Modeling Friction-Induced Dynamic Instabilities Using Flexible Multi-Body Dynamics with ADAMS

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ABSTRACT

This paper presents the use of flexible multi-body dynamics within ADAMS for simulating nonlinear system dynamic events. The nonlinear dynamic system being modeled in this case is a friction-induced dynamic instability. The ADAMS model consists of rigid and flexible bodies. The flexible body is represented by an assumed set of vibration modes which are obtained from a finite element modal analysis of the flexible body. The modal representation of the flexible body is then included in the ADAMS model through GSE statements which couple the rigid body dynamics equations to those of the flexible body dynamics equations. Simulation results of the friction-induced dynamic instability are presented. The simulation results confirm the hypothesis that friction-induced dynamic instability occurs when two vibration modes that are coupled by friction forces coalesce in their natural frequencies. The numerical results are in good agreement with observations from tests.

Introduction

This paper presents preliminary results on the use of flexible body dynamics with ADAMS in simulating nonlinear system dynamics events in which component flexibility plays an important role. One such event is the case of self-sustaining oscillations of a flexible (*i.e.*, not infinitely rigid) axle/leaf spring assembly, where such oscillations are induced by friction between the brake shoes and the brake drum. The mechanism giving rise to self-sustaining oscillations is the dynamic coupling of the brake shoe radial motion (also referred to as the *normal oscillation*) and of the spring wrap up mode or axle torsion mode of the brake-axle-suspension system (also referred to as the *tangential oscillation*). Friction between the brake shoes and the brake drum causes coupling between the normal oscillation and the tangential oscillation, which in turn results in modal (frequency) coalescence under certain conditions. When modal coalescence occurs, a vibration mode momentarily becomes unstable and this vibration mode eventually becomes a stable limit cycle.[1] The stable limit cycle is characterized by large amplitude, self-sustaining oscillations.

The approach taken in the present modeling effort is to perform time-domain, nonlinear dynamic simulations of the brake application process and to determine under what conditions dynamic instability will most likely occur. Several options are available for modeling the brake-axle-suspension system. These range from simple rigid body models to fully nonlinear finite element models with friction-contact elements. During the initial stages of this study, a simple ADAMS model of the brake-axle-suspension system consisting of rigid lumped masses was developed. This simple model was useful in the sense that it enhanced the understanding of the modal coupling mechanism that drives the dynamic instability. However, the multi-rigid-body model was not able to reproduce some of the observations in the test, and this shortcoming was due to the existence of complex vibration modes that were not captured by the multi-rigid-body model. At this stage of the modeling process, two options were available for improving the brake system model, namely: 1) developing a fully nonlinear finite element model with friction-contact elements; or 2) creating a multi-body dynamics model consisting of rigid and flexible bodies. In general, the proper modeling approach depends on the type of problem being considered. For instance, if we consider the instability problem that is governed by the vibration modes of the brake shoe and the brake drum, a fully nonlinear finite element analysis is required to simulate the event. An analysis of this type would involve lengthy execution times and would require tremendous amounts of data storage. On the other hand, if we consider the instability problem that is governed by the flexibility of the axle and the leaf spring suspension, the brake components can be treated as rigid bodies and a flexible multi-body dynamics approach can be utilized to simulate the nonlinear system dynamic event. In contrast with nonlinear finite element analysis, a flexible multi-body dynamic analysis in ADAMS involves execution times which are orders of magnitude less than nonlinear finite element analysis and requires much less data storage. Hence, in simulating dynamic instability in which the flexibility of the axle/leaf spring subassembly plays an important role, the most cost-effective approach is to use multi-body dynamics in modeling the brake-axle-suspension as a system of rigid and flexible bodies.

Model Description

A schematic drawing of a three-dimensional ADAMS model of an S-cam brake assembly is shown in Figure 1. The brake assembly consists of the following parts modeled as rigid bodies: 1) brake spider; 2) S-cam and cam shaft; 3) slack adjuster; 4) push rod; 5) air chamber; 6) leading shoe and roller; 7) trailing shoe and roller; and 8) brake drum. These parts are connected by mechanical constraints or by internal forces, the magnitudes of which are dependent on the system configuration (in other words, the dynamic system is autonomous, except for the prescribed nominal air chamber pressure).

The rotation of the brake shoes with respect to the brake spider is related to the rotation of the S-cam through the cam-to-shoe roller contact which is modeled by point-to-curve (PTCV) constraints. Moreover, the shoe rollers are allowed to lift-off the cam. The interaction of the drum and brake shoes are modeled by discrete forces along the lining of each brake shoe. Each contact force is treated as a unilateral, nonlinear spring and damper so that the normal force between the drum and the shoe at a particular interface point can only be a compressive force or zero. The tangential or friction force between the drum and the shoe is a function of the normal force and the relative velocity between the drum and the shoe, evaluated at the contact point. Specifically, the friction force is the product of the normal force and the coefficient of kinetic friction, and the direction of the friction force is determined by the tangential velocity of the drum relative to the shoe.

The air pressure force between the air chamber and the push rod consists of the nominal (specified) air pressure and the feedback air pressure coming from the air spring action. The air pressure force is a nonlinear function of the displacement of the push rod, and this nonlinearity is crucial in determining the operating conditions at which dynamic instability is most likely to occur. The remaining forces are modeled as linear springs and dampers.

Two brake assemblies as described above are attached to the ends of a front axle. The front axle also supports the air chamber through the air chamber bracket, and the front axle is attached to the vehicle by leaf springs. The front axle, air chamber brackets and the leaf springs are modeled as a subassembly of flexible bodies. This subassembly of flexible bodies is combined with the multi-rigid-body dynamics model by employing an assumed modes method (AMM) of formulation for the dynamics of flexible bodies.[2] The assumed modes are obtained from a modal analysis of a finite element model of the axle-bracket-leaf spring subassembly as shown in Figure 2. The modal flexibility representation of the subassembly is then included in the ADAMS model by means of GSE statements which combine the rigid body dynamics equations with those of the flexible body dynamics equations. The resulting system of equations form a set of nonlinear equations wherein the rigid body coordinates are coupled with the flexible body's modal coordinates.

Simulation Results

Time-domain simulations were performed on the ADAMS model of the brake-axle-suspension system described above. A light brake application process was simulated by stepping up the nominal air chamber pressure to a steady state value. Four different values of the kinetic friction coefficient were investigated. In each of the four cases, there was a phase shift (time delay) between the brake actuator at the driver's side and the brake actuator at the passenger's side in order to excite the axle torsion (anti-symmetric leaf spring bending) mode.

Figure 3 shows the friction force between the brake shoe and the brake drum at a discrete contact point. This figure shows that the fluctuations in the friction force increase as the coefficient of kinetic friction increases. For low values of the kinetic friction coefficient, the transient oscillations are eventually damped out. In contrast, for high values of the kinetic friction coefficient, the transient oscillations give rise to high amplitude, self-sustaining oscillations. An eigenvalue analysis of the brake system reveals that the frequency of the brake shoe radial motion (also referred to as *normal oscillation*) is close to the frequency of the axle torsion/anti-symmetric leaf spring bending mode of vibration (also referred to as *tangential oscillation*) over a limited range of the steady state air chamber pressure. The presence of friction couples these two modes of oscillation, and when the frequencies of these two modes coalesce, the system becomes momentarily unstable and enters into a stable limit cycle. It is worthwhile to note that this phenomenon is a characteristic of a nonlinear dynamic system, therefore this type of system behavior can not be predicted by the use of linear structural dynamic analysis codes.

Figure 4 shows the pitch plane orientation of the brake spider. This figure shows that the brake spider (and the whole brake assembly) starts to "rock" when the brake shoes engage the brake drum. Again, we can observe the destabilizing effect of the kinetic friction coefficient. Figure 5 shows the corresponding fore-aft deflection at a spider-axle attachment point, and Figure 6 shows the most dominant vibration modes that contribute to the deflection of the axle/leaf spring assembly. This figure shows that the axle torsion/anti-symmetric leaf spring bending mode (dashed curve denoted by Q4) is indeed the mode that coalesces with the brake shoe radial motion.

Conclusion

In this paper, the use of flexible body dynamics within ADAMS in order to simulate nonlinear dynamic instability has been presented. The flexible body is described by an assumed set of modes which are derived from a finite element modal analysis of the flexible body. This set of assumed modes is included in the ADAMS model through the use of GSE statements which couple the rigid body dynamics equations with those of the flexible body dynamics equations. The modal description of the flexible body plays an important role in characterizing the dynamic instability. The presence of friction couples two vibration modes whose natural frequencies are dependent on system parameters. Under certain conditions, the natural frequencies of the two coupled modes may coalesce, and consequently, the system momentarily becomes unstable and enters into a stable limit cycle. This limit cycle is characterized by high amplitude, self-sustaining oscillations.

References

1. W. W. Tworzydlo, E. B. Becker and J. T. Oden, "Numerical modeling of friction-induced vibrations and dynamic instabilities", ASME Publication DE-Vol. 49, pages 13-32, 1992.

2. A. A. Shabana, "Substructure synthesis methods for dynamic analysis of multi-body systems", Computers & Structures, vol. 20, no. 4, pages 737-744, 1985.

front model=q15r

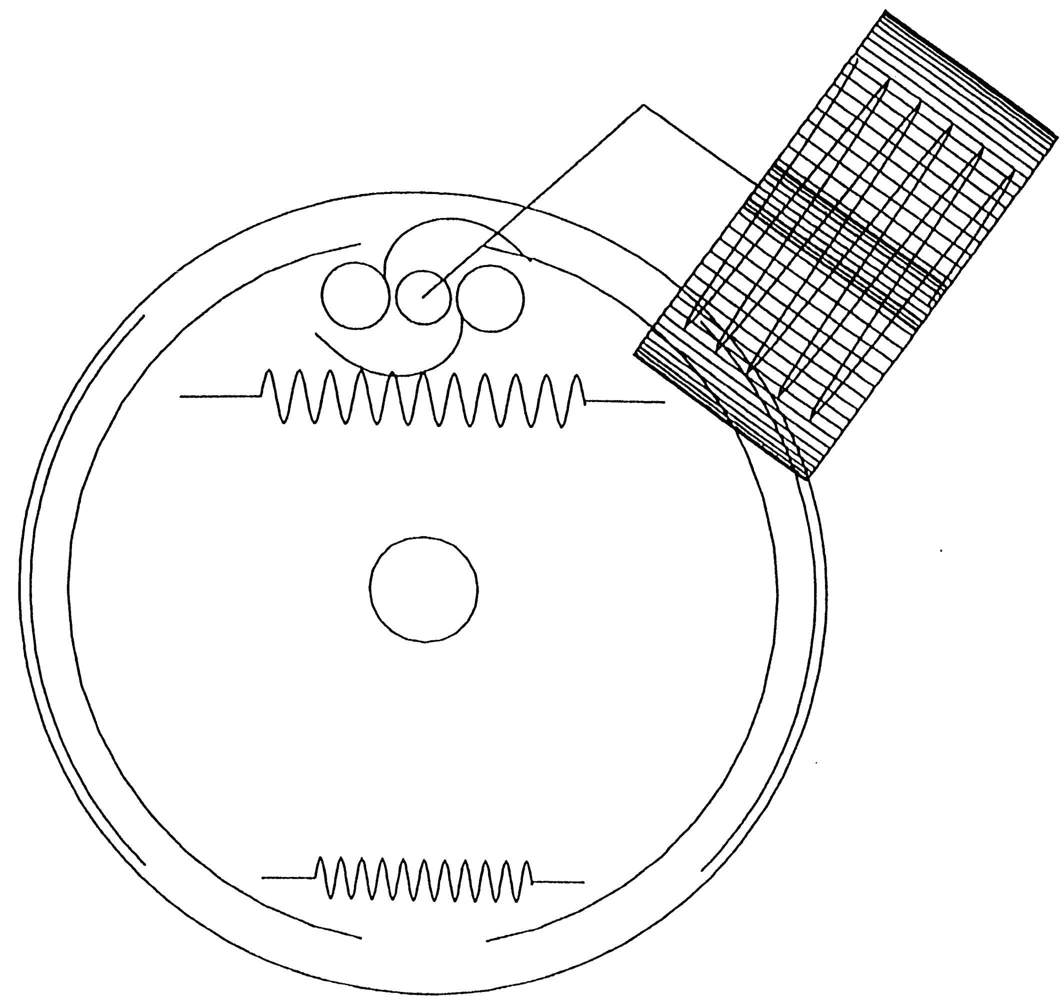
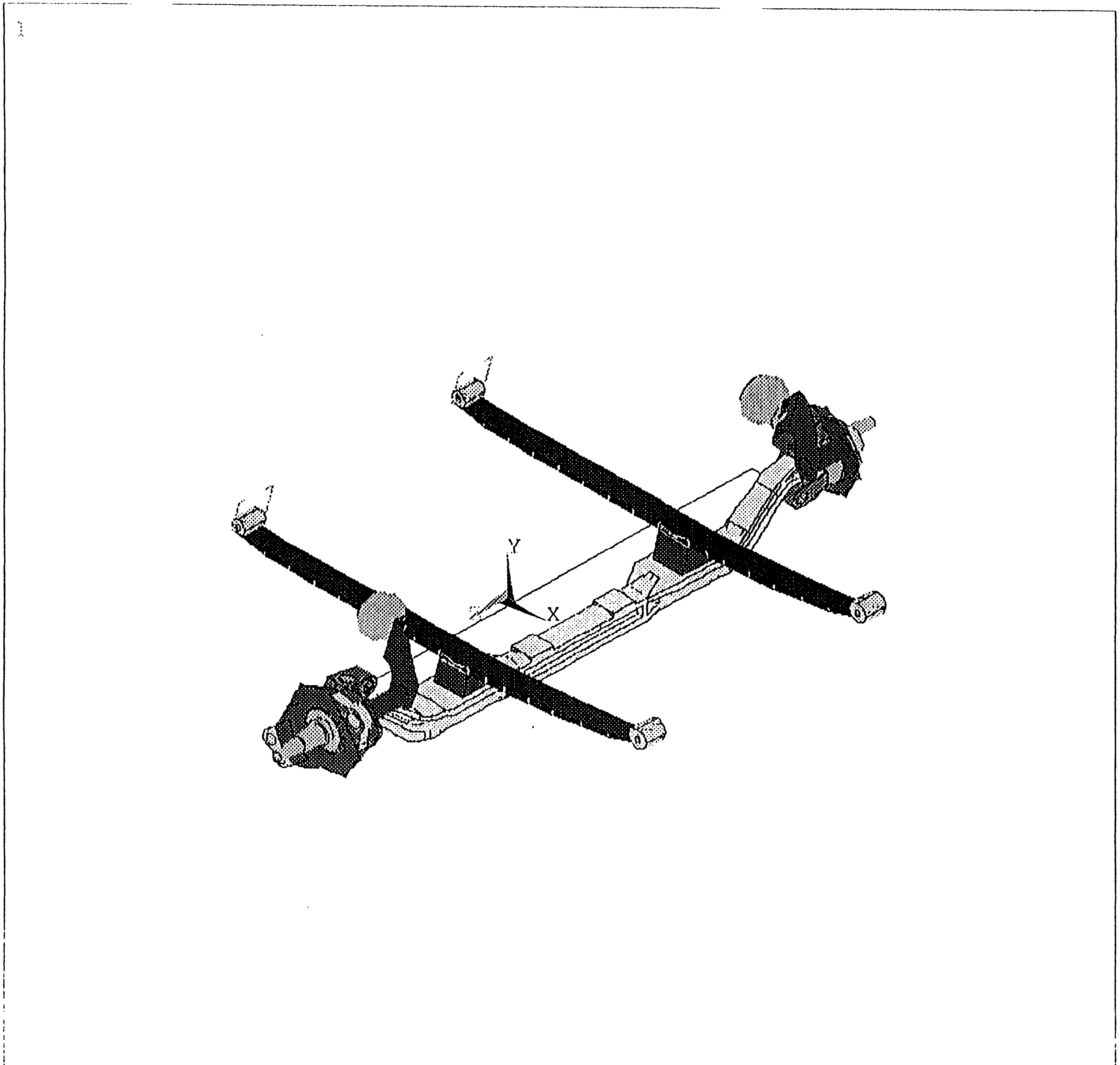


Fig 1



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EDGE

FRONT AXLE/SPIDER/SUSPENSION SYSTEM

Fig 2

Brake Shoe/Brake Drum Friction Force vs. Kinetic Friction Coefficient

friction force between leading brake shoe and brake drum

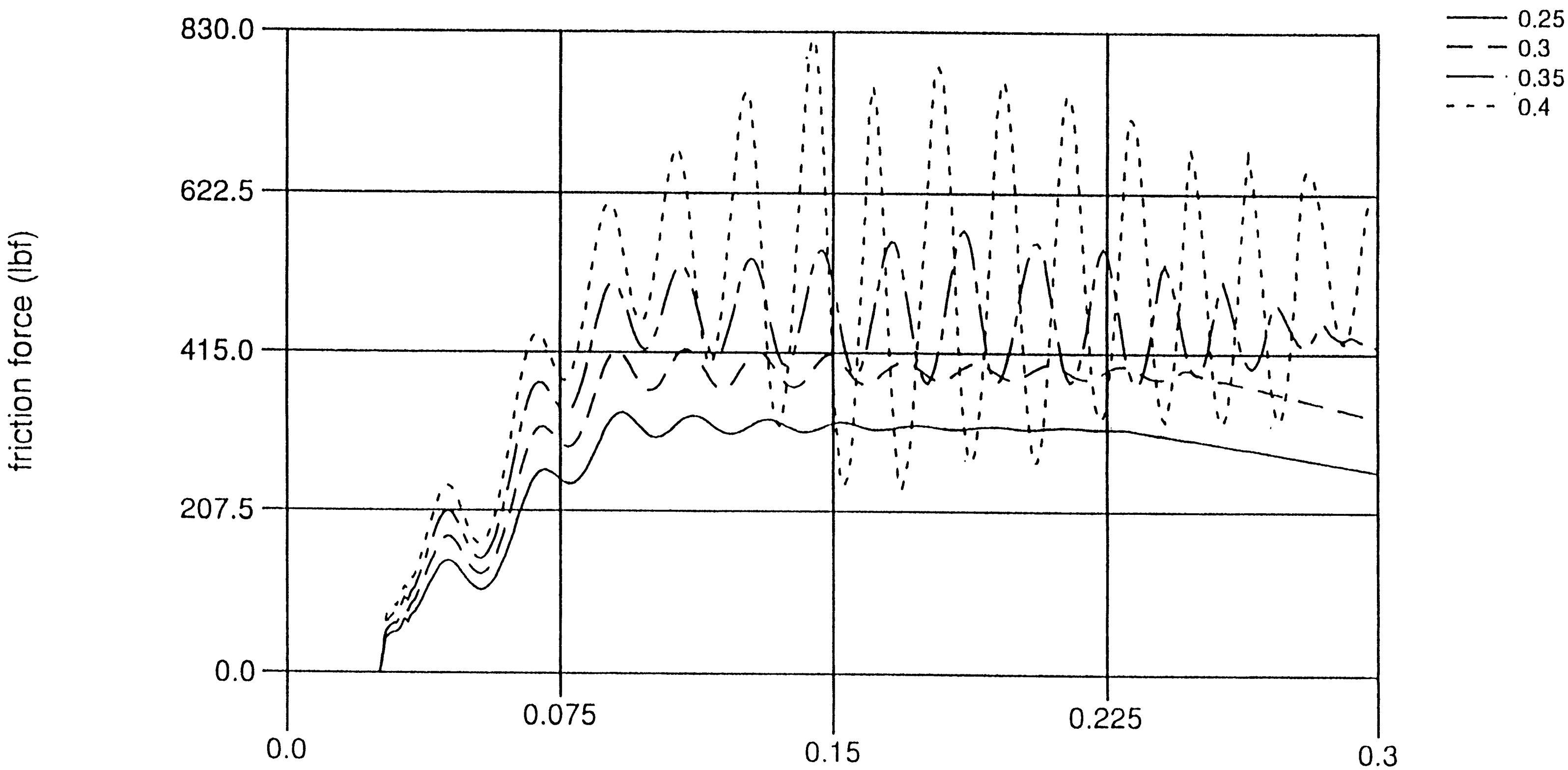

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Fig. 3

Brake Spider Pitch Angle vs. Kinetic Friction Coefficient

pitch plane orientation of brake spider

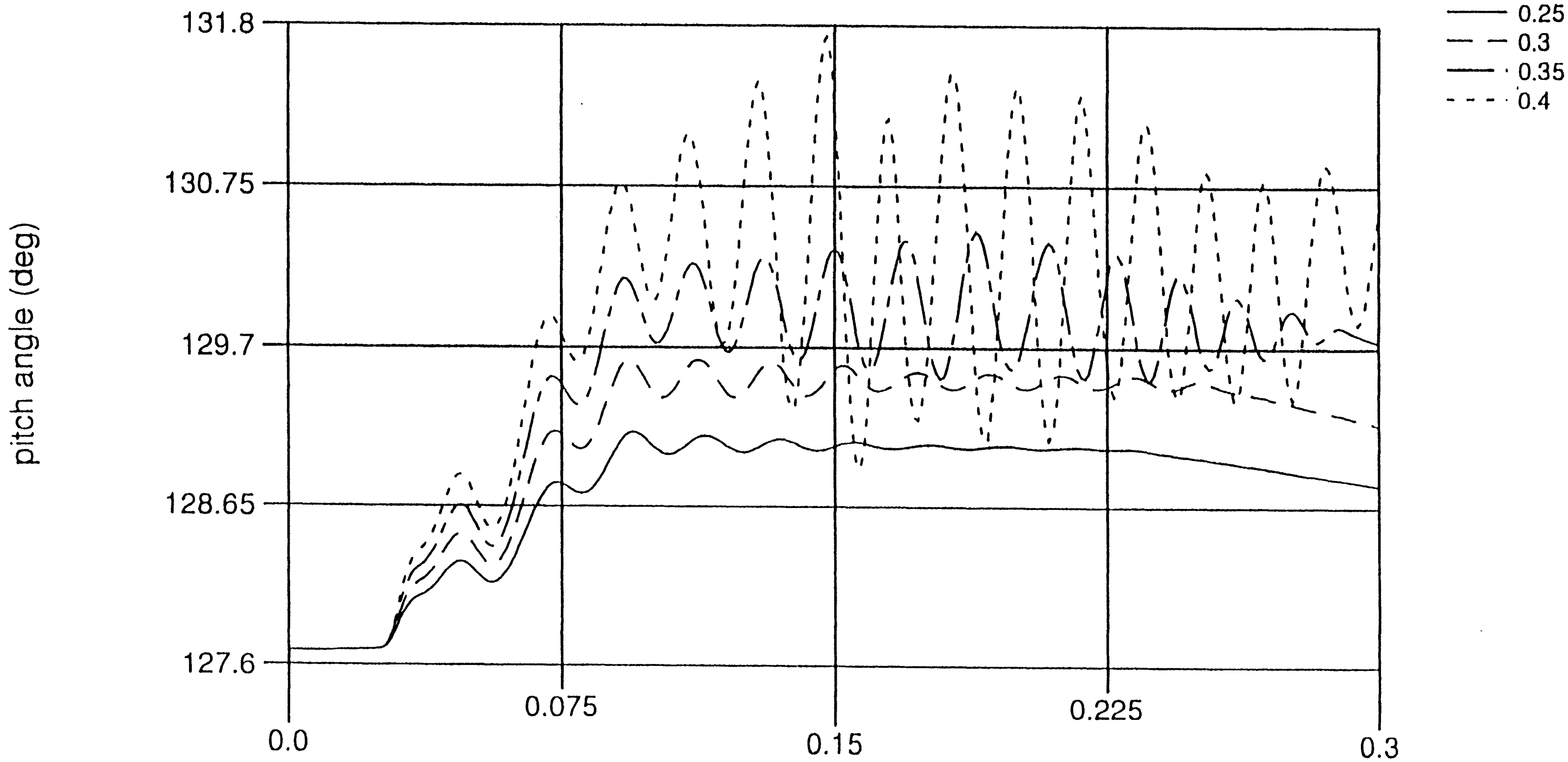

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Fig. 4

Axle/Suspension Fore-Aft Deflection vs. Kinetic Friction Coefficient

fore-aft deflection of node 1438 (LHS top spider attachment)

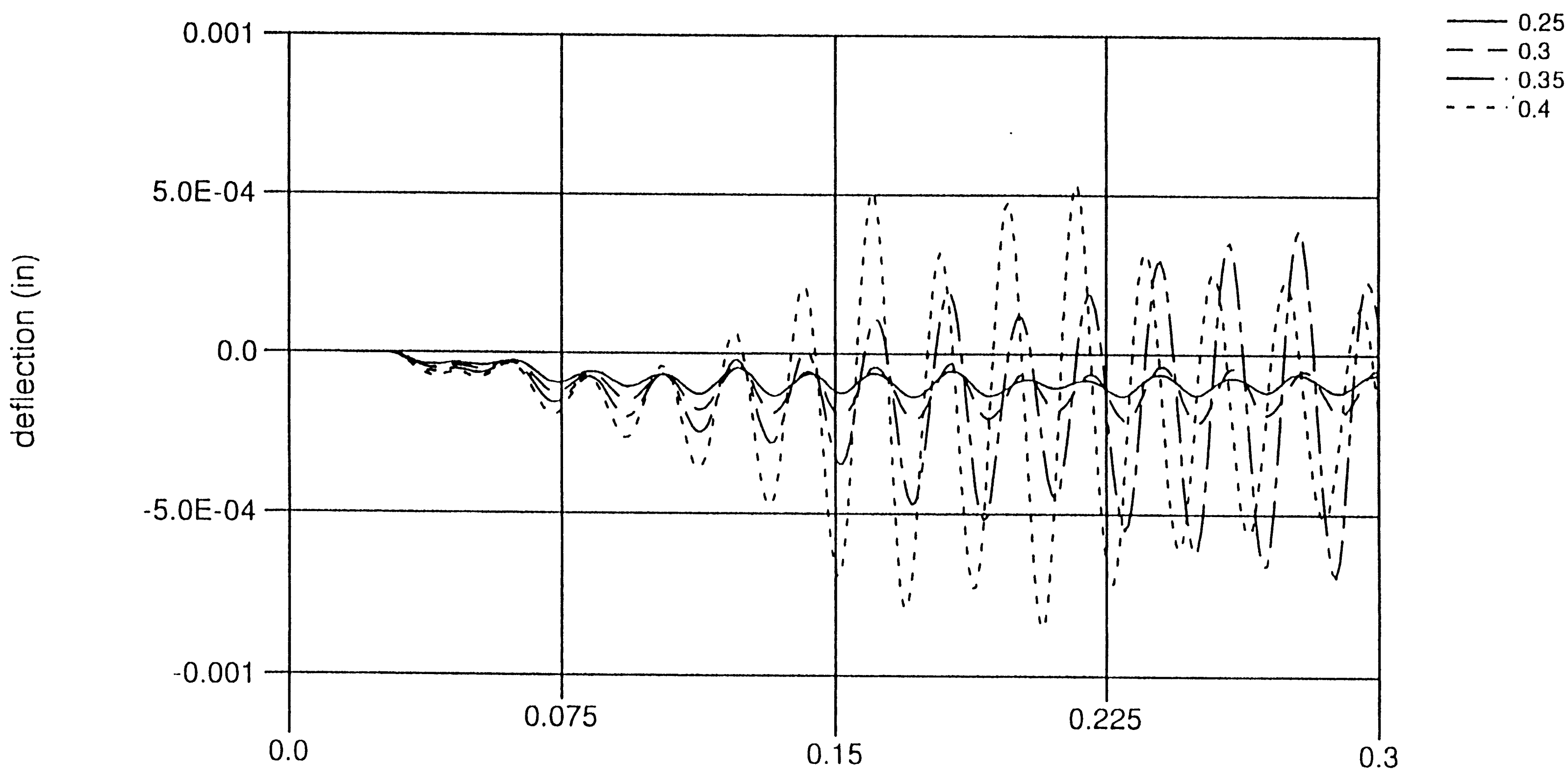

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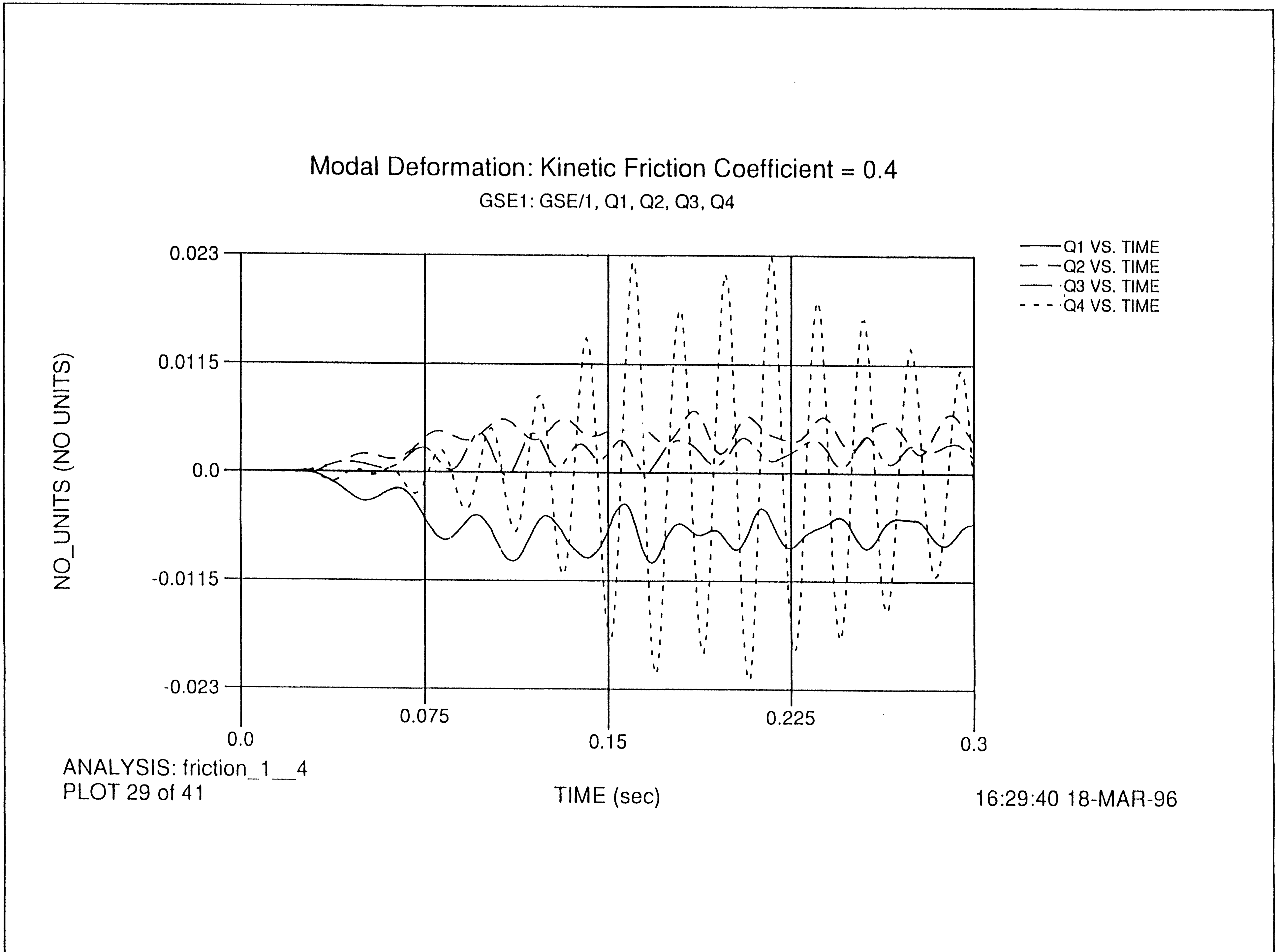
Fig 5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Q2 VS. TIME
Q3 VS. TIME
Q4 VS. TIME
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0.0
-0.0115
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0.0
0.075
0.15
0.225
0.3
TIME (sec)
ANALYSIS: friction_1__4
PLOT 29 of 41
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Fig. 6