX, linear

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damping force C, and a friction relative velocity force. The spring, K is attached parallel to the damper, C and connected to mass, M. The mass is in contact with a moving band. The left side of the mass is connected to a spring pivot. The initial length of the spring is Xo with a pivoted point. The band is moved by rollers rotating at a constant speed.

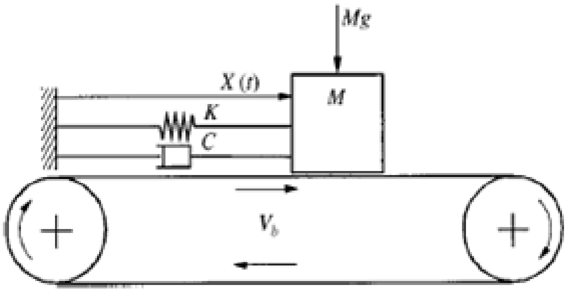

Fig. 1: Mass at position *X* (*t*) on a belt that moves at constant speed

In the case of stick-slip oscillations friction force is time-dependent during stick and velocity-dependent during slip. The friction relative velocity formula was mentioned in Guran, A. et al (2001).

$$F = \mu_s N(\mathrm{Sgn}(\dot{X} - V_b) - \sigma(\dot{X} - V_b)) \quad (2)$$

Where = gradient, =belt velocity, N=normal force and =mass velocity

The corresponding response was conducted by using Runge-Kutta method which was built in MATLAB software as a (ode45).

3. Stick-Slip with different parameters:

The simulation required a few trials to determine the mass, damping and stiffness values to be used with the system equation (2) in order to get the same result as that of Ko and Brockely's (1970) experiment. The values to achieve a stick-slip motion for the system as shown in Figure 1 are, Mass (M) =1 Kg, spring stiffness (K) =5 N/mm, Damping C=1.5 (N.mm)/Sec, Static friction coefficient =0.5,, normal load=10, initial displacement=0.0, belt velocity=0.5 mm/sec and mass initial velocity= 0.5 mm/sec. Those values are the baseline condition for this analysis. Using signum function in MATLAB software simplifies the required method to solve the stick-slip equation. The advantage is that the system can be integrated with any standard ODE-solver available in mathematical packages as MATLAB or ODE-solver of existing software libraries. Moreover, the system can be integrated without need to halt and this minimizes start-up costs. Figure3 shows the time-varying position X (t) of the mass when there is no external harmonic excitation.

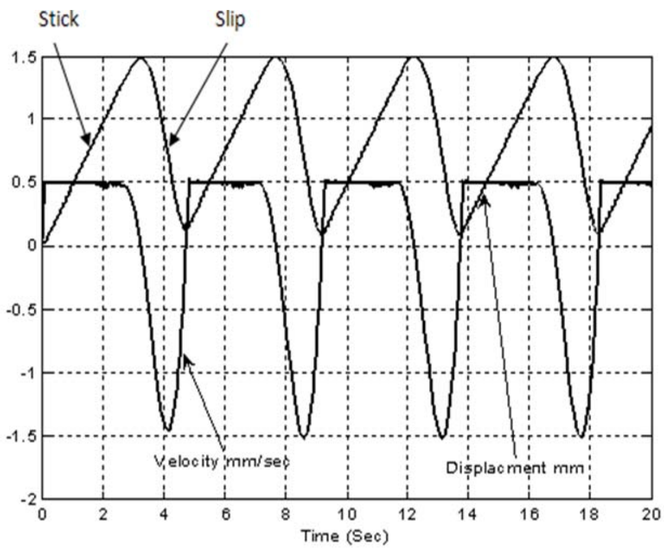

Fig. 3: Displacement (mm) and velocity (mm/sec) versus time

From Figure 3, there is no change in the mass velocity during the stick. As the system changed from stick to slip, the mass velocity increased until it reached the maximum value 1.5mm/sec at displacement approximately equaled 0.75mm. This displacement value (0.75mm) indicated that the mass was at the middle of displacement X (t) trying to return to the zero position, meaning that all the potential energy transformed to kinetic energy, leading the mass to reach maximum velocity. It is important to note that the velocity of the mass during the stick period will not exceed that of the belt. This is because the energy storing spring cannot accelerate the mass to a velocity exceeding the maximum velocity during the stick period, and the energy provided by the belt cannot accelerate the mass to a velocity beyond its own. It is of course possible to start the system with a bigger mass velocity than belt velocity. However, viscous damping and dry friction will drain the energy until a stationary state is achieved with the belt velocity equal to mass velocity. Figure 3 was returned to plot in a phase plane in order to understand the velocity-displacement relation as in figure 4, where the stick appears as horizontal line.

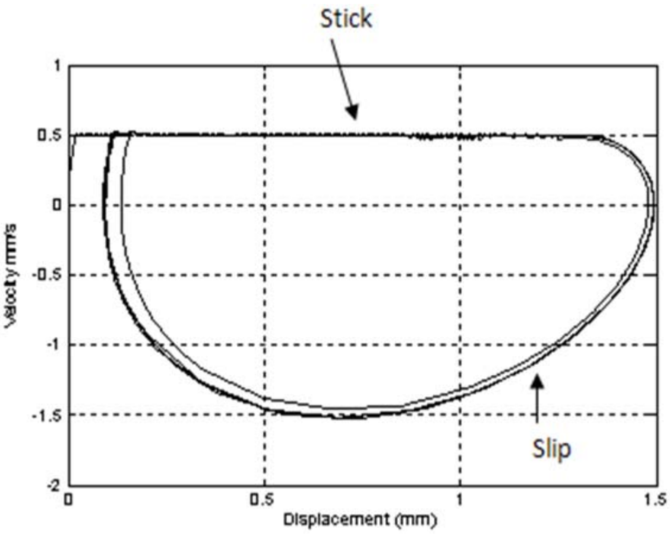

Fig. 4: The velocity-displacement phase plot of stick-slip motion

The position (response) of the mass would change linearly with time as long as the mass continued to stick. When the displacement is 1.4mm, the stick stopped and the mass started to slip. During the stick, the static friction magnitude increased until the maximum value. Since the velocity of the mass did not change with the displacement during the stick, the minimum and maximum velocity occurred during the slip.

The simulation was conducted with different mass values (3, 2, 1) Kg while the other parameters stayed constant, figure 7. The result showed that increasing the mass had no significant effect on the amplitude (peak-peak). Increasing the mass value increased the kinetic energy required to pull back the mass. For M=3Kg, the frequency is 0.16Hz while for M=1Kg the frequency is 0.23Hz. This indicated that the number of oscillations increased with decrease in mass. However, increasing the mass reduced the vibration of the system but increased the deformation (peak-zero).

The simulations for different damping coefficient values were conducted in order to investigate the effect of damping on stick-slip motion, Figure 8.

The result showed that decreasing the damping from 2N.mm/sec to 0.5N.mm/sec increased the number of oscillations or vibrations from 0.154Hz to 0.3125Hz. The mass did not return to zero position because decreasing the damping decreased the force required to pull back the mass.

4. Friction relative velocity:

The formula 4.2-3 was solved by using ODE45 in order to find the value of friction coefficient versus relative velocity. These values were used in the ABAQUS software to represent the contact friction versus slip velocity in a beam-disc system.

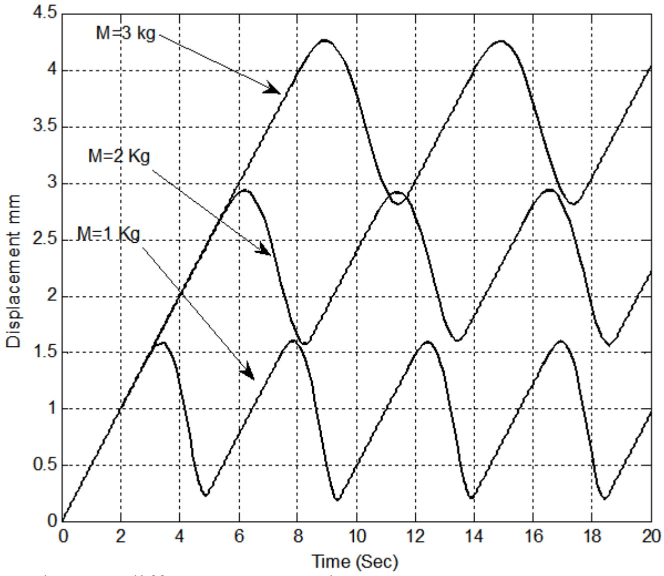

Fig. 7: System response due to different mass values: σ=0.5; =0.5; M=3Kg; N=30N; C=1.5N.mm/sec; =0.5mm/sec; K=5N/mm

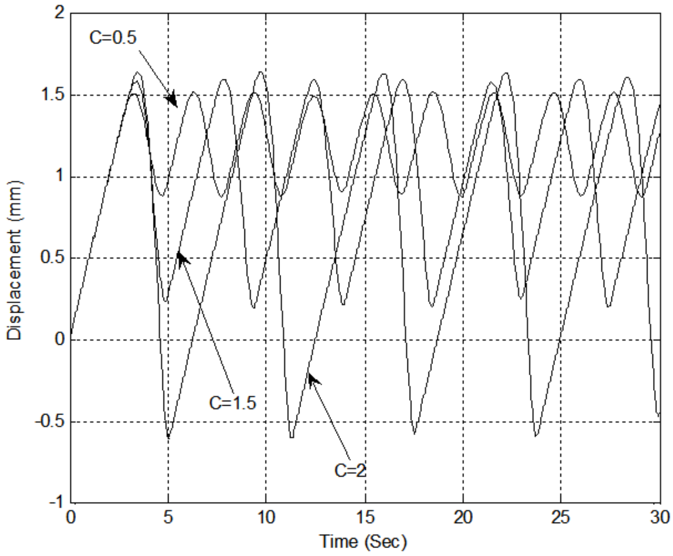

Fig. 8: System response to different damping coefficient values; σ=0.5; =0.5; M=1Kg; N=10N; =0.5mm/sec; K=5N/mm

Figure 9 shows that at zero relative velocity, when the belt and the mass moved together (stick), the friction coefficient increased until it reached the highest value 0.5. The friction coefficient value decreased due to the relative velocity between the mass and the belt in a slip. When the mass returned to stick, the relative velocity equaled zero and the friction coefficient value could be between 0.5 and zero. At relative velocity of 0.6, the friction coefficient was assumed to be constant and equaled to 0.4 (coulomb friction value). This assumption was belt on Stribeck curve which showed a constant value at relative velocity of 0.6 mm/sec. Wensrich, C. (2006) also used the same value for the static and kinetic friction coefficient.

After that the friction coefficient would increase until the slip occurred again. However, by increasing the relative velocity, the friction coefficient would decrease until at relative velocity 0.6mm/sec, the friction coefficient value become constant. The results indicated that the friction behavior was a consequence of the dynamic of the system and not of the interaction properties. At zero sliding velocity, the friction coefficient changed with time. The initial negative slope corresponded to negative damping, and thus caused oscillation to grow until a balance of dissipation was attained.

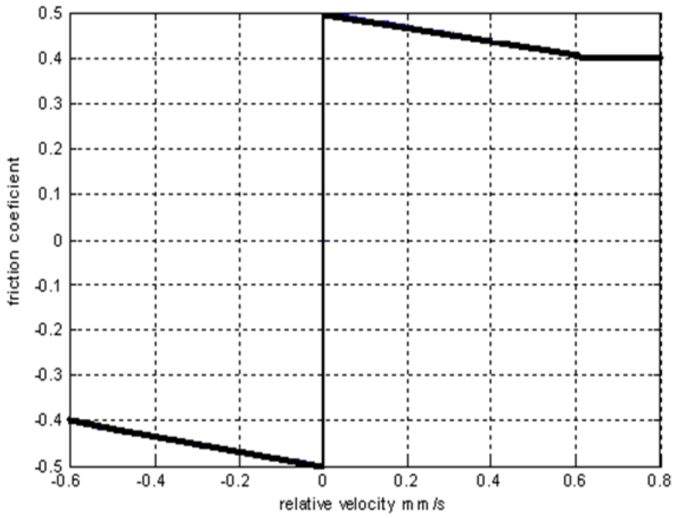

Fig. 9: Friction coefficient versus relative velocity mm/second

5. Stick-Slip motion for two masses with two degree:

Stick-slip friction contact was applied to two masses on a moving belt. The aim of the study is to understand more about the effect of stick-slip friction on two masses in two directions. Kinematic contact was applied between the lower surfaces of the masses and the upper surface of the belt. Mass1 was connected to mass2 and the wall by spring 0.06 N/mm. Using ABAQUS software allowed the study of stick-slip motion in two dimensions. Figure 9 was entered to the simulation by toggling on the slip rate.

The wall was constrained in three directions and there was no interaction between the belt and the wall. The mass density, Young's modulus and Poisson's ratio were 7.725e3 kg/m^3, 207 GN/m^2 and 0.25 respectively. The belt velocity was 50 mm/sec. The mass dimension was 10×10 mm. The belt density, Young's modulus and Poisson's ratio were 7.52e3 Kg/m^3, 120 GN/m^2 and 0.3, respectively.

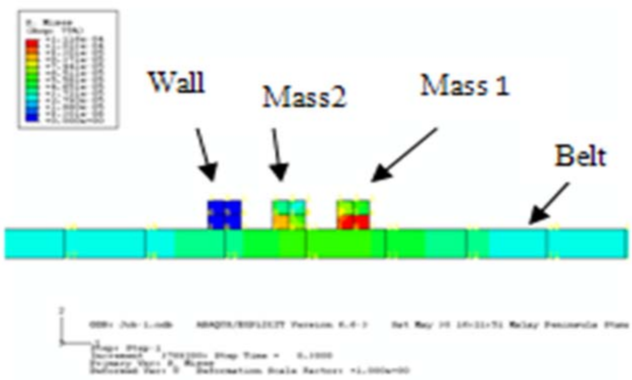

Fig. 10: two masses on moving belt

Figure11 indicates the surface contact separation due to change of static to kinematic friction coefficients with the velocity for mass1. Friction coefficients always tend to facilitate an unstable mode. As the friction coefficient further increased, the real parts of eigenvalues used to gauge the degree of instability also, increased.

Figure 12 indicates that separation happened when the mass changed the direction of motion. Figure 11 indicates the separation at 0.075 Sec and Figure12 indicates the high amplitude in the vertical direction (0.006 mm). It can be said that static friction changing static to kinetic friction increased the potential energy of the system which discharged as surfaces separate in the vertical direction (to kinetic energy). However, increasing the degree of freedom also affects stick-slip motion to appear as surfaces separate. Stick-slip motion effect will not always appear at every mass (Maximum) displacement. In other words, not all the stick slip motion causes surfaces separation or noise but it depends on the ratio of friction coefficient.

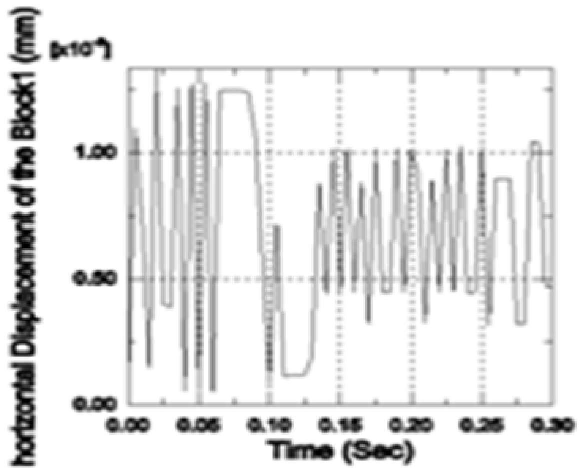

Fig. 11: The response in horizontal direction (U1) of the mass1 at node1

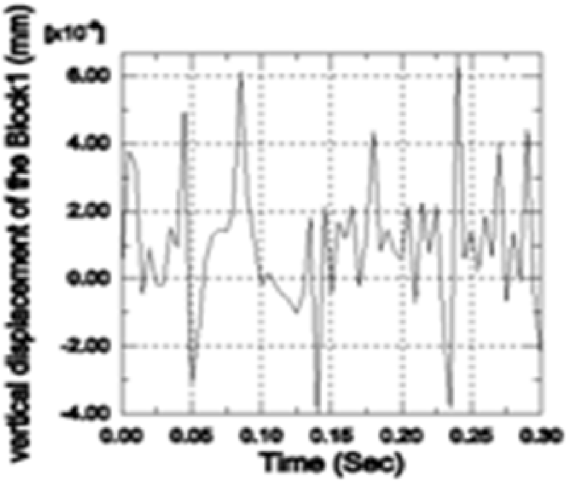

Fig. 12: The response in the vertical direction for the mass1

6. Beam on rotating disc application:

The beam on disc was modeled in ABAQUS software in order to observe the effect of friction on the response. The upper surface of the beam was constrained in the vertical direction, and its left end was constrained in the entire direction. The disc rotation velocity was 15.2 rev/min, clockwise. Figure 9 was entered to represent the interaction between the contact surfaces and Kinematic contact method was applied. The disc was molded as rigid and un-deformable (26mm) but the beam was deformable 1×100 mm, as illustrated in Figure 13.

The friction force as shown in Figure14 keeps the mass in equilibrium position (0, 0). The force which keeps the mass in equilibrium is the value of the static friction. The static friction will match the driving force at the equilibrium position until its limit is overcome. Once the static value is overcome, the frictional force reverts to the kinetic value. Figure14 shows the amplitude of oscillation increasing with time. This increment indicates increased instability with time due to self-excitation. However, the vibration is periodic but the effect of nonlinearity made the amplitude different, as the friction ratio.

7. Plate on disc at high frequency:

Plate on disc was modeled in ANSYS software in order to study the effect of the friction at high velocity.

The plate element was shell63 while the disc element was solid45. The plate dimension was 3.3×25.5cm while the disc dimension was the same as in figure 15.

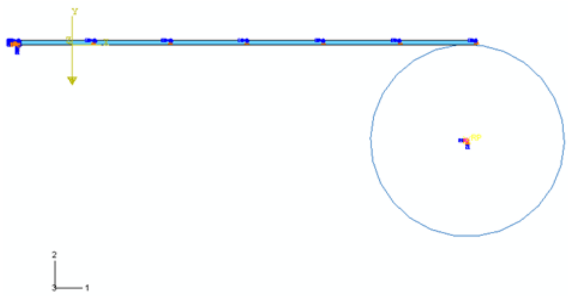

Fig. 13: Beam on rotating disc with rotating velocity 15.2 rev/min

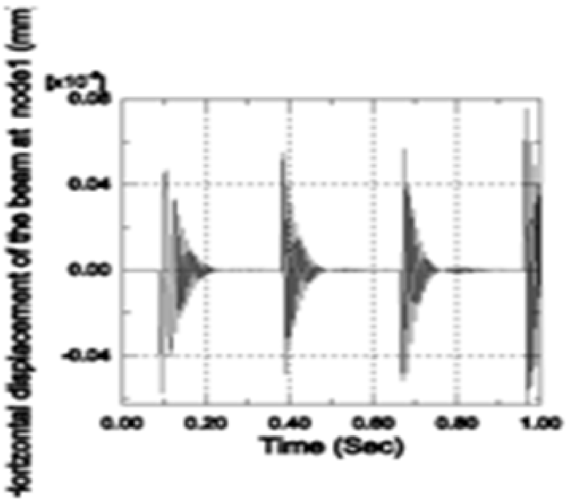

Fig. 14: Horizontal displacement of the beam at mass1

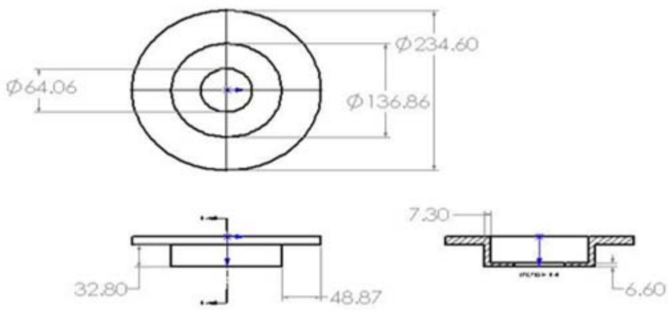

Fig. 15: Illustrated disc brake dimensions (all the dimensions in mm)

The contact between the surfaces was molded by using matrix27 as a contact element, figure 16.

The effect of friction coefficient on the squeal was analyzed by applying a range of friction coefficient values from 0.1 to 0.4, with increment of 0.1 as shown in the figure17. The analysis was conducted with contact stiffness value of 200 MN/m, which was assumed to be constant and distributed uniformly among the contact nodes. The system showed two frequencies (4441.9Hz and 3268.9Hz) at friction coefficient 0.1. The frequency value indicated a third diametral mode. These two modes began to converge until they merge (2919.8Hz) at friction coefficient 0.2. The merging continued beyond the friction coefficient 0.2, until discontinuity was achieved with friction coefficient 0.3.

The effect of a non-conservative (as a friction force) tends to couple the two modes. These effects make the system able to exchange energy in a way that causes the unstable behavior of the brake system to continue. It can be observed that the effects of friction coefficient enforce the system to generate the squeal at lower frequency from the stable separate modes.

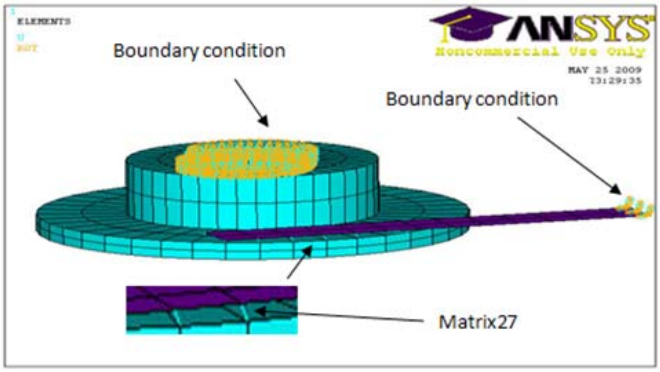

Fig. 16: Geometry of disc plate system

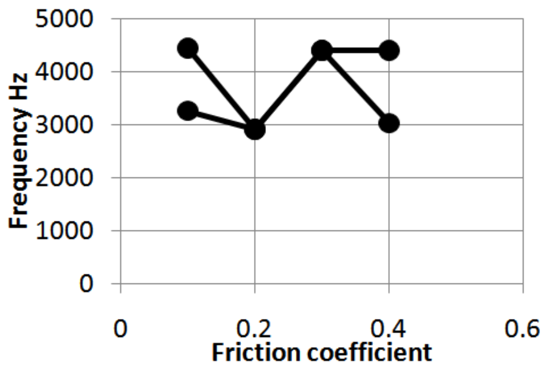

Fig. 17: frequency versus friction coefficient at K=200 MN/m for the third system diametral mode shape

The instability as a function of friction coefficient was conducted for the third mode shape, as in figure18. The figure shows that instability increased from zero at friction coefficient 0.1 to 258.07Hz at friction coefficient 0.2. The system after friction coefficient 0.2 became stable again. At friction coefficient 0.3, the system showed a small degree of instability 3.899Hz. At friction coefficient 0.4, the system returned to stability system where the real part equaled zero.

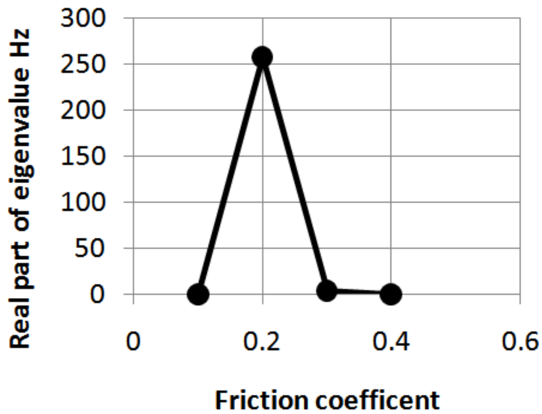

Fig. 18: The effect of friction coefficient on system instability at K=200 MN/m for the third system diametral

mode shape

It can be seen that instability decreased even when the contact stiffness was constant. This indicates that the separation could happen with increase in the friction coefficient because increase in the friction coefficient would increase the stress between the contact surfaces. Another reason for decreased instability is the node's position at the center of the pad, Fieldhouse (1991).

The mode shape of the unstable mode is shown in Figure 19. The test result showed that the squeal generated due to the coupling between the third diametral mode of the disc with the bending mode of the plate.

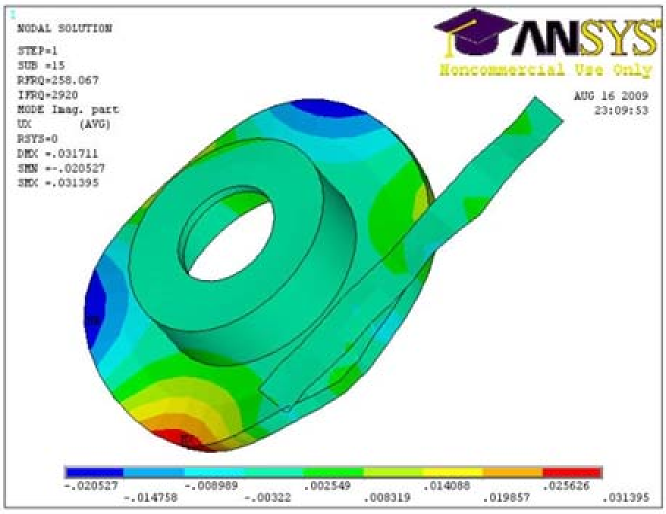

Fig. 19: Coupling mode shape at friction coefficient 0.2 and contact stiffness 200Mn/m

Conclusion:

The result showed that stick-slip motion at low frequency is the cause of noise. Increased mass value results in decreasing the noise. This indicates that mass contributes to change in the noise level. Increased damping showed increases the system stability. The result indicates that increasing the friction always decreases the vibration at low frequency while at high frequency, increasing the friction generates noise. The difference between the low and high frequency is due to the difference in friction contribution in the system equation of motion. At high frequency, the friction is a part of contact stiffness matrix while at low frequency, the friction is a part of the damping matrix. Changing the stiffness matrix including the friction coefficient to unsymmetrical made the real part to be positive at certain high frequency. Including the friction as a damping matrix during low frequency is the reason to generate positive real part or the instability.

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