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# Benefits of Semi-Active Damping in Up-Armored Vehicles for Ride, Handling, and Rollover Prevention

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## ABSTRACT

*The need for up-armored vehicles has increased over the years. This has put a greater emphasis on suspensions that can provide improved ride and handling capabilities while facing the additional weight. One of the challenges with these vehicles traditionally has been increased likelihood of rollover. Increased rollover is due to high center of gravity, kinematics of the overloaded suspension, and the low damping that is needed to satisfy 6-Watt ride speed performance criteria. The Lord magneto-rheological (MR) suspension system addresses these issues by improving the ride quality and handling characteristics thereby increasing safety and mission effectiveness. During handling maneuvers, algorithms inside the controller unit apply corrective forces to minimize peak roll angle and peak roll rate. The benefit of this has been tested on a vehicle comparing the stock passive dampers to the MR dampers over NATO Lane change events. Furthermore, the controller has the capability to continuously calculate the propensity of a rollover event. This can provide an audible alarm to the driver allowing the driver to learn the boundaries of safe handling for the vehicle.*

## INTRODUCTION

Over recent years, controlled suspension solutions have begun to evolve into the commercial and military marketplace. Several types of solutions are widely available today in the luxury passenger automotive market. Similar types of solutions are being tested and evaluated in the military realm where numerous up-armored vehicle platforms are in need of better ride and handling characteristics. As the need for up-armored vehicles increases, there will always be a need for controlled suspension solutions to provide better ride to increase mission time, better handling for improved safety, and better reliability to improve mean time between part failures. MR suspensions meet this demand with cost-effective and performance-driven solutions.

## MR CONTROLLABLE SUSPENSIONS

Magneto-rheological (MR) dampers are piston/cylinder arrangements where MR fluid is the working fluid in the device. By controlling the magnetic field across the flow path of the fluid the magnitude of the passively generated force can be modulated at high frequency.

The Lord MR Suspension system is comprised of MR dampers, position and inertial sensors, wiring harness, and a central controller that makes decisions in real-time. The system response time (from sensor signal to algorithm calculations to force at damper) is less than 10 milliseconds. The controller complies with MIL-STD-1275, -461, and -810, is J1939 data bus compatible, and can accommodate up to eight damper stations.

## CONTROL ALGORITHMS

Most controllable suspensions using real-time control use control schemes which are, in part, derived from skyhook control. The 'skyhook' damper generates an inertial force opposing the absolute velocity of the sprung mass, while conventional dampers generate a force that is functionally dependent on the relative velocity across the damper. This conventional damping force can oppose the velocity of the sprung mass or can accelerate the sprung mass depending on the instantaneous dynamics of the overall system [1]-[4].

To generate a "skyhook" force on the sprung mass of a vehicle requires an active system, where power must be supplied to the controlled force generating element. It has long been recognized, that a skyhook inertial damping force can be generated most of the time by simply modulating the passive force of a conventional damper [5]. In other words, by controlling the magnitude of the damper force (but not its sign), an inertial damping force can be generated during the majority of an operating cycle. Such control, as inherent in Lord's semi-active solution, requires far less power, weight, space, and cost than a fully active system and delivers performance nearly as good under most operating conditions.

## VEHICLE PERFORMANCE METRICS

Several performance metrics were used to evaluate and compare stock passive dampers with MR dampers. For NATO lane change tests, procedures are followed as defined in AVTP 03-160 [6]. These tests are driver handling maneuvers in a double lane change, in which, lanes are defined by traffic cones set according to the vehicle dimensions. The driver attempts to maintain constant forward speed through the course until one or more cones are knocked over or until wheel lift occurs. A failure is deemed at a particular speed if a vehicle cannot pass the course at that speed within three attempts.

For vertical ground input events, the common metric of vehicle performance is the vertical 6 watt driver absorbed power over controlled terrains (RMS courses). Absorbed power is a frequency weighted integration of a seat pad or seat base accelerometer [7, 8]. A vehicle's upper limit of operation for long periods of time is 6 watts. This metric is usually centered on the driver location, but can be applied to the passenger and cargo/crew locations as well. The speed at which absorbed power reaches 6 watts for a given location is dependent on the vehicle type and test course as this is determined by multiple passes over a specified RMS terrain at constant forward speed. The resulting data is fit with a curve fit of the form $y = ax^b$, from which the 6 watt crossing speed is obtained.

For half-round tests, the common metric of vehicle performance is peak vertical acceleration over half-round bumps. The typical range of half-round sizes for up-weighted military vehicles is 4, 6, 8, 10, and 12 inch radius bumps. The peak vertical acceleration value is determined by filtering the signal from a seat base accelerometer with a 30 Hz low pass filter (typically 8$^{th}$ order Butterworth) [9]. Peak acceleration value is the larger of the positive and negative peak magnitudes and the accepted upper limit is 2.5 g's [7].

## INDEPENDENT VEHICLE TEST

Independent vehicle tests were performed on a 22,600 pound GVW vehicle comparing the stock dampers to the MR suspension system. Testing comprised of RMS events, various half-rounds, NATO lane change tests, and 10,000 miles of durability.

### *NATO Lane Change Test Results*

Double lane change tests were conducted in accordance with the North Atlantic Treaty Organization (NATO) Allied Vehicle Test Procedure (AVTP) 03-160W [6]. The test surface had a surface coefficient of friction of 0.8 or greater. The course was setup based on actual dimensions of the vehicle tested.

The vehicle was operated through the test course at speeds in 5mph increments until the end limit was approached at which point speed was increased in 2mph increments. Tests stopped when loss of control was experienced, which was defined as failure to maintain the vehicle within the test course lanes.

Passive tests were conducted first with the test vehicle in both left-turn first and right-turn first directions. Due to added stability from the MR system it was discovered that the system could provide even better results without an anti-roll bar which would result in further savings to the customer. Therefore, the roll bar was removed for MR testing. Official MR Tests concluded with the test vehicle achieving a 10.1% speed improvement in the left-turn first direction and a 12.6% speed improvement with MR on a right-turn first direction leading to an overall speed improvement over passive of 11.4% without an anti-roll bar.

Furthermore, it should also be noted that the end limit of the vehicle with MR system installed was due to engine performance while the end limit of the passive damper vehicle was due to failure of traversing the course without making contact with the cones.

Several signals were recorded as part of the data acquisition, one of which was the yaw rate at the vehicle CG. Peak yaw rate for various speeds can be seen in Figure 1. Results show that the MR damper equipped vehicle had an average 9% decrease in yaw rate.

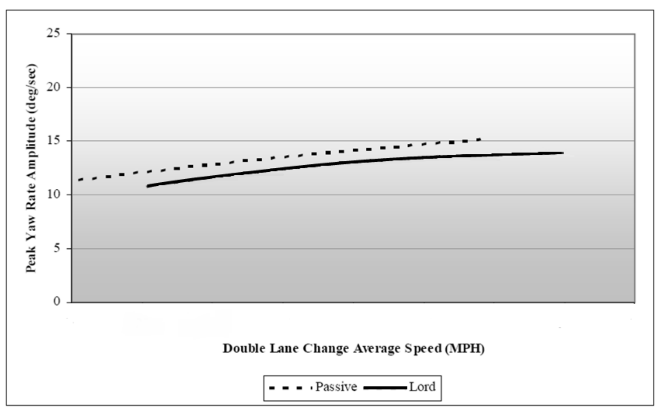



**Figure 1:** Passive versus MR peak yaw rate performance comparison.

The roll rate was also recorded at the CG of the vehicle and the comparison is shown in Figure 2. The roll rate decreased an average of 46% with the addition of the MR suspension system.

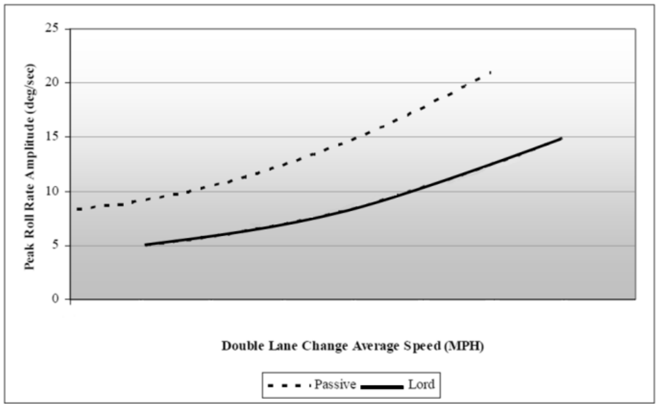



**Figure 2:** Passive versus MR peak roll rate performance comparison.

Roll angle information calculated from the controller was also compared at 50mph and this is shown in Figure 3. This shows the roll angle comparison between the stock vehicle and the MR suspension vehicle with anti-roll bar removed on a left turn first direction. As can be seen, there is significant decrease in roll angle for the MR equipped vehicle.

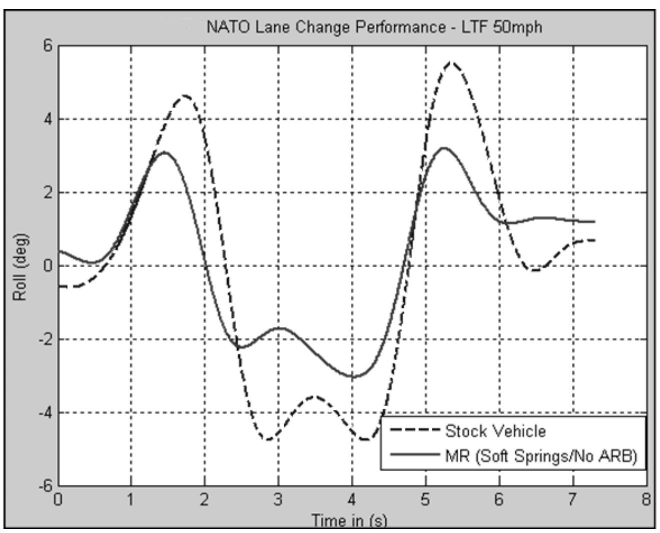



**Figure 3:** Passive versus MR roll angle performance comparison at 50mph.

### *RMS Test Results*

Tests were conducted with stock dampers and MR dampers on RMS courses ranging from 1.0" to 3.6" RMS. These results are mentioned here to demonstrate that handling and rollover improvements are achieved without detriment to other ride quality metrics. The tests revealed up to 26% improvement in 6 watt crossing speed, with an average of 19% improvement in ride quality with the MR system over these ride quality terrains. Furthermore, at the passive 6-watt crossing speed, the MR system reduces absorbed power an average of 37% over the tested range of terrains.

### *Half-Round Test Results*

Similarly, tests were run on half-round obstacles ranging from 6-inch to 10-inch half-rounds comparing stock dampers to the MR system. MR dampers provided up to 46% improvement in speed, with an average of 19% improvement in overall ride quality on the half-rounds.

## ROLLOVER ESTIMATOR

An additional feature of the MR controlled dampers is the potential for a rollover warning system using the same sensors that are already on the vehicle for overall semi-active control. A vehicle's rollover likelihood has much to do with the loading of the vehicle and the height of the c.g. The available suspension displacement sensors allow determination of the loading of the vehicle. Detection of the roll frequency from these same sensors allows the calculation of the c.g. height as detailed next.

Assume $\delta_0$ is the displacement across the suspension for nominal loading of the vehicle with suspension stiffness $k_{eff}$. Thus

$$m_0 g = k_{eff}\delta_0 \tag{1}$$

where $m_0$ is the nominal mass of the vehicle.

If the load increases then the suspension displacement $\delta$ becomes,

$$\delta = \delta_0 - \frac{m'g}{k_{eff}} \tag{2}$$

where $m'$ is the increased load on the vehicle. Combining Eqs. (1) and (2) allows determination of the added load from,

$$m' = (1 - \frac{\delta}{\delta_0})m_0 \tag{3}$$

The nominal roll frequency $\omega_{r_0}$ is related to the roll stiffness $k_\tau$ and the nominal roll moment of inertia about the c. g., $J_{r_0}$ through the relationship,

$$\omega_{r_0}^2 = \frac{k_\tau}{J_{r_0}} \tag{4}$$

The nominal roll moment of inertia is related to the nominal mass and c. g. height through,

$$J_{r_0} \approx m_0 \frac{w}{2} h_0 \tag{5}$$

From Eq. (4) we can determine how a change in the roll frequency $\Delta\omega_r$ is related to a change in the roll moment of inertia $\Delta J_r$. Furthermore, from Eq. (5) we can determine how $\Delta J_r$ is related to a change in mass $\Delta m$ and a change in c. g. height $\Delta h$. The result is,

$$\frac{\Delta h}{h_0} = -2\frac{\Delta\omega_r}{\omega_{r_0}} - (1 - \frac{\delta}{\delta_0}) \tag{6}$$

Obviously this procedure will require calibration for a given vehicle, but it is possible to determine load changes and c. g. height changes using the existing sensors.

Detection of the roadway slope $\theta$ might require an additional low frequency, pendulum-type sensor. This slope might also be determined from proper signal processing of the right/left vertical acceleration signals. In any case, acquiring the slope information is beyond the scope of this paper.

The most complicated part of the Rollover Estimator is the coupling of lateral acceleration of the vehicle with the current slope of the terrain. Figure 4 shows the rear view of a vehicle in a "positive" configuration where it is negotiating a left turn on a road with positive camber, i. e. $\theta > 0$.

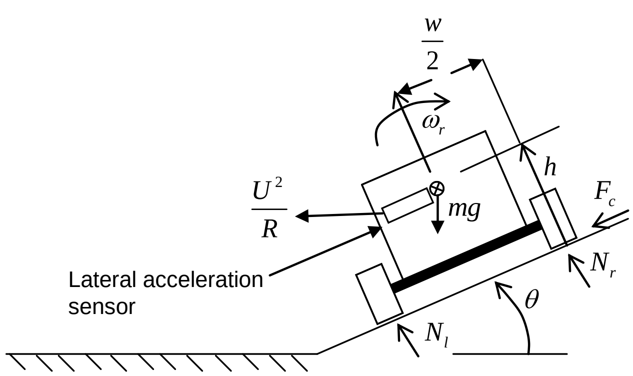



**Figure 4:** Rear view of a vehicle executing a left turn on a positive camber road

As the vehicle travels faster in the turn, if the traction is sufficient, then it will ultimately tip about the outside wheel and roll over in the positive roll direction as indicated by $\omega_r$. At the inception of rollover, the normal force at the left wheel will approach zero and the equations of motion at this instant are,

$$\begin{aligned} &F_c \cos\theta + N_r \sin\theta = m\frac{U^2}{R} \\ &N_r \cos\theta - F_c \sin\theta = mg \\ &F_c h - N_r \frac{w}{2} = J_r \dot{\omega}_r \end{aligned} \tag{7}$$

The lateral acceleration sensor will read in g's,

$$\bar{a}_l = \sin\theta + \frac{U^2}{Rg}\cos\theta \tag{8}$$

The concept of the Rollover Estimator is to determine what the sensor reading would be at the instant of rollover and then warn before this condition is exceeded.

The first two equations of Eq. (7), can be solved for $F_c$ and $N_r$ with the result,

$$\begin{aligned} &F_c = mg(\bar{a}_l - 2\sin\theta) \\ &N_r = \frac{mg}{\cos\theta}[1 + \sin\theta(\bar{a}_l - 2\sin\theta)] \end{aligned} \tag{9}$$

Using these in the third equation of Eq. (7) yields the condition for rollover as,

$$\bar{a}_l > \frac{\frac{w/2}{h}\frac{1}{\cos\theta}(1 - 2\sin^2\theta) + 2\sin\theta}{1 - \frac{w/2}{h}\tan\theta} \tag{10}$$

Thus, if the slope of the roadway can be detected and if the height of the c.g. is known with some confidence, then the onset of rollover is indicated when the lateral acceleration sensor has a reading that exceeds that of Eq. (10). A factor of safety would be used in actual application.

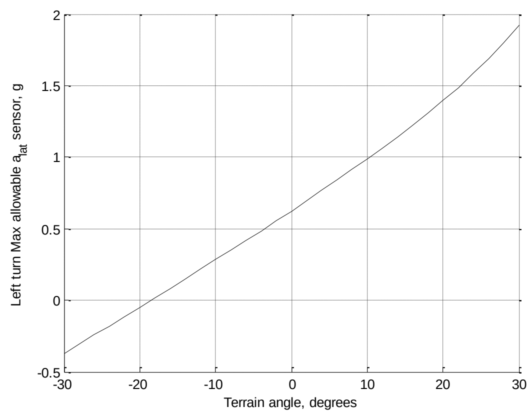
For the same configuration as shown in Figure 4 a similar analysis can be performed for a right hand turn. For this case the "off-camber" turn would occur when the terrain angle is positive and an "on-camber" turn would occur when the terrain angle is negative. Additionally, the lateral acceleration due to curving to the right would be in the opposite direction to that shown in the figure and the sensor would read,

$$\bar{a}_l = \sin\theta - \frac{U^2}{Rg}\cos\theta \tag{11}$$

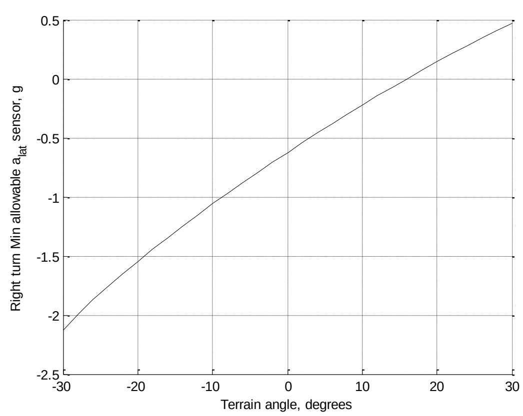
The result of the analysis is similar to Eq. (10) but this time,

$$\bar{a}_l < \frac{\frac{w/2}{h}\frac{1}{\cos\theta}(1-2\sin^2\theta)-2\sin\theta}{1+\frac{w/2}{h}\tan\theta} \qquad (12)$$

In words, the sensor reading must be less than the RHS of (12) to avoid rollover. Figure 5 shows the results for a left hand turn and a right hand turn over a range of terrain angles for $h = 1.4m$ and $\frac{w}{2} = 1.0m$.




(a) Left hand turn




(b) Right hand turn

**Figure 5:** Critical lateral accelerometer sensor readings for a left and right hand turn

The interpretation of these figures is as follows: For a left hand turn, if the terrain angle is zero, then rollover is imminent if the lateral acceleration sensor reads higher than about 0.6g. As the terrain angle increases (becomes more positive) the sensor can read a higher value before rollover is indicated and vice versa as the terrain angle becomes negative.

For a right hand turn, if the terrain angle is zero, the rollover is imminent if the lateral acceleration sensor reads less than about -0.6g. As the terrain angle becomes more negative (corresponding to a positive camber turn for this case) the sensor can read a larger negative value before rollover is indicated.

## CONCLUSIONS

It is well understood and documented that semi-actively controlled suspensions provide ride quality approaching that of fully active systems and are far superior to passive suspensions. There are ample commercial systems available for road vehicles. However, it is relatively new to apply the same semi-active technology to off-road and military vehicles.

The primary way to realize semi-active devices is through the use of MR fluid and such devices are called MR dampers. Such dampers require only the modulation of an electro-magnetic field for their operation. The overall system requires sensors and signal processing capabilities very much like their fully active counterpart. The MR suspensions described in this paper were developed primarily for ride control where passenger comfort and job performance are the primary focus.

An additional feature that requires no additional sensors is improved roll control of the vehicle. Using the same control strategies that are operational for ride control, the system can also provide significant improvement in roll control at no additional cost in hardware. Furthermore, using the same sensors and some additional signal processing, a Rollover Warning System can be implemented where the vehicle operator is alerted to the potential for rollover with sufficient time to react to the threat. A rollover estimator, such as this, is best developed and fine tuned to a particular application by active engagement from the Army and OEM. This will ensure the estimator is well designed for each application.

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