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ABSTRACT

The University of Texas at Austin Center for Electromechanics (UT-CEM) has been developing advanced suspension technology for high-speed off-road applications since 1993. During the course of the program, advanced simulation techniques, verified by hardware demonstrations, were developed and refined. Based on this experience, UT-CEM conducted a detailed simulation-based comparison of passive, semi-active, and full-active suspension systems for an 18,000 kg (20 ton) 8 x 8 vehicle. Performance metrics are proposed to compare crew comfort, crew effectiveness, on-board equipment effectiveness, and power/energy consumption. This paper presents the methodology and rationale for metrics used in the study, simulation results, and data from this trade study. Results indicate significant advantages offered by well-designed active systems compared to both passive and semi-active, in all metrics.

INTRODUCTION

Strategic considerations frequently impose severe volume constraints on combat vehicle designers, usually to meet airlift requirements such as C-130 transportability. For 18,000 kg (20 ton) wheeled combat vehicles, an 8 x 8, trailing arm topology is frequently considered as being the most compact practical approach. Additionally, due to volume constraints, these vehicles typically do not employ anti-roll bars. Consequently, to reasonably limit the scope of the study, only an 8 x 8 trailing arm topology was modeled. Trailing arm suspension systems are compatible with differential torque steering ("skid steering") or hybrid steering approaches that use limited Ackerman steering in combination with differential torque steering.

Mobility goals for advanced, high-mobility hybrid electric fighting vehicles frequently include the ability to travel cross-country at speeds from 48 km/h (30 mph) to 64 km/h (40 mph) on 3.8 to 5 cm (1.5 to 2 in.) RMS terrain. This capability increases the tempo of battle, improves survivability by providing quick dash speeds, and exploits recent investments in C4I (command and control, communications, computers, and intelligence) that enable managing battles at high speeds. These mobility goals require advanced suspension technologies, such as semi-active or active.

This paper reports on a study to compare performance for passive, semi-active, and full-active suspension systems for 18,000 kg (20 ton) 8 x 8 High Mobility, Low Volume, Wheeled combat Vehicles (HMLWV). The vehicle used in the study employed a low volume trailing arm suspension system with wheel motors and large tires (~1 m diameter) for good off-road tractability. Additionally, ride-height adjustment was assumed for all suspension approaches to accommodate highly variable loads typical of combat vehicles. This paper does look at roll control during steering, but details of how to accomplish smooth steering and good vehicle control with differential torque steering or hybrid steering concepts is beyond the scope of this study.

Because all three systems (passive, semi-active, and active) had the same volume constraints and all had ride height control, the systems exhibited a relatively high degree of similarity, which simplified comparisons. The passive suspension system consists of an air-over-hydraulic strut (spring) in addition to a passive damping orifice (valve). The semi-active system contains a similar spring plus controlled servo valves, which allows control of the damping rate only. This type of suspension arrangement does not allow energy to be added to the system. The active system replaces the servo valves with an electromechanical actuator, which operates in all four control quadrants and does allow the addition or removal of energy from the system. This system also includes an air-over-hydraulic strut to support the sprung mass. Although active suspension systems can be designed without passive spring elements, supporting vehicle static mass with passive elements presents significant system mass and power consumption benefits [1, 2]. In all three systems, ride height adjustment is accomplished by adding fluid to the air-over-hydraulic passive spring. All three systems exhibited the same swept volume and identical tires, with the distance between the tires and vehicle hull of approximately 11.4 cm (4.5 in.).

Traditional designs for passive suspension systems require trade-offs to be made: stiff springs provide good roll and pitch control but have harsh rides; soft springs give comfortable rides but poor roll and pitch control [3]. Semi-active systems improve these parameters to a degree. These systems can only develop a force after relative motion between the sprung and unsprung masses has started. This means that semi-active systems can improve the response to vertical terrain disturbances and reduce the rate at which body roll occurs during a turn, but within limits because only damping forces can be applied. Total roll can only be limited with semi-active systems by changing the stiffness of the passive spring (sometimes to infinitely stiff by momentarily locking the actuator), which increases ride harshness during turns. Active suspension systems have the ability to provide both excellent ride and excellent handling. Heave (vertical motion) control can be highly divorced from roll/pitch control, allowing arbitrarily stiff roll control with negligible impact on ride harshness.

SYSTEM CONFIGURATION AND SPECIFICATIONS

SPRING RATE

A key to all three suspension systems is choosing the proper passive spring rate. This is especially true for vehicles with a trailing arm topology. With trailing arm topologies, tires experience camber change equal to vehicle body roll, which degrades the tire contact patch with the ground. Both passive and semi-active systems rely on passive springs to control total body roll (and camber change) during turns. The goal of spring selection for the passive and semi-active HMLWV was to choose springs as soft as possible (to provide a smooth ride) but sufficiently stiff to keep tire camber change with respect to ground small enough to provide adequate high speed on-road safety. The limit for acceptable camber change was set by a survey of camber change for commercially available vehicles, assuming that this ensured adequate high-speed safety for the HMLWV. Consequently, a survey of vehicle specifications and literature suggested a tire camber limit of 2.6° under high gravity (> 0.6 g's lateral acceleration) turns. For example, the camber change for a high mobility multi-wheel vehicle (HMMWV) is 2.6° at 0.63 g's lateral acceleration. Using a loaded weight (18,000 kg, or 20 tons) and center of gravity height for the assumed HMLWV vehicle yields a minimum acceptable spring rate that equates to approximately 1.45 Hz natural frequency. This natural frequency is comparable to other combat and off-road vehicles, and is approximately equal to the conventional passive HMMWV. However, simulation results revealed that the spring stiffness was not as significant as originally anticipated for the semi-active system performance.

Since active suspension systems have the ability to fully eliminate body roll, and therefore fully eliminate tire camber change, the spring rate becomes divorced from roll issues. Passive spring rate is chosen based on its ability to fully support the sprung mass (requires no energy consumption to support static weight) and is optimized to minimize the actuator size (balancing force authority required for all system control requirements). UT-CEM was also able to exploit previous experience in the design of active suspension systems for the HMMWV [4] and for a 13,600 kg (15 ton) Advanced Technology Transit Bus (ATTB). The result was that the same system natural frequency was chosen for the HMLWV as was used on the HMMWV and ATTB.

DAMPING RATE

The next parameter requiring specification is the damping rate. For passive systems, damping rates available for similar-sized combat vehicles were reviewed and vehicle manufacturers were consulted. To obtain desirable off-road performance for large vehicles in severe conditions, ideal damping is 80% to 120% of critical. 100% was selected based on the loaded weight of 18,000 kg (20 tons). The passive results were not overly sensitive to damping rates over this range of values.

Combat vehicle manufacturers with semi-active suspension experience were contacted to review valve performance specifications. To reduce latency issues with turning valves off and back on again, the simulations assumed a multi-valve configuration. It was assumed that individual valves could move from zero force to full damping force in 0.01 sec, which was the most optimistic information provided by the vehicle manufacturers. This number is 8 to 10 times faster than valves that could be found from vendors. Typically, bandwidth represents a significant performance limitation for semi-active systems. Since the semi-active bandwidth used in simulations for this study was probably better than actual hardware could provide, performance results for the semi-active system were probably better than could be obtained in actual hardware. These issues are still being investigated.

Actuator damping and friction losses can significantly degrade performance for active systems. For this study, actual active-actuator damping and friction parameters (obtained from experimental data) were used in the simulation. These losses include seal coulombic and viscous friction terms, force and response performance, and passive spring performance. These values are also compatible with verified performance on previous UT-CEM active system demonstrations (HMMWV and ATTB). Consequently, the active suspension results may be more realistic than the overly optimistic semi-active simulation results.

CONTROLS

The active suspension system uses control algorithms developed by UT-CEM. The control parameters have been proven in past suspension demonstrations and on a single-wheel HMWLV test rig constructed at UT-CEM. Performance of the HMLWV actuator under control of these algorithms has also been demonstrated on the test rig. [4, 5, 6, 7]

For the study, control algorithms for the semi-active system were based on UT-CEM active algorithms. However, the net actuator power from all control terms was truncated to zero (a semi-active system cannot add energy to the system; it can only damp/remove energy). The resulting semi-active control system encompasses algorithms available in literature for semi-active systems and adds additional proprietary control terms developed by UT-CEM (even though the capability of the semi-active actuator to accomplish the high-bandwidth needed for these terms is unproven). Consequently, the final semi-active results may be somewhat optimistic because the algorithm is not based on actual actuator limitations.

PERFORMANCE METRICS

To be useful to a wide audience, simple performance metrics are desirable. Performance metrics were chosen to represent:

- impact of vehicle sprung mass motion on:
    - driver and crew comfort
    - ability of crew to perform crew functions
    - ability of sensors to operate effectively
- impact of unsprung mass motion on vehicle control and safety
- energy consumption for suspension components

Some of these metrics have validated standards (such as absorbed power and ride-limiting speed); others are based on reasonable hypotheses; still others are useful only for relative comparisons [8,9].

DRIVER AND CREW COMFORT

Average absorbed power at driver location and rear crew location is a customary indicator of crew comfort. The accepted U.S. Army standard for ride-limiting speed is 6 W. Not all 6 W rides are the same, however. Acceleration spikes (such as impacting bump stops or sharp terrain features) can be very grueling but often do not result in major increases in average absorbed power. For this reason, absorbed power should be considered in conjunction with RMS acceleration. Based on active suspension testing on the HMMWV at UT-CEM on relatively small amplitude terrain (RMS 2.5 to 3.8 cm, or 1 to 1.5 in.) without sharp terrain features, 15 to 16 W can be endured for a short period of time. For large amplitude terrain, such as that used in this modeling, drivers will typically endure approximately 12 W for short durations (based on UT-CEM testing experience at the U.S. Army Yuma Proving Ground (YPG), 12 W is a good indication of cross-country dash speed).

CREW EFFECTIVENESS

Although it does not appear that accepted, simple standards exist, UT-CEM experience in general vehicle testing suggests that vertical RMS acceleration at crewmember location is a good indication of crew effectiveness. Results indicated that the rear crew member location experienced the highest vertical accelerations and, therefore, was used as the crew member of interest for this study. Vertical acceleration seems to be more indicative than absorbed power of the ability of the crew to read displays and operate electronics. Terrain that produces 6 W absorbed power will typically produce approximately 0.33 g's of vertical acceleration. Only limited crew actions can be accomplished during a 6 W ride (too rough). It seems probable that crew functions would be improved significantly by decreasing the RMS acceleration level by 50%, or from 0.33 to 0.17g. It also seems probable that increasing RMS acceleration by 50%, or from 0.33 g's to 0.5 g's, would represent major degradation in even the limited crew functions that can be accomplished during a 6 W ride.

EQUIPMENT EFFECTIVENESS

In addition to acceleration issues, equipment and sensor performance can be limited by velocity. UT-CEM is unaware of accepted standards for equipment effectiveness, but is still searching. For sensors with a narrow field of view (e.g. optics), keeping the image in the viewfinder can be difficult at higher sprung mass vertical velocities. The velocity can be a more significant issue than accelerations because high frequency sprung mass accelerations may not result in large amplitude motions and the image can remain in the viewfinder. High sprung mass vertical velocity also creates problems with “pixel smear” in digital optic systems. In this evaluation, RMS vertical velocity was used to allow relative comparisons between suspension systems, but now it is believed that standard deviation may be more meaningful.

VEHICLE CONTROL, SAFETY AND HANDLING

A major factor that limits vehicle controllability and safety is the percentage of time that a tire loses contact with the ground. In this evaluation, standard deviation of tire deflection about its static deflection was used to assess controllability and safety. A normal distribution for tire deflection was assumed and the area under the curve was used to indicate the probability that the tire deflection remained negative (to indicate tire-ground contact). Transient loss of ground contact is normal in harsh off-road conditions. While UT-CEM is unaware of accepted standards, it seems that losing tire-ground contact less than 10% of the time on individual tires will cause only minor degradations in vehicle control. From the analysis, front tires experienced the most significant loss of contact. This will adversely affect the controllability in Ackerman steered vehicles because the front tires play the biggest role in steering. Loss of contact will also degrade steerability of skid-steered vehicles, but to a smaller degree because all tires can contribute to turning.

ENERGY CONSUMPTION

Both passive and semi-active systems allow more sprung mass excitation than active systems. Furthermore, these systems then remove this sprung mass energy through wasteful damping. UT-CEM active systems are much better at preventing sprung mass excitation and, when necessary, remove energy from the vehicle body through regeneration. The regenerative nature of the active system allows smaller components to be chosen for the actuator. Having minimal energy consumption is desirable, and this performance metric is comparative between the suspension systems.

MODEL

The following is a summary of the simulation approach taken by UT-CEM. For most trade studies, a medium fidelity Simulink model is used. This model is relatively fast-running (30 to 120 min) and it has been shown to capture trends and relative magnitudes very well on the HMMWV and ATTB programs. A high fidelity coupled VisualNastran4D/Simulink model is used to benchmark the Simulink results. In this model, vehicle dynamics were modeled with VN4D (similar to the way DADS models vehicles) and the control system was modeled in Simulink. While detailed results are obtained with the VN4D/Simulink model, the cost is a slow-running model (6 to 15 hr per simulation run). The UT-CEM HMLWV test rig is used to provide a level of experimental validation for the coupled VN4D/Simulink model and is discussed later in this section.

To allow for acceptable run times for trade studies, the Simulink model represents one half of the HMLWV vehicle (Figure 1). This half-vehicle approach is common for suspension terrain interaction studies and was proven effective in previous UT-CEM active suspension demonstrations, where the model accurately predicted ride improvements that were later verified by testing.

The terrain input was based on a scaled version of a test track at Waterways Experiment Station. Since terrain displacement data is at 6 in. intervals, additional small amplitude, high frequency disturbances were added to represent rocks and surface roughness. The scaled terrain is approximately 300 m (1,000 feet) long, 25 cm (20 in.) peak to peak and 5 cm (2 in.) RMS and is shown in Figure 2, with the vertical scale in inches.

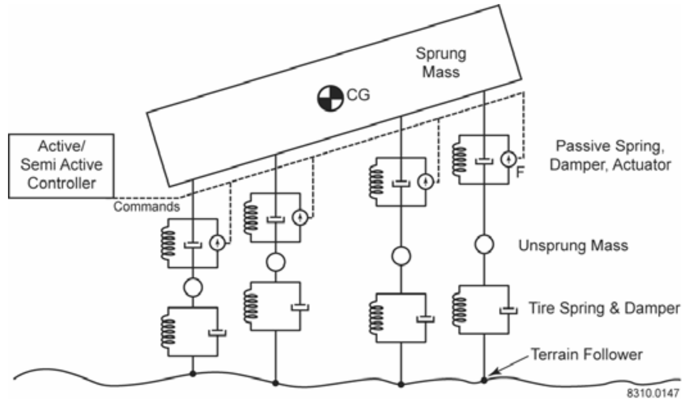


Figure 1. Diagram of half-car vehicle model

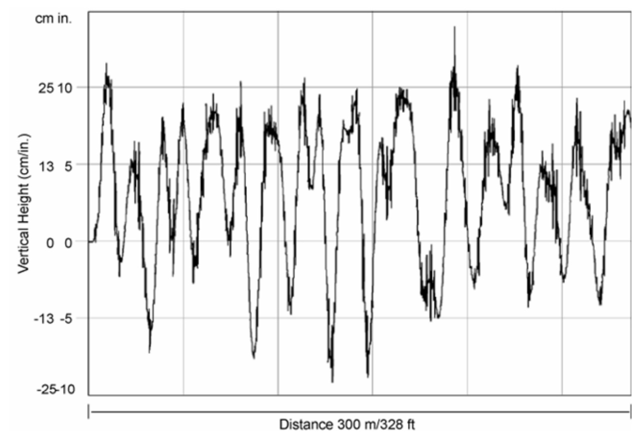


Figure 2. Terrain profile (vertical scale in inches; ~ 300m long)

Since the HMLWV is a hypothetical vehicle, models cannot be fully validated in the normal sense. However, the VN4D/Simulink modeling approach was verified by comparing data and model results for the UT-CEM HMLWV wheel station test rig, which was designed and built at full scale to approximately represent a single degree of suspension motion (vertical) and allows for capturing interaction of suspension and traction applied through the wheel motor (Figure 3). Figure 4 shows the VN4D representation of the test rig.

The VN4D Test Rig model included detail tire models, friction models for the active suspension actuator, and accurate representation of the test rig geometry. Typical results comparing model and experimental displacement data are shown in Figure 5. Note that most details of the wheel and sprung mass displacements show good agreement between the model and simulation, although the model does somewhat over estimate the displacement peaks. The scenario compared is at 48 kph (30 mph), representing a vigorous test and a significant challenge for the simulation

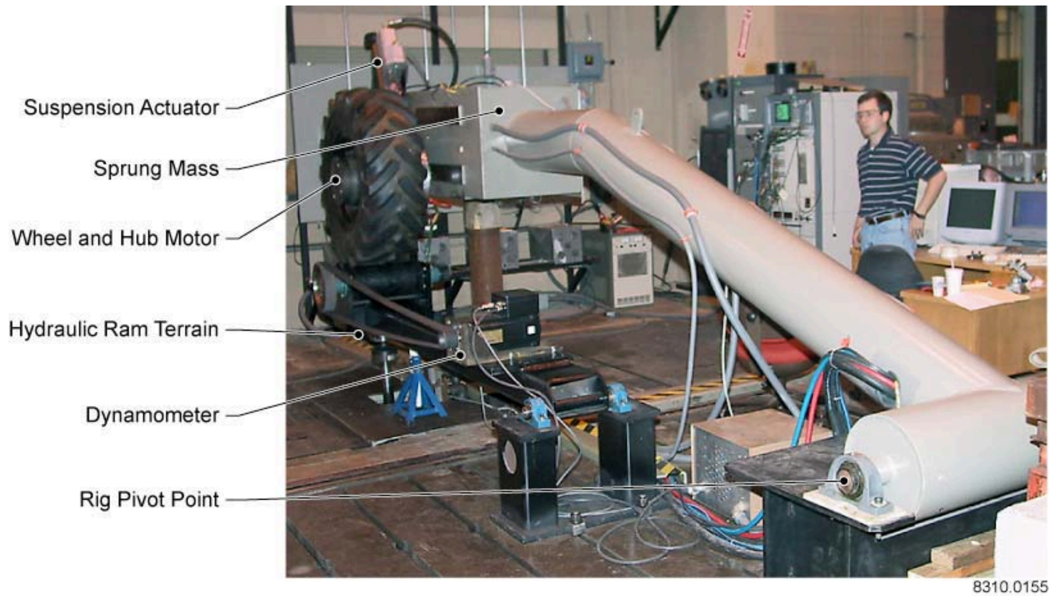


Figure 3. HMLWV test rig

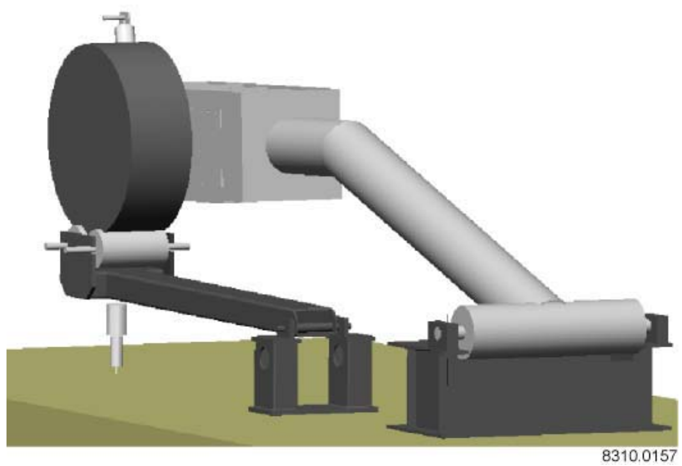
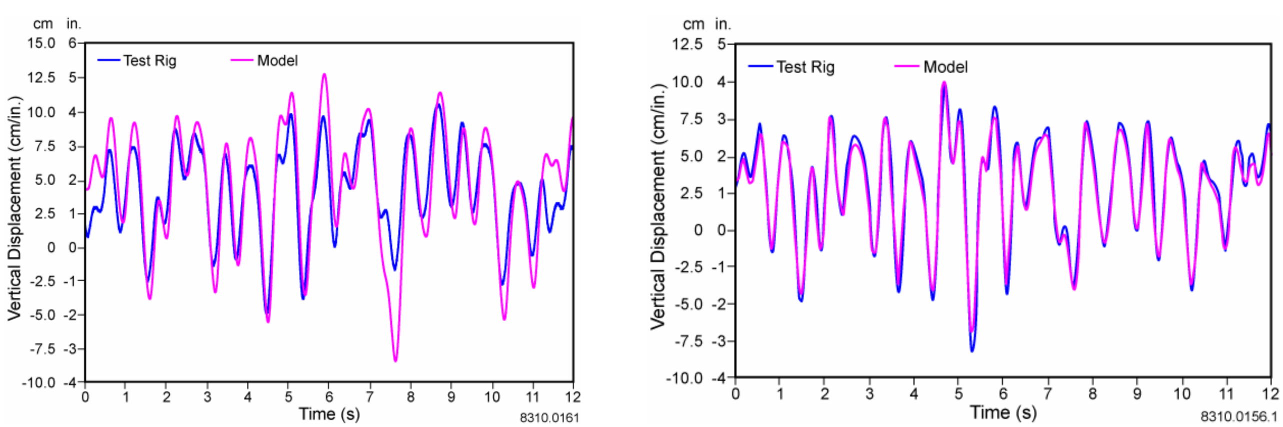


Figure 4. VisualNastran4D representation of HMLWV test rig



Body Comparison *Wheel Comparison*

Figure 5. Model vs. experiment on 48 kph (30 mph) test

The Simulink model is of moderate fidelity and the terrain is complex with rich frequency content. Consequently, coupling between vehicle dynamics, complex terrain frequencies, and realistic control algorithms often create trend lines that are not smooth (as in real vehicle test results). This is especially true of metrics dependent on derivatives, such as force, velocity and acceleration. Complex control algorithms (with conditional statements), friction stick-slip phenomena (e.g. in active system actuator), and varying degrees of viscous damping often combine to make the model stiff (and slow and temperamental).

RESULTS

Figure 6 is a plot of absorbed power versus vehicle speed for all three suspension systems. Ride (6 W) and dash (12 W) limits are also shown for reference purposes. The passive system has a ride limit of 29 km/h (18 mph) and a dash limit of 37 km/h (23 mph). The semi-active system shows improvement to 40 and 43 km/h (25 and 27 mph) for ride and dash limits respectively. At 64 km/h (40 mph), the active system approaches the ride limit but is still below 6 W. The trends in Figure 6 are comparable to similar trends seen in actual test data taken at Yuma Proving Grounds for absorbed power on the UT-CEM active suspension HMMWV (Figure 7).

The crew effectiveness plot is shown in Figure 8. For both the passive and semi-active systems, it is probable that crew use of monitors and many electronics will degrade significantly, compared to the active system. The active system shows roughly a 50% improvement over the passive and semi-active systems up to 40 km/h (25 mph). At higher speeds, the active system has much greater improvements. At some of the lower speeds for the active suspension vehicle, speed couples with the vehicle dynamics and limits improvements to approximately 30%.

The sensor vertical RMS velocity is plotted versus speed in Figure 9. Since there are no known acceptable limits for this parameter, the passive system serves as the baseline for the sensor effectiveness metric. The semi-active system has an improvement of approximately 30% over the passive system, while the active system shows roughly a 60% to 80% improvement.

Figure 10 is a plot showing the probability of maintaining tire contact. The 90% probability is highlighted to represent the hypothesized acceptable limit. The passive and semi-active systems are acceptable up to 36.8 km/h (23 mph) but are likely to degrade significantly and unsatisfactorily thereafter. The active system is acceptable up to 48 km/h (30 mph) with occasional minor degradation beyond.

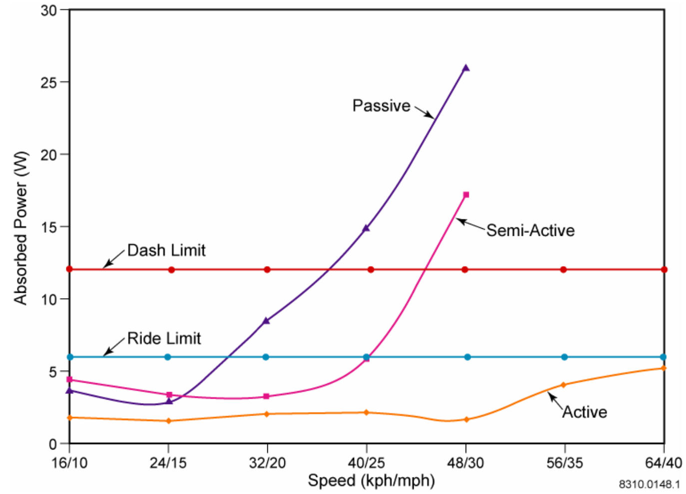


Figure 6. Crew maximum absorbed power

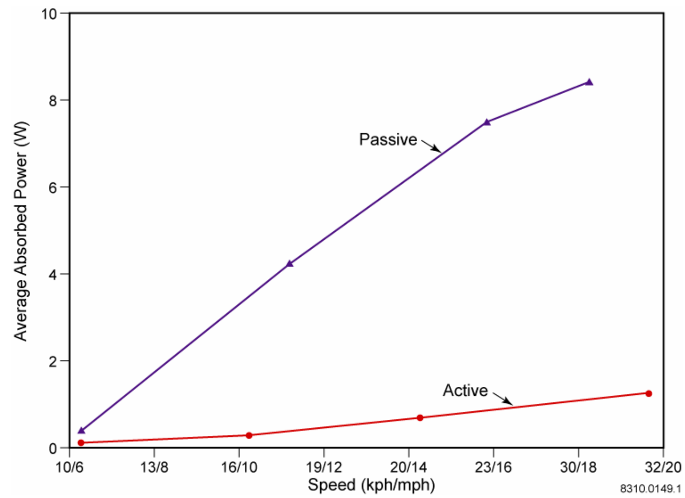


Figure 7. UT-CEM active HMMWV measured absorbed power

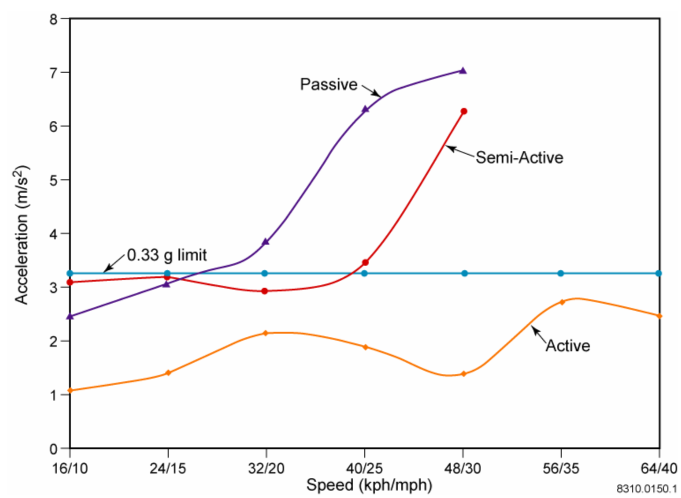


Figure 8. Crew effectiveness

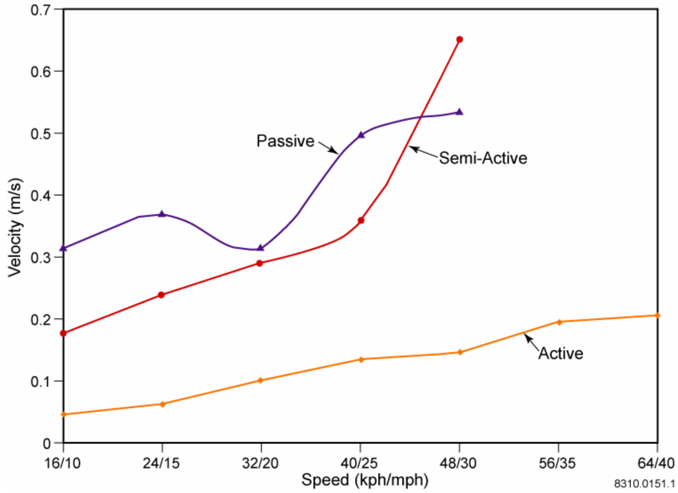


Figure 9. Plot of sensor effectiveness

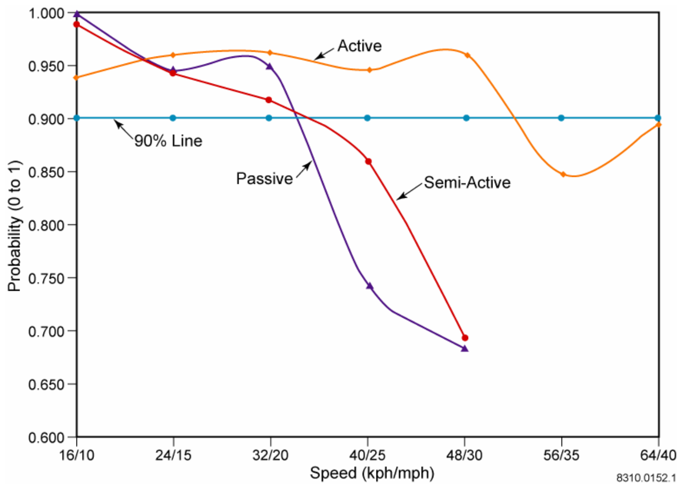


Figure 10. Tire-ground contact probability

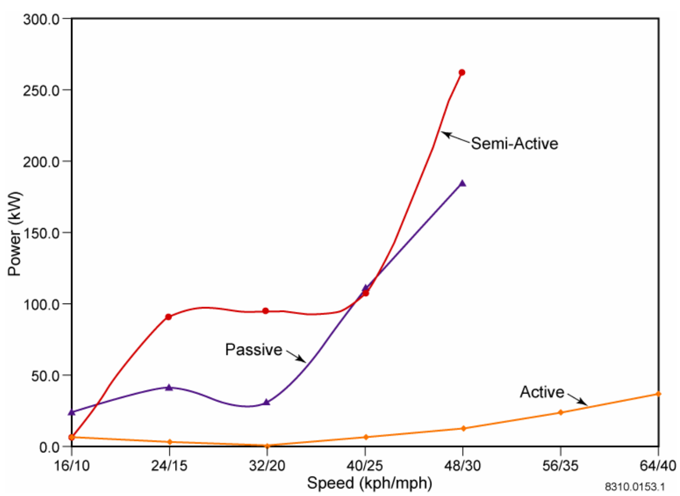


Figure 11. Average power consumption by suspension components

OBSERVATIONS AND CONCLUSIONS

When compared with the absorbed power and RMS acceleration plots at 16 and 51 km/h (10 and 32 mph) for the active system, it appears there is room for a trade-off between tire contact and ride quality/crew effectiveness. This can be accomplished by tuning the "wheel hop" algorithm, developed at UT-CEM but not part of the controls used in this trade study.

The power versus speed plot for power consumption by suspension components is shown as Figure 11. The passive system has moderate power consumption up to the ride limiting speed. The power consumption increases significantly between the ride limiting and dash speeds. The semi-active system is comparable to the passive at low speeds, but requires even more power as the vehicle approaches the dash speed. The ability of the active system to regenerate rather than burn energy during damping makes active energy consumption much lower than either the passive or semi-active at all speeds.

For the HMLWV vehicle, based on this set of simulations, the relative performance between the passive and active suspension systems agrees with HMMWV test results obtained by the U.S. Army at Yuma Proving Ground. Additional investigation into performance metrics and accepted standards is needed (and is ongoing).

The large 50 cm (20 in.) wheel travel was a key factor for all three suspension systems in the ability of the vehicle to traverse this terrain at these speeds. All of the simulation results presented here were for straight-ahead motion. Body roll during turning consumes suspension travel for both passive and semi-active systems, but not for active suspensions. Consequently, differences between active, semi-active and passive systems may be more pronounced when turning maneuvers are superimposed on rough terrain.

The stiffer spring used by the semi-active system (compared to the active system) was well offset by UT-CEM effective damping algorithms and was not as significant as originally anticipated. The semi-active system suffered from its inability to apply positive power at the crucial times when needed most during high speed rough terrain. Additional work is needed to accurately estimate responsiveness of semi-active controlled valves. The assumed high responsiveness (imposed zero delay times and rate limits that seldom were invoked) of these valves was crucial to the performance of the semi-active system. Recently obtained information on average valve actuation delays implies semi-active results are overly optimistic and need to be re-run. In addition, the semi-active control algorithms used in this study are not degraded by actuator issues and may produce optimistic performance results. Transition from control of the electromechanical actuator to the hydraulic variable damping valves could well result in performance degradation of the semi-active system relative to these simulations.

It is known that application of traction torque at the end of a trailing arm couples with suspension motion. This effect can be cancelled with active suspension but only reduced with passive or semi-active systems (cancellation of this traction torque requires positive power - a motor actuator force). Consequently, further degradation in passive and semi-active performance may occur in actual vehicle operation involving vehicle acceleration and deceleration.

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REFERENCES:

1. F. B. Hoogterp, J. H. Beno, D. A. Weeks, "An Energy Efficient Electromagnetic Active Suspension System," SAE Publication No. 970385, Society of Automotive Engineers, February 1997.
2. J.H. Beno, D.A. Weeks, D.A. Bresie, A.M. Guenin, J.S. Wisecup, and W. Bylsma, “Experimental Comparison of Losses for Conventional Passive and Energy Efficient Active Suspension Systems,” SAE Publication No. 02AC-114, Society of Automotive Engineers, March 2002.
3. M. Appleyard and P.E. Wellstead, “Active Supsensions: Some Background,” *IEEE Proc, Control Theory Appl.,* Vol 142, No. 2. March 1995.
4. D.A. Weeks, J.H. Beno, A. M. Guenin, and D. Bresie, "Electromechanical Active Suspension Demonstration for Off-Road Vehicles at the University of Texas Center for Electromechanics," SAE Publication No. 2000-01-0102 and SP-1519, March 2000.
5. K.T. Schuetze, J.H. Beno, W.F. Weldon, S.V. Sreenivasan, "A Comparison of Controller Designs for an Active, Electromagnetic, Off-road Vehicle Suspension System Traveling at High Speed," SAE Publication No. 980924, Society of Automotive Engineers, February 1998.
6. D. A. Weeks, J. H. Beno, D. A. Bresie, S. K. Ingram, A. M. Guenin, "Control System for Quarter Car Heavy Tracked Vehicle Active Electromagnetic Suspension," SAE Publication No. 971086, Society of Automotive Engineers, February 1997.
7. K.T. Schuetze, J.H. Beno, W.F. Weldon, S.V. Sreenivasan, "A Comparison of Controller Designs for an Active, Electromagnetic, Off-road Vehicle Suspension System Traveling at High Speed," SAE Publication No. 980924, Society of Automotive Engineers, February 1998.
8. L.F. Stineleather, G.O. Hall, and A.O. Radxe, "A study of vehicle vibration spectra as related to seating dynamics," SAE Publication No. 720001, Society of Automotive Engineers, 1972.
9. R.A. Lee and F. Pradko, "Analytical analysis of human vibration," SAE Publication No. 680091, Society of Automotive Engineers, 1968.