An Energy Efficient Electromagnetic Active Suspension System

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ABSTRACT

The technology thrust to develop an effective electromagnetic actuator for application in an active suspension system has precipitated a fresh look at the active control schemes in an effort to reduce the required force levels of the actuator. The resulting "near constant force" control algorithm is described and its ability to greatly reduce vehicle sprung mass motion is documented through simulation and single wheel station laboratory test stand results. The vehicle power and energy requirements associated with this unwanted vehicle vertical are analyzed and comparisons between the corresponding passive and active systems are presented. The success of the active system leads naturally to the conclusion that a passive suspension equipped vehicle will become power limited at a much lower speed than will this active system when traversing severe cross-country terrain.

INTRODUCTION

The possibility of utilizing an electromagnetic (EM) actuator in a high-wheel travel active suspension system has recently gained considerable interest in both the commercial and the military sector [1-2]. The bus developers are interested in the potential benefits of active suspension for ride comfort and vehicle roll control as well as for the accompanying height and attitude control. Height and attitude control can be used to greatly simplify potential solutions to the handicap access requirements the bus manufactures must meet. The interest in an EM actuator based system is predicated on the availability of an energy efficient, and appropriately packaged system, that if exists would eliminate the need for a complex hydraulic system.

A passive suspension system is simply a traditional suspension system composed of springs, dampers (i.e. shock absorbers), and various structural linkages [3]. The springs and dampers of a passive system are appropriately tuned for the expected input frequencies, to attenuate the terrain input (as seen by the tires). The purpose of the shock absorbers (or dampers) is to dissipate the energy that is input into the chassis by the ground through the suspension system. The more energy that is allowed to be input to the chassis, the more energy that the shock absorbers must dissipate, and the more energy that the vehicle's power train must provide. A semiactive suspension system is defined here as a passive system which has adaptive damping characteristics. The system has the traditional springs and dampers, but the

vehicle motion is constantly monitored, and the damper characteristics are reduced (or set to zero) when the damping forces would increase chassis motion. It has been demonstrated that semiactive suspension systems can moderately increase the ride quality of a vehicle while simultaneously slightly decreasing the energy dissipated by the suspension system [4-5].

An active suspension system represents a distinct departure from traditional suspension system concepts. Springs and dampers are no longer required (though for practical reasons springs are often still included). The suspension forces that are normally provided by the springs and dampers are now provided by some form of actuator (e.g. hydraulic, pneumatic, or EM) based on the output of vehicle motion based control algorithms. The active suspension control research for rough terrain negotiation has primarily focused on more sophisticated approaches to the attenuation of the ground induced inputs [6-7]. The success of active suspension has been quite impressive in the areas of on-road stability and handling and has been demonstrated repeatedly. The cross-country ride problem, however, has not proven as easy to conquer to date.

The military's interest in active suspension systems also begins with significantly improving vehicle ride and stability, but for the military, cross-country ride is the focus of interest. The Army's thrust to remove hydraulics from its ground combat vehicles precipitated its interest in EM actuator based active suspension systems. The technology thrust to develop an electric drive combat vehicle has only served to heighten this interest since adequate electric power would be available on the vehicle and the addition of a hydraulic system for active suspension would only add to the overall vehicle complexity.

The intended application of such a system would be to obtain maximum high-speed cross-country performance desired by military applications. The major barrier to making such an application practical has been the weight and power requirements of an electromagnetic actuator based system. Since the actuator weight for such a system is driven largely by the actuator force requirements and therefor indirectly by its power demands, minimizing the required active suspension power will also greatly reduce the system weight.

An active suspension control algorithm is presented that requires significantly less power (and correspondingly lower forces) than do other algorithms. This "near constant force"

strategy attempts to prevent the excitation of the vehicle's sprung mass rather than to attenuate the motion once it has begun. The resounding success of this algorithm is demonstrated through a series of simulation and laboratory test stand results. The laboratory test stand results include test data obtained from a single wheel station, rotary EM actuator, supporting a sprung mass of 4500 kg as well as test data from a reduced scale, 4 wheel representation of a high mobility off-road vehicle. Simulation results corresponding to the test data are also presented.

The power and energy requirements of this active suspension configuration are actually shown to be significantly lower than those for the comparable passive suspension configured vehicle. In addition the acceleration levels of the sprung mass are also dramatically reduced resulting in the operational maximum cross-country speed being significantly increased over, not only the passive system, but over other proposed active suspension configurations as well.

NEAR CONSTANT FORCE ACTIVE SUSPENSION CONTROL ALGORITHM

The underlying philosophy for the "near constant force" (or NCF) active suspension system algorithm is quite simple. Since the purpose of a suspension system is to isolate the chassis (or sprung mass) from the motion of the wheel (or unsprung mass) which is trying to follow the terrain, a perfect suspension system (in terms of isolation) would always generate only the static force to the chassis. With a constant vertical force on the chassis, the chassis would then be subjected to zero vertical acceleration and therefor a perfect ride. In practice, of course, this is simply not practical since the vehicle must go up and down hills and the suspension travel is limited to a few inches at best.

The idea persists, however, that the best active suspension control algorithm would be the one that would provide the least variation in suspension forces while minimizing the prospect of exceeding suspension travel. Thus was born the NCF control algorithm. The perfect solution of this problem would require prior knowledge of the terrain, but the development of the NCF algorithm was developed without the assumption of this prior knowledge and yet has been shown in the laboratory to perform at an extremely high level of efficiency.

A model of the original laboratory test stand for which the NCF control scheme was developed, is shown in figure 1. This model represents (as does the test stand) one of the fourteen roadwheel stations and one-fourteenth of the hull mass of a main battle tank (the M1 Abrams). The road-wheel is mounted on a trailing arm and the hull mass is constrained so that it can move only in the vertical direction. The suspension vertical force to the hull is actually applied as a torque between the road arm and the hull at the road arm pivot. Since this model is restricted to vertical motion, it can be viewed the same as a 1/4 car model would be for a four wheeled vehicle. Figure 2 presents the predicted motion of this model's hull mass for the standard torsion bar suspension system on the M1 tank to the terrain input shown. This terrain is extremely rough as indicated by its terrain roughness index of 8.9 cm rms. The gradual hills were removed from the terrain prior to the simulation to comply with the travel

Figure 1 One-dimensional tank suspension model

limits of the laboratory terrain generation hardware. This simulation represented a vehicle velocity of 16 kph and it can be seen from the plots that for this very stiff passive suspension system, the hull tends to follow the terrain.

Any practical active suspension design will include a passive spring to statically support the vehicle and to reduce the overall power requirements of the active system and the EM actuator based system is no different. The demonstration rotary actuator built on a main battle tank scale, incorporated air spring for this purpose. This permitted the spring rate to be kept quite low while still providing the required static force. The first task of the NCF algorithm is to negate the effect of this low rate spring which varies with the arm angle of the trailing arm suspension system. The NCF algorithm was developed based on the premise that it would have available only measurements that could be obtained from the vehicle itself. These could include arm angle and velocity making the cancellation of the spring force a simple matter, and could also include, as needed, chassis (or hull) vertical, pitch, and roll accelerations, velocities, and displacements (or estimations thereof). The resulting actuator

Figure 2 Simulated hull motion at 16 kph for passive M1 torsion bar system

force from the NCF algorithm for a 1/4 car model (i.e. considers chassis vertical motion but not pitch or roll) is given by:

$$F_a = -F_s(\theta_{arm}) - k_1[y - y_{ref}] - k_2 y' - k_3 \theta_{arm} - k_4 \theta'_{arm} \quad (1)$$

where $F_s(\theta_{arm})$ is the force required to offset the air spring's additional force (as indicated, F_s is a function θ_{arm}), y is the vertical displacement of the chassis, θ_{arm} is the angle of the roadarm, y_{ref} is the height of the desired chassis vertical trajectory, and k_1 , k_2 , k_3 , and k_4 are appropriate gains. Of course the displacement of the chassis, y , and of the desired reference trajectory, y_{ref} , are not readily available to the controller and so the gain k_1 is set equal to zero in the NCF algorithm. The term is left in equation (1) to emphasize the fact that it is ideally what is desired to be controlled. However much can be discerned about the nature of the term $[y - y_{ref}]$, from the more easily measured signals y' and θ_{arm} and feedback based on these terms is used instead of on $[y - y_{ref}]$.

Figure 3 shows the hull motion for a 16 kph simulation as did figure 2, but in this case an active suspension system was assumed and the suspension force is determined by the one-dimensional NCF algorithm. Also, as was done for figure 2, the gradual hills had been removed from the terrain. Figure 4 shows the actual results from the laboratory test stand corresponding to the predicted results of figure 3. The comparison is quite comforting and lends credibility to the attempt to greatly decrease hull motion in response to severe cross-country terrains.

The appropriate choice of gains in (1) allows the NCF algorithm to actually respond to hills while effectively filtering out the higher frequency inputs inherent in rough cross-country terrain. Figure 5 presents the simulated hull response to the same terrain used in the previous examples, but where the gradual hills have been left in the terrain. This simulation was performed at a vehicle speed of 64 kph and shows almost a total attenuation of the higher frequency inputs.

Figure 3 Simulated hull motion at 16 kph for NCF active suspension system

Figure 4 Laboratory test stand hull motion at 16 kph for NCF active suspension system

The version of the NCF algorithm presented in (1) is concerned with minimizing vertical chassis motion only. The extension of the NCF algorithm to a full 3-dimensional vehicle model must also consider the chassis pitch and roll motions in calculations. The NCF actuator force for this full vehicle model can be similarly be written as:

$$F_a = -F_s(\theta_{arm}) - k_1[y - y_{ref}] - k_2 y' - k_3 \theta_{arm} - k_4 \theta'_{arm} - k_5 \theta - k_6 \theta' - k_7 \alpha - k_8 \alpha' \quad (2)$$

where θ and θ' are the chassis pitch angle and pitch rate, α and α' are the roll and roll rate, and the additional k 's are appropriate gains as before. This 3-d version of the NCF active algorithm was implemented to control a 30 % scale, 4 wheeled active suspension test rig representing the Army's High Mobility Multi-purpose Wheeled Vehicle (or HMMWV). Preliminary results from this test rig were similar in terms of chassis motion reduction, to those discussed earlier for the single wheel station system.

Figure 5 Simulated hull motion at 16 kph for NCF active suspension with hill following employed

POWER COMPARISONS FOR ACTIVE AND PASSIVE SUSPENSION SYSTEMS

It is clear that when the sprung and unsprung masses are moving up and down, that some of the power that was intended to drive the vehicle forward is being expended in the vertical direction. This power is, of course, supplied by the vehicle's engine and is delivered through the inefficiencies of the drive train to provide motion to the vehicle. Power expended in exciting the vehicle vertically is totally wasted, and if it becomes excessive will significantly limit the overall performance of the vehicle. It will be demonstrated that this is in fact the case for the passive suspension system on the M1 when trying to negotiate very rough terrain.

The unsprung mass (roadwheel or tire) is intended to follow the terrain and thus must be continuously accelerated in the vertical direction when traversing rough terrain. The unsprung mass, however, is usually less than 10 % of the sprung mass and thus is a rather minor drain on the system power than is the sprung mass at the same excitation levels. The active suspension system cannot, therefore, effectively reduce the unsprung mass motion of the power required of the engine to produce that motion.

The sprung mass on the other hand has no requirement to follow the high frequency component of the terrain input, and in fact, the suspension systems very purpose is to keep it from doing so. Unfortunately, the passive system, as was demonstrated in figure 2, is not very effective at accomplishing this. In the passive system the power that is input into the unwanted vertical (as well as pitch and roll) motions is ultimately dissipated by the shock absorbers and lost. In order to compare the vertical motion power between the active and passive suspension systems, vertical sprung mass power was computed as:

$$\text{Power} = F v = F a v \quad (3)$$

where F is force, v is velocity, m is mass, and a is acceleration. The resulting comparisons that are shown below are for a single

Figure 6 Sprung mass vertical power requirements for active and passive systems at 16 kph

wheel station as represented in figure 1, where the sprung mass represents 1/14th of an M1's sprung weight (the M1 has 14 road-wheels).

Figures 6 and 7 depict the power required to accelerate the sprung masses for the passive and NCF active suspension systems at 4.47 m/s and 17.8 m/s. The simulated results are over the same severe cross-country terrain that was used in the previous examples. The passive system employed the standard M1 torsion bar and rotary damper whereas the active system was modelled with the electromechanical suspension active suspension system described in [1] and utilizing the NCF control algorithms described above. At the lower speed, figure 6 shows that the rms and peak powers associated with both systems is quite small. At 17.8 m/s, however, the difference in power between the two systems is dramatic as demonstrated in figure 7. The passive system requires 540 kW peak and 71 kW rms power just to accelerate the sprung mass of a single wheel station along its vertical trajectory. Projecting this rms power over the 14 road-wheels of the M1 would result in a total sprung mass acceleration rms power of 994 kW, or almost 90 % of the M1 engines rated gross power. It is realized that not all the M1 wheel stations are identical (only 6 of the 14 have dampers) and vertical motion of the sprung mass is somewhat abated over the length of the vehicle, but this power figure does not include contributions from the hull pitch and roll motions either.

It should be noted that the passive M1 could not attain nearly this speed (i.e. 17.8 m/s) across this terrain even if the vehicle and crew could stand the accelerations generated by such a run. The vehicle is in fact power limited long before this speed even though this is approximately the level surface top speed of the M1.

Figure 8 portrays the peak and rms power levels for this same pair of suspension systems as a function of vehicle speed. The terrain is the same cross-country terrain that was used previously and the plots were generated from simulation results using the same single wheel station model. The item of major interest here is the extreme divergence in power requirements at higher vehicle speeds between the passive and the NCF algorithm based active suspension system. Thus high speed travel on such a

Figure 7 Sprung mass vertical power requirements for active and passive systems at 64 kph

terrain simply cannot be accomplished by this passive suspension system (or probably any other practical passive system). It will require instead, not only an active suspension system, but an active system that significantly reduces unwanted vehicle hull motions.

Figure 8 Power consumption between active and passive systems as a function of vehicle speed

ENERGY USAGE

When considering energy consumption of the passive and active suspension systems, the vehicle drive train and active suspension power delivery system efficiencies must be considered. At the 17.8 m/s speed, the passive system required 994 kW rms of power to provide the sprung mass vertical acceleration of the total vehicle. Since the power to accomplish this motion must be provided through the vehicle's drive train, the associated additional power losses can also be attributed to the excessive vehicle motion. If it is assumed that the vehicle automatic transmission and differential each operate at 90 % efficiency, these losses are approximately 233 kW.

These power train losses would represent the total power lost if the energy imparted to the sprung mass were completely recoverable, however this is not the case. Separate simulations show that an individual M1 shock absorber can dissipate as much as 95 to 100 kW of power when the vehicle is driven at 17.8 m/s over this same terrain. It should be noted that there are six shock absorbers on the M1 (on the front two and the rear wheel stations on each side) but the other wheel stations also have some inherent damping associated with them also.

The rms power required for the active system to accelerate the sprung mass was shown to be only 5 kW. This power can be applied by one of two paths to the sprung mass. As in the passive case, the power can be delivered through the vehicle drive train with the partially misdirected goal of vehicle forward motion in mind. Alternatively, the power to accelerate the sprung mass can be delivered directly through the active suspension system actuators. If in this latter case, one assumes that the active suspension power conversion device (for either mechanical to hydraulic or mechanical to electrical) and the suspension force actuator are both 90 % efficient, then the overall efficiency of the power delivery process is 81 % regardless of whether it is delivered through the vehicle drive

train or through the active suspension system. Thus the total power required from the engine to provide the sprung mass motion for the NCF active system is only 6 kW even if none of the sprung mass motion energy is recoverable.

The power losses attributable to the unsprung mass vertical motions have not been considered here but they are not avoidable since the wheels must be made to follow the ground in order to be able to provide the necessary hull (or chassis) support. Ground contact is also required, of course, to permit adequate vehicle stability and handling. In either case, the wheel motions of both the active and the passive systems should be quite comparable making the power lost to unwanted sprung mass motion the major power differential between the two systems.

CONCLUSIONS

The attempt to negotiate severe cross-country terrain at high speeds with conventional suspension equipped vehicles has been shown, to not only impart excessive motion to the vehicle hull or chassis, but to also require such high levels of available engine power that the higher speeds are simply not possible. The "near constant force" (or NCF) active control algorithm has been defined, and when coupled with an EM suspension system has been shown to reduce the sprung mass motion power by a factor of about 200. The NCF approach to active suspension control requires much lower actuator force levels than do previous active control schemes, thus making the prospects of a competitively sized EM actuator quite plausible.

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