ACTIVE SUSPENSION SYSTEM ENERGY AND POWER REQUIREMENTS FOR MILITARY APPLICATIONS

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Report prepared for AECV Brainstorming Meeting, Washington, DC, October 17, 1995

Later submitted to unknown conference

PR - 219

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I Introduction

Considerable effort has been devoted to developing active and semi-active suspension systems for off-road and on-road vehicles [1-15]. For land combat vehicles, the goal has been to significantly increase cross-country speed over rough terrain to improve battlefield survivability and combat effectiveness.

For this paper, suspension systems are categorized as passive, semi-active, or fully active. Passive systems are considered to consist entirely of springs and dampers connected between vehicle wheels (or track suspension components) and the vehicle body (sprung mass). Suspension components can be hydraulic, pneumatic, or solid mechanical. Displacements imparted to the system from the terrain result in compression/elongation of the springs, causing a varying force on the sprung mass. This imparts motion to the sprung mass. Although springs, of necessity, impart a non-constant force on the sprung mass in response to road input (which seems contradictory to ideally smooth rides), the varying force can be of smaller amplitude and lower frequency than the instantaneous ground variations. Damper components are intended to dissipate energy gained by the sprung mass as a result of the varied spring force. Analysis show that simple, non-controlled damper components, despite their averaged benefits over period of time, often result in the damper applying an instantaneous force on the sprung mass that adds to sprung mass motion. To help alleviate this problem, which is not completely avoidable with passive spring-damper suspension systems, shock absorbers typically have two different damping rates, depending on the direction of relative travel between sprung and unsprung masses.

Semi-active systems represent the first level of improvement, and are considered to be systems that have variable and controlled energy dissipation mechanisms, i.e., variable damping rates, in addition to passive springs. Damping components still connect the wheels (or track system) to the vehicle body. Most systems are hydraulic. One of the most successful control schemes, at least in simulation, envisions damping being applied based on the velocity of the sprung mass with respect to an absolute reference frame (dubbed "sky-hook" damping). Determining sprung mass velocity with respect to an absolute reference frame has often been difficult to fully realize in practice, resulting in limited performance benefits for rough terrain, off-road semi-active systems. Actuators required for a semi active suspension system are often valve systems to control fluid flow rates (to control vehicle damping rates) and, therefore, require little actuator power.

Fully active suspension systems contain active force generating components, that can add or subtract energy from the system. The force generating components connect the wheels (or track system) to the vehicle body. The net force applied between the sprung and unsprung mass is varied to achieve any desired combination of damping and applied forces based on relative or absolute motions of the sprung and unsprung masses. Generating forces of comparable magnitude to the springs in a passive system, requires powerful actuators. This often leads to the concept of using supplemental passive springs to support the vehicle static weight, thereby reducing the required actuator force producing capabilities (and size of the actuator). Recognizing that passive springs necessarily impart motion to a sprung mass in response to terrain variations, force generating actuators must be controlled to offset passive spring force fluctuations to result in reduced sprung mass motions. Without supplemental springs to support vehicle static weight, required actuators are usually considered to consume relatively large amounts of power. Actuator power requirements are usually considered to be a concern even for the spring assisted systems.

It is common in the literature (including previous articles by the authors of this paper) for researchers to evaluate anticipated performance gains between active and semi-active systems; compare power requirements to drive semi-active systems (typically only the power to operate a hydraulic valve) with power requirements to drive a fully active force generator; and draw conclusions, often concluding that semi-active systems represent the sensible compromise between performance, power needs, and complexity [2,8-12,15]. This paper examines the power situation more carefully, in the context of the power chain for the entire vehicle, and draws the conclusion that vehicle on-board power requirements for high speed cross country travel is insufficient *unless* very effective suspensions (of much better performance than may be possible with semi-active suspensions) are in use. Before power requirements can be analyzed, it is appropriate to present a realistic picture of performance benefits that are possible with fully active suspensions.

II. University of Texas Electromechanical Suspension System

Although it is possible to hypothesize benefits that are obtainable with fully active suspension systems, a more realistic assessment of the possibilities can be realized by examining an actual experimental system. While other systems also may have achieved high levels of success, this section will summarize the performance gains that have been obtained in simulation *and* in laboratory test results for a fully active single wheel ("quarter-car model") proof-of-principle demonstrator at the University of Texas Center for Electromechanics (UT-CEM). The UT-CEM demonstration was focused on an M1 Main Battle Tank (MBT) and was conducted at full scale. Conceptually, a road wheel station for an M1 tank is shown in figure 1, where the sprung mass is approximately 5 tons (i.e., the weight supported by one of the tank's 14 road wheels). The system developed and tested at UT-CEM replaced the M1 torsion bar with a electromagnetic torque motor, supplemented by an air spring. The function of the air spring is to support sprung mass static load, reducing the torque requirement of the torque motor. The torque motor is controlled in such a manner that net force exerted on the sprung mass (at the attachment points of the torque motor and the air spring) is approximately constant, regardless of road wheel motion. Feedback control on the torque motor corrects for the minor motions imparted to the sprung mass by the slightly non-constant force. Of course, there are delay times associated with this feedback control signal, which are considered when modeling the system.

The special purpose torque motor used in the actual system was designed and fabricated by UT-CEM for the project. Additionally, in the course of the project, UT-CEM developed a new approach to active suspension system control, referred to as a Near Constant Force Suspension (NCFS), that allows simplified control strategies and enables smooth rides over rough terrain without the requirement for advanced terrain knowledge, or "look-ahead." Although the system is still undergoing tests and control algorithm refinement, the project has proven very successful. Details for the project are provided in a separate paper [14].

Figure 2 shows the results of the NCFS system simulation on the most severe of three terrain profiles provided by the U.S. Army Tank and Automotive Command (TACOM). The terrain profile represents a 500 foot track with a 3.486" rms displacement. Vehicle speed for this calculation was 10 mph. The original terrain profile included some gradual hills, which have been removed to keep terrain displacements within the 20 inches of M1 suspension travel (the algorithms to accommodate large hills without the need for "look-ahead" have not yet been programmed into our suspension models or the demonstration system). From the plot, sprung mass motion appears to be mild enough to enable the crew to continue to use fire control systems, even on this severe terrain. For comparison, provided in figure 3, suspension performance on the same track was simulated with a single road wheel model that incorporated a passive torsion bar and rotary damper system. For the comparison plot, damper and torsion bar rates were selected to represent an M1 tank suspension (approximately critically damped) and vehicle speed was 10 mph. Note that sprung mass movement is quite large, essentially mimicking the terrain profile. TACOM experience indicates that crew endurance limits would limit sustained top speed to under 15 mph for this terrain track.

Figure 4 depicts a diagram of the test rig layout used to obtain experimental verification of the NCFS concept. Figures 5 and 6 show experimental results obtained with the test rig for road disturbance input profiles identical to the terrain depicted previously. The vehicle speeds for these results were 10 mph and 40 mph respectively. As mentioned, the system does not employ any "look-ahead" capabilities. Table 1 provides data related to the 40 mph test.

Table 1: Statistics for 40 mph Test Results

	Terrain Displacement (in)	Block Displacement (in)	Road wheel Acceleration (gees)	Block Acceleration (gees)	Ground Force (pounds)
Minimum	-11.37	-1.33	-10.47	-0.18	698
Maximum	8.57	1.96	9.27	0.30	13069
Mean	0.00	0.00	0.00	0.00	10415
RMS	3.37	0.51	1.56	0.05	10550

Note . Acceleration results involve numerical derivatives of displacement data, which tends to produce somewhat "noisy" results. Nevertheless, the relative magnitudes between road wheel and block accelerations should be approximately correct (i.e., both calculations involve numerical derivatives that probably produce comparable amounts of "noise").

Although several important conclusions can be drawn from the modeling and experimental results described in this section, the important conclusion for this paper is that fully active suspension systems can enable dramatic vehicle performance improvements: four-fold cross country speed increases with *simultaneous* six-fold sprung mass RMS acceleration reductions over very severe terrain were demonstrated on the UT-CEM test-rig.

III. Power Flow for Passive vs. Active Suspension Systems

Using the above results as an indication of the possibilities presented by active suspensions, it is now appropriate to address power needs associated with passive vs. active suspension systems. Figure 7 depicts power flow in a wheeled vehicle with passive suspension components. As expected, all power originates with vehicle prime power, depicted here as an internal combustion engine (ICE). Power flows through a variable speed transmission, fixed ratio gears (shown here, for a rear wheel drive vehicle as a differential), and to the drive wheels. At this point, power flows from the power train into several different branches, including tire losses, aerodynamic losses, forward motion of the total vehicle mass (including sprung and unsprung mass), and the "suspension branch." Since power flows to the suspension branch only when the vehicle is moving, it is apparent that the suspension branch receives power from the drive train, at the end of the power chain, as shown.

Within the suspension branch, power is used to accelerate the unsprung mass in such a manner as to follow the vertical travel required by the terrain. Additionally, power flows to the suspension components (shown here as passive springs and dampers) where energy is dissipated. Finally, power flows to the sprung mass and causes sprung mass motion in directions other than in the intended vehicle forward direction. This unwanted sprung mass motion is primarily in the vertical direction. The suspension components determine how much power flows to the sprung mass. The total energy dissipated in the suspension components is a function of how much energy is imparted to both sprung and unsprung masses.

Figure 8 depicts the power flow in a wheeled vehicle with fully active suspension components. Here a force actuator replaces the passive spring and damper system and is shown to draw power from the vehicle primary power supply (the ICE engine). Recognizing that the force actuator will likely be electric or hydraulic, a power converter has been included. For hydraulic actuators this converter would be a pump to convert shaft mechanical energy to hydraulic pressure; for electric actuators it would be a shaft driven electric generator. Additionally, to facilitate analysis, the force actuator is considered to be lossless and actuator losses have been separated into a hypothetical component, "actuator loss mechanism."

Comparison of figures 7 and 8 reveals several interesting features. First, although it is usually ignored, passive suspension systems draw power from the vehicle prime power. Furthermore, as shown in figure 5, power for the passive system flows through several components, each with their characteristic efficiencies, such as the vehicle transmission (efficiency approximately 85% for automatic transmissions) and differential (efficiency approximately 90%). Similarly, power for the active suspension actuator originates from vehicle prime power and flows through a power conversion process (efficiency approximately 90%) and an actuator loss mechanism (efficiency approximately 90%). Consequently, if passive and active suspension systems produce identical vehicle performance, both require comparable amounts of power from the ICE and both systems exhibit comparable losses in transporting energy through the various components in the power flow sequence. If passive and active systems operate in identical manners (i.e., the active system is programmed to act as a spring-damper system), the passive system dissipates energy in its damper and the active system dissipates the same amount of energy in its damping action. An electric active suspension, for example, could accomplish damping by dissipating energy in its electrical resistance. However, power demand on the ICE and energy consumption in the suspension would be equivalent.

IV. Power Comparisons for Passive Suspensions and Fully Active Suspensions

Figures 7 and 8 also indicate that the power to move sprung and unsprung masses originates with the vehicle ICE. Of course, the objective of the vehicle power system is to propel the vehicle (sprung and unsprung masses) in the forward direction. However, power is also required to move sprung and unsprung masses in directions perpendicular to forward motion. Most unwanted motion is in the vertical direction and this vertical motion is the object of the present discussion. This section presents simulation results for the UT-CEM EM suspension single wheel test rig with its active suspension and for the same test rig with simulated M1 tank suspension components (torsion bar spring and rotary damper, assumed to be critically damped), which will be referred to as a stiff suspension.

Referring to figures 2 and 3, it appears that much more power is required to produce sprung mass vertical motion in the M1 tank stiff system than for the active system. Considering the apparent small amounts of relative motion between the wheel and sprung mass for the stiff system (i.e., both the terrain and sprung mass in figure 3 appear to follow similar trajectories), the power requirement for the actual suspension components (i.e., for energy dissipation in the damper) will be negligible compared with the power required to produce the sprung and unsprung mass vertical motions.

In contrast, the fully active system appears to have negligible power associated with the sprung mass vertical motion. Consequently, its major power requirement is for producing unsprung mass vertical motion. In this case, however, power to produce mass vertical accelerations can flow through two branches: through the transmission and differential or through the power conversion and actuator loss mechanism. Both branches exhibit comparable efficiencies.

Now power requirements of the stiff and active suspensions can be compared. Both active and stiff systems exhibit identical motion of the unsprung mass (i.e., a wheel following terrain). Additionally, the power train leading to the unsprung mass motion in both cases is comparable: the stiff suspension (figure 7) shows that the power flows through two components with efficiencies of 85% and 90%, while the active suspension (figure 8) shows that power also flows through two components with similar efficiencies. Consequently, the power associated with motion of the unsprung mass (including the power chain leading to the unsprung mass motion) is nearly identical in both active and stiff systems. The major difference in power requirements for stiff and active systems is power associated with sprung mass motion.

Figures 9 and 10 depict power required to accelerate sprung masses for stiff and active suspension systems at two different speeds, 10 and 40 mph, over the same terrain depicted in figures 2, 3, 5, and 6. These results were obtained from simulations for the single wheel system shown in figure 1. As before, the stiff system employed the M1 torsion bar and rotary damper. For the active system, the M1 torsion bar and damper was replaced by the UT-CEM NCSF Electromechanical Active Suspension System. For these simulations, the stiff suspension was constrained to maintain contact with the terrain, even though ground pressure was allowed to be negative (which would normally imply that the vehicle became airborne). This constraint was not necessary for the active suspension since ground pressure never became negative. Note that the scales are identical on each plot, but that the right scale (for the active system) and the left scale for the passive stiff system are very different. Vertical sprung mass power was computed as:

$$\text{Power} = F\,v = m\,a\,v \tag{1}$$

where "F" is force, "v" is velocity, "m" is mass, and "a" is acceleration. It should be emphasized that these plots relate to a single wheel station, supporting 1/14th of an M1 tank weight, whereas the M1 has 14 road wheels.

As shown in figure 9, for a 10 mph vehicle speed, rms and peak power associated with both systems is small. However, there is a very significant difference for the 40 mph vehicle speed. At 40 mph over this terrain (figure 10), the stiff suspension requires 540 kW peak and 71 kW rms just to accelerate the *sprung* mass of a single wheel station through its motion. With the assumption that there is an averaging effect among the 14 road wheels, the rms power may be the most significant figure of merit. However, since the M1 has 14 road wheels and a 1500 hp engine, even the rms power associated with the vertical acceleration of just the sprung mass, 994 kW (1330 hp), requires so much power that the vehicle will not be able to maintain its intended speed. The active suspension system, however, only requires 5 kW (6.5 hp) of power from the ICE to produce sprung mass vertical acceleration.

Figure 11 depicts peak and rms power levels for the stiff and active suspensions at 10, 20, 30, and 40 mph vehicle speeds. From this plot, it is clear that the M1 does not have enough power to cross this terrain at high speeds *unless* the vehicle is equipped with an effective active suspension system.

V. Energy Usage

Previous sections considered power, but did not address energy consumption. As described in Section III and in figures 7 and 8, efficiencies within the power flow paths for active and passive suspensions are comparable. However, for high speeds over rough terrain, much higher levels of power are required for sprung mass motion for the stiff suspension. At the 40 mph level, for example, nearly 200 times more rms power is required for the stiff suspension. Multiplying the single wheel numbers by 14, the 994 kW of power required to provide sprung mass vertical acceleration for the stiff system, when reflected through a differential with an efficiency of 90% and an automatic transmission with an efficiency of 85%, results in a power draw on the ICE of 1299 kW to power *unwanted* sprung mass vertical motion. The losses within the power chain alone result in a power consumption of 305 kW. This would represent the total losses associated with this motion if energy that is put into the sprung mass vertical acceleration were totally recoverable, but it is not. On the other hand, rms power required to accelerate the sprung mass for active systems is 5 kW and when reflected back through power chain losses of 90% and 90% is 6 kW.

The above described energy consumption did not include other effects, such as the power chain losses associated with vertical motion of the unsprung mass. However, unsprung mass motion is the same in both the stiff and active system, so power chain losses will also be approximately the same. However, the two suspension systems do consume different amounts of energy in suspension damping. Simulations for stiff suspensions on the terrain at 40 mph show that damping consumes a total of 509 kW peak and 112 kW rms. These numbers include total suspension damping, and no attempt is made to apportion this between sprung and unsprung mass (whereas the power numbers presented in figures 9, 10, and 11 only consider sprung mass vertical motion). For active systems, "damping" is a more nebulous concept due to the actual control algorithm, but the power associated with velocity dependent (non-conservative) forces is 2.2 kW peak and 0.26 kW rms. Again, these numbers are for one wheel station and must be multiplied by 14.

VI. Conclusions

Several conclusion can be drawn from the material discussed in this paper.

1. Results presented in section II, based on actual experimental data, provide a realistic indication of the potential for improved cross-country mobility offered by effective active suspension systems. The results indicate that fully active suspension systems present an opportunity to dramatically increase cross country vehicle speed over rough terrain with simultaneous reductions in sprung mass vertical motions (over current systems at slow speeds).

2. Realistic high performance active suspensions systems actually reduce power requirements and energy consumption for high speed travel over rough terrain.

3. While it has always been obvious that improved suspensions are required from a ride quality viewpoint for increased speed over rough terrain, it now appears that power requirements also demand high performance suspension systems before high speed cross-country travel can be realized.

VII. Acknowledgments

The work reported here was supported by the U.S. Army Tank and Automotive Command, The Advanced Research Projects Agency and the State of Texas.

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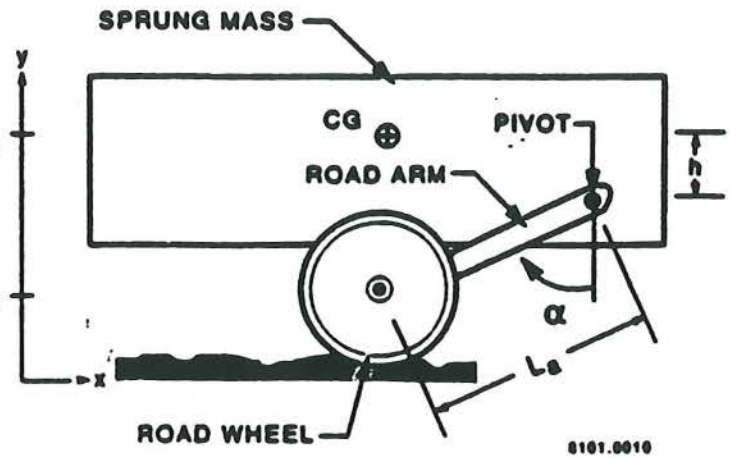

Figure 1. One of the fourteen M1 tank road wheel stations, showing the tank body, trailing arm (road arm), road wheel, and the road arm pivot.

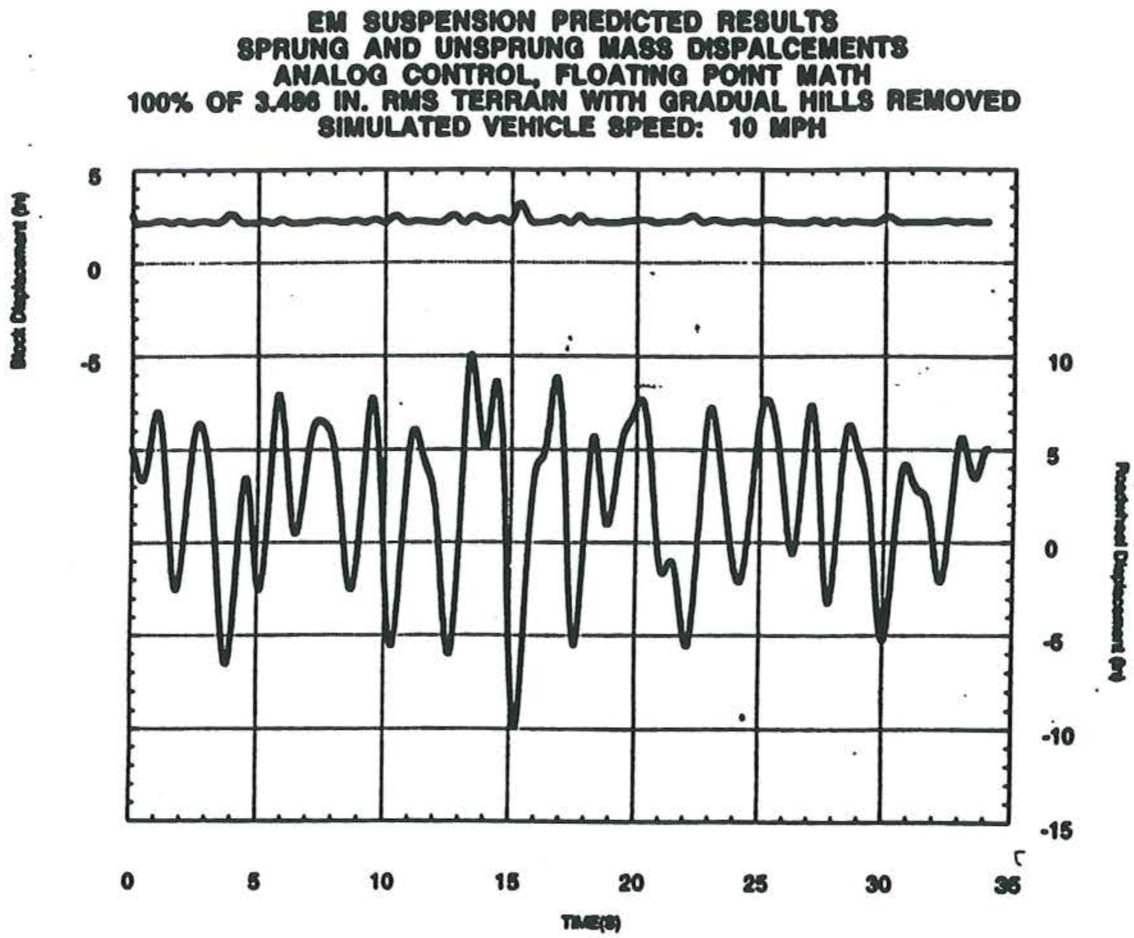

Figure 2. Plot shows simulation results, predicting displacement of the 5 ton sprung mass in a single wheel station model identical to figure 1 except that the torsion bar and damper has been replaced by the UT-CEM Near Constant Force Suspension (NCSF) system. The displacement of the road wheel (which follows the terrain) is also shown. For this plot, the simulated vehicle speed was 40 mph over the 3.4" rms terrain profile.

**EM SUSPENSION PREDICTED RESULTS FOR M1 TORSION BAR SUSPENSION
SPRUNG AND UNSPRUNG MASS DISPLACEMENTS
100% OF 3.486 IN. RMS TERRAIN WITH GRADUAL HILLS REMOVED
SIMULATED VEHICLE SPEED: 10 MPH**

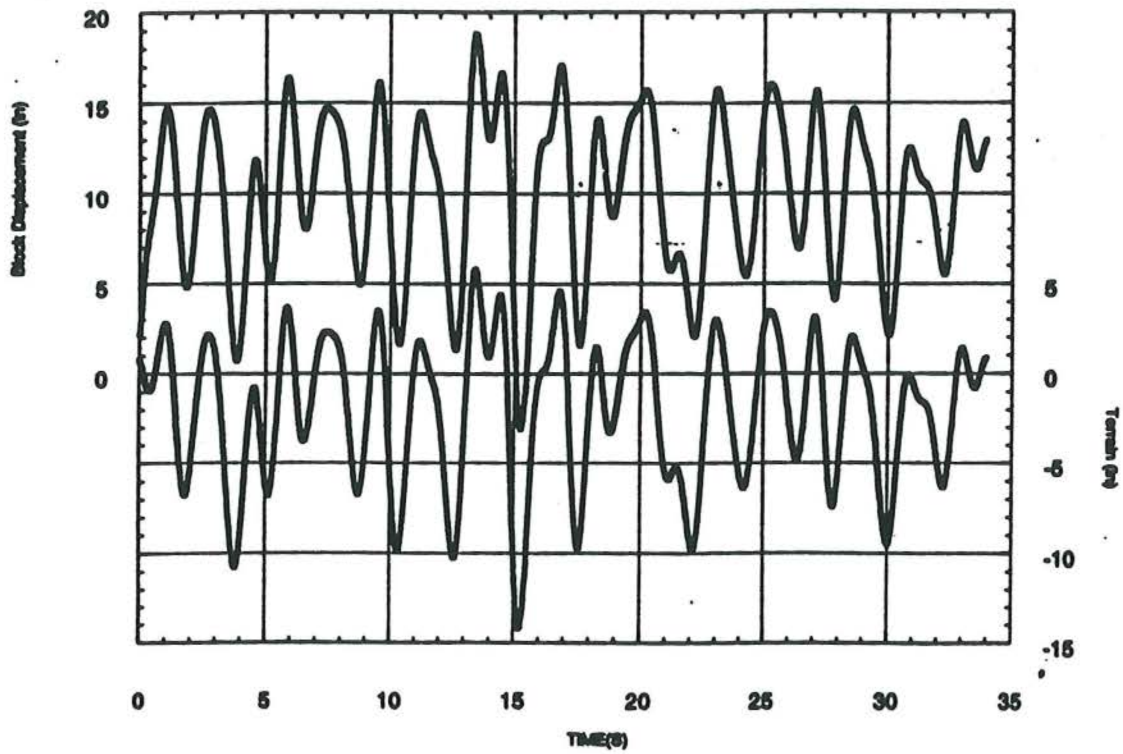

Figure 3. Plot shows simulation results of the single wheel test station depicted in figure 1. For this plot, the suspension components consisted of an M1 tank torsion bar and a rotary damper. Simulated vehicle speed is 10 mph. This plot is to provide a comparison of the M1 tank system with the results shown in figure 2.

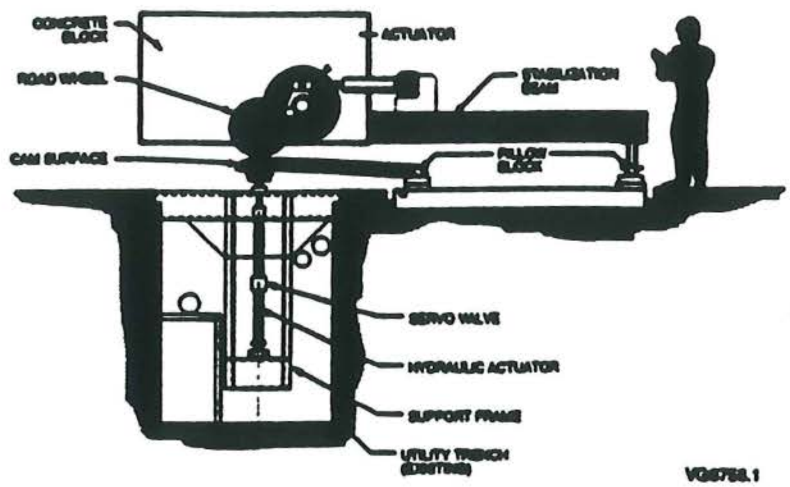

Figure 4. This diagram depicts the NCFS test apparatus general layout. The entire apparatus pivots about the pillow block at the far right of the diagram. Below ground level, a hydraulic actuator imparts vertical displacements to simulate road disturbances to an M1 roadwheel that rides on a cam (which is driven by the hydraulic actuator). The round actuator embedded in the 5 ton concrete block is a bi-directional electromechanical torque motor. The torque motor applies a torque through the roadarm/roadwheel assembly to support the block. The roadwheel is aligned with the center of mass of the block. Consequently, the block's mass is supported by the roadwheel and is free to pivot about the right most pillow block.

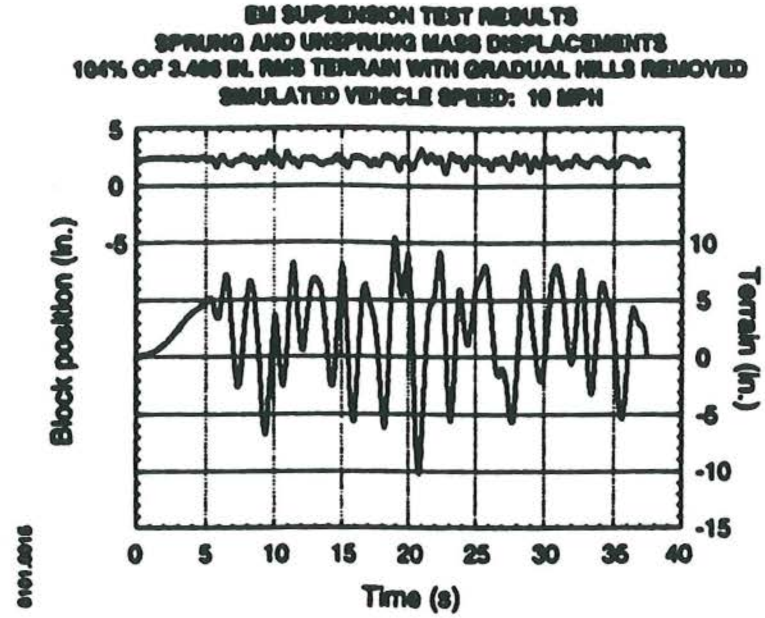

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Figure [redacted]. Plot shows displacement of a 5 ton concrete block, serving as the demonstrator sprung mass, and the displacement of the roadwheel as it is excited by a hydraulic actuator. Results agree very well with predictions from UT-CEM models. The terrain profile used here was provided by TACOM; the simulated vehicle speed for this plot is 10 mph, which is the approximate top speed that the M1 can traverse this terrain without exceeding sustained crew acceleration limits. For this test, the EM suspension system reduced RMS acceleration loading from .59 gees at the roadwheel to 0.04 gees at the sprung mass.

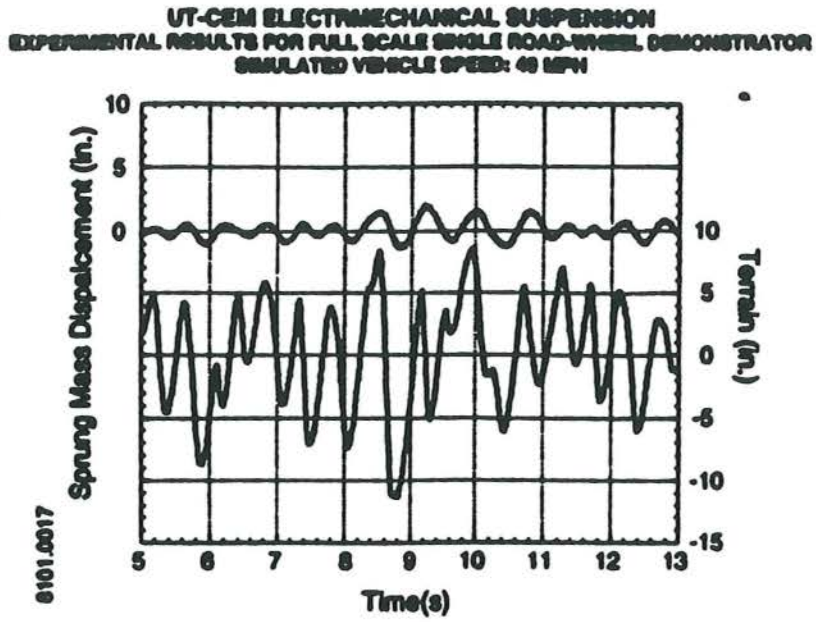

Figure 6. Plot shows displacement of a 5 ton demonstrator sprung mass, and the displacement of the roadwheel as it is excited by a hydraulic actuator at a simulated vehicle speed of 40 mph. For this test, the EM Suspension system reduced RMS acceleration loading from 1.6 gees at the roadwheel to 0.05 gees at the sprung mass.

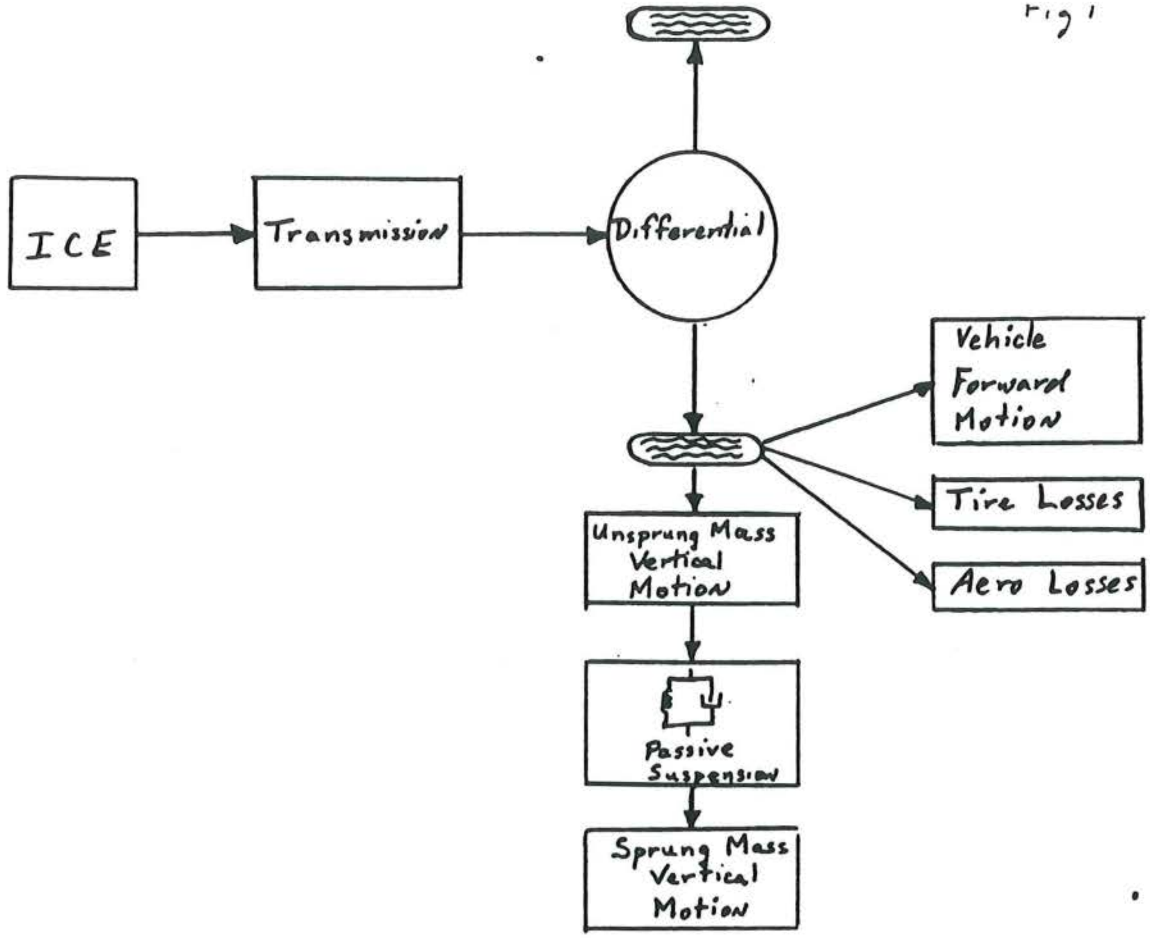

Figure 7. Diagram depicts the power flow in a conventional wheeled vehicle. Power is generated in the Internal Combustion Engine (ICE); flows through a variable speed transmission (typically, for military vehicles, this is an automatic transmission with efficiency of approximately 85%); flows through a fixed gear ratio component, shown here as a differential (typically with efficiencies of 90%); flows to the wheels and is used to provide vehicle acceleration, to overcome wheel rolling resistance, to overcome aerodynamic drag and to provide unsprung and sprung mass vertical acceleration.

Fig 8

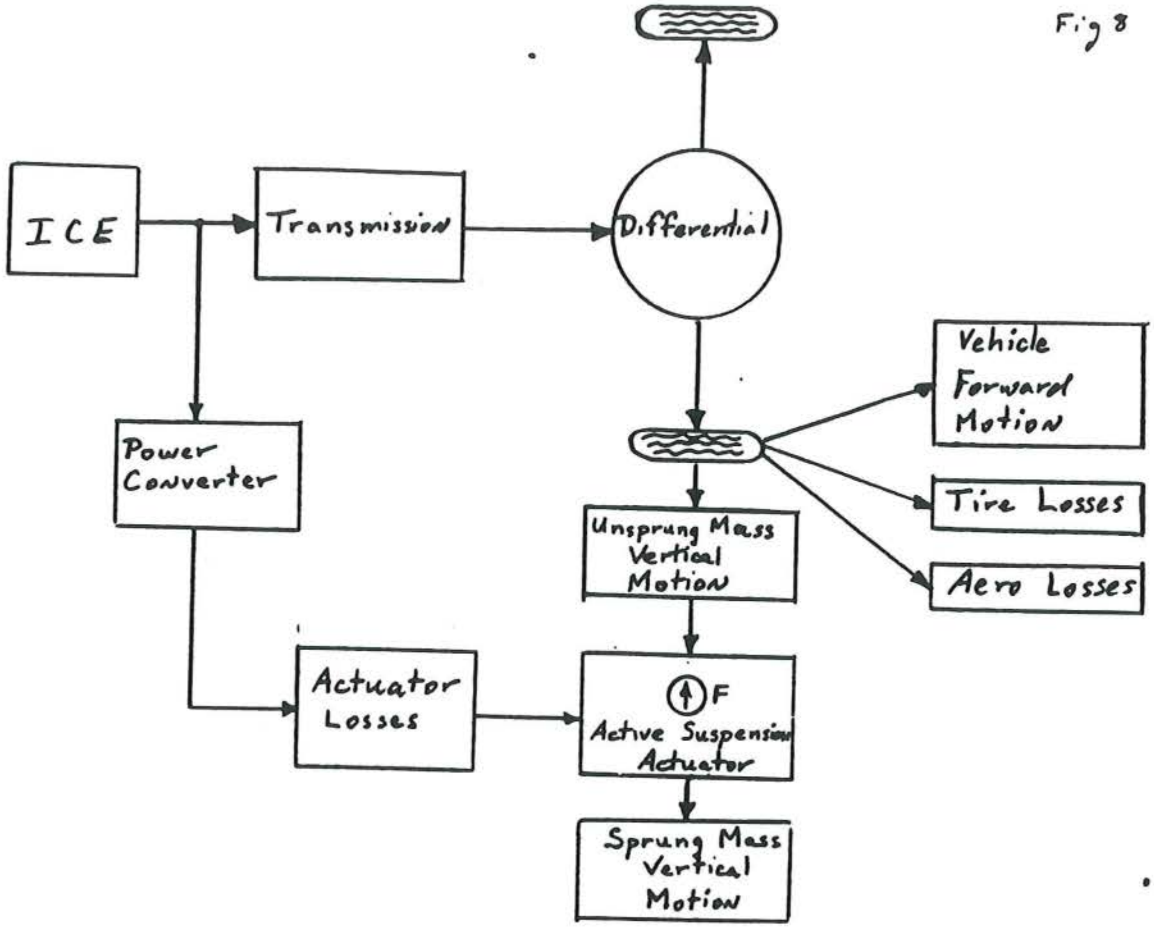

Figure 8. Diagram depicts the power flow in a wheeled vehicle that incorporates an active suspension. The diagram is identical to figure 7, with the passive suspension components replaced by an active force actuator. The actuator, as depicted here, is considered lossless, and the losses associated with the actuator have been separated into a separate box (this facilitates analysis). Additionally, a power flow path to the force actuator has been added. This path includes a power conversion device (e.g., a shaft driven generator if the actuator is electric) and the loss mechanism for the force generator. The conversion device and the actuator loss mechanism typically have efficiencies of approximately 90%.

Single Wheel Station Power at 10 mph
over TACOM 3.486" rms Terrain

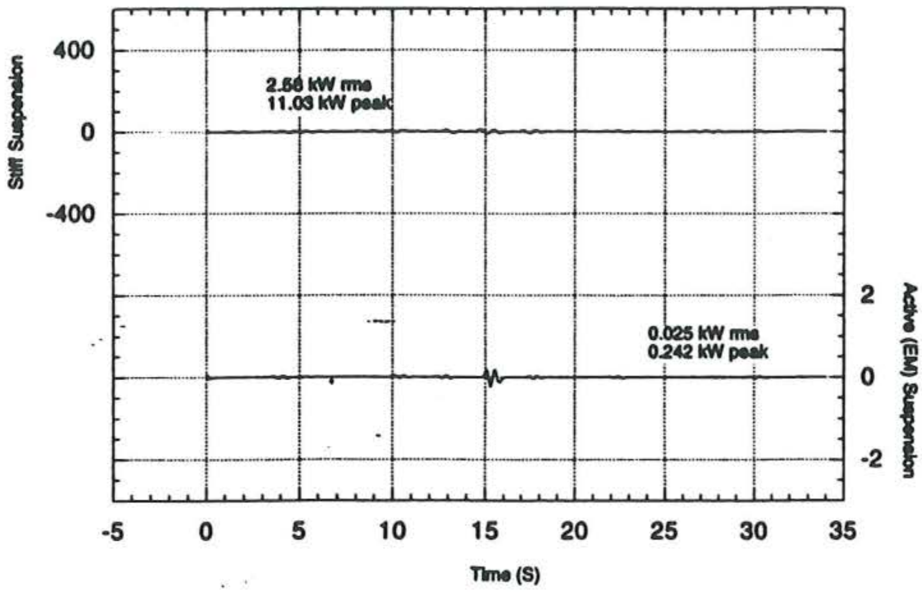

Figure 9. This plot depicts the power required to produce sprung mass accelerations for the single wheel station shown in figure 1. The results were obtained in simulation, with sprung mass power calculated as the product of force and velocity. The terrain profile was identical to that used in figures 2, 3, 5, and 6. The stiff suspension results apply to a system as shown in figure 1, with a suspension that consisted of an M1 torsion bar and rotary damper. For the Active (EM) Suspension results, the M1 torsion bar and damper were replaced with an NCSF Electromechanical Suspension. The scale of the plot was selected to be identical with Figure 8, for easy comparison. The simulated vehicle speed was 10 mph for these results.

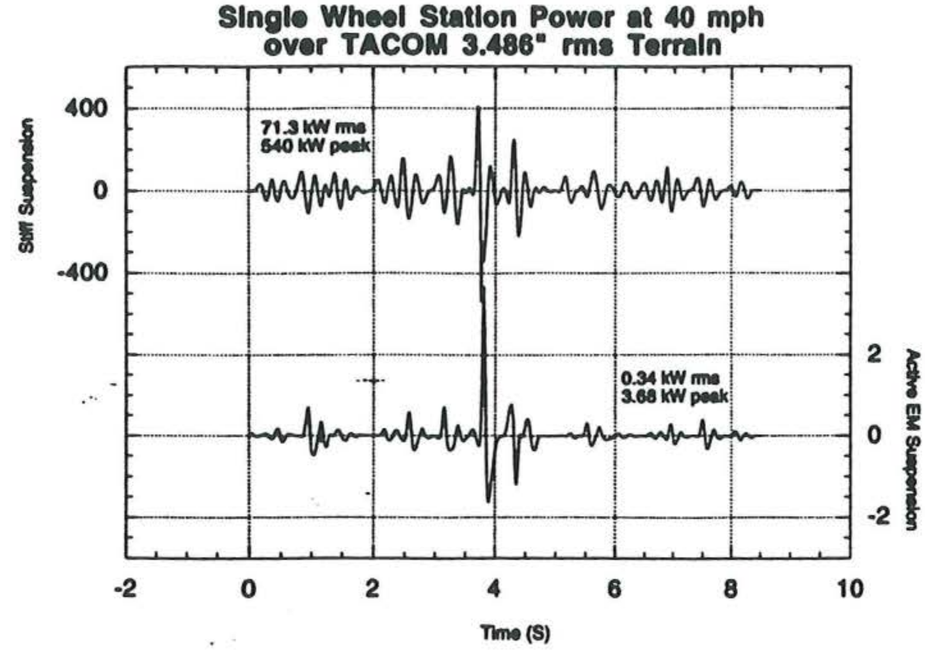

Figure 10. This plot depicts the power required to produce sprung mass accelerations for the same physical systems described in figure 8. However, the simulated vehicle speed for this calculation was 40 mph, over the same terrain.

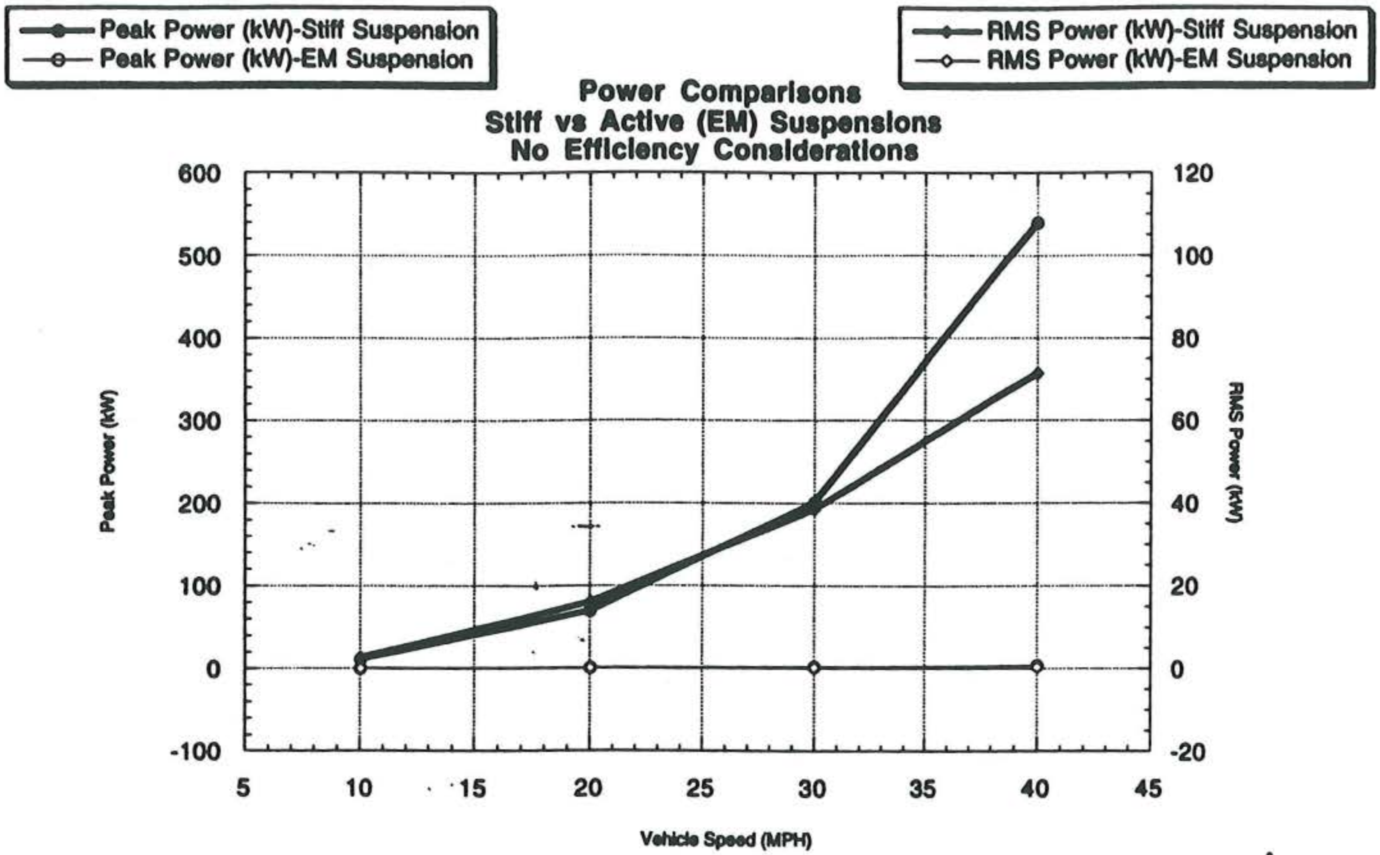

Figure 11. This plot tabulates results for peak and rms power associated with sprung mass vertical motion for vehicle speed so f 10, 20, 30, and 40 mph. The terrain is identical to that used previously.