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A TECHNIQUE FOR THE VALIDATION OF VEHICLE MODELS USING THE ROAD SIMULATOR

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A TECHNIQUE FOR THE VALIDATION OF VEHICLE MODELS USING THE ROAD SIMULATOR

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INTRODUCTION

One of the more important aspects of vehicle design is mechanical mobility. This term is a measure of how fast a vehicle can get from Point A to Point B without breaking and yet preserving the cargo and maintaining driver comfort. Mechanical mobility may be divided into five categories: Rough terrain mobility, soft terrain mobility, water mobility, vegetation mobility, and high speed mobility. Each of these five mobility divisions may be subdivided into ride comfort (including cargo), durability and handling. The TACOM road simulator has been used extensively to investigate the durability and ride characteristics of existing and proposed military vehicles relative to rough terrain and high speed mobility. The real payoff for simulation testing, however, lies not in the durability testing of production vehicles and subsystems but in using it as a tool to interface computer models with prototype hardware to evaluate subsystems prior to integration to form a prototype vehicle system; and finally the road simulator would be used to perform the initial performance and durability evaluation of the new vehicle prior to field tests.

The process from the vehicle concept to prototype field testing is an iterative process using the methodology proposed in this paper. The first step in the iteration is to formulate a mathematical model using the dynamic equations of motion for the suspension system and relevant components of a vehicle concept. This model is then subjected to the mission profile for which the vehicle is intended. The suspension parameters are then tuned to minimize a

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cost factor, which is a function of ride, handling, cargo comfort, speed, etcetera. The second step is to manufacture prototype suspension components having the parametric characteristics derived from the mathematical model. These components are then evaluated to determine their real parametric characteristics. The third step then is to plug the measured parameters into the mathematical model and reevaluate to determine the acceptability of any parametric changes. The fourth step is to build a prototype vehicle, install it on the road simulator and subject the vehicle system to the mission profile. The fifth step is the final adjustment of the mathematical model to match its response to the response of the vehicle on the road simulator. This model may now be used for any dynamic studies on the vehicle which precipitate from field tests.

This paper will be limited to a description of some of the techniques which have been used at TACOM within the Surface Mobility Division aimed at validating mathematical models.

CONCLUSIONS AND RECOMMENDATIONS

It is apparent from this study that although the simple model tracks the hardware fairly well, a more sophisticated model is needed. This study also shows that the actuator to vehicle interfaces and centering radius rods have a significant effect on the response of the vehicle. These effects should either be minimized during initial simulator design or should be included as inputs to the computer model. The techniques used for comparing responses are adequate except for the possible addition of computing a correlation coefficient. Some recommendations related to the mathematical model are:

1. Treat the frame as an elastic member.

2. Separate the following into separate mass-spring-damper systems.

    a. Cab.

    b. Engine.

    c. Cargo Box.

3. Include roll in the simulation.

4. Increase the number of transducer locations and hence the number of correlation points.

No ride optimization or parameter adjustment was attempted due to the oversimplification of the model which was a result of inadequate computer capacity. A computer with sufficient capacity to continue this study has been acquired by the Surface Mobility Division at TACOM. The XM705 program is complete so continued model validation should be performed using some other vehicle scheduled for evaluation on the road simulator.

BACKGROUND

The introduction to this paper has outlined a proposed methodology for the computer aided design and development of a vehicle from initial concept through the prototype phase and into field testing. This paper will focus specifically on the fifth step which is the comparison of the model response with the response of the vehicle system on the road simulator. The objective of this study is to initiate steps to fill that particular gap in the proposed methodology. Previous work within the Surface Mobility Division at TACOM has resulted in a continuous parameter tracking technique for determining spring rates and damping coefficients for non-linear suspension components which complements the classical direct reading techniques for measuring vehicle parameters. The work contained in this paper was initiated to supplement a durability test on the XM705 1¼-Ton, 4X4, Cargo Truck to be performed using the road simulator. The XM705 is shown mounted to the road simulator in Figure 1.

The instrumentation required for the durability test was sufficient to allow this "first cut" mathematical model validation to be made. The mathematical model was, therefore, designed to utilize the existing test setup.

MODEL FORMULATION

The vehicle model was limited to four degrees of freedom in the pitch and bounce modes. At the time this study was initiated, sufficient analog computer capacity to include roll freedom was unavailable. Since this was to be the initial attempt at tracking vehicle responses to model responses in real time and since roll could be eliminated on the road simulator, the elimination of roll freedom in the model would not invalidate the technique. Further simplifying assumptions were as follows:

1. The sprung mass is modeled as one rigid body.

2. The pitch angle, θ, is assumed small such that $\sin\theta = \theta$ and $\cos\theta = 1$.

The vehicle is coupled through the wheel spindles to the road simulator as shown by the arrows in Figure 1 thus eliminating the tires and unsprung masses from the system model. The model then may be represented as shown in Figure 2.

Equations of Motion.

The free body diagram of the sprung mass is shown in Figure 3.

Where:

$\ddot{y}_{CG}$ = Vertical acceleration of the center of gravity.

$\ddot{\theta}$ = Pitch acceleration.

a = Distance from center of gravity to the center of the front suspension = 6.15 ft.

b = Distance from center of gravity to the center of the rear suspension = 5.10 ft.

F_{FT} = Total front suspension force.

F_{RT} = Total rear suspension force.

m = Sprung mass = 139.75 slugs.

I_θ = Sprung mass pitch inertia = 750 slug-ft^2

The equations of motion for this system then are:

$$m\ddot{y}_{CG} = F_{FT} + F_{RT} \pm mg \tag{1}$$

$$I_\theta \ddot{\theta} = aF_{FT} - bF_{RT} \tag{2}$$

The total suspension force is a summation of the spring force, shock absorber force and frictional forces. The spring forces and shock absorber forces are obtained from curves shown in Figures 4, 5 and 6. The interleaf friction for the leaf springs was measured in the laboratory. This friction is proportional to the spring force. The constant of proportionality fell in the range 10 to 15 percent of the spring load, 10 percent for the rear springs with three leaves and 15 percent for the front springs with five leaves.

Equations 1 and 2 may now be rewritten

$$m\ddot{y}_{CG} = F_{FS} + F_{FD} + F_{FF} + F_{RS} + F_{RD} + F_{RF} - mg \tag{3}$$

$$I_o \ddot{\theta} = aF_{FS} + aF_{FD} + aF_{FF} - bF_{RS} - bF_{RD} - bF_{RF} \tag{4}$$

Where F_{FS} and F_{RS} = front and rear spring forces

F_{FD} and F_{RD} = front and rear shock absorber forces

F_{FF} and F_{RF} = front and rear friction forces.

Also from the above discussion:

$$F_{FF} = .15\ F_{FS}$$

and $$F_{RF} = .10\ F_{RS}$$

Samples of the analog computer circuits required for the model simulation are shown in Figures 7 through 12.

ROAD SIMULATOR

The road simulator used in this study and shown in Figure 1, imparts vertical excitations to the wheel spindles of the vehicle using four linear electro-hydraulic servo controlled actuators. These actuators are capable of 200 inches per second linear velocity, have twelve inches of usable stroke and respond to 100 hertz with a double amplitude of .0005 inches.

As mentioned previously, the vehicle is attached through its wheel spindles to the actuators. The left front attachment is indicated by the arrow in Figure 1. The interface hardware allows six degrees of freedom at each wheel in order not to unrealistically load the axles and suspension. The vehicle is attached through the wheel spindles (tires and wheels removed) to provide a more positive attachment to the actuators which in turn prevents vehicle ejection from the road simulator due to high amplitude inputs. One major benefit related to this paper, of removing the tire is that a reliable tire model is no longer required to validate the suspension model.

SINE WAVE RESPONSE

The computer model and the road simulator were excited simultaneously to a sine wave of five inches or maximum attainable double amplitude. The frequency was increased from .5 hertz to 8 hertz. Figure 13 shows the responses of the model and the vehicle at their centers of gravity to these excitation signals. Several conclusions may be drawn from comparing the recordings and also by considering observations made during each of the test runs:

1. The noise superimposed on the acceleration signal from the road simulator is a result of mechanical noise (system clearance) in the interfaces between the spindles and the actuators.

2. The attenuation of the acceleration signal from the road simulator at three hertz was a result of the vehicle's center of gravity scribing an ellipse instead of a vertical straight line. This was caused by the inputs of the radius rods which were used to center the vehicle on the road simulator.

3. Overall, the model response compared well with the response of the vehicle on the road simulator.

The next step is to compare responses to terrain related random excitation signals.

RANDOM RESPONSE

The model and the vehicle on the road simulator were then subjected to a field vibration related random input. Maximum actuator excursion was twelve inches. The results of this random response are shown in Figures 14 and 15. Figure 14 shows the time histories of the vertical frame accelerations at a point centered above the front suspension. The vehicle response lags the model response due to hydraulic lag and filter lag. A low pass filter (40Hz) was inserted at the accelerometer output to clean up the signal for ease of presentation and comparison. Figure 15 shows the power spectral density curves computed from the time signals in Figure 14. The resonant points at two and five hertz agree well. The reduced amplitude of the vehicle response at the five hertz resonant point was caused by the effects of the radius rods as explained in the sine wave response discussion.

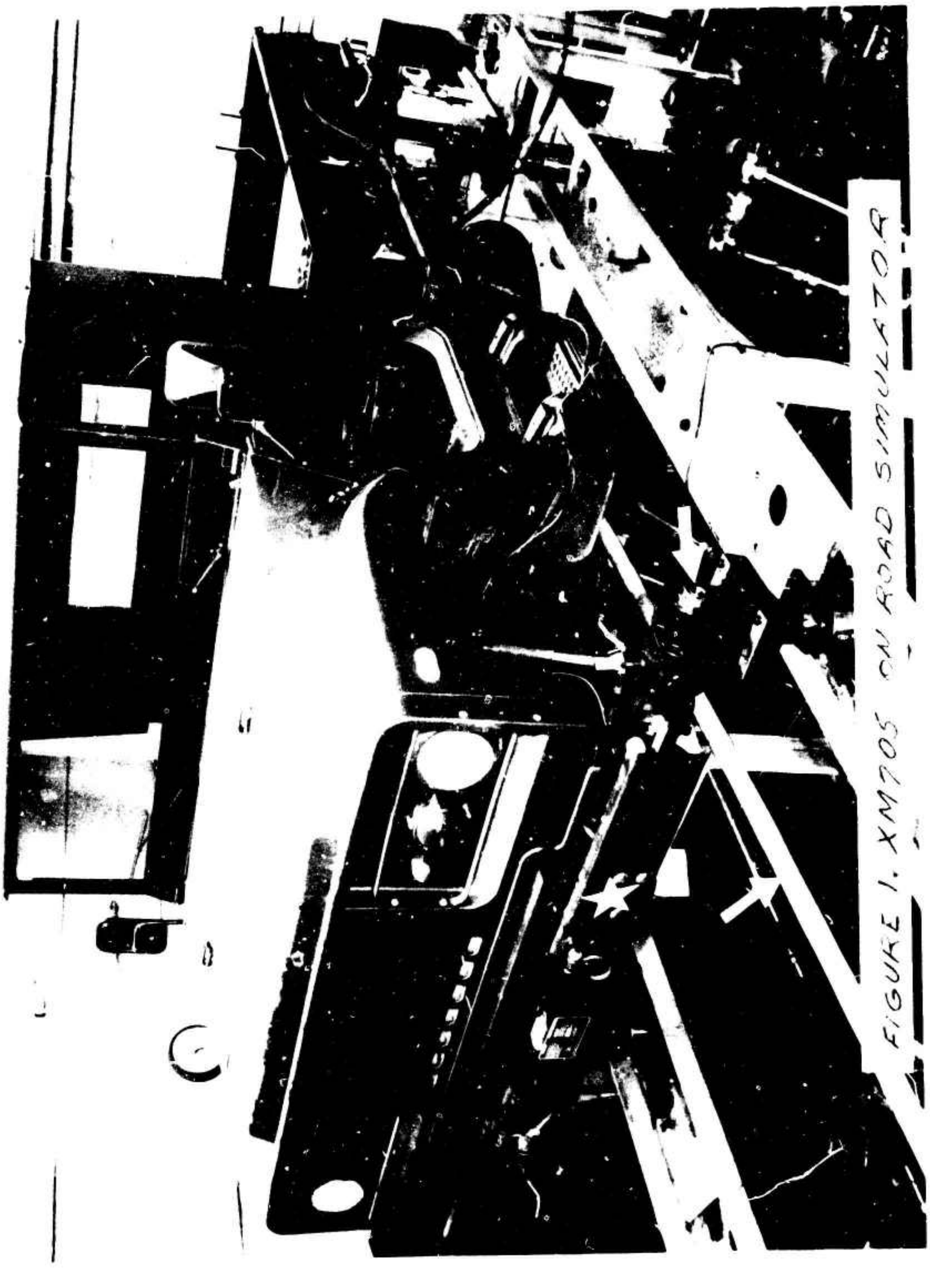


FIGURE 1. XM705 ON ROAD SIMULATOR

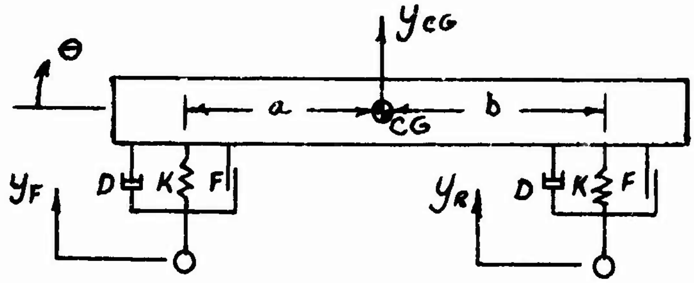


FIGURE 2. MODEL DIAGRAM

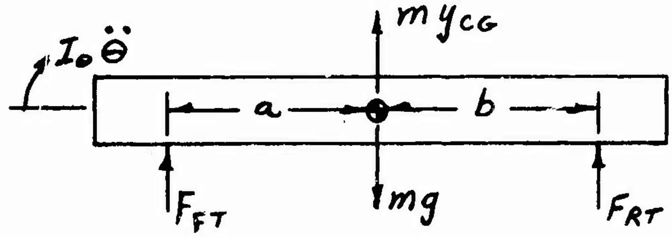


FIGURE 3. FREE BODY DIAGRAM

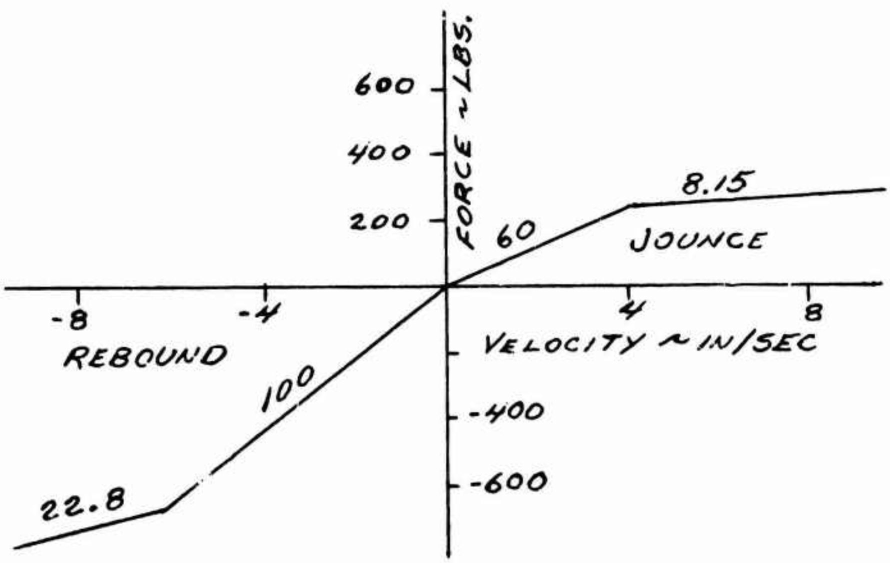
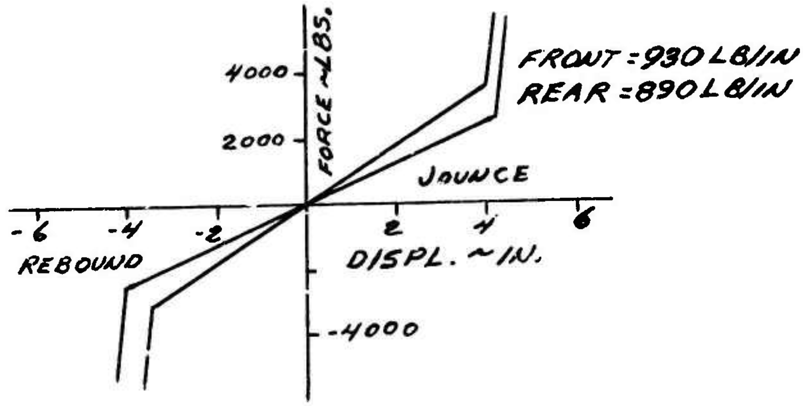


FIGURE 4. SHOCK ABSORBER CURVE



FIGURES 5&6. FRONT & REAR SPRING CURVES

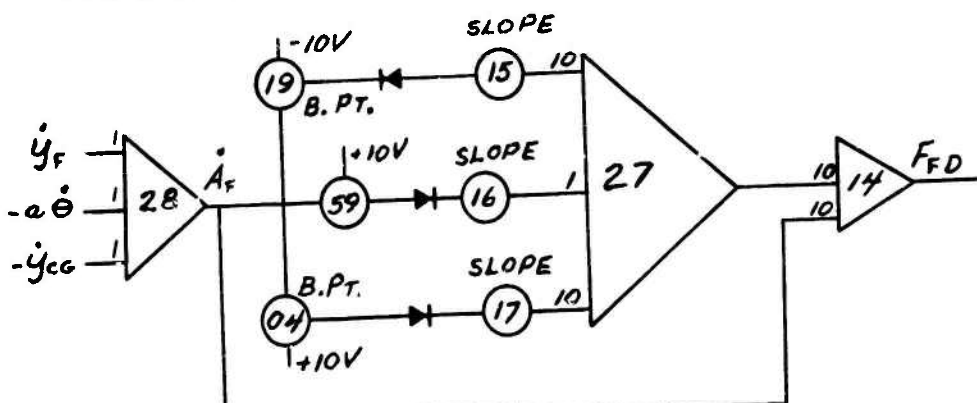


FIGURE 7. SHOCK ABSORBER ANALOG DIAGRAM

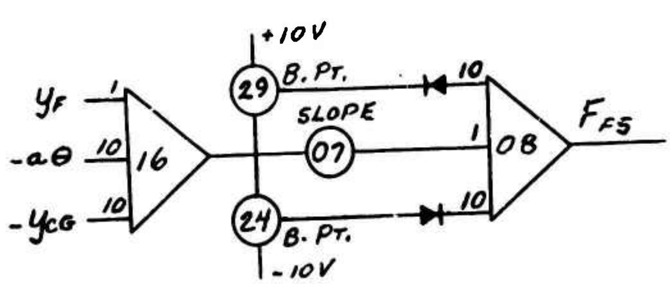


FIGURE 8. SPRING FORCE ANALOG DIAGRAM

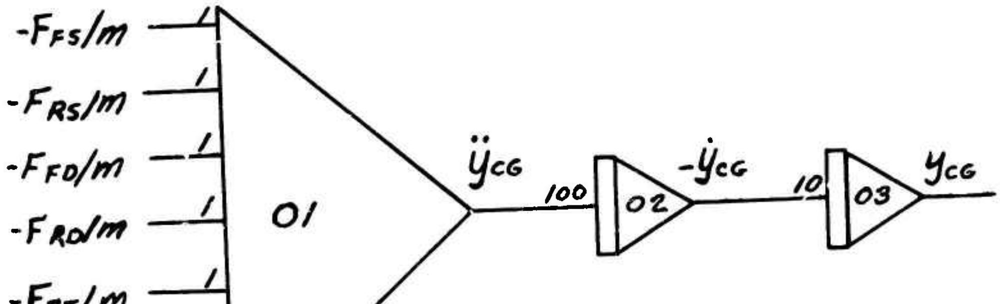


FIGURE 9. SPRUNG MASS BOUNCE ANALOG DIAGRAM

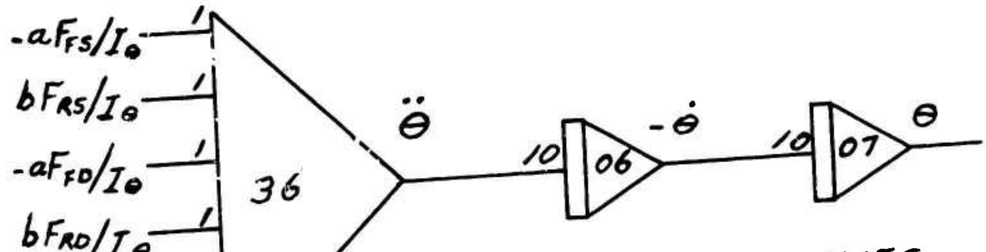


FIGURE 10. SPRUNG MASS PITCH ANALOG DIAGRAM

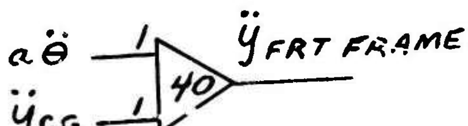


FIGURE 11. FRAME ACCELERATION ANALOG DIAGRAM

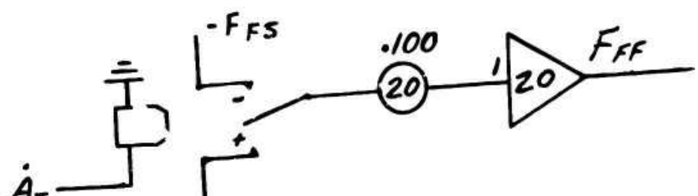


FIGURE 12. FRICTION FORCE ANALOG DIAGRAM

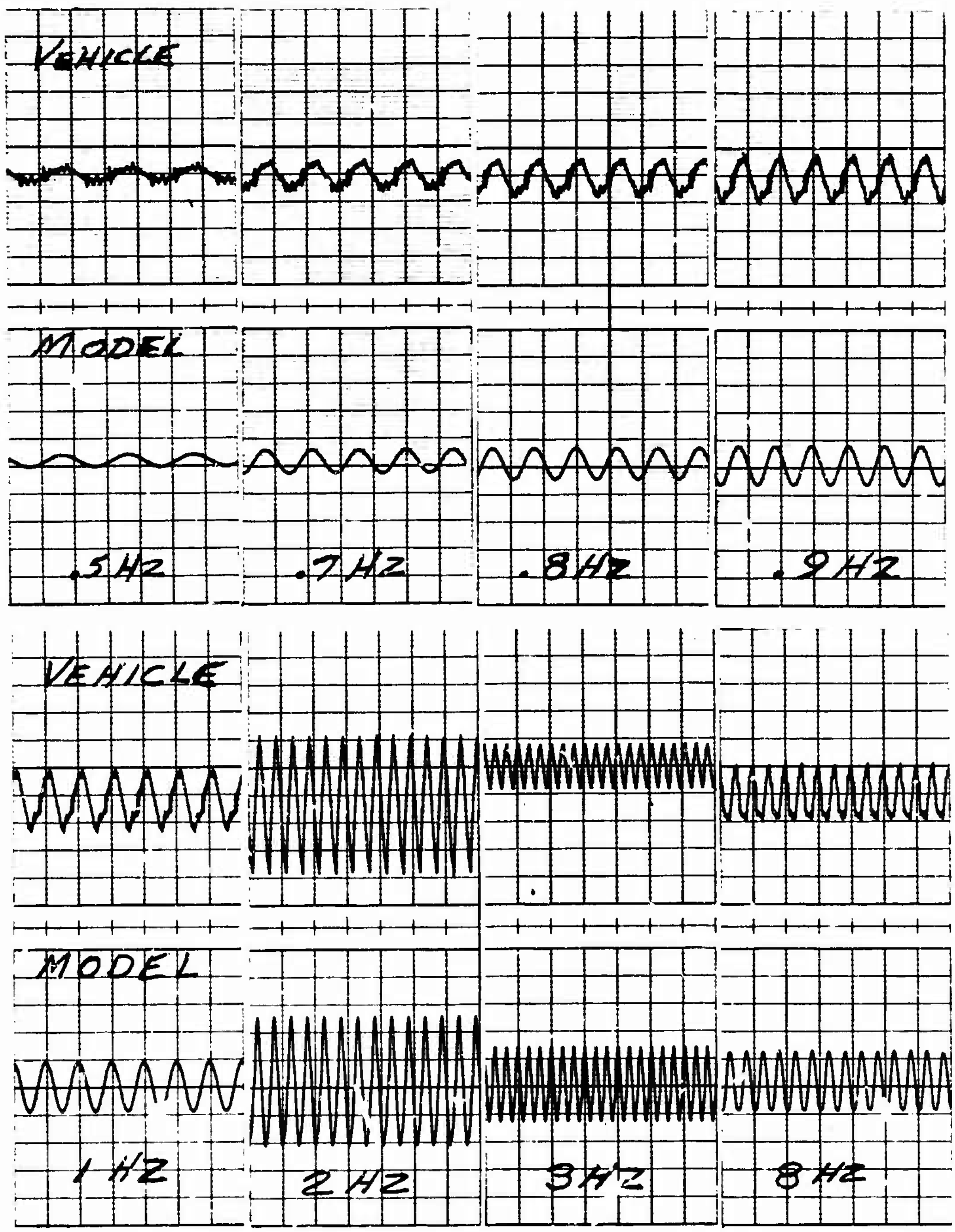


FIGURE 13. CENTER OF GRAVITY ACCELERATION – 0.1g/LINE

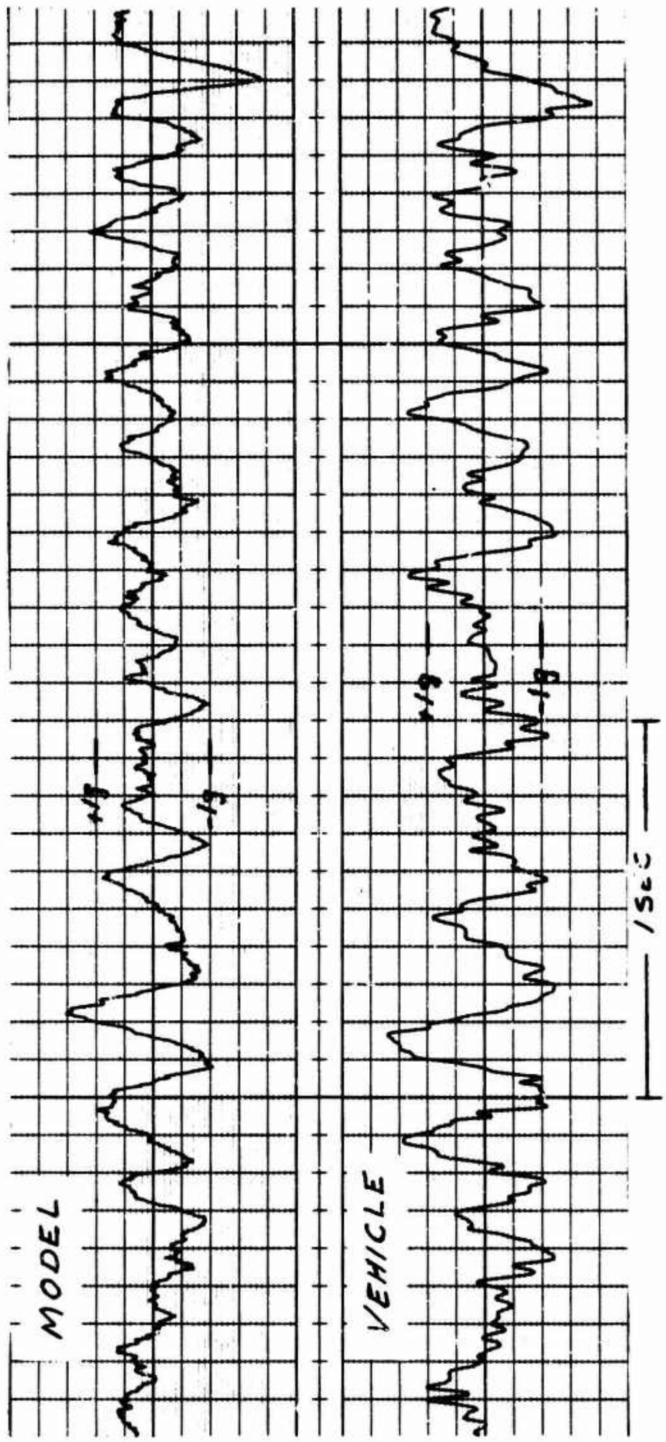


FIGURE 14. FRAME ACCELERATION RESPONSE

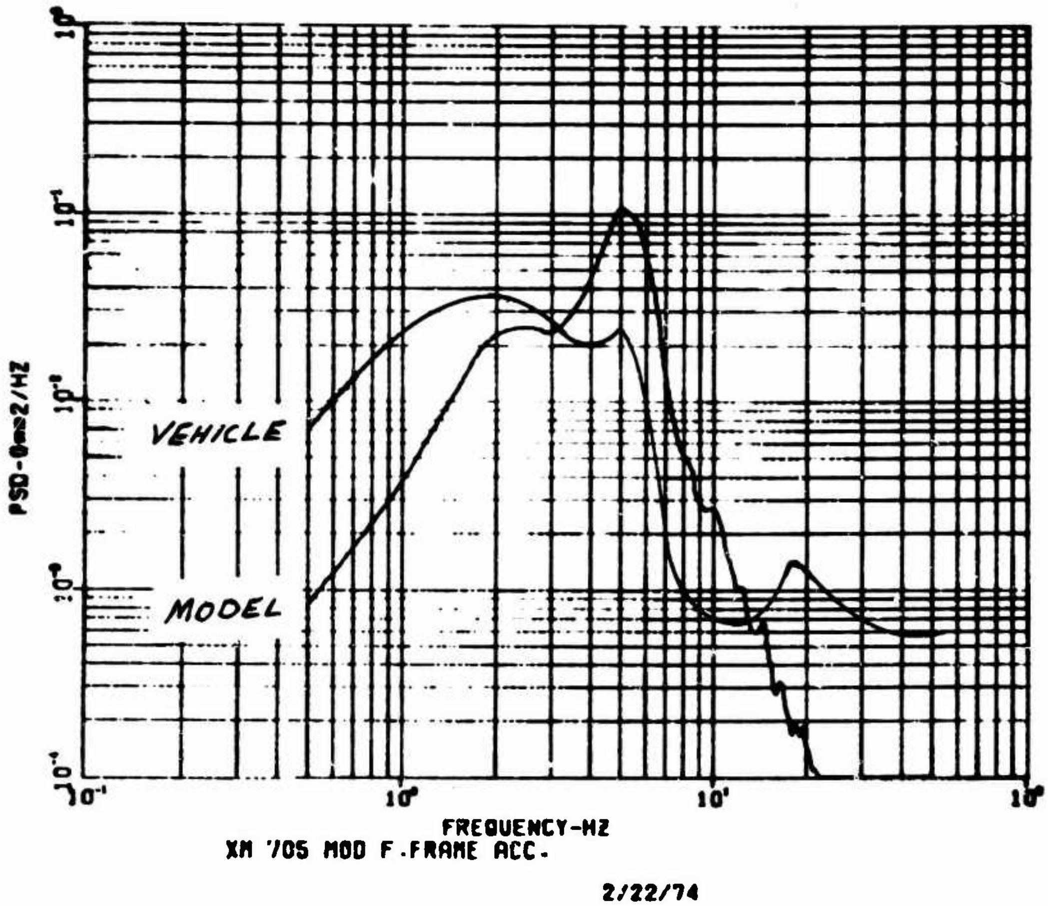


FIGURE 15. FRAME ACCELERATION PSD