A New Method Development to Predict Brake Squeal Occurrence

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ABSTRACT

A new method to predict brake squeal occurrence was developed by MSC under contract to Ford Motor Company. The results indicate that the stability characteristics of this disc brake assembly are governed mainly by the frictional properties between the pads and rotor. The stability is achieved when the friction coefficient of the pads is decreasing as the contact force increases. Based on the results, a stable brake system can be obtained without changing the brake structure by incorporating the appropriate frictional coefficient in the brake system. The method developed here can be also used as a tool to test the quality of any brake design in the early design stage.

INTRODUCTION

Brake squeal is the most common type of noise whose frequency content is 1000 Hz or higher with excessive high and irritating sound pressure levels [1]. The squeal occurs when a system experiences very large amplitude mechanical vibration. It is believed that there are two possible sources of the brake squeal: the first is the stick-slip phenomenon [1, 2] and the other is the instabilities of the brake assembly [3, 4].

Complex eigenvalue analyses have been used to study the stability of a brake system and predict the occurence of the brake squeal in [1,5], among others. In these analyses, the detailed model of a brake system can be achieved by using a fine finite element mesh. However, the results are usually not satisfactory due to the difficulties in modelling the friction between the rotor and pads. Some work has been also undertaken in [6, 7] where the phenomenon of brake squeal is treated as a self excited vibration problem due to friction between the pads and rotor.

This paper present a new approach to predict the occurrence of brake squeal. It is based on the theory that the stability of the brake system reflects the likelihood of squeal for the brake assembly modelled. The brake structure includes rotor, caliper, piston and pads. The dynamic behaviors of rotor and caliper which remain linear in the physical system are calculated via MSC/NASTRAN. An eigenvalue analysis is performed to obtained the modal characteristics of these components. They are used as constraint equations on the pad boundaries. MSC/DYNA is used to perform the transient analysis of the pads with the modal characteristics of rotor and caliper. Nonlinear frictional effects are also included and present when two surfaces are in contact. The dynamic responses of the pads are used to predict the stability of the brake system.

FINITE ELEMENT MODELLING OF A BRAKE ASSEMBLY

1. The MSC/NASTRAN model consists of two components: rotor and caliper as shown in Figure 1a. It is assumed that these components remain linear and undergo only small deformations. The model of rotor consists of 446 QUAD4 elements and 407 grid points. The caliper consists of 138 solid elements and 283 grids points.

2. The MSC/DYNA model consists of two components: pads and piston as shown in Figure 1b. The pads are modeled using solid elements for the frictional material and plate elements for the steel backing plate. The piston is modeled using stiff bar elements connecting the load application point to grid points on the inner pad in the vicinity where the actual piston would contact the pad. The pad models consist of 67 HEXA and 4 PENTA elements.

3. The overall brake system model is shown in Figure 1c. Please note that the coordinate system used for the analysis does not correspond to the vehicle coordinate system.

ANALYSIS

1. An eigenvalue extraction is performed to obtained all modes up to 14,000 Hz by using MSC/NASTRAN (Lanczos method).

2. A fortran program developed by MSC produces the component mode file by reading the MSC/NASTRAN output file and extracting the pertinent information file which will be appended to the MSC/DYNA model. The component modes generated for the MSC/DYNA model are represented with CSPR (linear springs) elements, CONM2 (concentrated masses) elements and MPCs (multiple point constraint equation).

3. The nonlinear transient response of the pads is analyzed by MSC/DYNA, an explicit finite element program. The time histories of displacement at selected nodes are stored and plotted to examine the stability characteristics.

EXAMPLES

Nine examples were analyzed, using the same brake structure but different friction properties for the pad material as shown in Figure 2. In order to describe these models, the following notations are used:

COF is the coefficient of friction
V is the relative tangential velocity
N is the contact force

The friction models can be divided into four groups:

Group I: The coefficient of friction is a constant and does not depend on tangential velocity and contact force.
(Model A)

Group II: The coefficient of friction is a function of tangential velocity.
(Models B and C)
Model B: COF decreases as V increases.
Model C: COF increases as V increases.

Group III: The coefficient of friction is a function of contact force.
(Models D and E)
Model D: COF decreases as N increases.
Model E: COF increases as N increases.

Group VI: The coefficients of friction is dependent of both tangential force and contact force.
(Models F, G, H and I)
Model F: COF decreases as V increases; and COF decreases as N increases.
Model G: COF decreases as V increases; and COF increases as N increases.
Model H: COF increases as V increases; and COF decreases as N increases.
Model I: COF increases as V increased; and COF increases as N increases.

The time histories of displacement at the three nodes, 2071, 2089 and 2119, located on the backing plate at the top, middle and bottom of the pad respectively (Figure 1), using friction models F and G are shown in Figures 3 and 4. It is shown that the time histories of displacement at these nodes provide the same information about the stability of system. Hence, we will plot displacement response only at one node, 2071, in the following discussion since it provides us enough data to predict the occurrence of brake squeal.

In the fist and second groups (models A-C), where the coefficient of friction is independent of contact force, the amplitudes of displacement increase slowly as the time elapses. It is also shown in Figure 5 that the responses generated from these models are quite similar. Although the amplitudes of displacement increase mildly, these systems are unstable.

In the third group (models D-E), where the coefficient of friction is a function of contact force, a different phenomenon appears as shown in Figure 6. In model D, where the coefficient of friction decreases as the contact force increases, the displacement amplitude almost remain the same except that in the early stage for developing the amplitude to certain values. On the other hand, in the model E, where the coefficient of friction increases as the contact force increases, the displacement amplitude increase rapidly.

In the fourth group (models F-I), where the coefficient of friction is a function of both tangential velocity and contact force, the same phenomenon as that in the group III is observed as shown in Figure 7. The displacement amplitude almost remain the same in case that the coefficient of friction decreases as the contact force increases (models F and H) and the displacement amplitude increases rapidly in case that the coefficient of friction increases as the contact force increases (models G and I). It also can be seen from Figure 7 that the responses are not much different between the models F and H even though the relations between coefficient of friction and tangential velocity are totally different in these two models. The same argument is also applied to the models G and I. Thus we can say that the relation between the coefficient of friction and the tangential velocity is not important in determining the stability characteristics of the brake.

In conclusion, the relation between the velocity and coefficient of friction is not important in determining the stability characteristics of the brake whereas the relation between contact force and coefficient of friction plays an important role. If the coefficient of friction is independent of contact force (models A-C), the system is unstable but the displacement increases slowly. If the coefficient of friction increases as the contact force increases, the displacement diverges rapidly (models E, G and I). If the coefficient of friction decreases as contact force increases, a relatively stable response is achieved (Model D, F and H).

CONCLUSIONS

For the modeled brake structure, the stability are not sensitive to velocities but mainly governed by the frictional characteristics between the pads and rotor. By selecting the pads, whose coefficient of friction decreases as the contact force increases, as shown in models F and H in Figure 2, the stable dynamic response of the brake system can be achieved without changing the brake structure. The method developed here can be also used as a tool to test the quality of any brake design in the early design stage.

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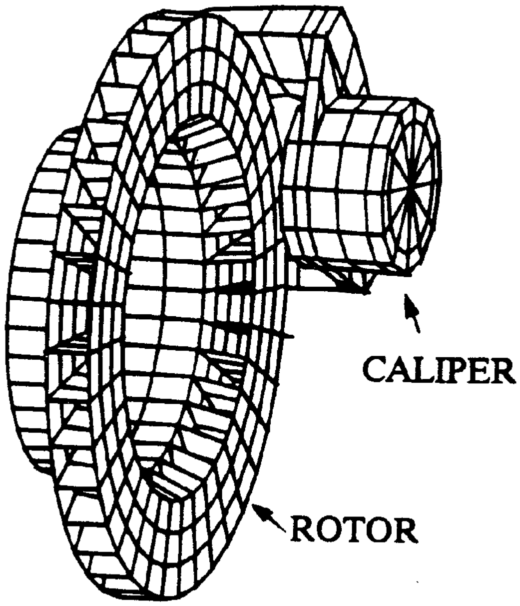
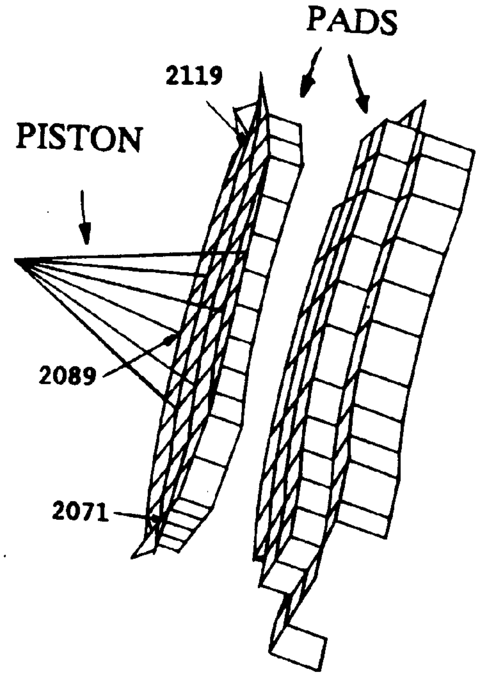
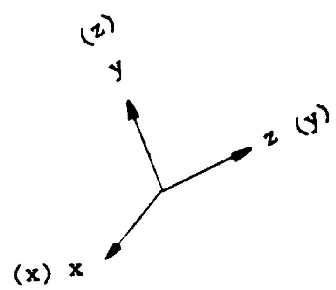
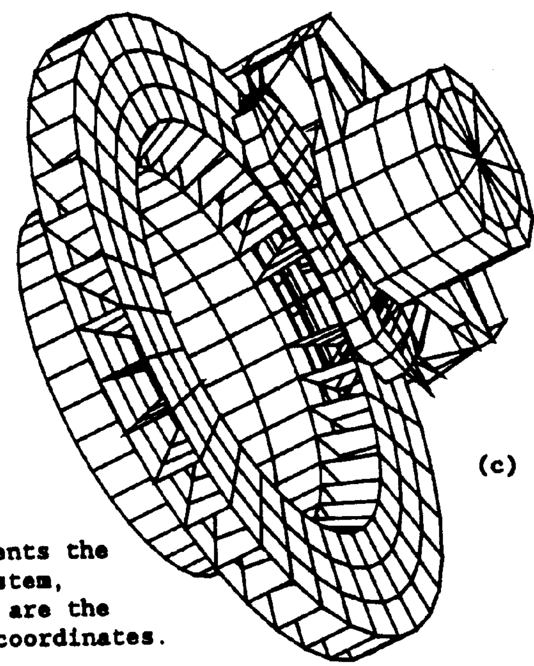


(a) MSC/NASTRAN Model (Rotor and Caliper)



(b) MSC/DYNA Model (Piston and Pads)





(c) Overall Brake Model

Note x, y and z represents the analysis coordinate system, while (x), (y) and (z) are the corresponding vehicle coordinates.

Figure 1. Finite Element Model of a Brake

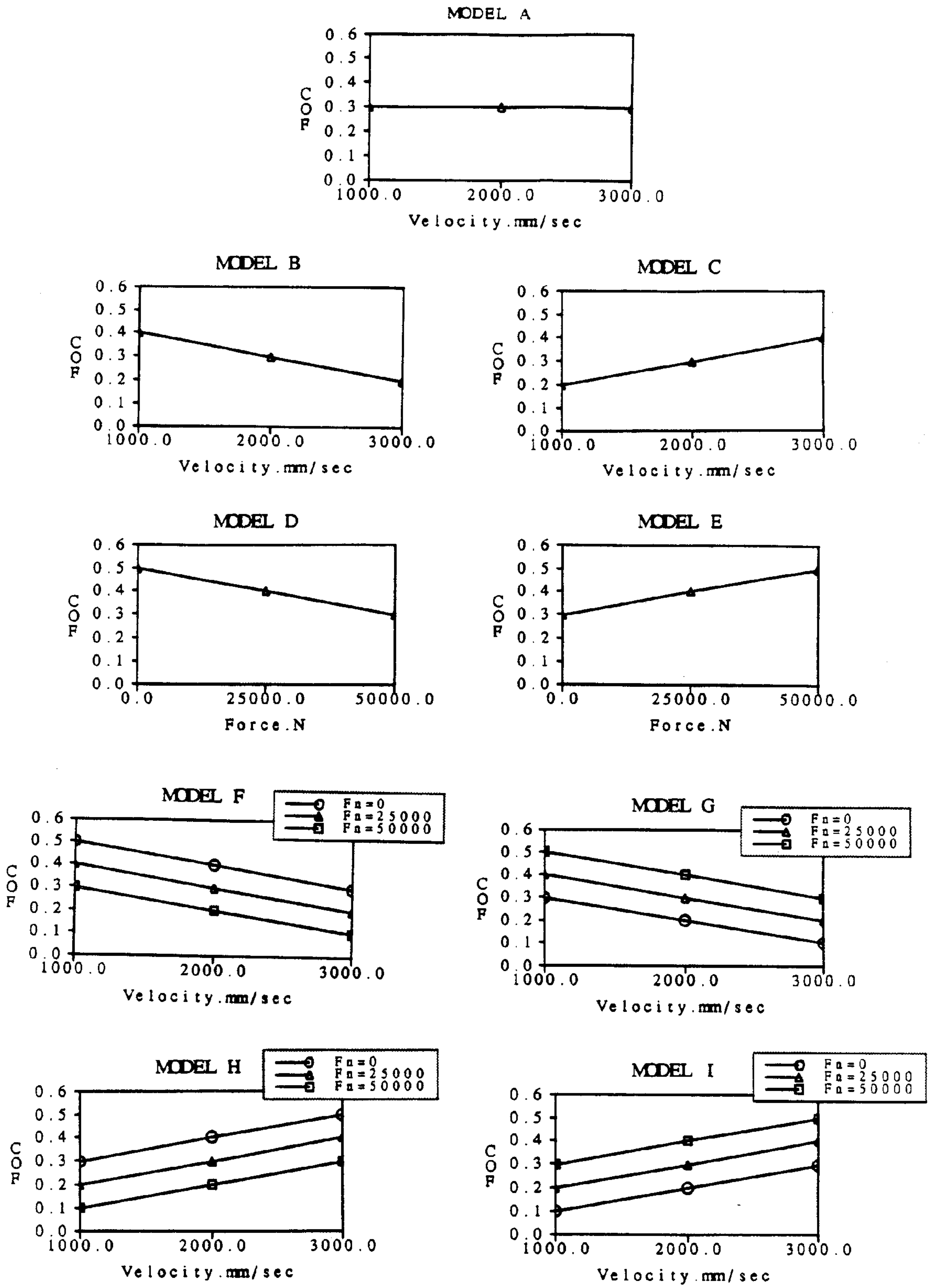


Figure 2. Coefficient of Friction (COF) in Function of Tangential Velocity and/or Contact Force

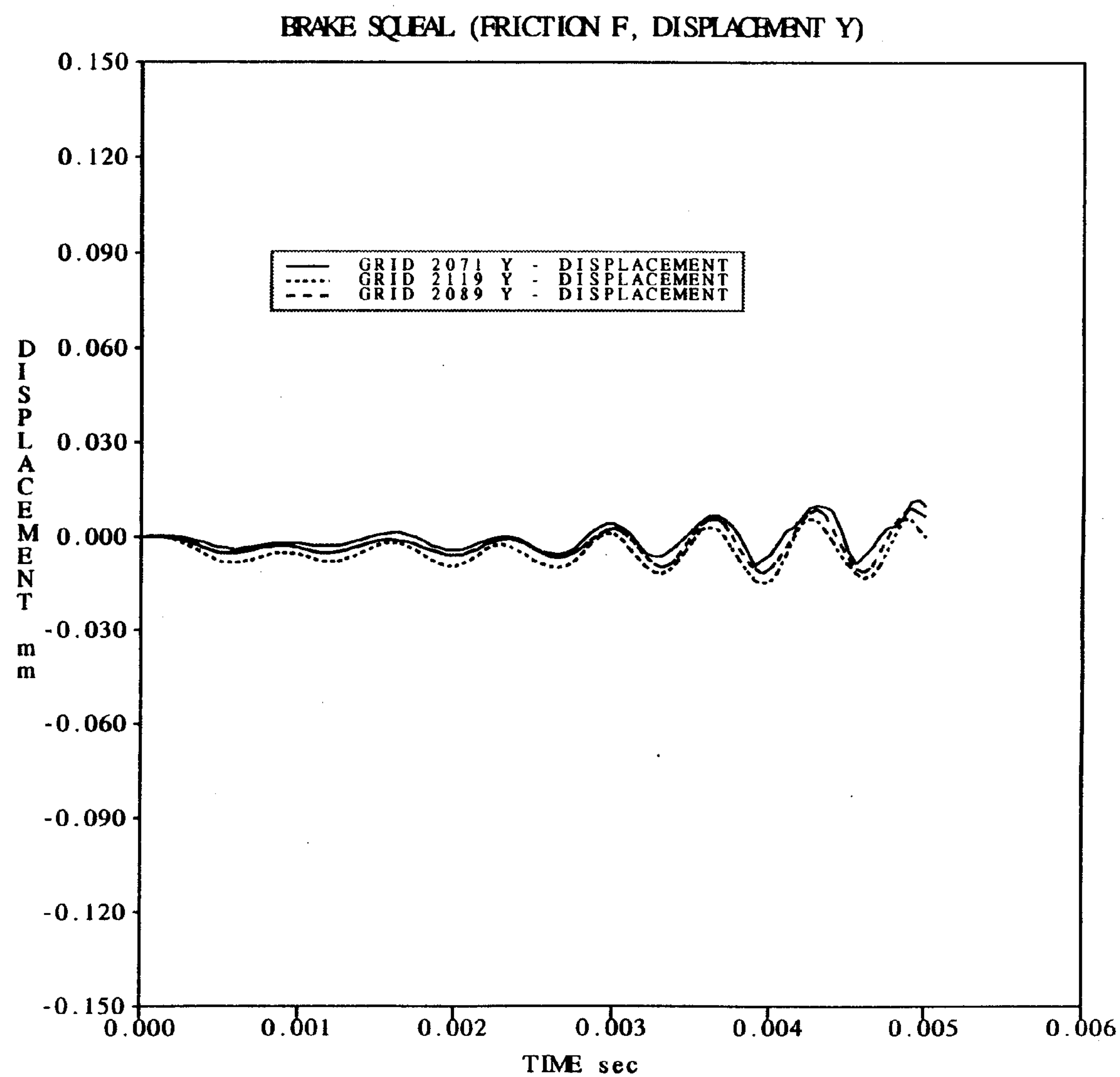


Figure 3. Time histories of Displacement at Selected Nodes (Friction Model F)

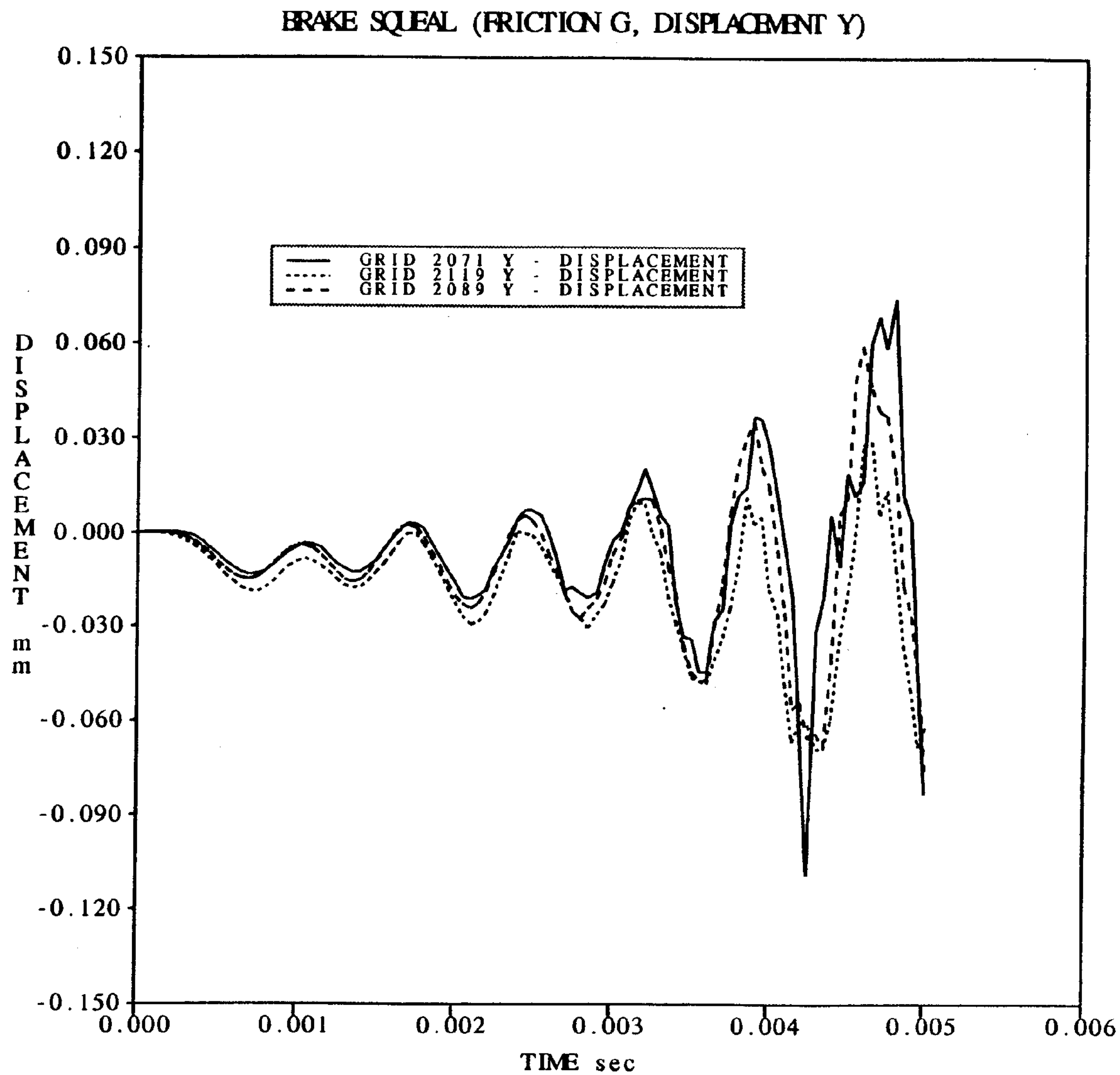


Figure 4. Time histories of Displacement at Selected Nodes (Friction Model G)

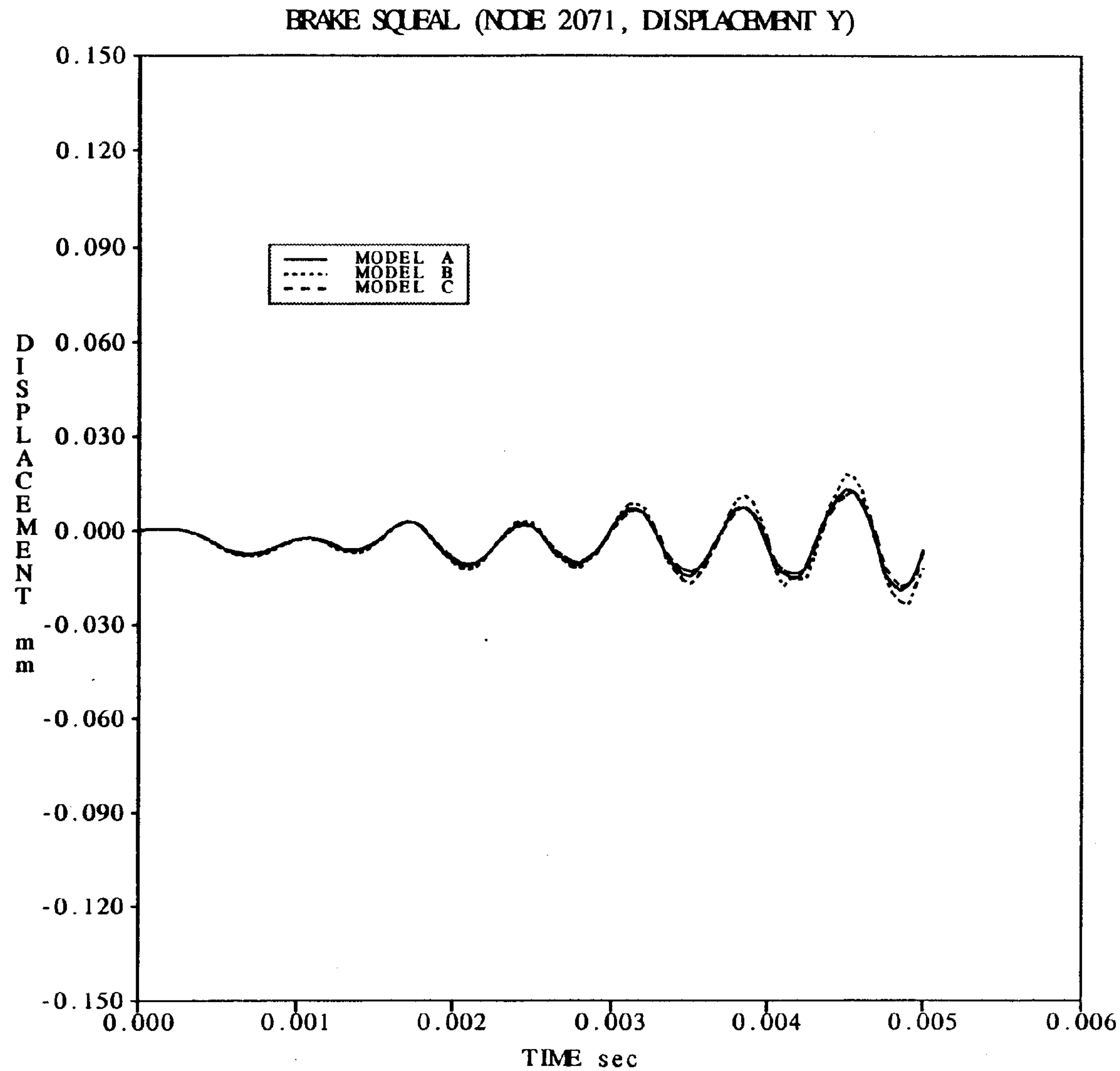


Figure 5. Time histories of Displacement at Node 2071. (Friction Models A, B and C)

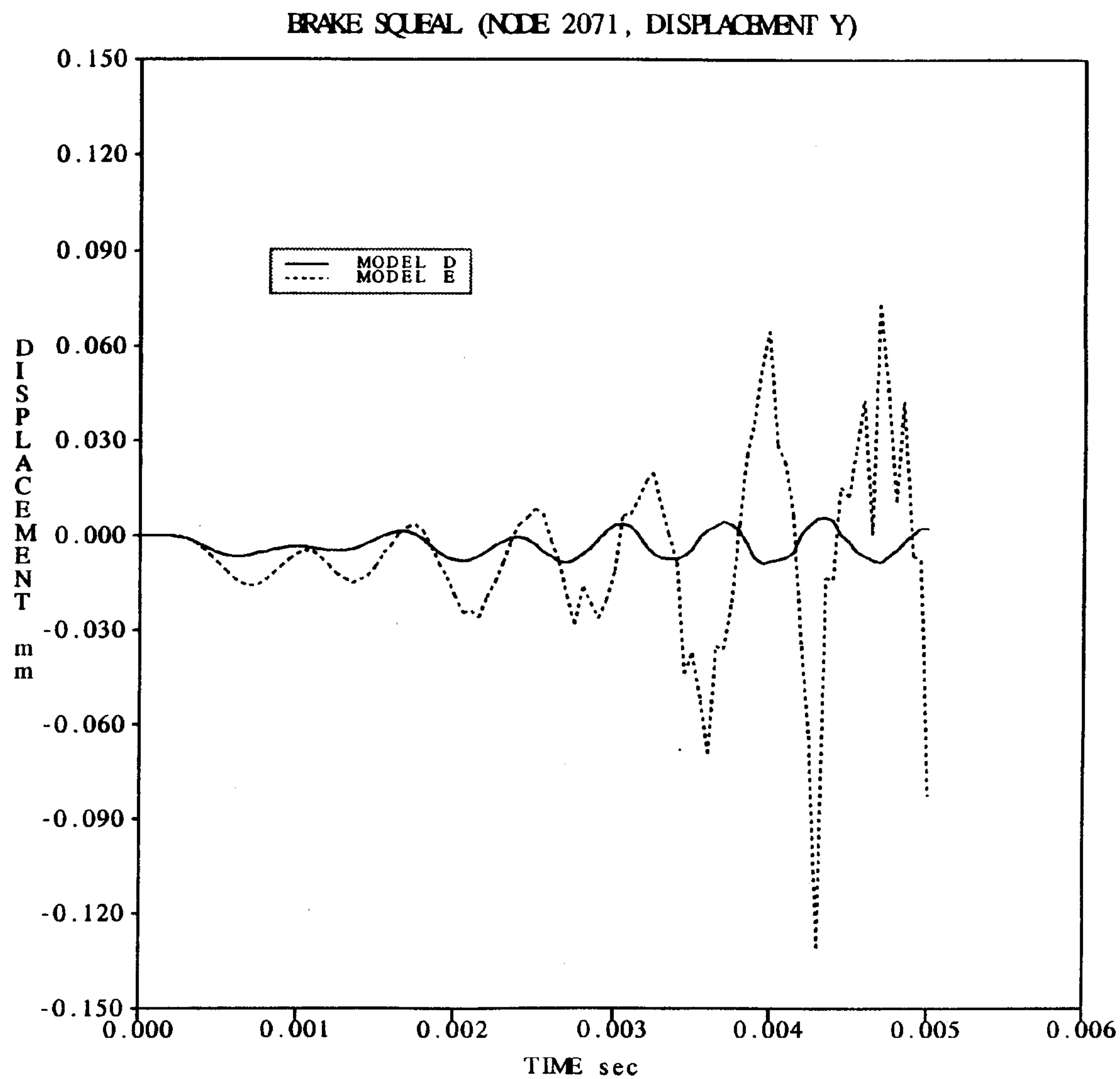


Figure 6. Time histories of Displacement at Node 2071. (Friction Models D and E)

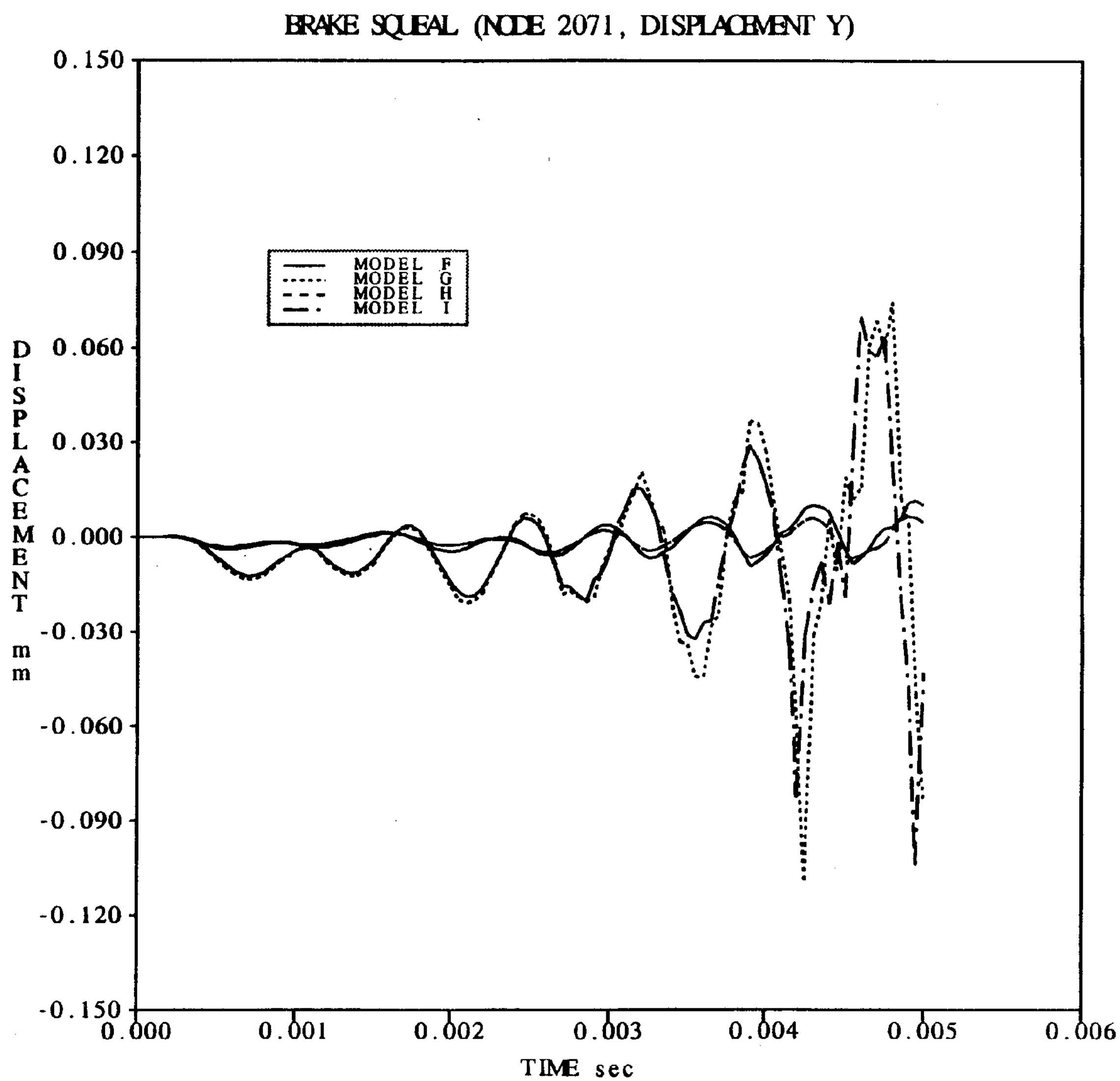


Figure 7. Time histories of Displacement at Node 2071. (Friction Models F, G, H and I)