SUSPENSION TRADE STUDIES FOR HYBRID ELECTRIC COMBAT VEHICLES

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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. Recently, 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 were developed 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.

UT-CEM active suspension technology has recently been licensed to Northrop Grumman Canada Corp for the military field of use.

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, this typically results in an 8 x 8, trailing arm topology. Additionally, due to volume constraints, these vehicles typically do not employ anti-roll bars.

Mobility goals for advanced, high-mobility hybrid electric fighting vehicles frequently include the ability to travel cross-country at speeds exceeding 48 km/h (30 mph) and frequently up 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.

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. In all three systems, ride height adjustment is accomplished by adding fluid to the air-over-hydraulic passive spring.

Traditional designs for *passive* suspension systems require a trade-off to be made: a stiff spring provides good roll and pitch control but has a harsh ride; a soft spring gives a comfortable ride but poor roll and pitch control. *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 a *semi-active* system can improve the response to vertical terrain disturbances and reduce the rate at which body roll occurs during a turn¹, but within limits because only the damping force can be applied. *Active* suspension systems have the ability to provide both excellent ride and excellent handling. Heave (vertical motion) control can be fully divorced from roll/pitch control, allowing arbitrarily stiff roll control without impacting 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. Although *semi-active* systems can reduce roll rate, both *passive* and *semi-active* systems rely on passive springs to control total body roll (and camber

¹ 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.

change) during turns. The goal of spring selection for the HMLWV was to choose a spring 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 ($> 0.6g$ lateral acceleration) turns. For example, the camber change for a high mobility multi-wheel vehicle (HMMWV) is 2.6° at $0.63g$ lateral acceleration. Using the 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 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 the *active* system has the ability to fully eliminate body roll, and therefore fully eliminate tire camber change, the spring rate becomes divorced from roll issues. The 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 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, damping is typically 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 s, 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 HMLWV test rig constructed at UT-CEM. Performance of the HMLWV actuator under control of these algorithms has also been demonstrated on the test rig.

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.

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 vehicle testing at YPG suggests that vertical RMS acceleration at rear crew location is a good indication of crew effectiveness. 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 0.33g 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 0.33g to 0.17g. It also seems probable that increasing RMS acceleration by 50%, or from 0.33g to 0.5 g, 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 a mean and 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. 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. The validation is ongoing, but excellent correlation has been obtained so far.

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.

Figure 1. Diagram of half-car vehicle model

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 25 cm (20 in.) peak to peak and 5 cm (2 in.) RMS.

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 2 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 2 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 3).

Figure 2. Crew maximum absorbed power

Figure 3. UT-CEM active HMMWV measured absorbed power

The crew effectiveness plot is shown in Figure 4. 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%.

Figure 4. Crew effectiveness

The sensor vertical RMS velocity is plotted versus speed in Figure 5. 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 5. Plot of sensor effectiveness

Figure 6 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. 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.

Figure 6. Tire-ground contact probability

The power versus speed plot for power consumption by suspension components is shown as Figure 7. 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.

Figure 7. Average power consumption by suspension components

OBSERVATIONS AND CONCLUSIONS

For the HMLWV vehicle, based on this first phase 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.

The trends identified in this study may also apply to tracked vehicles. Similar to the HMLWV 8 x 8 vehicle, tracks involve multiple trailing-arm wheel stations, significantly more than the four wheels associated with small vehicles. The increase in number of wheel stations from an 8 x 8 vehicle to the 10 or 12 wheel stations that would be associated with tracks may not be significant because the increase in number of wheel stations from a small four-wheeled vehicle to the 18,000 kg (20 ton) 8 x 8 proved to be insignificant.

The HMLWV large-diameter soft tires and tracks both tend to smooth out small amplitude bumps, but tracks may do more smoothing than wheels. The fact that track road wheels are small and hard may partially offset the additional smoothing tendencies of tracks. Other track issues (such as track tension) need to be studied to determine if *passive* tensioning is adequate or if *active* tensioning is beneficial. Optimal damping of track vibrations may prove to be a significant benefit of *active* suspension systems for high-performance tracked vehicles.

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