Laboratory Testing of an Active Electromagnetic Near Constant Force Suspension (NCFS)

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ABSTRACT

In March of 1995, the University of Texas at Austin Center for Electromechanics (UT-CEM) began work on developing active suspension control algorithms for four-wheeled, off-road, rough terrain, vehicles. To serve as a test platform to validate simulations, a four corner test-rig, representing a military HMMWV at one third scale, was designed and fabricated. Multiwheel control algorithms were developed, based on single wheel concepts previously described in SAE publications. The four-wheel test-rig performance compared well with single wheel test-rig performance, showing that the active suspension concepts developed by UT-CEM, which do not require advanced terrain knowledge (i.e., no "look-ahead"), are compatible with full vehicle control.

INTRODUCTION

UT-CEM is developing an Electromechanical Active Suspension System (EMASS), with a goal of increasing cross-country mobility for military vehicles. The necessity for advanced suspension systems for combat vehicles is highlighted by the fact that a modern main battle tank has a top road speed of approximately 20.1 m/s (45 mph), but is limited to approximately 11.2 m/s (25 mph) or less when traveling cross-country. Considering the status of current passive system performance and the limitations of existing materials, further increases in cross-country mobility will require the replacement of existing passive suspension approaches with active or semi-active concepts [1]. The EMASS uses electromechanical actuators to allow rapid response active control. The system utilizes a control approach that is based on maintaining a nearly constant vertical force on vehicle sprung mass, despite terrain fluctuations, to minimize sprung mass accelerations. The near constant force suspension (NCFS) is described in detail in a previous SAE publication [1]. This approach has proven simple, effective, and reliable in the laboratory and does not require prior terrain information (i.e. no "look ahead"). Initially, a full scale, laboratory demonstrator of a single M1 main battle tank wheel station was chosen to demonstrate the ability of an EMASS to smooth harsh terrain details and demonstrate EM actuator performance. The analysis, design, and initial testing of the single wheel system are detailed in companion SAE articles [1][2]. This paper focuses on efforts to expand the single wheel algorithms developed previously, to allow full control of a four-wheel, off-road vehicle, with emphasis on enabling high off-road speeds. In conjunction with this four-corner algorithm development, a subscale, four-corner test-rig was designed and fabricated to enable algorithm testing in hardware, and improvements were made to the single wheel algorithms.

This paper will be of interest to the community of active suspension engineers for three reasons. First, the applicability of NCFS suspension control approaches (that have been previously developed for single wheel test-rigs) for full vehicle control is presented. Second, the ease with which the single wheel algorithms transferred to four-wheel test-rig control is presented. Third, the scaling approaches used by UT-CEM in designing a subscale four-corner test-rig are presented.

BACKGROUND

In passive spring-mass-damper suspensions, the spring and damper produce force on the sprung mass in response to relative motion between the sprung and unsprung mass. The result is the transmission of some of the terrain height variations to the sprung mass. Consequently, the passenger ride quality degrades when attempting to negotiate rough cross-country terrain to the extent that passenger acceleration loading becomes the major limitation in high speed cross-country travel for military vehicles.

An active NCFS approach, combined with damping based on sprung mass absolute velocity ("sky hook damping"), demonstrated on a laboratory based single wheel test-rig at UT-CEM had previously shown potential for greatly reducing sprung mass accelerations and significantly improving cross-country performance [2]. Control approaches for the UT-CEM test-rig are based on maintaining a nearly constant supporting force for the sprung mass [2]. The concepts can be understood by considering that to achieve a perfect ride from an active suspension, the equations for motion of the sprung mass can be solved for the actuator force necessary to maintain a constant force on the sprung mass. The result referred to as a CFS, requires obtaining and processing the road arm angle and its derivatives in real time. A less demanding approach, referred to as a NCFS, utilizes a close approximation to constant force, combined with feedback to correct for unwanted sprung mass motions. For the trailing arm suspension geometry in figure 1, it can be shown that a near constant force is produced by maintaining a constant output torque from the actuator [1]. Consequently, a practical alternative to the CFS approach is a NCFS system that produces constant torque, supplemented by feedback systems to correct for minor sprung mass motions that result from the fact that the actual force on the sprung mass is slightly non-constant.

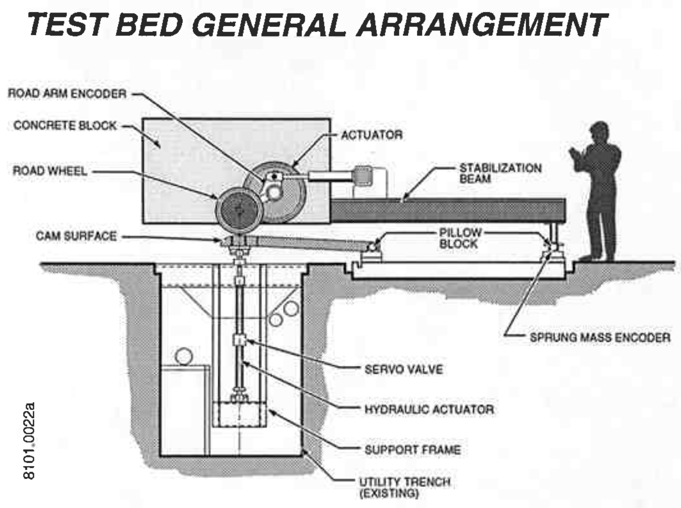

Figure 1. This diagram depicts the EMASS NCFS single wheel test apparatus general layout. The entire apparatus pivots about the pillow block at the far right of the diagram. Below ground level, a hydraulic actuator imparts vertical displacements to simulate road disturbances to a M1 road-wheel. The round actuator embedded in the 4,500 kg (5 ton) concrete block is a bi-directional electro-mechanical (EM) torque motor. The torque motor applies torque through the road arm/road wheel assembly. Also, shown attached to the road arm, is an air spring. The air spring provides static support for the block to minimize the EM actuators torque and power requirements.

The UT-CEM EMASS system employs an air spring to support sprung mass static weight. Motivation for this approach was to reduce size and capacity of the electro-mechanical actuator. Constant combined spring and actuator torque is achieved when the EM actuator is used to cancel the fluctuations in air spring torque that occur with road arm rotations (which cause the spring to compress or elongate). This results in a good "near constant force" approach. In the NCFS, some sprung mass motion results due to imperfect spring cancellation, friction and weight variation. Hoogterp reported greatest success in reducing sprung mass motion through use of a sprung mass damping term [3]. Consequently, NCFS can be enhanced by damping sprung mass motion with respect to an absolute reference.

EM actuators are particularly suited for the active NCFS, because they are inherently force transducers, whereas, their hydraulic counterparts are usually employed as displacement transducers. Force transducers produce force in proportion to a command and produces no force otherwise (i.e. back-drivable). In comparison, displacement transducers must be closed loop positioned through a load cell to produce the desired force (i.e. not back-drivable). As a result, when force control is desired, use of a displacement transducer unnecessarily complicates the system by requiring an additional control loop [4].

SINGLE WHEEL SYSTEM AND EXPERIMENTS

UT-CEM's single wheel EMASS demonstrator utilizes a back-drivable EM actuator as its active component. A schematic of this system is shown in figure 1. The direct drive actuator is coupled to M1 tank trailing arm suspension components and supports a 5,000 kg concrete block. Initially, the program objective concentrated on the feasibility and design of an EM actuator which meets both the suspension system force requirements and a 250 kg mass budget. To minimize the system weight and static power consumption, a passive air spring provides static support for the sprung mass. A NCFS approach, with absolute spring damping, was used to control sprung mass motion. In addition, a sprung mass displacement proportional term was used to prevent static ride height from drifting. Simulated single wheel performance for a standard M1 tank torsion bar system on a 8.85 cm rms cross-country terrain while traveling at 4.5 m/s is presented in figure 2. The benefit of the EMASS control approach over the standard M1 tank torsion bar suspension is illustrated through its ability to achieve superior cross country performance at 17.9 m/s. Measured performance using a hydraulic terrain simulator for the single wheel system traversing a 8.85 cm rms terrain at 17.9 m/s is shown in figure 3. The cross-country terrain produced by the hydraulic terrain simulator in the laboratory, produced a road-wheel acceleration exceeding 100 m/s^2 peak, while the sprung mass acceleration remained less than 3.0 m/s^2 peak and less than 0.5 m/s^2 rms [2].

MULTIWHEEL TEST-RIG AND EXPERIMENTS

Multiwheel active suspension efforts focused on development of a system for a military HMMWV with the desired goal to improve cross-country performance. A full size HMMWV has approximately 25 cm of total wheel travel. At UT-CEM, a 1/3 scale HMMWV demonstrator with the equivalent of 42 cm total wheel travel was built to test and evaluate multiwheel EMASS control algorithms and explore the potential performance improvement that additional travel might provide.

UT-CEM's 1/3 scale HMMWV, multiwheel EMASS demonstrator features four independently controlled direct drive EM actuators for active components. The suspension is designed to mimic the HMMWV's dual A-arm suspension geometry with each back-drivable actuator making the connection between the upper control arm and the chassis (see bottom of figure 4). Recognize that an A-arm attached to each torque motor on the four-corner test-rig (that simulates a wheeled

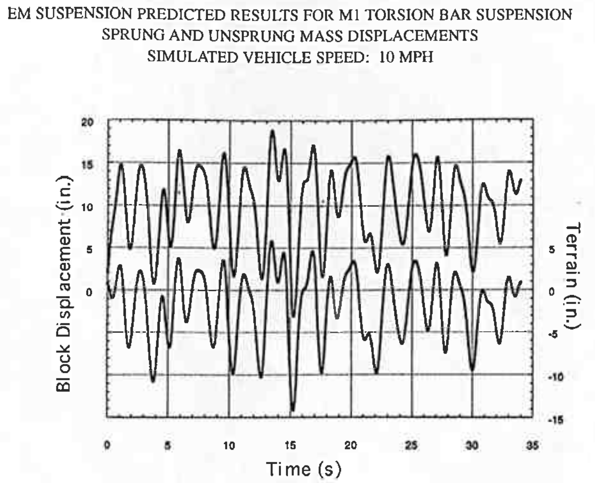

Figure 2. Plot shows simulation results of the single wheel test station shown in figure 1. For this plot, the EMASS components were replaced with passive components equivalent to a M1 tanks torsion bar and rotary damper. Simulated vehicle speed is 4.47 m/s (10 mph). This plot provides a performance baseline for the M1 to compare EMASS results to. The sprung mass vertical displacement ranges from -6.66 cm to 50 cm.

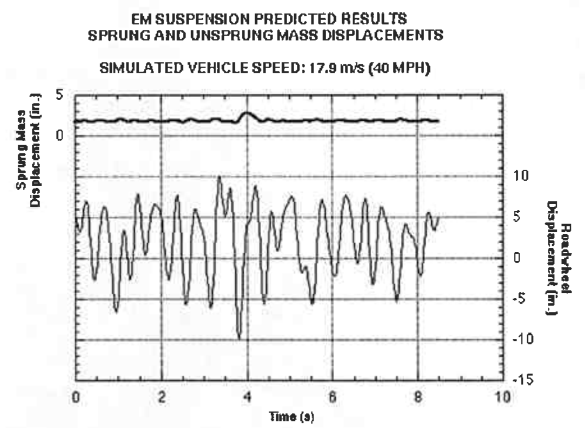

Figure 3. Plot shows the displacement of the 4,500 kg (5 ton) sprung mass when the road wheel is excited by the hydraulic terrain simulator using the present NCFS control approach. In this approach, sprung mass velocity which is used to damp sprung mass displacement is calculated from the integral with respect to time of sprung mass acceleration. The displacement of the road wheel as it travels over the simulated 8.85 cm (3.486 in.) rms terrain at 17.9 m/s (40 mph) is also shown. On this test, EMASS system reduced the rms acceleration of the sprung mass to less than 0.5 m/s^2, while the road wheel exceeded 16 m/s^2 rms.

vehicle) is very similar to the trailing arm configuration used on the single wheel test-rig (that simulates a tracked vehicle). Both the tracked vehicle road arm and the wheeled vehicle A-arm are simple levers that convert torque to roughly vertical force. As with the single wheel system, each corner of the vehicle is statically supported with an air spring and NCFS is achieved through spring rate cancellation. The EM actuators are rated for a peak torque of 35 N-m and a continuous torque of 15 N-m. Each corner of the test-rig is attached to an independently programmable hydraulic actuator by which the terrain is input. The assembled HMMWV system is shown in the top of figure 4. The model is approximately 1 m long and has a sprung mass of approximately 100 kg. Each air spring/EM actuator assembly can accommodate a total wheel travel of 14 cm equating to approximately 42 cm total travel on a full size vehicle. The nominal ride height for the scale system is centered in the range of travel and sprung to unsprung mass ratios are comparable to the full scale HMMWV.

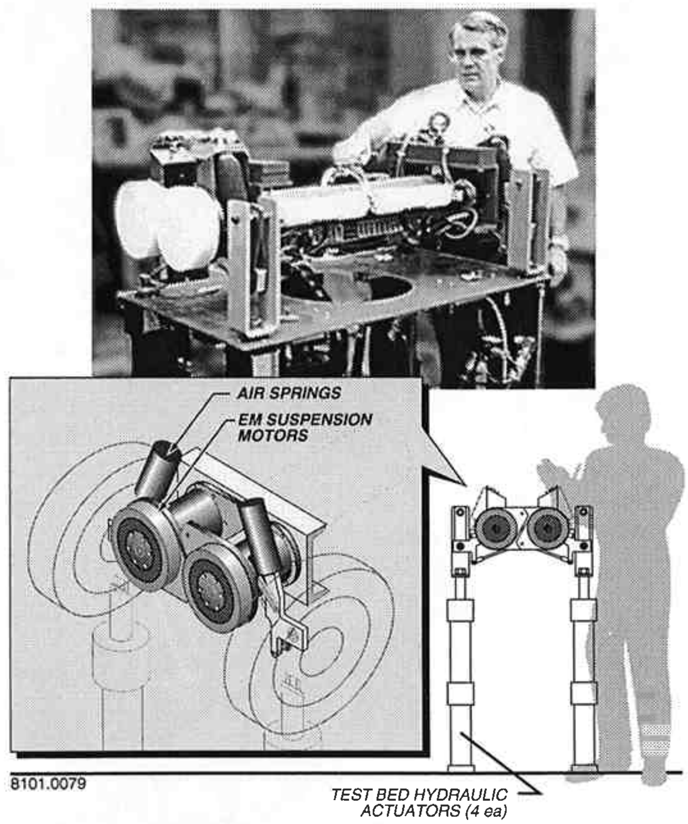

Figure 4. These diagrams show a schematic of two of the four wheel actuator assemblies on UT-CEM's 1/3 scale HMMWV. An EM actuator forms the attachment point and rotational pivot for the upper control arm. Upper and lower control arms attach to and maintain camber of a wheel spindle. An airspring provides static support of the vehicle through its attachment to the top of the wheel spindle. The tires and wheels were omitted from the test-rig and the wheel spindle attaches directly to the hydraulic terrain simulator hydraulics through a swinging yoke. The picture shows the completed 1/3 scale four-wheel HMMWV test-rig.

Since the multiwheel demonstrator was built at 1/3 scale, scaling laws were developed to insure that the subscale and full-size systems would have like performance. Recognizing that gravity acting on a full scale system and on the subscale test-rig would be identical, it was decided to keep acceleration scale independent (i.e., the same in each system). It was also decided to have length scale linearly. This led to the following relations between the model (i.e., subscale) and full scale systems:

letting

s = scale
x = displacement
v = velocity
a = acceleration
t = time
M = mass
F = force

and given the desired scaling relationship that

$$x_{model} = s * x_{actual} \quad (1)$$

and

$$a_{model} = a_{actual} \quad (2)$$

From kinematics, for constant acceleration,

$$x = 0.5 * a * t^2$$

so

$$t_{actual} = \sqrt{\frac{2 * x_{actual}}{a_{actual}}}$$

and

$$t_{model} = \sqrt{\frac{2 * x_{model}}{a_{model}}}$$

substituting

$$t_{model} = \sqrt{\frac{2 * s * x_{actual}}{a_{actual}}}$$

yields

$$t_{model} = \sqrt{s} * t_{actual} \quad (3)$$

since

$$v_{model} = a_{model} * t_{model}$$

substituting

$$v_{model} = a_{actual} * \sqrt{s} * t_{actual}$$

establishing

$$v_{model} = \sqrt{s} * v_{actual} \quad (4)$$

Similarly, the relations for the model's mass and force were developed. The scaling relations developed for the model are summarized in table 1. Of primary importance, this analysis indicates that time varies with the square root of scale. As a result, scale controller cycle times should be roughly half (0.577) those used in the full scale system and that scale terrain profiles must negotiated in about half (0.577) the time as full scale terrain profiles.

Table 1. Scaling laws developed for 1/3 scale HMMWV EMASS demonstrator

Parameter	Dependency	Scale Value
Distance	scale	0.333
Velocity	square root of scale	0.577
Acceleration	1	1.0
Time	square root of scale	0.577
Mass	scale cubed	0.037
Force	scale cubed	0.037

The NCFS control algorithm presently used on the four-wheel EMASS demonstrator was directly transferred from the single wheel demonstrator. To facilitate four-wheels, required looping through the control routine four times with appropriate multiplexing of the input/output (IO) information, thereby maintaining independent control of each wheel. As a result, the control cycle time per wheel was lengthened four times. This caused an initial concern, because the scaling laws presented above indicate that to achieve the same performance as the single wheel system the 1/3 scale controller cycle time should be 0.577 times shorter. In spite of the relatively long servo cycle time, the four-wheel system generally performed well. That the scale system performance is acceptable at 8 ms cycle time indicates that a full size system should perform equally well with a 16 ms cycle time.

The only problems that could be attributed to cycle time were associated with calculating the numerical derivative of A-arm position for cancellation of parasitic frictional damping. Characterization of the 1/3 scale HMMWV air springs revealed considerable viscous damping and stiction in the system. To achieve a NCFS, it would be necessary to cancel this additional viscous damping and stiction. Without this cancellation, the system has the equivalent of 47% of critical damping for a sprung mass natural frequency of 0.8 Hz. This additional damping and stiction results in greater sprung mass displacement. Because the longer controller cycle time, the maximum gain that could be applied to calculated velocity without causing controller oscillation was an order of magnitude less than what was necessary to completely cancel the parasitic friction of the system. As a result, performance evaluations were made without the benefit of active damping cancellation.

An additional concern was that because each wheel was controlled independent of the others, that the controllers might oscillate or fight each other. A 2-axis tilt meter was installed on the 1/3 scale HMMWV system to provide additional tilt/roll and tilt rate/roll rate feedback, if necessary, to damp inter-controller oscillations. As of this time, it has not been necessary to use this additional control feature.

Laboratory EMASS demonstrator performance was evaluated while negotiating cross-country terrain similar to what was used in the single wheel tests. Measured sprung mass displacement and the hydraulically simulated terrain input for the 1/3 scale HMMWV EMASS test-rig is presented in figure 5. The terrain's amplitude was reduced to the model's 1/3 scale suspension travel of 14 cm peak to peak and 2.66 cm rms The terrain inputs to the two front wheels are identical and the two rear wheels are delayed 32 m/s to account for the effect of a 17.9 m/s forward velocity. Front and rear wheel sprung mass natural frequencies are the same and the system relies on the sprung mass damping to damp fore/aft rocking motion. Measured performance from the 1/3 scale demonstrator shows similar performance at high speed cross-country terrain as the single wheel demonstrator achieved. The EMASS system reduced the rms acceleration of the sprung mass to less than 1.38 m/s^2, while the road-wheel exceeded 14.8 m/s^2 rms in spite of the aforementioned friction and cycle time concerns.

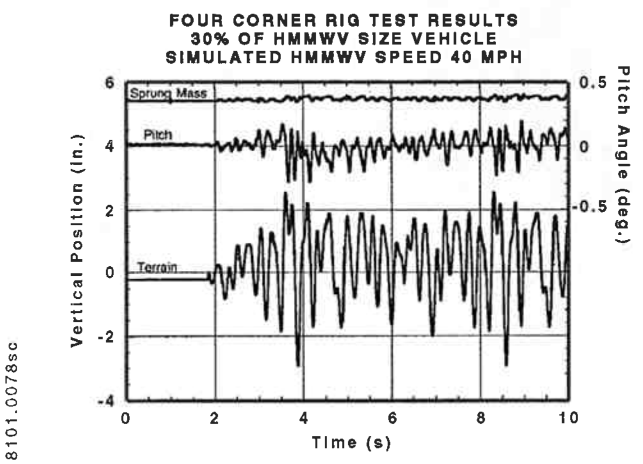

Figure 5. Plot shows the displacement of the 1/3 scale HMMWV 100 kg (220 lb) sprung mass when the road-wheel is excited by the hydraulic terrain simulator using the present NCFS control approach. The displacement of the road-wheel as it travels over the simulated 2.66 cm (1.05 in.) rms terrain at 17.9 m/s (40 mph) is also shown. On this test, EMASS system reduced the rms acceleration of the sprung mass to less than 1.38 m/s^2, while the road-wheel exceeded 14.8 m/s^2 rms.

Future efforts will attempt to offset viscous friction effects through incorporation of A-arm velocity transducers for active cancellation of the friction. In addition, new efforts will focus on EM actuator mass reducing concepts, such as gear reduction systems and permanent magnet actuators, as well as development of algorithms for following major terrain features.

CONCLUSIONS

A multiwheel test-rig, representing a 1/3 scale military HMMWV was designed and fabricated. Single wheel EMASS control algorithms utilizing a NCFS approach were transferred and successfully demonstrated on this multiwheel test-rig. Multiwheel performance similar to what was demonstrated for the single wheel was achieved. No additional roll/tilt control or damping proved necessary for successful control of the four-wheel system. In addition, minimal effect on overall EMASS performance resulted from the equivalent of nearly an order of magnitude increase in controller cycle time and from substantial viscous friction in test-rig mechanics.

Future efforts will address undesirable damping in system mechanics, actuator size and weight, and control algorithms necessary to follow major terrain features.

ACKNOWLEDGMENT

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