# UPGRADED SUSPENSIONS SYSTEMS IMPROVE MOBILITY AND SURVIVABILITY OF GROUND VEHICLES

## ABOUT THE AUTHORS

**John LaPlante Chief Technology Officer, Active Shock, Inc Manchester, NH jlaplante@activeshock.com:** co-founder of Active Shock Inc. a company formed to develop and market semi-active suspension systems and components. He has been involved in the off-road racing industry for over 15 years and has experience in all aspects of passive and semi-active suspension systems with extremely high forces and long strokes. His previous experience includes program manager for the HTMMP (Helo Transportable Multi Mission Platform), an off-road, 4WD technology test bed for the U.S. Marine Corps which set new standards in rough road performance by incorporating a long-travel, high capacity suspension system adapted from off-road racing vehicles and a light-weight, fuel efficient turbo-diesel engine. In addition he served as program manager for the JTEV (Joint Tactical Electric Vehicle) developed jointly with Aerovironment Inc. The JTEV featured a series hybrid electric drive system powered by a small turbo-diesel engine driving a 60 KW alternator. A 360 volt battery pack was used to store energy for the hybrid operation and also to provide a "silent running" electric only capability with a range of up to 15 miles. Four-wheel drive was achieved using one motor per axle driving a custom designed gear reduction and differential.

**Don Flynn Director of Business Development, Active Shock, Inc Manchester, NH dflynn@activeshock.com:** Graduated from the United States Military Academy with a concentration in Nuclear Physics and was commissioned as an Armor / Cavalry officer. Don served on active duty for 7 years in the US Army. He held a variety of leadership positions both domestically and overseas. After leaving the Army he has held a variety of leadership positions in Supply chain management through program management and business development with GE, BAE Systems and now Active Shock. He is certified Six Sigma Black Belt and holds an MBA from Union College.

## ABSTRACT

Problem outline: Adding armor to protect vehicle occupants leaves conventional suspensions incapable of delivering ride quality and handling needed for mission objectives.

The current up-armored HMWWV has exceeded the design weight for the M1113 variant by a considerable amount. The rated GVW is 11,500 lb, yet the mission ready weight often reaches 14,500 lb. This results in greatly reduced reliability and frequent breakage of suspension components. When the suspension is overloaded, the ride height is reduced and the vehicle is no longer capable of safely traversing rough terrain. In addition, the lack of available compression travel results in severe bottoming that transmits extremely high shock loads into the suspension, chassis and ultimately the occupants.

A semi-active suspension upgrade kit for the up-armored HMWWV was developed with the objective of achieving comparable performance for the weight range of 8300lbs curb weight to 14,500lbs combat loaded.

Objective: The performance target was equivalent handling, stability and ride quality as an M1113 HMMWV at rated GVW. The project developed a bolt on suspension retrofit kit for existing vehicles which is also compatible with new vehicles builds.

The kit maintained ride height throughout the range of predicted loads and provided sufficient damping force to achieve comparable handling and performance to the target. Occupant comfort was improved through a reduction in absorbed power over RMS terrain courses. Testing demonstrated improved stability and balance during slalom and increased speed during NATO lane change maneuvers. This translated into a reduced danger of rollover, better control during aggressive or emergency maneuvers and a 50% reduction in unscheduled maintenance costs.

The system has application for other vehicles that traverse rough roads or have large changes from curb weight to GVW.

Conclusion: A semi active suspension significantly improves vehicle mobility and reduces maintenance cost. Complete study provided below.

# UPGRADED SUSPENSIONS SYSTEMS IMPROVE MOBILITY AND SURVIVABILITY OF GROUND VEHICLES

## Project Objective

The intent of the effort is to develop a suspension upgrade kit for an up-armored HMWWV that will achieve comparable performance for the entire weight range of a GMV: 8300lbs curb weight to 14,500lbs combat loaded. The baseline performance comparison / requirement is the equivalent performance with respect to vehicle handling and stability as well as ride quality as exhibited by an OEM M1113 HMMWV at rated GVW in the test cases:

1. Over a standard RMS course measuring absorbed power.
2. Over half round obstacles of 4", 6", 8", 10", 12" measuring peak acceleration.
3. Through ISO lane change maneuvers and road-holding skid pad measuring lateral acceleration and vehicle roll.

At a minimum threshold, the proposed suspension system will allow a modified HMMWV to achieve the same off road mobility as a stock HMMWV while increasing its reliability and maintainability.

## Operational Requirements

At the beginning of the project, a summary requirements document was written to guide the design process where the main performance goals were defined as follows:

- Performance Comparable to Stock Setup at Stock GVW
    - But with Vehicle Weight of 8,300lbs to 14,500lbs
    - Over Standard RMS Road Course
    - Over 4", 6", 8", 10", and 12" ½ Round Obstacles
    - Handling Road Maneuver Courses
- Default Mode Comparable to Stock Setup
- Bolt On/Bolt Off Kit, Can return to Stock Setup

- Environmental
    - -45degC to 70degC Ambient
    - 36inch Water Depth
    - 10,000ft Altitude
- Maintenance
    - 15,000mile Threshold, 30,000mile Objective
    - Only Inspection and Standard Lubrication
- Power
    - 24Volt Input
    - Reasonable Protections for Electrical Faults
    - Draws <200watt Continuous and 800watts peak

## HMMWV Suspension Kit Description

The solution selected was a combination of components and subsystems that could be readily retrofitted to existing vehicles. The system components included semi active dampers, ride height control system along with replacement springs, brackets and associated hardware to improve suspension durability.

### Semi-Active Damper

The stock dampers are replaced with semi-active units possessing a much higher load and thermal capacity.

The dampers are conventional twin tube design but use an electronically controlled variable orifice valve to control damping force in real time. The valve mechanism contains an embedded voice coil linear actuator that augments a mechanical spring providing the base closing force of the pressure relief valve surrounding the orifice. This hybrid valve design has been custom engineered to meet specific bandwidth and other system dynamics parameters to be naturally stable and highly responsive (<10 millisecond transient response time) to a feedback control system using standard digital microprocessor based hardware and software components.

The second major element of the semi active damper is the digital control system and the algorithm that is designed to optimize ride quality, vehicle handling and control based on changing terrain conditions and transient events. The control system continuously samples incoming instrumentation data at a rate of up to 1,000 times per second from a position sensors embedded in the damper and accelerometers contained in the vehicle mounted corner electronic control unit to calculate position and velocity state information. The vehicle's steering column is also fitted with a Hall Effect sensor to monitor any command steering inputs from the driver that would potentially require a change in damping coefficient, e.g. a rapid cornering maneuver. By feeding these input state variable parameters into a control model with a multi-variable gain map and state table (Figure 1), the controller can determine whether or not a change in damping coefficient is required and commands the voice coil actuator to open or close the variable orifice valves in each damper thereby adjusting the damping force at each wheel.

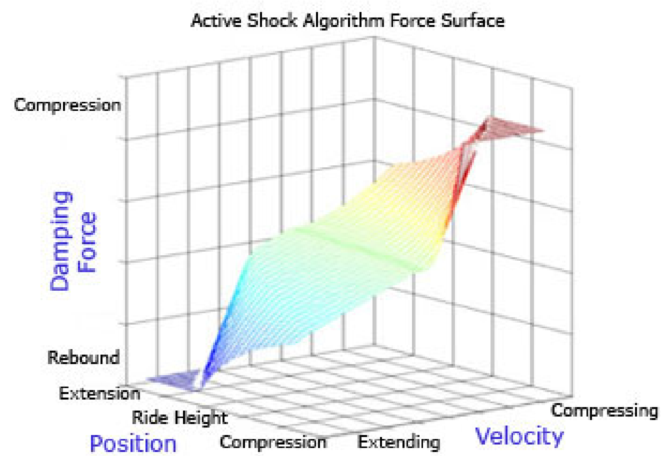


Figure 1

The software control system is comprised of a series of vehicle body dynamics control algorithms as well as individual wheel corner algorithms designed to collectively optimize the ride quality and handling of the vehicle, in a number of different driving modes ranging from normal highway driving to rugged off-road terrain. The system is also designed to recognize and instantly adjust damping for special transient situations including changes in vehicle weight, aerial jump landings, or aggressive cornering at high speeds. If the controller or actuators were to lose power or experience an unusual system failure or breakdown, the dampers are designed to revert into a passive fail-safe mode with a normal benign damping coefficient, which will enable the vehicle to be driven safely until it can be brought in from the field for maintenance.

A master controller board is mounted to the chassis which has an integral X and Y axis accelerometer and accepts inputs from the air spring pressure sensors and the steering angle sensor. This allows the algorithm to detect turning, braking and acceleration and send out high level signals to each corner to modify their behavior via a CAN bus. Each corner is self sufficient during high shock events with the primary goal being to prevent bottoming.

During lower amplitude handling events, the behavior of each corner is dictated by the master controller. The corners are commanded to work in unison to provide improved handling and stability.

**Ride Height Control System**

The ride height control system is designed to maintain the vehicle at a desired ride height independent of changes in vehicle payload and weight. Double convolute air springs are used at each corner and provide up to 2/3 of the total spring force at the nominal ride height when the vehicle is at the maximum weight of 14,500 lb. The front air spring is mounted on a self-aligning sliding strut that attaches to the upper a-arm. See Figure 2.

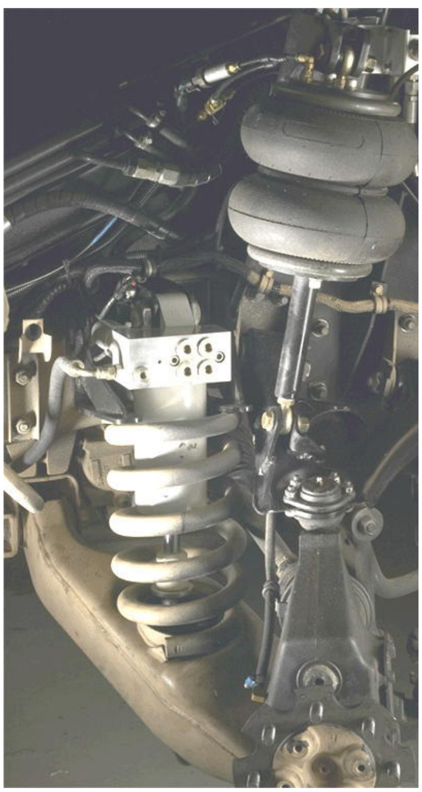

Figure 2 - Front Suspension

The strut is used primarily to ease packaging constraints; it allows the air spring to be moved up away from the upper a-arm to provide additional tire clearance. It also provides alignment of the airspring; the upper a-arm goes through a large angle change that exceeds the airspring's rated angular misalignment capacity. The strut assures that the upper and lower bead plates remain parallel while the spherical joint at each end of the strut provides angular misalignment capacity.

The rear air spring is mounted directly to a redesigned upper a-arm and the load is reacted by a new bracket that mounts to the existing a-arm bolt pattern. The rear A-arm geometry and lack of wheel steering movement allowed the simpler mounting scheme to be used without violating the air spring manufacturers design rules for spring travel or misalignment. See Figure 3

Figure 3 - Rear Suspension

The air spring load is automatically controlled to either maintain a predetermined ride height or allow the user to select from several preset driving modes, for instance low height for fast on-road driving or high ground clearance mode for off-road use. A dedicated controller receives position information from the Master and evaluates the data to determine when to add or release air to the air spring. The algorithms goal is to maintain the desired ride height despite changes to payload, but not to react to fast driving events that cause wheel displacement. Effectively the position information is very heavily filtered to determine the average wheel position which is then compared to the desired ride height. In addition there are a number of safety and fail safe elements to prevent the system from responding incorrectly in various situations.

The air is compressed using a small on-board DC electric air compressor. The air is then sent through a desiccant style air dryer to the air springs via a solenoid valve mounted in a dedicated valve block. When the air is vented from the spring, it is directed back through the air dryer in the opposite direction, thereby removing the moisture from the desiccant and ensuring that the desiccant does not become water saturated. The solenoid valves are actuated directly from the controller without the need for any intermediate relays or power switches.

The pressure in each air spring is also monitored by the damper system master controller and commands are sent to the damper corner controllers. The combination of pressure and ride height allow the corner controller to calculate the load supported by the air spring as well as the steel spring and thereby determine the total spring load at each corner of the vehicle.

Finally, the hydraulic bump and droop stops are replaced with strain rate dependent polyurethane stops. These have the benefit of extreme rising spring rates when they are compressed close to the limit of travel. Hydraulic bump stops are useful if properly designed but when the suspended load is increased to the degree presently found in the HMMWV, they no longer provide sufficient force and allow metal to metal contact when the suspension bottoms out.

Figure 4 shows a schematic layout of the major components described above. Each corner has a steel spring, air spring, semi-active damper, position sensor and accelerometer. The various control units, air spring, CAN and power buses as well as user interface options are also shown.

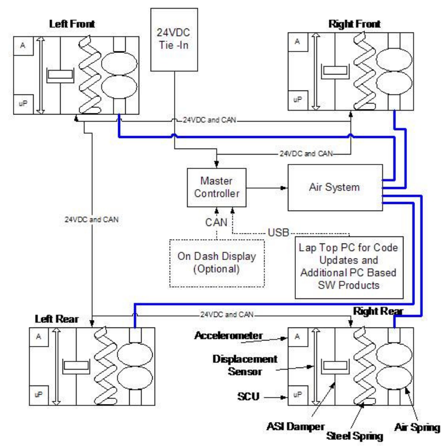


Figure 4 – System Schematic

**Backward Compatibility**

Throughout the design effort, backward compatibility and ease of installation were of paramount importance. The shock and spring brackets are all designed to work with either the new kit components or the original parts. If one semi active damper were to fail, that unit could be replaced with a stock unit. The other three could be left operational or replaced as the situation dictates. No further action would be needed. The only action required to disable the shock or air spring system would be to turn off power to the controllers. The air springs could also be disconnected from the automatic controller and manually inflated in the event of failure or damage to the main air system.

**Stress Analysis and Load Cases**

To accommodate the semi active dampers and air springs, additional chassis brackets needed to be designed and several of the stock vehicle brackets required a re-design. A basic finite element stress analysis was performed on the most highly stressed of these components. All of these brackets were designed with 4130 steel, heat treated to 135,000 psi yield strength condition.

## Performance Validation Testing

The HMMWV suspension kit was subjected to a broad array of lab and field tests by three separate organizations: in house by the manufacturer, at TACOM using a 4 post shaker table and at an independent field test facility. The results are summarized in the following sections and are broadly split into Design Validation and Performance/Durability Testing in both the lab and the field.

## Design Validation

### Field testing

The vehicle was ballasted with a combination of steel plates and 80 lb bags of concrete to allow variation of load in the field. The test vehicle is shown in figure 5

Figure 5 – Test Vehicle

The initial test performed was a simple curb drop where first the front end, then the rear end were driven up onto an 8 inch curb, then driven off onto flat ground. This allowed the behavior of the damper and the algorithmic responses to be examined and final tuning performed on the valve parameters to achieve target percent critically damped. The single event of dropping off a block is similar to the control theory method of exciting a system with a step response, then observing the return to equilibrium. The path the system takes including rise time and number of cycles to settle out can be used to empirically determine the percent of critical damping.

The spring rates of the air springs were also measured in the vehicle. The mounting geometry such as motion ratio and angular motion of an air spring can have a large impact on the final spring rate at the wheel. The vehicle was placed on instrumented weight plates and the air system was run through a series of pressures and weights to fully characterize the spring rate and preload relationship with air pressure and position of the wheel. This information was then combined with the steel spring rate and preload and is used by the corner controllers to measure the wheel load and calculate the natural frequency at each wheel in real time when the vehicle is running. These parameters are then used throughout the algorithm to calculate the desired damper force output for a given dynamic condition.

### On-road handling

On-road handling and stability testing were conducted on the unused apron of a local airport. Base line test were conducted on the vehicle at full combat weight fitted with stock suspension. The vehicle was subjected to lane change and slalom course test runs with the driver debriefed and the speeds recorded. Testing revealed the tires were overloaded resulting in up to 90 degrees of phase lag between driver inputs to the steering wheel and vehicle response. While it is possible to drive a vehicle with this sort of behavior in controlled situations such as a slalom or lane change course, the driver is forced to use feed-forward techniques and basically teach himself how and when to steer based on visual cues rather than from feeling the response of the vehicle. While suitable in a controlled test environment, the technique is not viable in emergency situations such as obstacle avoidance or typical urban driving.

The vehicle was then fitted with the suspension kit and subjected to the same test regime as the stock setup. As part of the system functionality, the semi active damper algorithm looks at the driver input via the steering wheel sensor and differentially stiffens each corner damper at the appropriate time in either compression or rebound to greatly reduce body roll and to help the driver balance the vehicle. The configuration variables that control this response were tuned to help control the transient response of the suspension springs and tires and create a reasonable and safe amount of understeer. The final configuration was capable of reducing the steering response phase lag from 90 degrees to approximately 20 degrees greatly improving driver control.

**Off-road ride quality**

Shock transmissibility was evaluated by measuring the peak vertical acceleration at the driver's seat when the vehicle was driven over rigid half-round obstacles of 4, 6, 8 and 10 in radius. This test was also very useful for tuning the algorithm since it exercises most of the ride quality features in a single controlled, easily repeatable event. The combination of the semi active damper and the air spring ride height control resulted in excellent performance over the full range of half –round heights. The air springs keep the vehicle at normal ride height at all payloads, plus the damper has sufficient force authority to effectively control bottoming on the larger sizes without being unduly stiff over the smaller events due to excessive compression damping. Once the wheel has passed over the half-round, the algorithm reduced the damping to allow it to fall away to full rebound in preparation for landing. On landing the compression end stop kicks in if needed to prevent bottoming then the rebound damping brings the vehicle back up to ride height at close to critical damping. The half round tests provided excellent feedback to the driver and observers on the efficacy of the algorithm.

**Rough road testing**

After performance testing was completed the vehicle equipped with the upgraded suspension kit was run in normal operating mode at 14,500 lb combat weight. The test course was as rough as possible given locally available test venues. A test loop composed of representative terrain features was defined and vehicle speed was increased until shock temperatures reached approximately 100 deg C or the driver felt that he was unable to safely operate the vehicle. Testing was nearly continuous with periodic inspection stops. If the driver or observer detected any faults, the vehicle was stopped and inspected. Repairs or redesigns were evaluated on a case by case basis with the goal of maximizing run time on all components.

In total over 1500 miles were run with approximately 60% at full speed over rough terrain. During this time many failures occurred, both minor and major, up to complete loss of the left rear wheel due to stub axle failure. In all cases the failure was diagnosed and either repaired or replaced with a redesigned component and testing continued.

## Performance/Durability Testing

### TACOM Testing

The vehicle was subjected to two rounds of testing on the TACOM 4 post shaker table. TACOM personnel conducted a full battery of tests to fully quantify the performance. This was followed by a simulated 1000 mile durability test using primarily Belgium Block road course data as the input stimulus. The following chart (Table 1) shows the test results for the upgraded suspension equipped vehicle as compared to a baseline HMMWV with stock suspension over a variety of simulated road courses and obstacles at the 10,200lb Curb Weight (CW) and 14,500 Gross Vehicle Weight (GVW).

Table 1- % Improvement over Stock

| Test Description | Curb weight 10,200 lb | Combat weight 14,500 lb |
|---|---|---|
| Belgium Block | 32% | n/a |
| Perryman II | 16% | 20% |
| #in step In phase | 54% | 30% |
| 1/2 rounds AVG | 65% | 54% |
| ASYM RMS | 51% | 16% |
| SYM RMS | 56% | 8% |

The test results were excellent for ride quality over asymmetric road course, ½ rounds and the symmetric road courses. In general the ride quality scores in terms of absorbed power measured at the driver's seat were much lower than the stock vehicle. When this is combined with the improved vehicle handling and stability it is clear that the suspension kit is performing as intended.

### Third Party Test Results

In addition to the in house testing and the TACOM shaker table testing, a full round of performance and durability tests were performed at an independent third party test facility. Ride quality testing was conducted that was similar in nature to the TACOM shaker table tests, however they were run on maintained test courses under realistic conditions.

### Ride Quality - Root Mean Square (RMS) Roughness Course

The vehicle was fully instrumented and data was gathered in accordance with a pre-described test plan to produce absorbed power values for each of the test conducted. The vehicle was tested on four RMS courses at both Curb Weight (CW) and Gross. Absorbed power values were gathered and plotted against vehicle speed and evaluated. The results are shown in Figure 6.

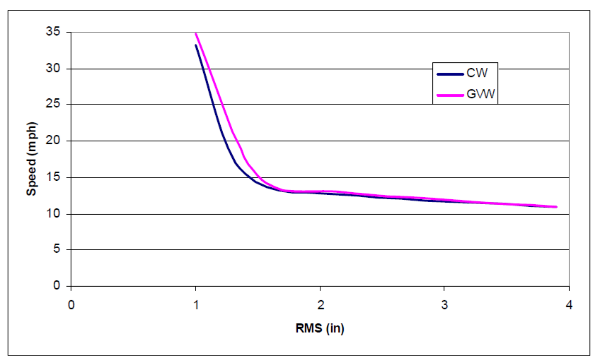


Figure 6 – Six Watt Speed vs. Course Roughness

Typical passive suspension systems there is compromise to be made between ride and handling (Dixon , 2007) In the case of the stock HMMWV suspension, it can be optimized to either ride well at CW or GW, but not both. The addition of the upgraded suspension system allowed the vehicle to achieve similar absorbed power scores regardless of the weight trim of the vehicle.

### Ride Quality -Half Round Testing

Table 2 shows the vehicle speed over a three differently sized half round obstacles. An example of the vehicle under test is can be seen in Figure 7.

Table 2
**2.5-g Speed Results**

| Vehicle Configuration | 2.5 *g* Speeds (mph) 6-inch Half-Round | 8-inch Half-Round | 10-inch Half-Round |
|---|---|---|---|
| CW | 15.2 | 13.1 | 8.6 |
| GVW | 15.3 | 13.8 | 11.6 |

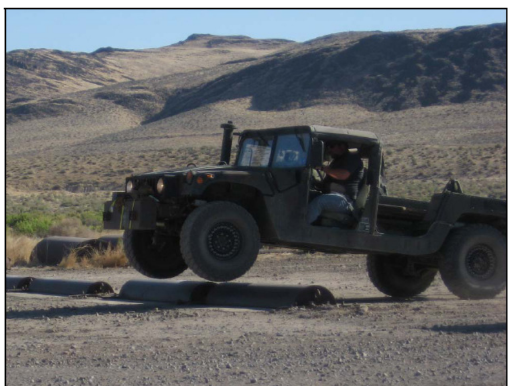

Figure 7 – Half Round Testing

The table shows the speed at which 2.5 g of vertical acceleration was recorded at the driver's seat location. As was demonstrated in the RMS testing, the vehicle was able to negotiate the obstacles at nearly identical speeds whether at CW or GVW. In addition the system reduced the number of damping cycles after the vehicle cleared the half round feature than would be observed on a vehicle with standard suspension.

### Performance Testing – Double Lane Change

The vehicle was driven through a NATO double lane change maneuver to ascertain the handling characteristics in an emergency maneuver at both CW and GVW. The goal was to determine the maximum speeds at which the vehicle could complete the maneuver without contacting the cones marking the outline of the course. Table 3 shows the results of the test.

Table 3
**NATO Double Lane Change Speeds by Vehicle Configuration**

| Payload Configuration | Speed (mph) | Max. Lateral Acceleration (*g*) | Max. Yaw Rate (deg/s) | Max. Roll Rate (deg/s) |
|---|---|---|---|---|
| CW | 57.4 | 0.67 | 12.7 | 7.5 |
| GVW | 54.9 | 0.58 | 15.3 | 8.2 |

Speeds through the course were remarkably similar as were the yaw, lateral acceleration and roll rates measured at the vehicles center of gravity.

### Durability Testing

The durability test phase was conducted over the typical percentages of on-road, off-road and cross-country terrain called out in the HMMWV ORD. A minimal instrumentation set was retained for this testing with the emphasis on consistent driving and documentation of miles driven and Test Incident Reports (TIRs). The test was conducted with the vehicle loaded to GVW.

The test vehicle completed 12,129 miles of durability testing with no suspension system or vehicle frame failures.

## Maintenance Cost Analysis

The standard HMMWV operating at 14,500lb GVW has potential for greatly accelerated suspension wear and decreased reliability when compared to a more lightly loaded vehicle. This decrease in reliability has a monetary loss in terms of replacement cost for failed parts, but more significantly, it severely restricts the availability of vehicles and the capability of the vehicle to meet the desired mission. Furthermore, missions that require the vehicle to be fielded for extended periods of time, away from a service depot, may be unfeasible. There are also stability issues with up armored HMMWV's that in certain conditions impair the vehicle's speed and maneuverability. This lack of stability has been attributed to a number of troop incidents resulting from vehicle rollover. The press has reported that some troops have resisted adding armor to their HMMWVs because of these problems.

## Maintenance Cost

It is difficult to quantify the total cost incurred due to the additional weight and resulting stress placed on the stock HMMWV. The data gathered for part replacements is inconsistent. In addition, the stock vehicle used as a control sample in the durability test program was unable to complete the entire 12,000 miles due to severe failures in the frame and structure at approximately 6000 miles. This negated a side by side comparison.

Additionally, the Army does not assign costs for labor, recovery, etc. One could look at the total loss cost of the truck as one measure. However, for the purposes of establishing a rough order of magnitude estimate, the following is a more detailed comparison of the most affected components and the amount of time consumed in addressing the replacement/repair. Table 4 shows an estimate of the parts and time consumed in maintaining the HMMWV suspension for 12,000 miles. The components listed will either be replaced as part of the upgraded HMMWV kit or are stock parts whose service life will be returned to normal replacement intervals as a result of the improved ride quality provided by the system.

Table 4 – Part Cost Estimate

| Component | Mean Miles to failure | Cost per repacement | Time per replacement | Cost per 12k miles | Time per 12k miles |
|---|---|---|---|---|---|
| Rear shock replacement | 2000 | $200.00 | 1.6 | $1,200.00 | 9.6 |
| Rear Spring replacement | 2000 | $250.00 | 2 | $1,500.00 | 12 |
| Front Shock replacement | 6000 | $200.00 | 1.6 | $400.00 | 3.2 |
| Front Springs | 6000 | $150.00 | 2 | $300.00 | 4 |
| Rear Lower Control Arms | 3000 | $600.00 | 5.2 | $2,400.00 | 20.8 |
| Rear half shafts | 12000 | $700.00 | 3.6 | $700.00 | 3.6 |
| Upper Ball Joints | 1000 | $120.00 | 1.2 | $1,440.00 | 14.4 |
| Lower Shock Mounts | 5000 | $60.00 | 1 | $144.00 | 2.4 |
| Front Crossmember | 5000 | $250.00 | 4.5 | $600.00 | 10.8 |
| Rear Crossmember | 5000 | $250.00 | 2 | $600.00 | 4.8 |
| Rear Spring Mount | 12000 | $200.00 | 2.5 | $200.00 | 2.5 |
| Front Tie Rod Ends | 3000 | $60.00 | 1 | $240.00 | 4 |
| Total | | | | **$9,724.00** | **92.1** |

## Price of the System vs. the Cost of Maintenance

While it is difficult to quantify the benefits of increased mobility, one can compare the cost of the upgraded suspension kit to the savings enjoyed by the reduction in premature component failure and associated vehicle down time. Table 5 shows a cost recovery analysis based on volume kit pricing and the maintenance costs shown in Table 4.

Table 5 – Cost Recovery

| Kit Volume | Kit Price | Maintenance Costs (12K miles) | Recovery Period (miles of service) |
|---|---|---|---|
| 1000 | $ 20,000 | $ 9,724 | 24,681 |
| 5000 | $ 18,000 | $ 9,724 | 22,213 |
| 10000 | $ 15,000 | $ 9,724 | 18,511 |

The upgraded suspension kit is designed with a target life of 30,000 miles. Key components such as seals, sensors, bushings, etc. have been specified and laboratory tested to confirm the life cycle. If the maintenance costs shown can be avoided due to the installation of the kit it is possible to calculate the number of miles necessary to recover the initial purchase price of the kit through reduced maintenance spending. In any of the purchase price scenarios shown in Table 4 the period in miles of service needed to recover the purchase price is less than the useful life of the kit. Therefore the operator would enjoy a net savings for any mile driven over the recovery period mileage.

While no dollar value is assumed for cost of labor and down time, there is clearly a benefit derived from equipment that is completing a mission vs. equipment that is down for repair. There is also down time associated with retrofitting vehicles with an upgraded suspension kit. Table 6 shows the projected net benefit in vehicle uptime resulting from fitting the upgrade kit over the projected life of the suspension kit.

Table 6 – Uptime Benefit

| Kit Installation Time (hrs) | Maintanence time 12K miles (hrs) | Projected Uptime recovered by Kit Installation (hrs) |
|---|---|---|
| 30 | 92.1 | 62.1 |

**Cost/Benefit of Mobility**

It would be impossible to quantify a monetary value for the additional stability and maneuverability of the kit. However, greater survivability, greater acceptance of the up armored HMMWV, and greater confidence in the performance of the up armored HMMWV will be an enormous benefit to our troops.

## ACKNOWLEDGEMENTS

The authors would like to acknowledge the US Special Operations Command / SOAL K and Mike Ellis for the SBIR funding and assistance with the development of the advanced suspension system.

## REFERENCES

John C. Dixon (2007). *The Shock Absorber Handbook Second Edition*, West Sussex England: John Wiley and Sons Ltd.