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Development of Advanced TMA Designs, Phase II

FINAL REPORT
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Submitted by

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16. Abstract

This report addresses a new design concept for a Truck Mounted Attenuator used for work zone safety, which was conceived to redesign the TMA to achieve a cost-effective and practical solution to current TMA problems. These problems are related to truck dump bed usage, extension length behind the truck, and electrical connections subject to corrosion.

In Phase I, a new TMA concept was proposed, which was based on the use of hydraulic shock absorbers (HSAs) to reduce the size and maintainability requirements as compared with existing systems, yet still meet NCHRP performance requirements.

Phase II of this project focused on the initial development of this HSA system, which consists of two large hydraulic shock absorber cylinders fixed to the truck chassis on one end, extending rearward to an impact strike plate on the other end, and supported laterally by utilizing the dump truck body for support and stiffening.

In order to verify the feasibility of this proposed design to achieve the cited objectives, the focus was placed on a quarter-scale HSA system. This scale TMA was designed, fabricated, and dynamically tested to a dynamically equivalent one-sixteenth energy absorption level. Vertical testing of the 2” bore x 24” long cylinder pair, with cylinder and chassis support structure, was performed with a 4,000 lb weight dropped at increasing heights up to 7 ft.

The tests were successful, and results have proven that this system has the potential to be scaled up to full size, and be used as a truck mounted attenuator for work zone safety.

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SUMMARY

This is the Final Report for the Phase II Study on “Development of Advanced TMA Designs,” which addresses a new design concept for a Truck Mounted Attenuator used for work zone safety.

This project focuses on the need to redesign the TMA to achieve a cost-effective and practical solution to current TMA problems. These problems are related to truck dump bed usage, extension length behind the truck, and electrical connections subject to corrosion.

Phase I of this project presented an extensive literature search of existing TMA designs and literature, and documented various other energy absorption and storage system concepts for possible use in a TMA system.

Phase I also presented a dynamic analysis and simulation, which showed that the minimum stopping distance required for a TMA is about seven ft., which can be obtained by maintaining a constant deceleration of 20g of the impacting vehicle. This is specified as one of the dynamic requirements for a TMA in NCHRP Report 350, Level 3.

Based on these studies, a new TMA concept was proposed which was based on the use of hydraulic shock absorbers (HSA’s). HSA’s can potentially reduce the size and maintainability requirements of TMAs as compared with existing systems, yet still meet NCHRP performance requirements.

Phase II of this project focused on the initial development of this HSA system. This system consists of two large hydraulic shock absorber cylinders fixed to the truck chassis on one end, extending rearward to an impact strike plate on the other end, and supported laterally by the dump truck body for support and stiffening.

In order to verify the feasibility of this proposed design to achieve the project’s objectives, the focus was placed on a scaled-down HSA system. A quarter-scale HSA TMA system was designed, fabricated, and dynamically tested to a dynamically equivalent one-sixteenth energy absorption level. Vertical testing of the 2” bore x 24” long cylinder pair, with cylinder and chassis support structure, was performed with a 4,000 lb weight dropped at increasing heights up to 7 ft.

The tests were successful, and results have proven that this system has the potential to be scaled up to full size, and be used as a truck mounted attenuator for work zone safety.

The focus of Phase III will be to develop a final design, fabricate and dynamically test a full-scale TMA system mounted on a truck, to NCHRP 350 Level 3 requirements.

INTRODUCTION

Phase I of this project defined the Truck Mounted Attenuator (TMA) problem, reviewed the state of the art, and proposed a new Hydraulic Shock Absorber (HSA) concept for the TMA.

Phase II developed the HSA concept further, built, and tested a scale model of the system, and proven feasibility of the concept.

This report summarizes the efforts of Phase II.

Research Problem and Background

A TMA is designed to protect both work site crew and the traveling public. The TMA is placed at the beginning of a work zone to shield workers from roadway traffic. Various TMAs have been available for sometime to improve roadway work zone safety.

The NJDOT's Maintenance Operations unit recently purchased new TMAs that are rated for 62.5 mph. These Scorpion units are pictured in Appendix A.

These new TMA units have multiple drawbacks. When not in use as a TMA, a portion of the unit is folded over the dump bed making it impossible to use the truck for other purposes.

When the unit is used as a TMA in the open position, the unit extends fourteen feet from the back of the truck. This length of extension, without articulation, makes for difficult vehicle maneuvering, especially near a vertical curb or on tight curves during moving operations.

On tight curves the TMA may not be properly aligned to the lane.

This unit also has electrical connections which are subject to corrosion.

The Phase I project reviewed various existing Truck Mounted Attenuator (TMA) designs and literature, and documented other energy absorption and storage system concepts for possible use in a TMA system. Possible modifications of current TMAs were explored for practicality, but there were no satisfactory alternatives.

Therefore, a new concept for a TMA system design was proposed, based on the use of hydraulic shock absorbers (HSAs). HSAs can potentially reduce the size and maintainability requirements of TMAs as compared with existing systems, yet still meet NCHRP 350 Level 3 performance requirements.

The scale-down development of the HSA system was recommended for Phase II of this program, to verify the feasibility of the proposed designs to achieve the cited objectives. These objectives are to develop a better TMA design which is modular, has a reduced length, and is affordable, all while rated for a 62.5 mph automobile impact.

Scope of Investigation

As shown in Appendix A, different TMAs have been approved for operation at different standards and levels, and there are some designed especially for the high impact energy of NCHRP 350 TL-3 requirements.

The goal of this project is to study the feasibility of using a relatively shorter, lighter, cost effective TMA with improved energy dissipation capabilities, instead of a longer, heavier, more expensive current TMA. The new design would have no major damage upon impact, and therefore require no replacement for an impact, no requirement for expensive maintenance, high reliability, and high absorbing impact energy capability.

The proposed new TMA design will be developed to achieve the objectives cited and the specifications described in NCHRP Report 350 Level 3, as follows:

- Dissipation of the collision energy of standard passenger vehicles traveling up to 62.5 mph (100 km/hr) with a weight of 4,410 lb (2,000 kg).
- Nominal occupant ride-down acceleration of 20g or less.
- Reduced extended length from the current 14 ft.
- Reduced maintenance cost.
- Modular design.
- Improved electrical connections.

Further, the desire is to be able to mount the new TMA on a standard dump truck, of the type shown in Figure 1.

Figure 1. Standard dump truck used for TMA support.

Previously, a dynamic analysis was performed to investigate the dynamic behavior of a TMA, which showed the theoretical possibility of reducing its length to approximately 6 ft. The minimum length of the TMA can be achieved with a device that can produce the constant force of 100,000 lb that corresponds to the energy needed to be dissipated to meet the above requirements.

Passive multiple orifice HSA units will be employed as the main impact absorption devices. HSA's have characteristics suitable for high speed shock absorption in terms of load capacity, dimensions, and shock force response.

A deployment/retraction system will be designed as well, so that HSAs can be stowed under the truck to significantly enhance their maneuverability during moving operations.

Further, the impact face of the TMA is to be designed for structural rigidity as well as envelopment capability. Envelopment capability allows the TMA face to partially conform to the impact vehicle, to minimize its tendency to move laterally or glance off the TMA into traffic.

This Phase II project included the following significant tasks:

- Preliminary design of a TMA based on an HSA concept.
- Analysis and simulation based on a mathematical model of the TMA.
- Design of a reduced-scale prototype HSA for impact testing, without deployment/ retraction system, for proof-of-concept purposes.
- Design of test fixture and setup for reduced-scale prototype HSA test.
- Fabrication of scale TMA, and fabrication of test fixture modifications.
- Impact testing of reduced-scale prototype.
- Measurement and analysis of test results.
- Plan for design of full-scale HSA-based TMA, including mounting system, deployment/retraction system, impact face design, and hydraulic/electrical system.

TMA HSA DESIGN

The proposed TMA concept is based on the use of hydraulic shock absorber (HSA) technology, as applied to large cylinders attached to a dump truck in a TMA configuration.

Shock Absorber Units

The multiple-orifice hydraulic shock absorber (HSA) used in this design is illustrated in Figure 2.

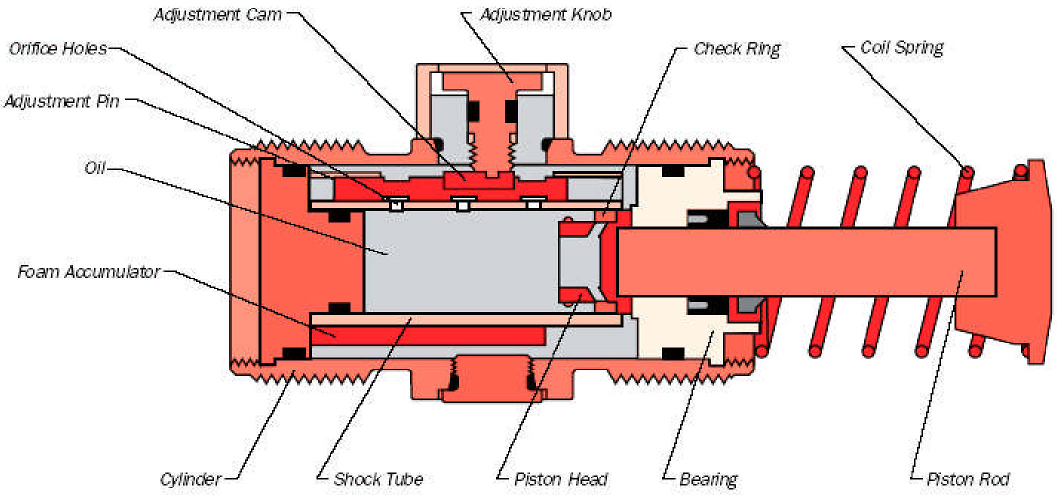

Figure 2. Multiple orifice shock absorber design. (Enidine Inc.)

The HSA is a passive device, which provides approximately a constant retarding during an impact, as seen in Figure 3. The stroke/total length of such absorbers is relatively short, and such devices are commonly used for dissipating large impact energy in a relatively small space.

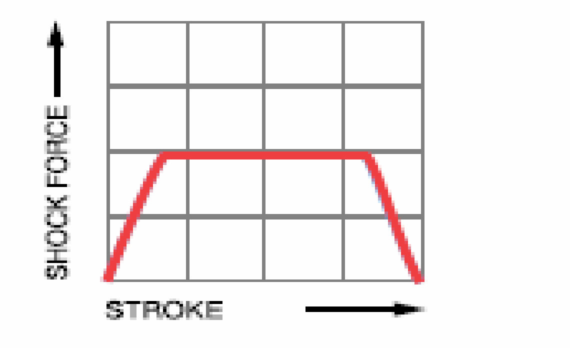

Figure 3. Graph of the absorbed shock force vs. stroke. (Enidine Inc.)

TMA Concept Development

TMA Concept 1

Initially, the idea was to use the HSA technology with cylinders extending rearward from the supporting truck, as illustrated in Figure 4.

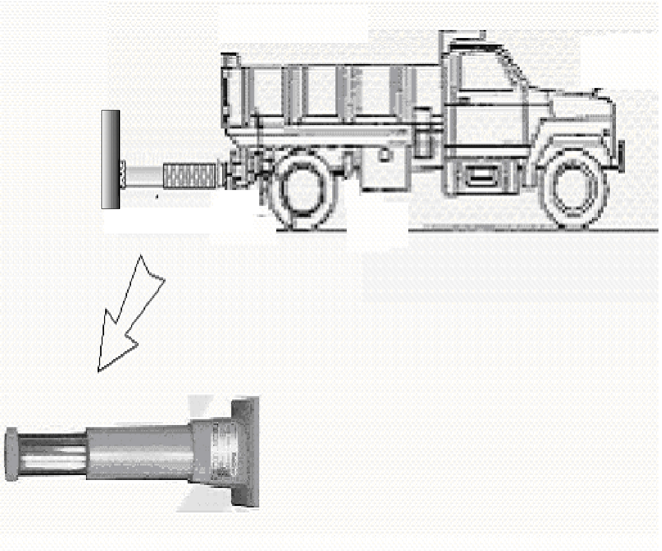

Figure 4. Early concept of shock absorber type of TMA.

Design issues were examined, including:

- Required length of HSA cylinder, with approximately 8 ft. stroke, to fit under truck chassis.
- Initial preference to keep cylinders in tension, not compression, to minimize non-axial loading.
- Lateral support of cylinders, to prevent cylinders from being subjected to bending through side loads, which they cannot resist without failure.

This resulted in the need to develop another concept.

TMA Concept 2

Enidine Inc. suggested another concept, in which the cylinders would be in tension, as seen in Figure 5.

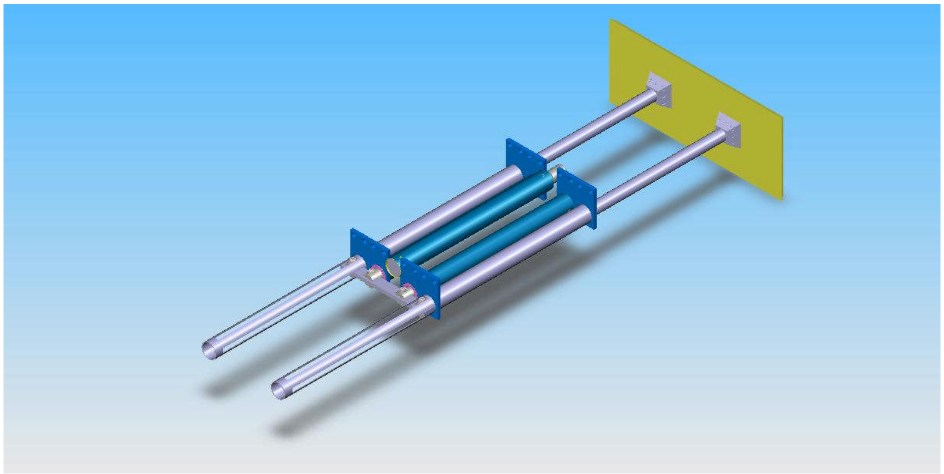

Figure 5. TMA concept with cylinders in tension. (Enidine Inc.)

Enidine developed an accompanying analysis for this shock absorber concept, which is shown in Appendix C.

However, this concept generated concerns that the overall length was too great for the relatively short supporting truck and dump body. Therefore, another design was developed.

TMA Concept 3

The third concept was a return to the cylinders in compression. However, issues related to lateral loading and overall support had to be addressed.

Duane Bennett of AHMCT conceived of a way to gain lateral support of the cylinders by using the dump truck bed. The truck bed would be detached from its pivots and actuating cylinder, and be permitted to slide axially rearward, constrained by a lateral guide structure on the primary chassis, as seen in Figure 6 a & b. The truck bed would still be supported by the main chassis structural beams.

Each HSA cylinder would be mounted to the forward part of the truck chassis, with each piston rod able to extend rearward and fixed to the strike plate, which in turn is laterally supported by the truck bed.

The main cylinders would be nominally 8 in. dia. bore x 8 ft. stroke. An additional 8 ft. stroke hydraulic cylinder would be supplied, which would be used to deploy the cylinders rearward when needed, or retract the cylinders when not needed, such as for highway driving. This auxiliary cylinder would be valved to be inactive when the main cylinders are active.

This is the concept used for the detailed scaled-down prototype, which was designed, built, and tested. Specifically, a one-quarter scale unit was detail designed by Duane Bennett, and oriented vertically for testing purposes.

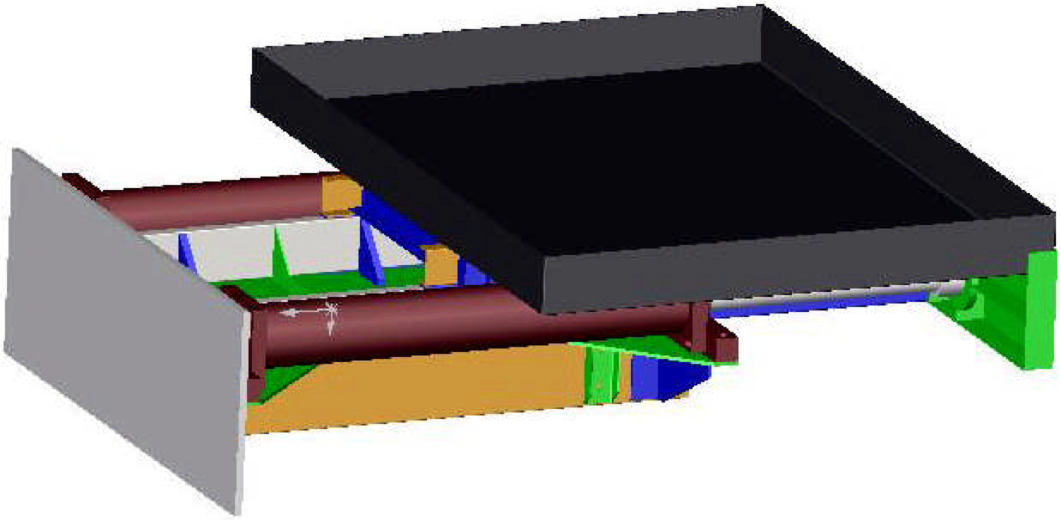

Figure 6a. TMA with cylinders in compression and truck bed support - upper (D. Bennett)

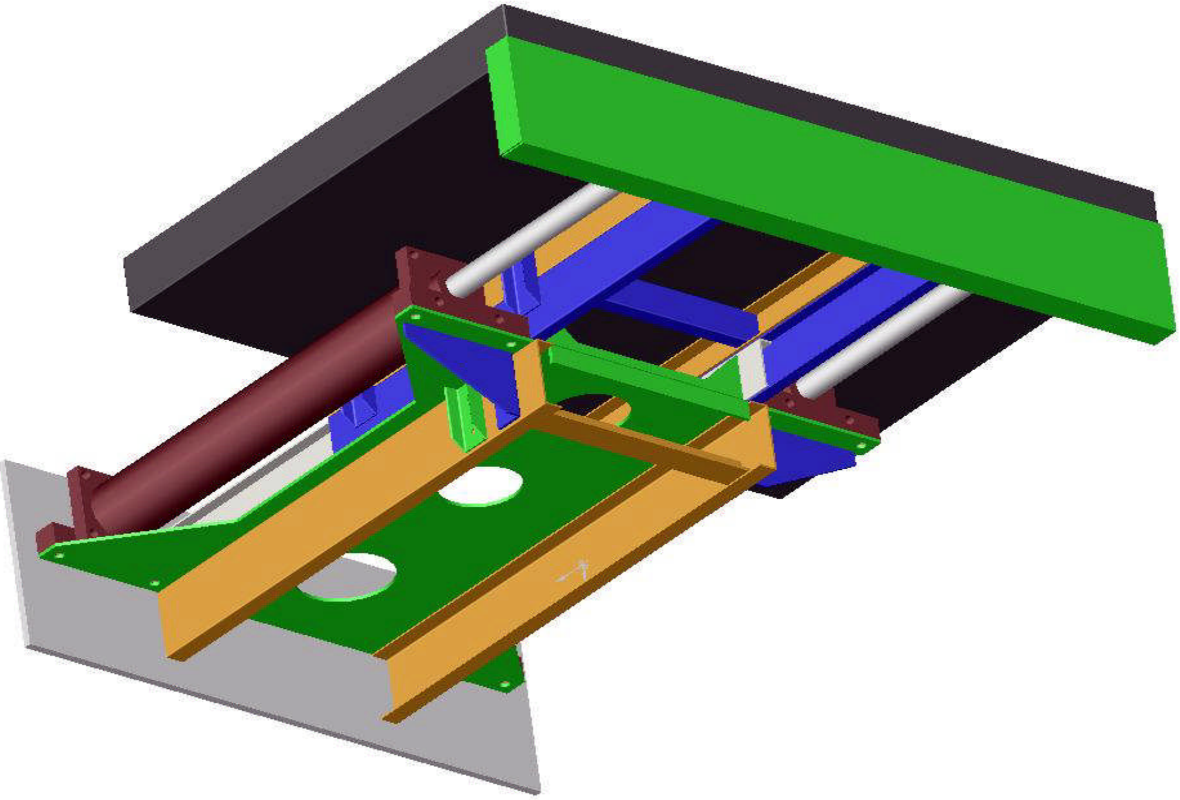

Figure 6b. TMA with cylinders in compression and truck bed support - lower (D. Bennett)

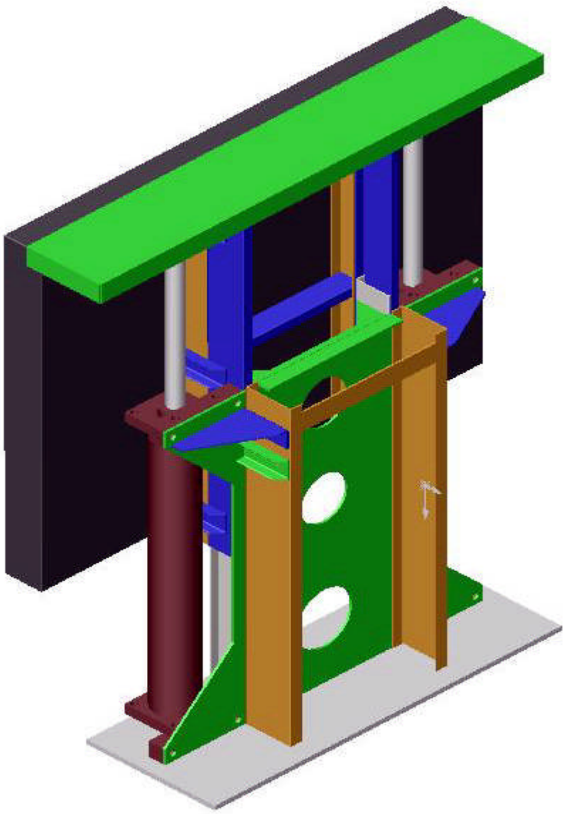

Figure 7. Vertical orientation for testing of quarter-scale TMA unit. (D. Bennett)

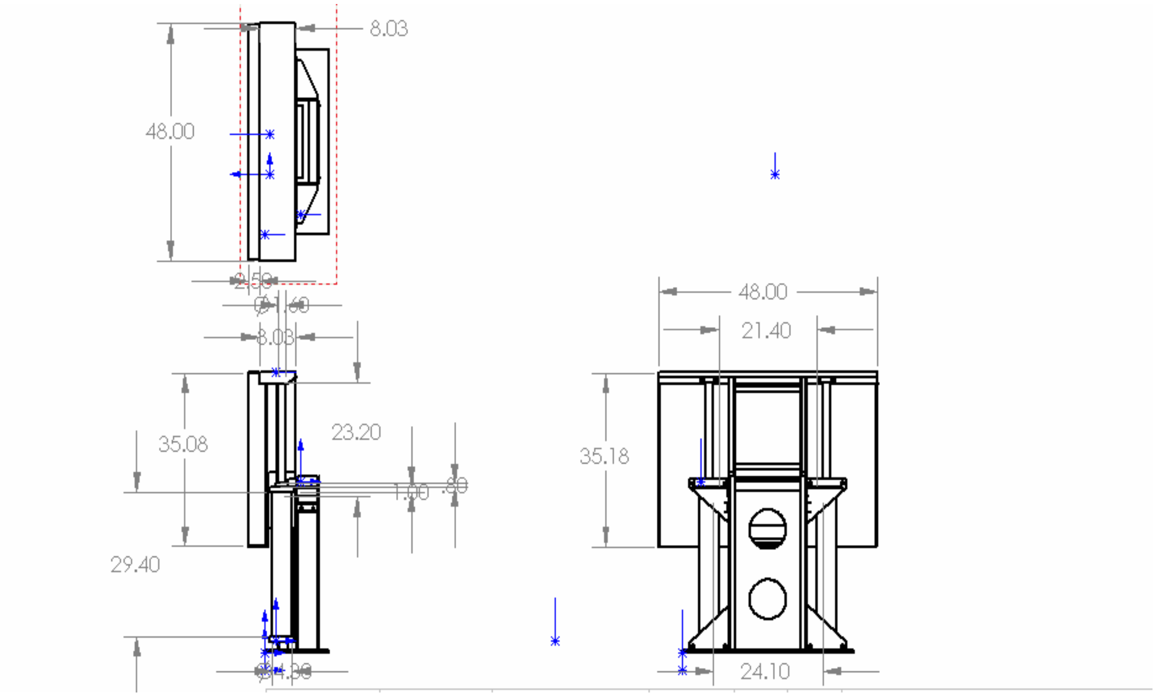

Figure 8. Drawing views of quarter-scale TMA unit for testing. (D. Bennett)

In Figures 7 and 8, shown in a vertical orientation, note the two HSA cylinder bodies and extended piston rods, the truck chassis, and the rectangular dump truck body used to support the strike plate and restrict the lateral motion of the cylinders.

This unit design was analyzed, to check for high stresses due to impact conditions.

ANALYSIS AND SIMULATION

Vehicle Crash/Impact Dynamics

Vehicle crash/impact mechanics were investigated in Phase I, to determine the the shortest stopping distance and impact force required to meet TMA design criteria.

NCHRP 350 Level 3 requirements, as noted in Appendix B, include:

- Initial impact velocity of vehicle 1,000 km/hr = 62.5 mph = 91.7 ft/sec.
- Impacting vehicle weight 2,000 kg = 4,410 lb.
- Supporting truck weight 9,000 +/- 450 kg =19,800 +/- 990 lb.
- Deceleration of the vehicle under an impact,
  less than 20g = 645 ft/s ^{2} = 0.216 km/hr^2.

Using conservation of energy equations, where the kinetic energy of the impacting vehicle equals the amount of energy absorbed by the shock absorber as it applies a constant force with a constant deceleration, this results in an ideal case in which:

- The minimum stopping distance = 5.70 ft.
- Constant Resisting Force = 88,429 lb.
- Total kinetic energy dissipated = 504,419 lb-ft.

The results show that, from a theoretical point of view, the shortest stopping distance is less than half of the extended length of current TMAs.

Of course, the ideal force and acceleration profiles cannot be achieved perfectly, but a TMA stroke length of about 8 feet should be feasible.

Dynamic analyses have been performed for the full scale impact model, as well as the quarter scale impact model. These are presented in Appendices D and E.

Analysis of Scaled-Down Design

Appendix F shows the analysis performed on the scaled-down design. The impact was mathematically modeled, and the resulting stresses and displacements determined. The net result was that, for the scale unit, there were no anticipated weak points or expected failure locations.

SCALE-MODEL DESIGN, FABRICATION, AND TEST SETUP

The quarter scale model assembly design and part detail drawings were developed by Duane Bennett. One such sample assembly drawing, showing both deployed and compressed positions, is shown in Figure 9. Note that this would be the top view, looking down on a truck body, but is erected vertically for impact testing.

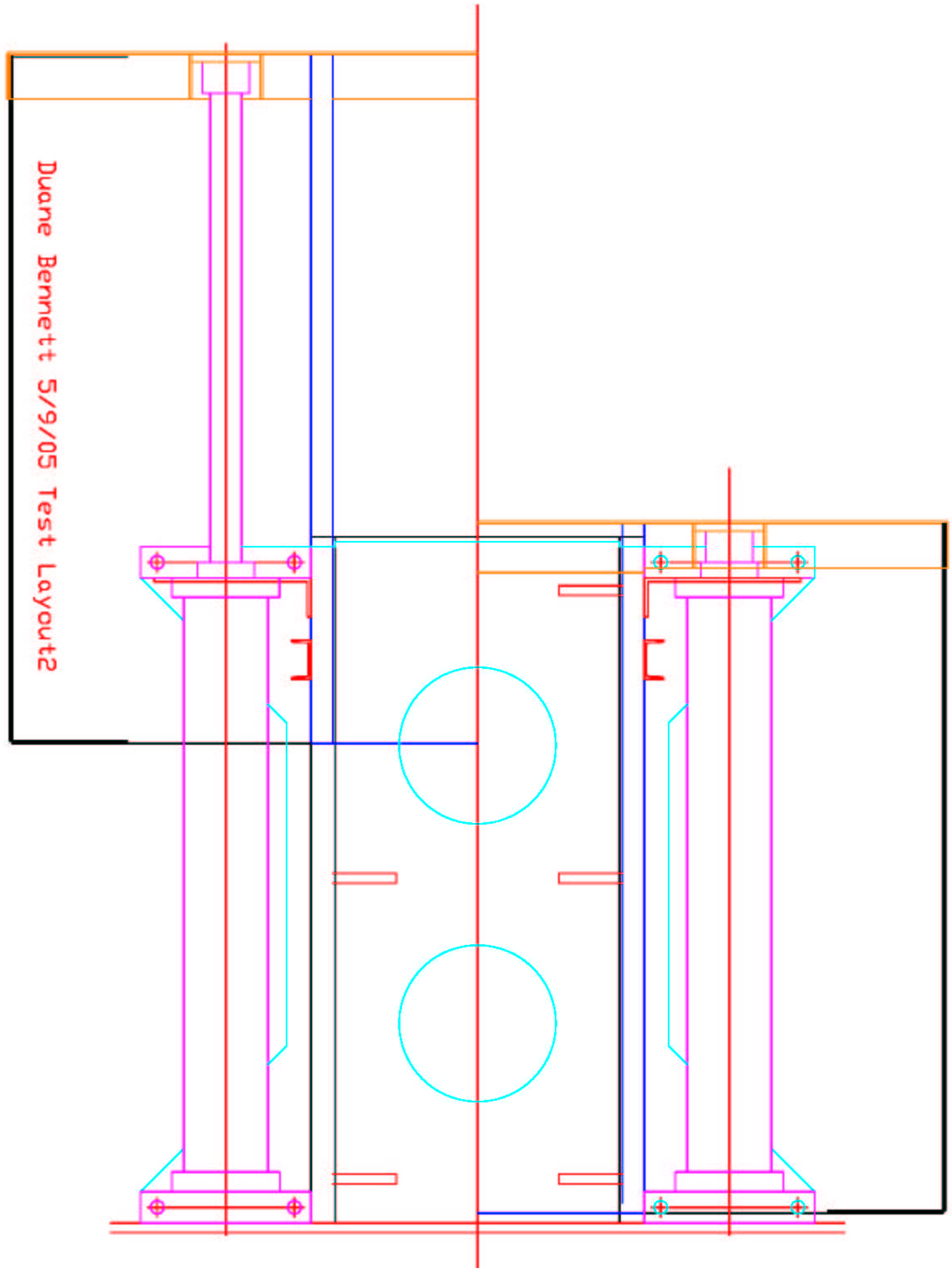

Figure 9. Quarter-scale TMA assembly drawing. (D. Bennett)

The two hydraulic shock absorber cylinders used for the quarter-scale tests were 2” bore x 24” stroke cylinders, manufacturing by Enidine Inc. A drawing of this cylinder is shown in Figure 10.

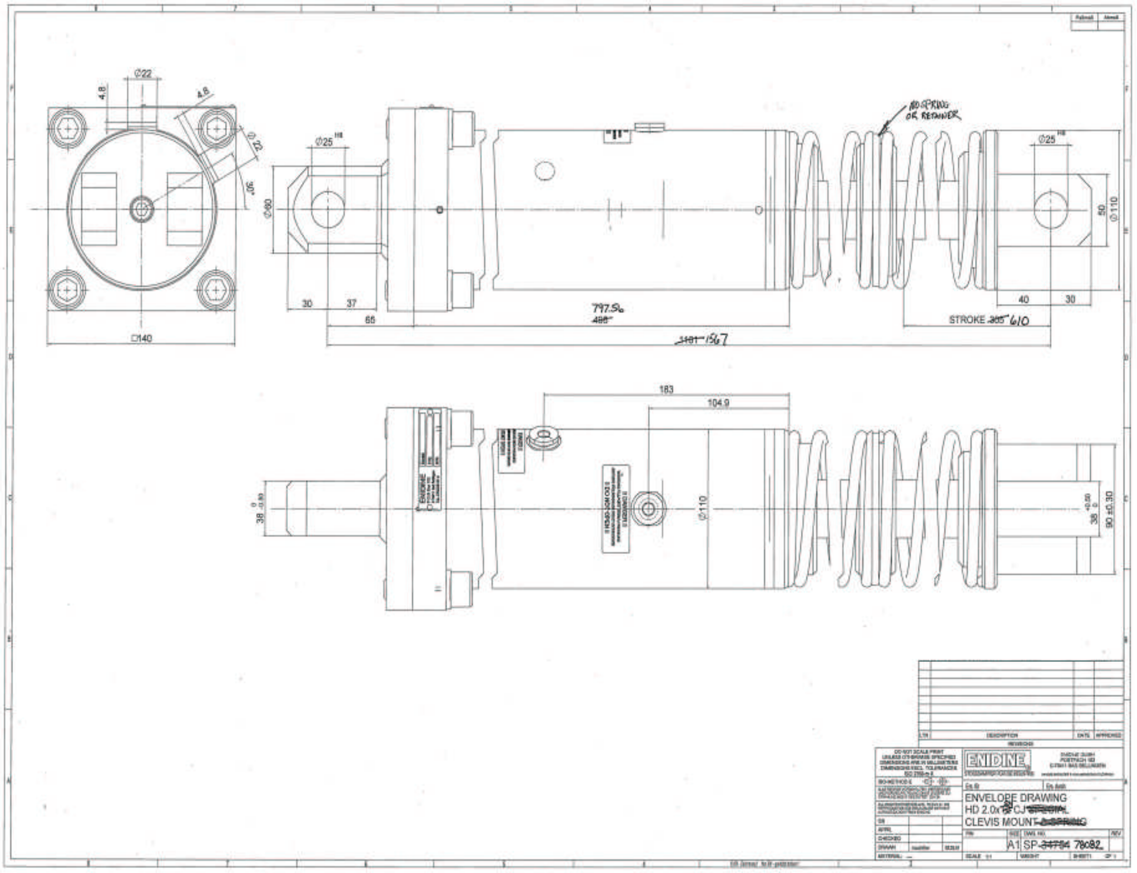

Figure 10. HSA cylinders used in scale test, 2 @ 2” bore x 24” stroke. (Enidine Inc.)

Based on the detail and assembly drawings, the quarter-scale test fixture components were fabricated and assembled, with the assistance of Enidine. The assembled TMA test unit, as installed in the drop test apparatus, is shown in Figure 11.

Figure 11. Fabricated TMA quarter-scale test unit, installed in drop test apparatus.

The drop test apparatus consists of a vertical rail and guided deadweight unit, which can be dropped from a predetermined height. This system is shown in Figure 12 a & b. The 4,000 lb deadweight, which is a steel box filled with lead weights and steel plates, and the quick release hook are shown in Figure 13 a & b.

This test will have a maximum drop height of 7 ft (84"), then the shock absorber compression stroke of 2 ft (24"). We will measure the force and velocity on impact of the 4,000 lb weight, and the displacement and velocity of the fixture strike plate during compression.

Figure 12 a&b. Drop test apparatus without a test unit, and with TMA installed. (Enidine)

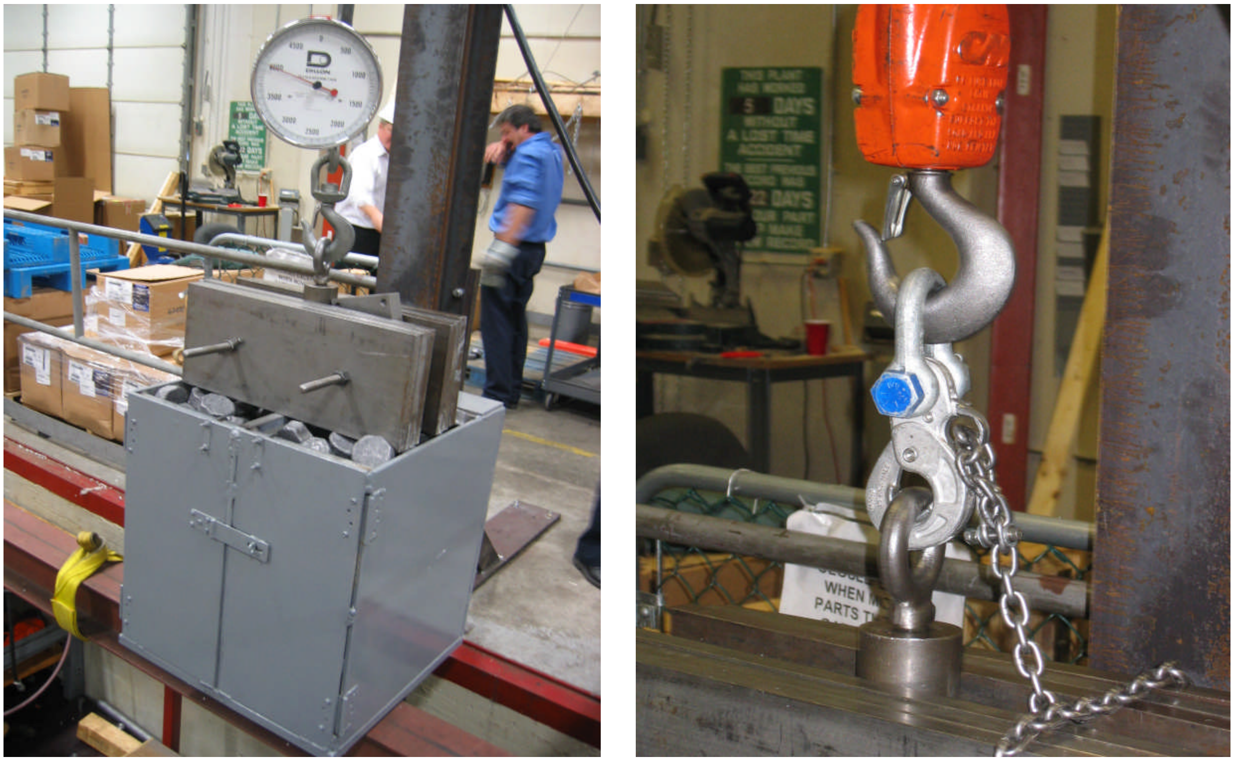

Figure 13 a&b. 4,000 lb drop weight and quick release hook (Enidine Inc.)

Data acquisition units and transducers normally used by Enidine for these drop tests are:

- Digitizing data acquisition from National Instruments Pct Plus 62.5 KHZ < 2lsb.
- Force gage from PCB Peizotronics mod. 214A, 0 to 40,000 lbs. ± 3% of reading. (The force gage is able to sense the high impact shock load accurately.)
- Displacement and Velocity data from a 30 inch retractable string pot. (Note: The use of a accelerometer was not recommended.)

The string pot is shown in Figure 14, and the data acquisition system is shown in Figure 15.

Figure 14. String pot set up to measure displacement and velocity. (Enidine Inc.)

Figure 15. Data acquisition system. (Enidine Inc.)

TESTING RESULTS

A series of drop tests were run using successively greater drop heights on the quarter-scale TMA. The energy was absorbed in all cases, such that the HSA design concept was proven feasible, at least at this scale.

Four primary tests were run, with 0.5 ft, 1.5 ft, 4 ft, and 7 ft drop heights:

Test Run no.	Weight (lb)	Drop Height (in.)
1	2,920	6
2	4,000	18
3	4,000	47
4	4,000	84

The results for the first three tests are shown in Appendix G, while those for the last is shown below. Figures 16 a & b show curves for both the test data and theoretical simulation. Note that the force curve at 7' or 84” did not represent a true progressive curve as desired, but the curve did represent a successful test.

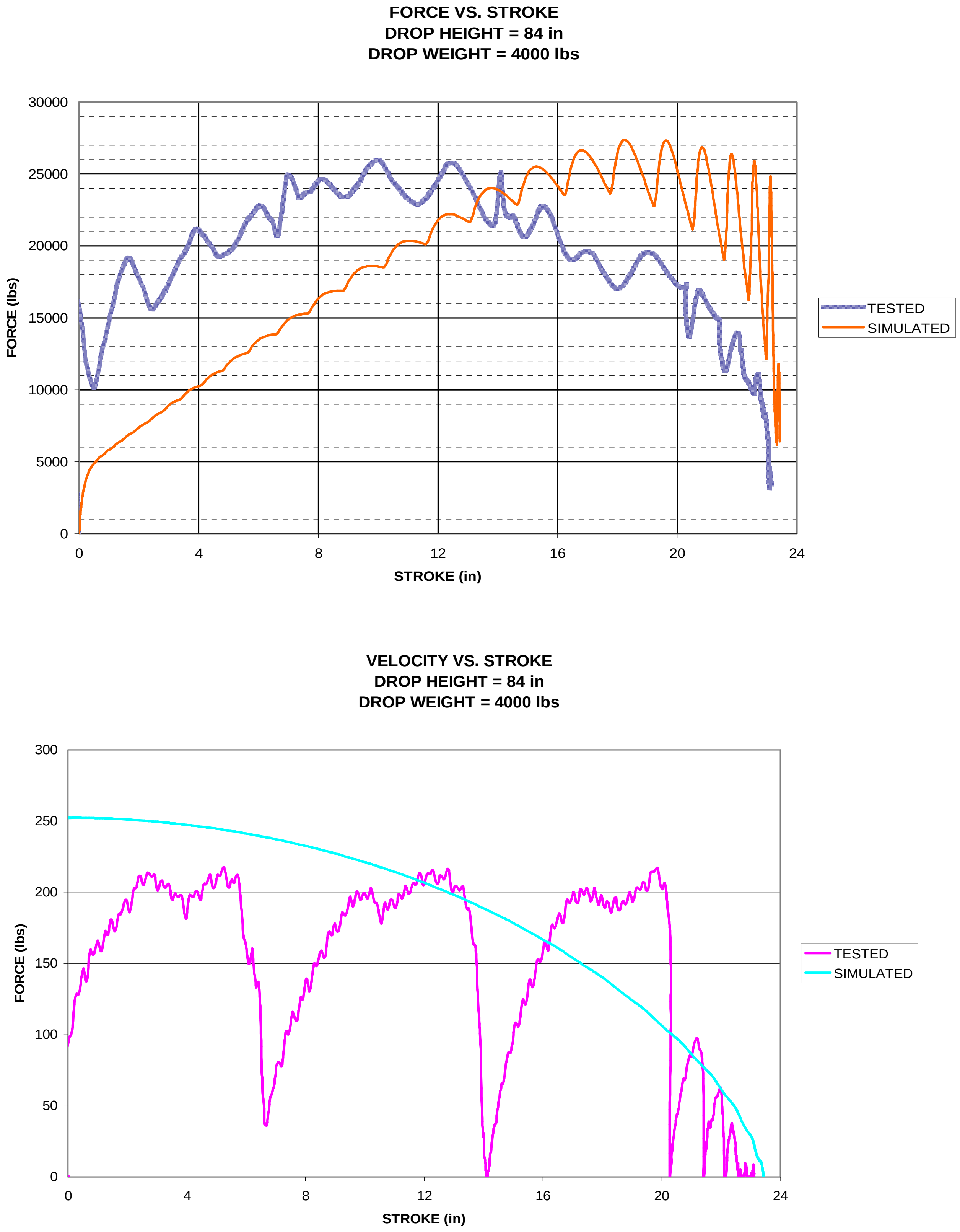

Figure 16 a&b. Test results for 84" drop test.

CONCLUSIONS

As discussed, the original concept design had to be modified to support and reinforce the HSA units as they were extended rearward. This was accomplished through the use of the dump truck body, which would also slide rearward on chassis guides and would structurally reinforce the cylinders against side loads.

This concept would require a dedicated truck for this use, and would not readily be convertible for use in other operations, such as salt spreading in winter months.

The proposed hydraulic shock absorber (HSA) based TMA system design retains most of the major advantages of other TMAs, while eliminating or reducing their disadvantages:

- The dump truck bed is not blocked, and is available for use in transporting materials (in a horizontal position).
- The deployed system extends rearward approximately 9 ft.
- The retracted cylinders remain under the truck bed.
- After the unit is in an impact situation, it is expected to be able to be reused without any major maintenance required. It is therefore “reusable,” without any need for costly spare parts or rebuilding necessary.

Beyond the design concept, dynamic testing of a quarter-scale model of the HSA system has been completed, demonstrating that this system exhibits:

- good mobility and stowage/extension for deployment,
- high energy dissipating capacity,
- maintaining less than 20g deceleration of an impacting vehicle during a collision,
- low maintenance, and
- no replacement after an impact.

Therefore, it is felt that the feasibility of this HSA concept has been demonstrated for potential use as a TMA.

REFERENCES

[1] Chung, J., Berkof, R., Cheung, Y. and Kim, C., 2003, “Alternate Designs of Truck Mounted Attenuators (TMAs)”, Research Report for NJDOT Project, Stevens Institute of Technology.

[2] Enidine Corporation, 2002-2003 “Enidine Adjustable Multiple Orifice Shock Absorber Design Data,” 7 Centre Drive, Orchard Park, NY 14127. http://www.enidine.com

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APPENDIX A CURRENT TRUCK MOUNTED ATTENUATORS

In this section, a brief compilation of current high-speed impact TMAs is documented.

First, a list of approved TMAs (from New York State Department of Transportation) is shown in the following table:

BRAND NAME	**MANUFACTURER LOCATION**	**NCHRP TESTING APPROVALS**		
		350 TL-3	**350 TL-2**	**230**
Alpha 60 MD	Energy Absorption Systems, Inc. Chicago, IL			Pass
Alpha 70K			Pass	
Alpha 1000				Pass
Alpha 100K		Pass		
Alpha 2001 MD				Pass
Hex Foam				Pass
Connecticut Crash Cushion	A.H. Harris & Sons, New Britain, CT		Pass	
Ren-Gard TMIA Model 815	Renco Highway Control Products, Phlugerville, TX		Pass	
SAFE-STOP^{TM}	Energy Absorption Systems, Inc., Chicago, IL	Pass		
Scorpion Model A	Traffix Devices Inc. San Clemente, CA		Pass	
Scorpion Model C		Pass		
Syro/Trinity Hexcel TMA	Syro Inc., Dallas, TX			Pass
Syro/Trinity MPS-350 III		Pass		
UMAD 70K	Traffic Maintenance Attenuators, Inc., Hatfield, PA		Pass	
UMAD 100K		Pass		

NCHRP 350 TL-3 TMA's are required whenever indicated in the Contract Documents or whenever the posted speed limit is lower, any of the devices listed above are approved for use.

All TMA's purchased new after October 1, 1998, must be approved as a NCHRP 350 TL-3 or TL-2 TMA.

Next, illustrations of the high-speed TMA units, which meet NCHRP 350 Level 3 requirements, follow. Such devices can sustain an impact at a maximum velocity of 100 km/hr (62.5 mph) and, during a collision, the ride down acceleration of the impacting vehicle is less than 20 g. (The supporting truck where Truck Mounted Attenuator is mounted weighs 9,000 +/- 450 kg. The testing impacting vehicle is 820 to 2000 kg.)

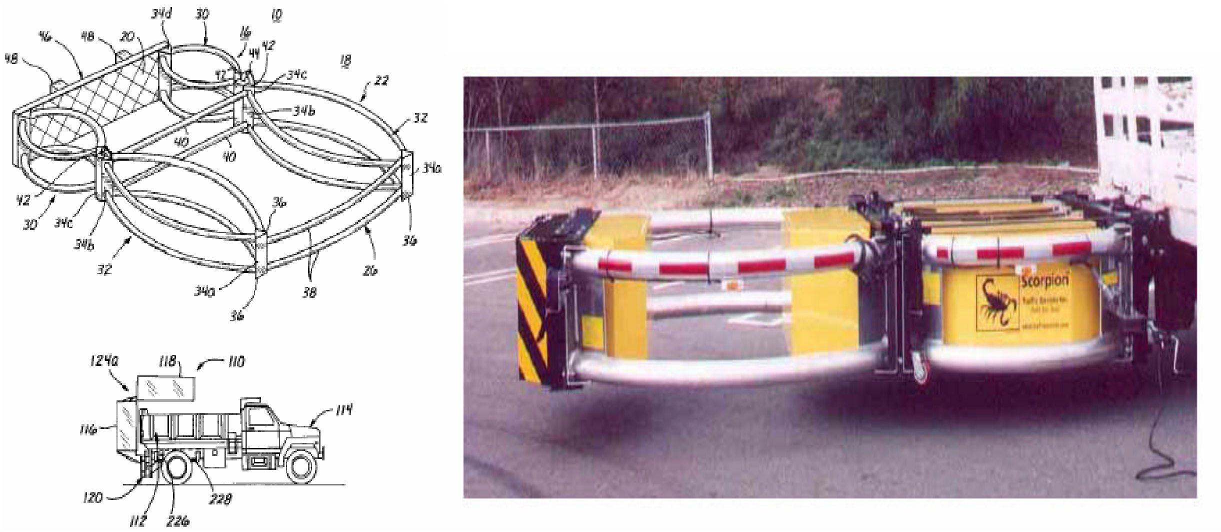

Figure 17 a&b. Scorpion TMA Model C. (Traffix Devices Inc.)

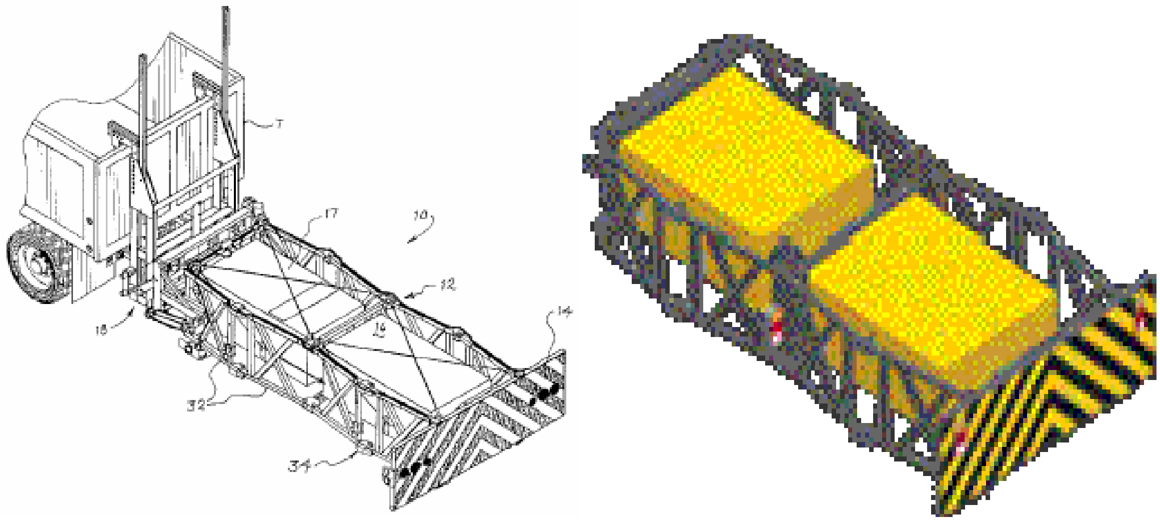

Figure 18 a&b. Safety Stop 180 ideal design and cartridge. (Energy Absorption Inc.)

Figure 19 a&b. U Mad 100K. (Traffic Maintenance Attenuators, Inc.)

Figure 20 a&b. MPS 350. (Syro Inc.)

Figure 21 a&b. Alpha 100K TMA. (Energy Absorption Systems Inc.)

APPENDIX B TEST SPECIFICATION OF NCHRP 350 LEVEL 3

The National Cooperative Highway Research Program (NCHRP) Report 350, Level 3, specifies the following test requirements for Truck Mounted Attenuators (TMAs):

- Test object maximum initial velocity is 100 km/hr (62.5 mph)
- During a collision, the ridedown acceleration of the object is less than 20g.
- The supporting truck to which the TMA is mounted weighs 9,000 +/- 450 kg (19,800 lbs)
- The impacting vehicle is 820 to 2000 kg (1,808 to 4,410 lb)

Other test specifications for TMA impact conditions include the following:

1. Vehicles with a mass of 820 kg (1,808 lb) impacting straight into the rear of the TMA at 1000 km/h (62 mph) shall remain upright with a theoretical occupant impact velocity of 12 m/s (39 fps) or less and the nominal occupant ridedown acceleration of 20g's or less per NCHRP 350, Test 3-50 evaluation criteria. The front of the truck shall be restricted from forward movement by positioning it against a solid wall or concrete block for this test.

2. Vehicles with a mass of 2000 kg (4,410 lb) impacting straight into the rear of the TMA at 1000 km/h (62 mph) shall remain upright with a theoretical occupant impact velocity of 12 m/s (39 fps) or less and the nominal occupant ridedown acceleration of 20g's or less per NCHRP 350, Test 3-51 evaluation criteria.

3. Vehicles with a mass of 2000 kg (4,410 lb) impacting straight into the rear of the TMA at 1000 km/h (62 mph) and an offset of W/3 with respect to the TMA centerline, shall remain upright with no significant roll, pitch, or yaw per NCHRP 350, optional Test 3-52 evaluation criteria.

4. Vehicles with a mass of 2000 kg (4,410 lb) impacting at 10 degrees into the rear of the TMA at 1000 km/h (62 mph) and an offset of W/4 at an angle of 10 degrees with respect to the TMA centerline, shall remain upright with a theoretical occupant impact velocity of 12 m/s (39 fps) or less and the nominal occupant ridedown acceleration of 20g's or less per NCHRP 350, optional Test 3-53 evaluation criteria.

5. To minimize potential damage to the truck, no portion of the TMA's structure or energy absorbing elements shall protrude forward of the underride during an impact.

In the absence of a common support vehicle, it is recommended that TMA tests be conducted with a support vehicle having a test inertial mass of 9000 +/- 450 kg. There are other criteria for basic design of impact cartridges, their supporting structure including struts, etc.

APPENDIX C ENIDINE APPLICATION ANALYSIS OF PRELIMINARY TMA SYSTEM

ENIDINE INCORPORATED

Application Analysis

For

Stevens Institute of Technology

TMA

Truck Mounted Attenuator

Enidine Model Number

HD 6.0M x 90

7 Centre Drive, Orchard Park, New York 14127 Phone: (716) 662-1900 • Fax: (716) 662-1909 Technical Hot Line: 1-800-852-8508 http: www.enidine.com	CUSTOMER: Stevens Institute of Technology	DATE: 06/17/2004
	PREPARED BY: Sean M. France	
	APPROVED BY:	

Report Content

Document Name	Description
Engineering Application Analysis	Description of application with recommendations
Concept Envelope Drawing	Dimensional data for proposed concept
Enidine Simulations	Shock force, G-Load, and Velocity curves

Engineering Application Analysis

Application Description:

Energy Absorption device mounted on a highway construction vehicle designed to FHA TL3 rating, reusable after impact and retractable during nonuse.

Enidine’s understanding of FHA TL3 rating :

Vehicle load range 1800 – 4400 Lbs.
Velocity max is 62.5 MPH
Maximum allowable G load is 20g

Calculated Results Per System:

	Max Condition	Min Condition
Total Energy	Et = 6,896,373 in-lbs	Et = 2,821,244 in-lbs
Shock Force	Fp = 90,149 lbs.	Fp = 36,879 lbs.
G-Load	G = 20.5	G = 20.5

Simulated Results Per Shock:

	Max Condition	Min Condition
Total Energy	Et = 3,448,187 in-lbs	Et = 1,4101,622 in-lbs
Shock Force	Fp = 45,490 lbs.	Fp = 17,849 lbs.
System G-Load	G = 18	G = 15

(Please see included simulations)

Recommendations:

The following recommendations are based on the calculations and simulations performed using the defined application parameters:

1. Two HD 6.0M x 90 model shock absorbers will satisfy the application requirements. These shocks must be used in extension mode. The strain energy exceeds the capacity of a compression mode.

Conclusions:

Enidine does not have all of the required data to propose a conclusion. The dimensional data of the construction vehicle is required to confirm whether the HD 6.0M x 90 will fit. Please see the attached envelope drawing.

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Phone: (716) 662-1900 • Fax: (716) 662-1909
Technical Hot Line: 1-800-852-8508
http: www.enidine.com

CUSTOMER:

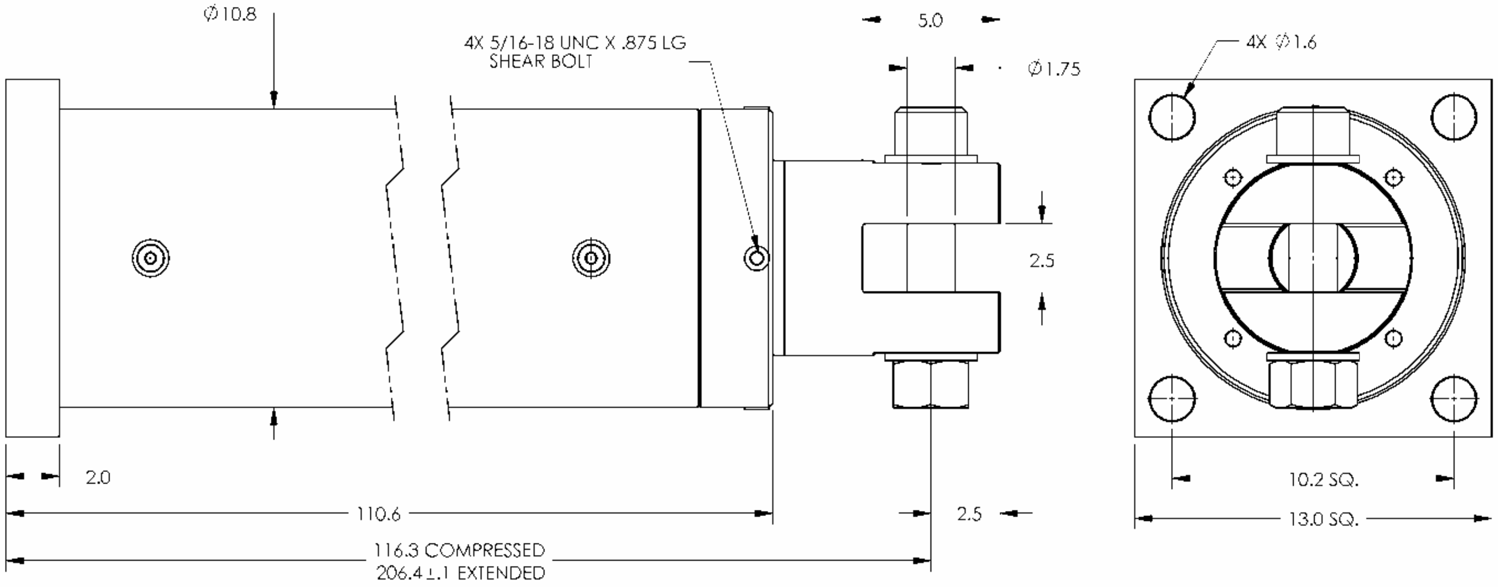
DATE: 05/20/04

PREPARED BY: Sean M. France

APPROVED BY: Steve M. Smolkovich

REVISIONS

REV	DESCRIPTION	DATE	APPROVED



3. DESIGN PARAMETERS:
   - NUMBER OF SHOCKS IN SYSTEM: 2
   - IMPACT WEIGHT: 1,800-4,400 LBS
   - IMPACT VELOCITY: 62.5 MPH
   - SHOCK ORIENTATION: PARALLEL TO VEHICLE
   - DIRECTION OF SHOCK TRAVEL: HORIZONTAL

2. MATERIALS AND FINISHES:
   - EXTERNAL SURFACES: SILICONE SEALED AND PAINTED STEEL
     - COLOR: TBD
   - PISTON ROD: CHROME PLATED STEEL

1. ALL DIMENSIONS ARE NOMINAL.

NOTES: UNLESS OTHERWISE SPECIFIED.

CAD GENERATED DRAWING,
DO NOT MANUALLY UPDATE

PART NUMBER: TBD
REV.: -

DO NOT SCALE PRINT
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
REMOVE ALL BURRS
INTERPRET DIM AND TOL PER ASME Y14.5M - 1994
LIMITS ON FINISHED DIMENSIONS:
TOL ON ANGLE ± -- 2PL. ± --- 3PL. ± ---

NOTICE: THE INFORMATION CONTAINED IN THIS DRAWING IS PROPRIETARY TO ENIDINE AND SHALL NOT BE DISCLOSED IN WHOLE OR IN PART, OR USED FOR ANY DESIGN OR MANUFACTURE, EXCEPT WHEN SUCH USER POSSESSES DIRECT WRITTEN AUTHORIZATION FROM ENIDINE.

THIRD ANGLE PROJECTION

APPROVED		
CHECKED	S. France	06/03/2004
DRAWN	C. Jackson	05/17/2004

MATERIAL
SEE NOTE 2

ENIDINE
An IMC Company

ENIDINE INCORPORATED
7 CENTRE DRIVE
ORCHARD PARK, NEW YORK 14127
PHONE: (716) 662-1900

ENVELOPE DRAWING
TMA

SIZE	CAGE CODE	DWG NO	REV
B	24403	TBD	.0.0

SCALE: 1:4 | WEIGHT: 900 LBS | SHEET 1 OF 1

CADFILE: TMA.SLDDRW

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Orchard Park, New York 14127 USA
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Telefon: +49-7635-81 01-0
Telefax: +49-7635-81 01-99
http://www.enidine.de

ENIDINE SHOCK ABSORBER PERFORMANCE SIMULATION

This sheet was prepared on 6/17/2004 at 12:40:40 PM

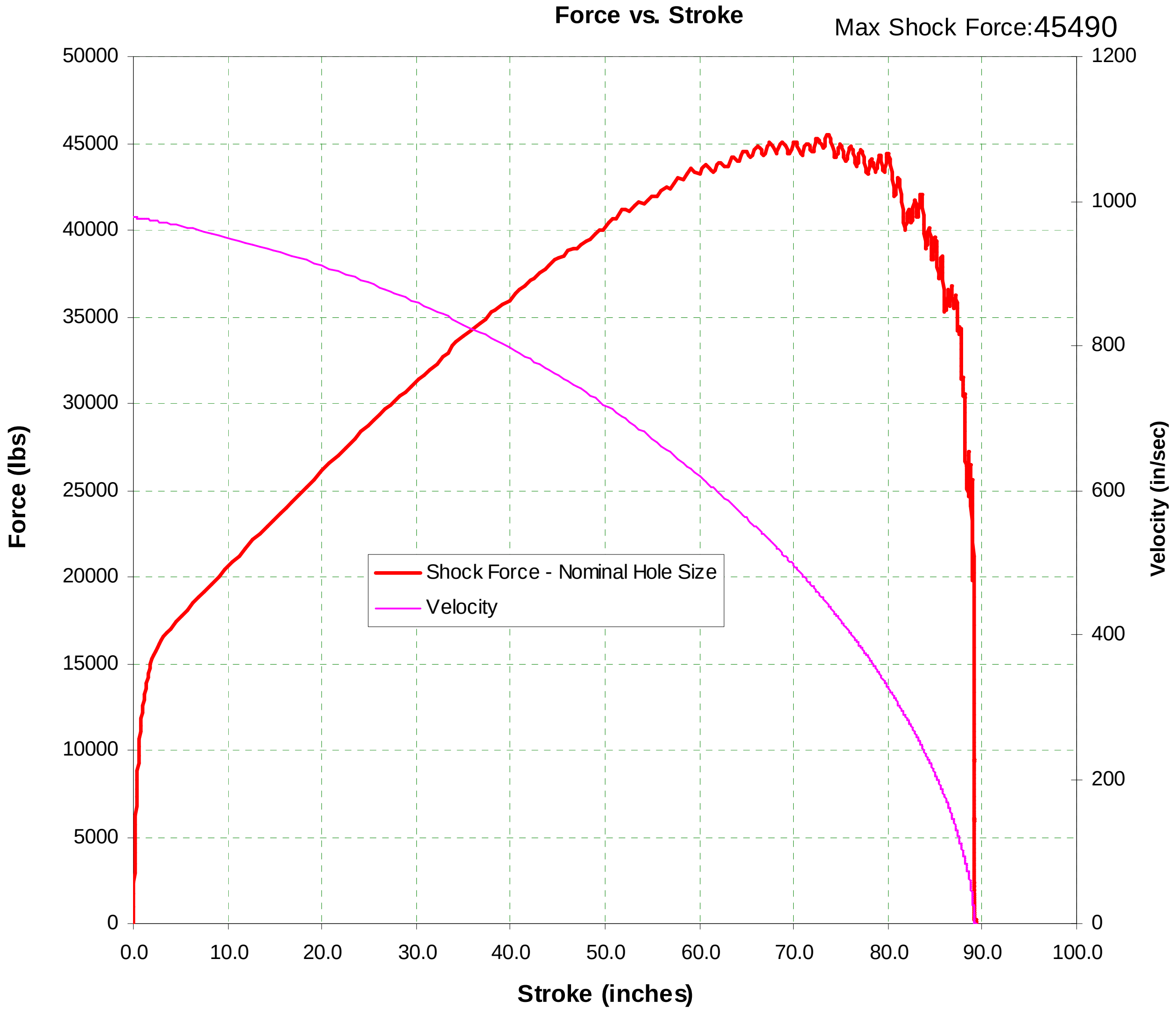
Enidine Model	HD 6.0M X 90
Enidine Part Number	TMA

Input Data Per Shock Absorber:

Impacting Weight [lbs]	2200.0
Impacting Velocity [in/sec]	1100.0
Propelling Force [lbs]	0.0

Technical Notes

2 UNITS IN TENSION 90" STROKE
4400Lbs @ 62.5 Mph



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Enidine Model HD 6.0M X 90

Enidine Part Number TMA

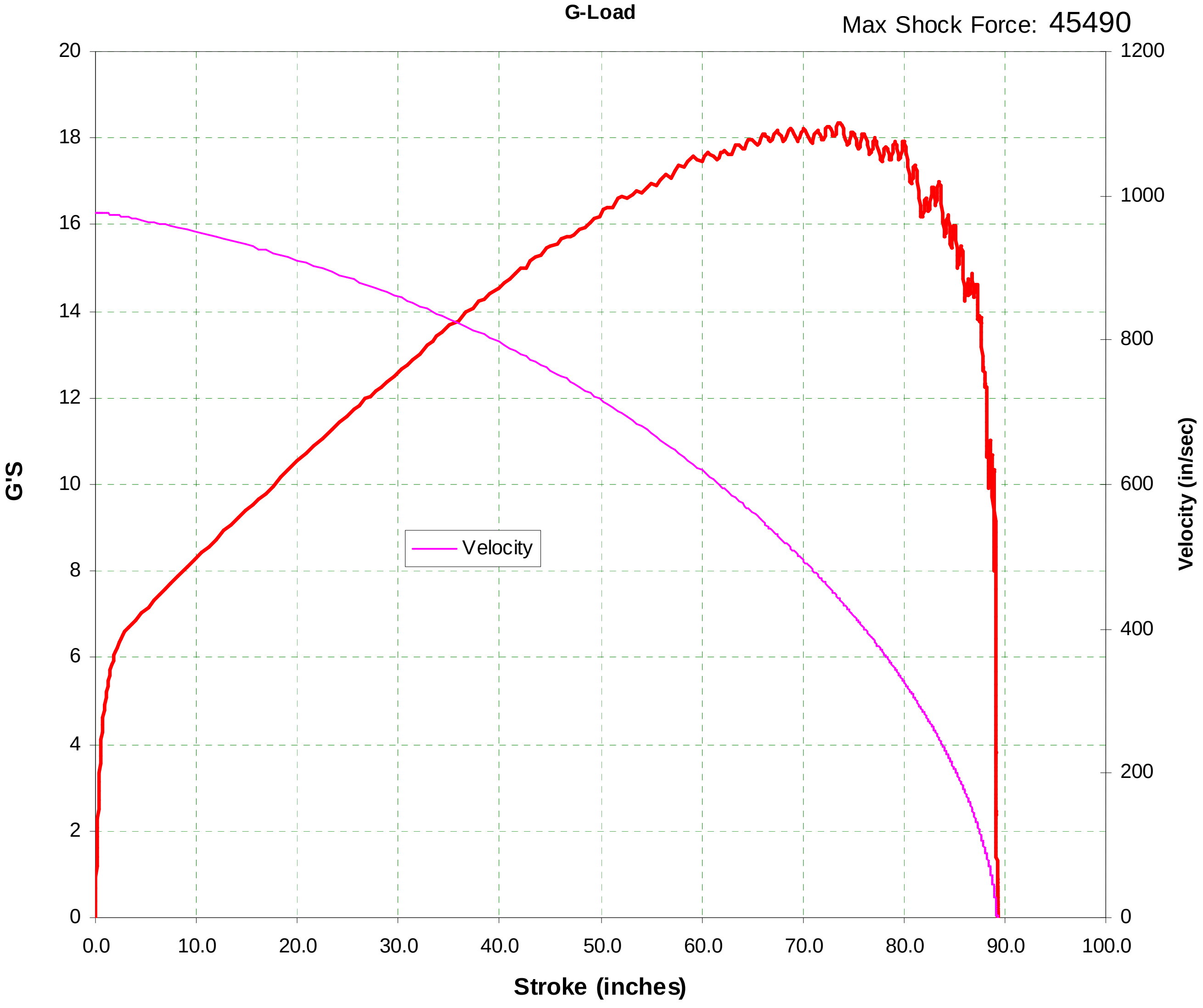
Input Data Per Shock Absorber:

Impacting Weight [lbs]	2200.0
Impacting Velocity [in/sec]	1100.0
Propelling Force [lbs]	0.0

Technical Notes

2 UNITS IN TENSION 90" STROKE
4400Lbs @ 62.5 Mph

Max Shock Force: 45490



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ENIDINE SHOCK ABSORBER PERFORMANCE SIMULATION

This sheet was prepared on 6/17/2004 at 12:43:05 PM

Enidine Model	HD 6.0M X 90
Enidine Part Number	TMA

Input Data Per Shock Absorber:

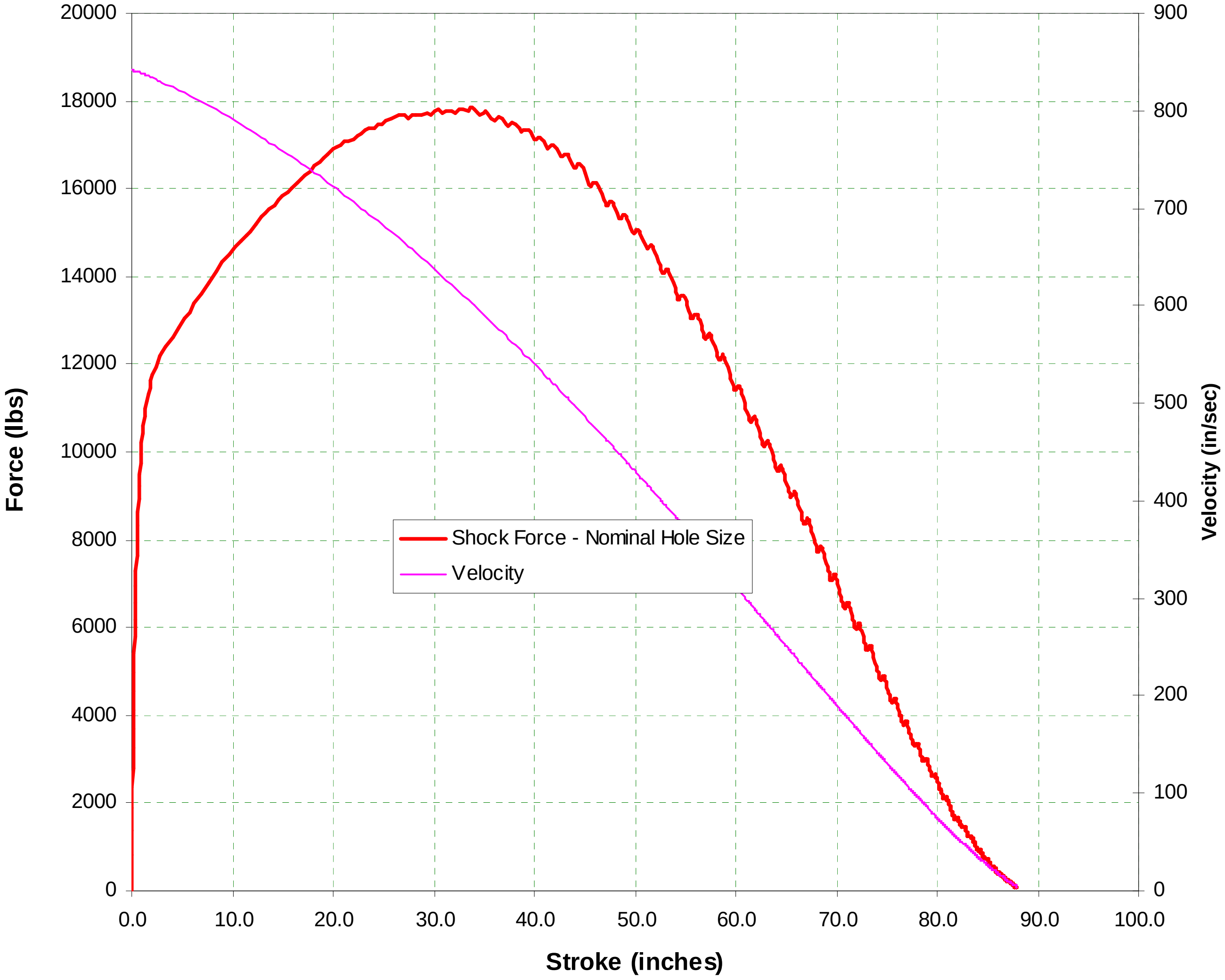
Impacting Weight [lbs]	900.0
Impacting Velocity [in/sec]	1100.0
Propelling Force [lbs]	0.0

Technical Notes

2 UNITS IN TENSION 90" STROKE
1800Lbs. @ 62.5 Mph

Force vs. Stroke

Max Shock Force: 17849



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ENIDINE SHOCK ABSORBER PERFORMANCE SIMULATION

This sheet was prepared on 6/17/2004 at 12:43:05 PM

Enidine Model HD 6.0M X 90

Enidine Part Number TMA

Input Data Per Shock Absorber:

Impacting Weight [lbs] 900.0

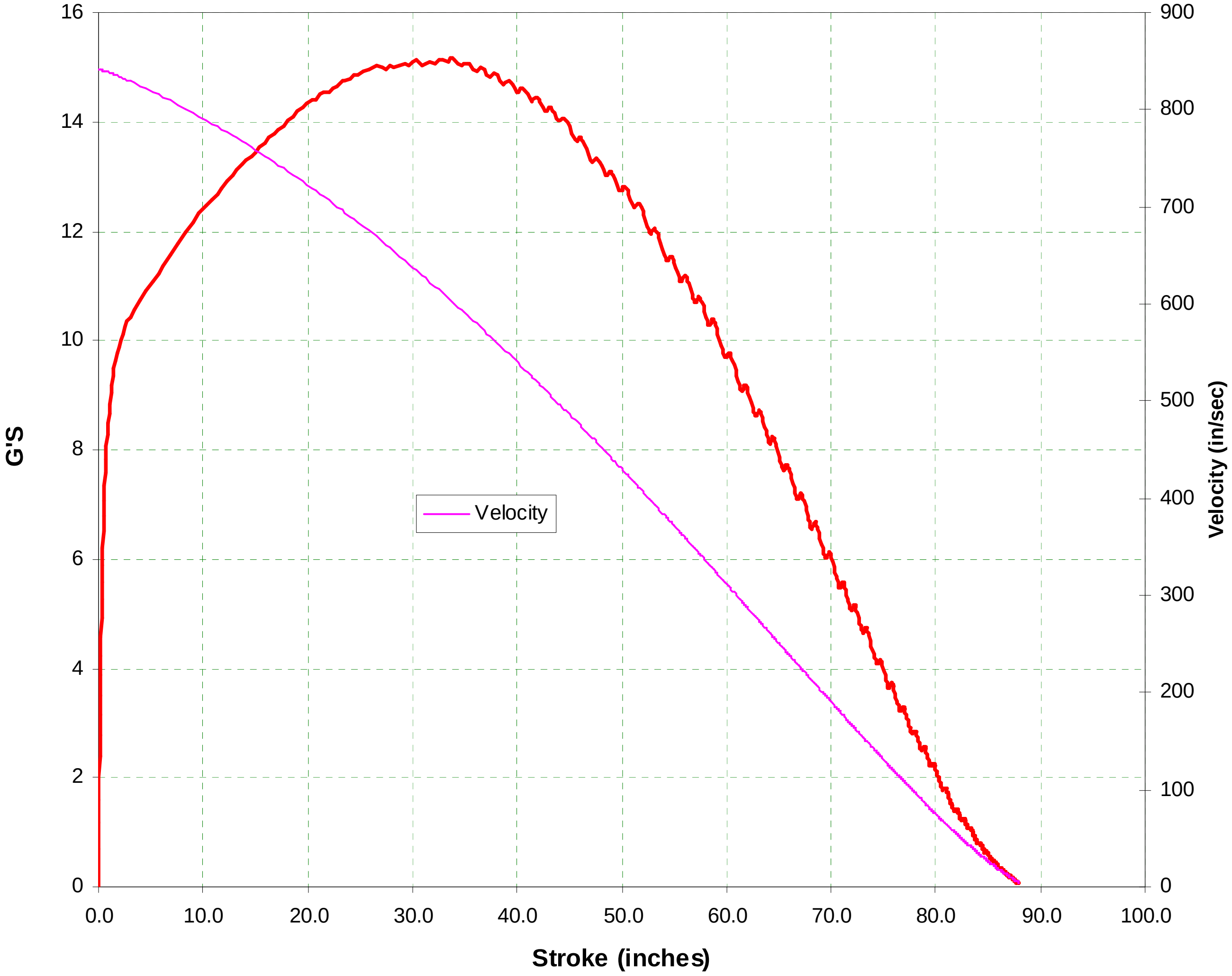
Impacting Velocity [in/sec] 1100.0

Propelling Force [lbs] 0.0

Technical Notes

2 UNITS IN TENSION 90" STROKE
1800Lbs. @ 62.5 Mph

Max Shock Force: 17849



APPENDIX D ANALYSIS & SIMULATION OF HSA IMPACT DYNAMICS: FULL SCALE

Vehicle Impacting Immovable Truck

In this case, vehicle mass, M_1, can be moved along x-axis only (this case is modeled by a spring and a mass in a horizontal position). The spring, as shown in Figure 1, is used to absorb the kinetic energy of the mass. Therefore, we can construct the dynamic model as follows.

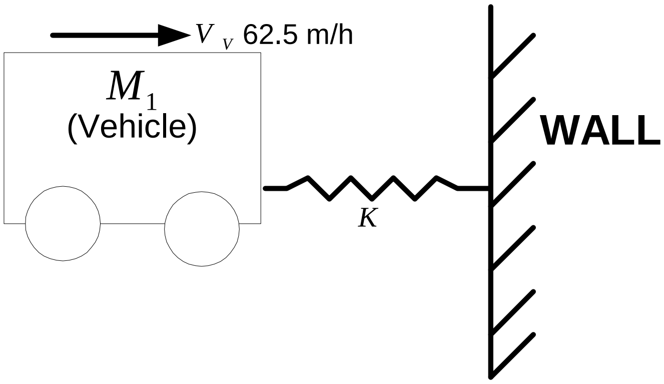


Figure 1. Spring "absorber" connection between the vehicle and the wall.

Dynamic Model

The differential equation of motion is given

$$\ddot{X} + \frac{K}{M_1} X = 0$$

where K is the stiffness coefficient of the spring (spring constant)

M_1 is the mass of the vehicle

Constraints

- Vehicle Mass: M_1 = 2,000 kg = 4,409 lb
- Spring coefficient K =3,100,000,000 N/km = 210,000 lb / ft

(calculated from the stiffness needed to maintain 20 g deceleration)

- Initial Velocity: $V(t=0) = 100 \quad km/hr = 62.5 \quad mph = 91.7 \quad ft/\sec$
- Initial Position: $X(t=0) = 0$ ft
- Displacement: X $\geq$ 0 and Velocity: $V \geq 0$

  (Impacting vehicle always has a positive motion, i.e. it doesn't move backwards, and the velocity is never negative)
- The acceleration of impacting vehicle $\geq$ -20g

Solution

The general solutions are as follows.

$X(t) = B\cos\,\omega t$ where B is a constant, $\omega = \sqrt{\dfrac{K}{M_1}}$

Velocity: V= $\dot{X}(t) = -B\omega$ sin ωt

Acceleration: A= $\ddot{X}(t) = -B\omega^2$ cos ωt

Substituting $V(t=0) = V_0$ and general solution, the result is $B = \dfrac{V_0}{\omega}$

The solution to the above equation is

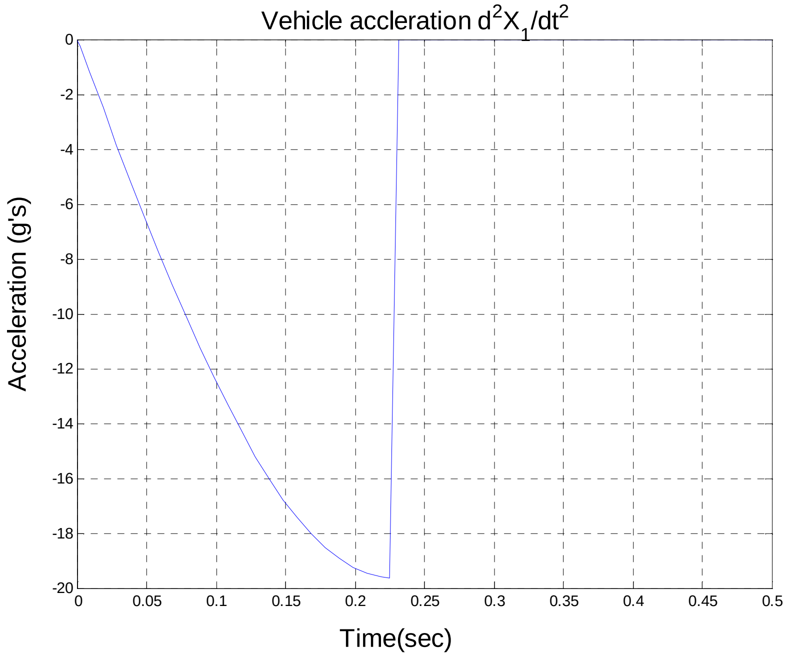
$$X(t) = \frac{V_0}{\omega}\sin(\omega t)$$

Simulation Results

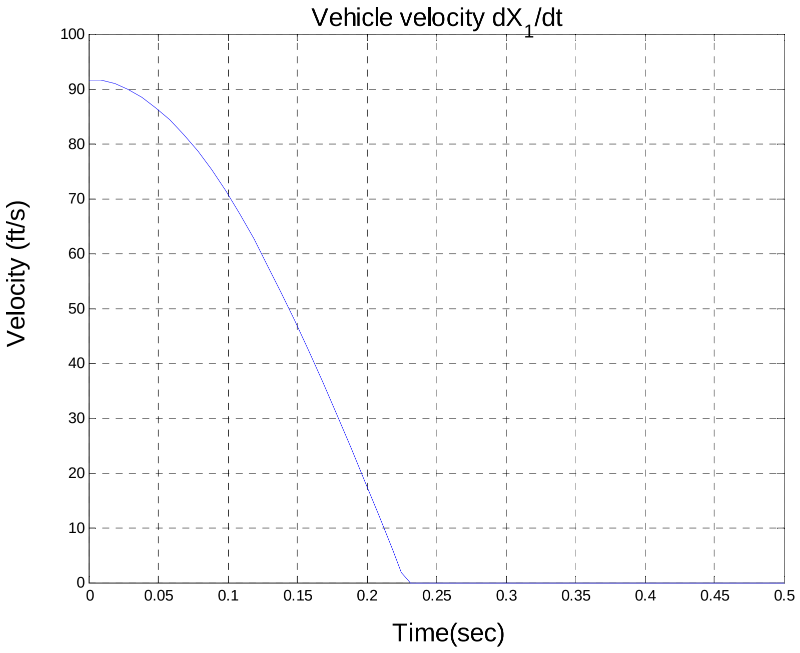
To maintain -20g acceleration, K =3,100,000,000 N/ km = 210,000 lb/ft is chosen;

Velocity: V = $V_0 \cos\,\omega t$; Acceleration: A = $-V_0\omega \sin\,\omega t$

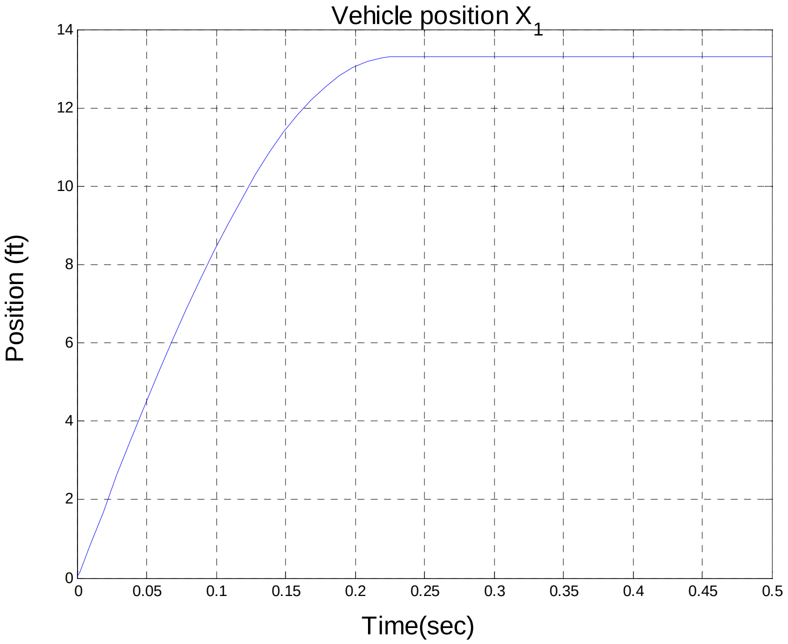
For vehicle M_1**:**


Vehicle accleration d^2X_1/dt^2
Acceleration (g's)
0
-2
-4
-6
-8
-10
-12
-14
-16
-18
-20
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
0.5
Time(sec)

(a)


Vehicle velocity dX_1/dt
Velocity (ft/s)
100
90
80
70
60
50
40
30
20
10
0
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
0.5
Time(sec)

(b)



(c)

Figure 2 a,b,c. Simulation results of immovable truck case.

As a result of the simulation, the maximum deceleration of vehicle was 20g. From 0.22 second, the spring was fully compressed. Also from around 0.22 second, the deceleration of the vehicle became constant after the vehicle stopped. The car stopped at 0.22 sec and within 14 feet for condition that the constant spring coefficient was used. In addition, the maximum force that stops the vehicle is mass × acceleration = 4410 lb × 20g, so it is 88,180 lb.

Vehicle Impacting Movable Truck

In this case, we allow the truck to slide during the impact. (in the previous case, we assumed the truck was equivalent to a rigid wall). This case is modeled by two masses and a spring in a horizontal position.

We consider two masses of a vehicle and truck. The initial velocity of the vehicle is 62.5 mph. In order to store the impact energy, we place a spring in between the vehicle and truck with the brake on. Each tire of the truck has friction acting against the ground. The coefficient of the friction (0.72) is found in a research paper [1]. So the results of simulation based on two dynamic models are shown in the following.

---

[1] Glenn Elert; Friction; 1998-2004 The Physic Hypertextbook^{TM}; http://hypertextbook.com/physics/mechanics/friction/

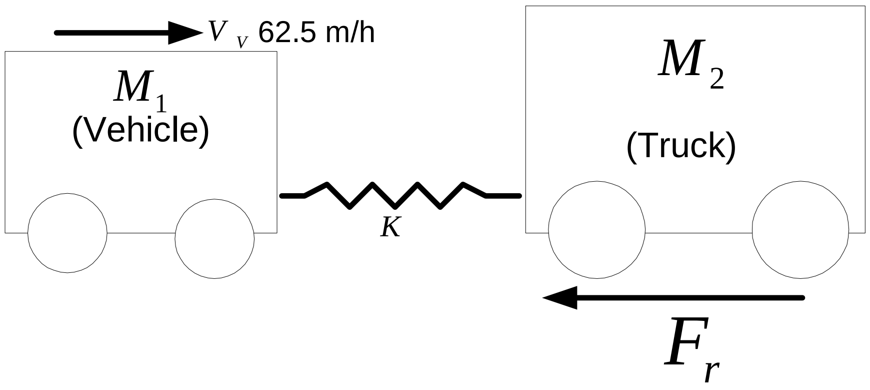


Figure 3. Spring connection between two masses.

Dynamic Model

The differential equation of motion is given

$M_1\ddot{X}_1 = K(X_2 - X_1)$ ……………………………(1)

$M_2\ddot{X}_2 = -K(X_2 - X_1) - F_r$ ………………………(2)

where F_r the sliding truck's tire friction acting against the road

M$_1$ and M$_2$ are the masses of the vehicle and truck respectively.

K is the stiffness of the spring.

Constants

- M_1: Vehicle mass = 2,000 kg = 4,409 lb
- M_2: Truck mass = 9,000 kg = 12,786 lb
- K : Stiffness coefficient of spring K = 3,100,000,000 N/ km = 231,000 lb / ft
- F_r: Friction force of truck
- Initial Velocity: $V_1(t=0) = 100 \quad km/hr = 62.5 \quad mph = 91.7 \quad ft/\sec$
- Initial Velocity: $V_2(t=0) = 0 \quad mph$
- Initial Position: $X_1(t=0) = 0$ ft, $X_2(t=0) = 0$ ft

- Displacement: X_1, X_2 ≥ 0 and Velocity: $V_1, V_2 \geq 0$

  (Impacting vehicle always has a positive motion, i.e. it doesn't move backwards, and the velocity is never negative)

- The acceleration of impacting vehicle ≥ -20g

Solutions

The general solutions are as follows.

$$X_1 = \overline{X}_1 \cos \omega t$$
$$X_2 = \overline{X}_2 \cos \omega t$$

where $\overline{X}_1, \overline{X}_2$, and ω are constants.

Velocity:
$$V_1 = \dot{X}_1 = -\overline{X}_1 \omega \sin \omega t$$
$$V_2 = \dot{X}_2 = -\overline{X}_2 \omega \sin \omega t$$

Acceleration:
$$A_1 = \ddot{X}_1 = -\overline{X}_1 \omega^2 \cos \omega t$$
$$A_2 = \ddot{X}_2 = -\overline{X}_2 \omega^2 \cos \omega t$$

Substituting the general solutions into the equation (1) and (2), we have the solutions in the following.

$$\overline{X}_1 = -F_r \left(\frac{K}{M_1 M_2 \left(\omega^4 - K^3 \right)} \right)$$
$$\overline{X}_2 = -F_r \left(\frac{K}{M_1 M_2 \left(\omega^4 - K^3 \right)} \right) \times \frac{K - M_1 \omega^2}{K}$$

where $\omega = K \sqrt{\dfrac{M_1 M_2}{M_1 + M_2}}$

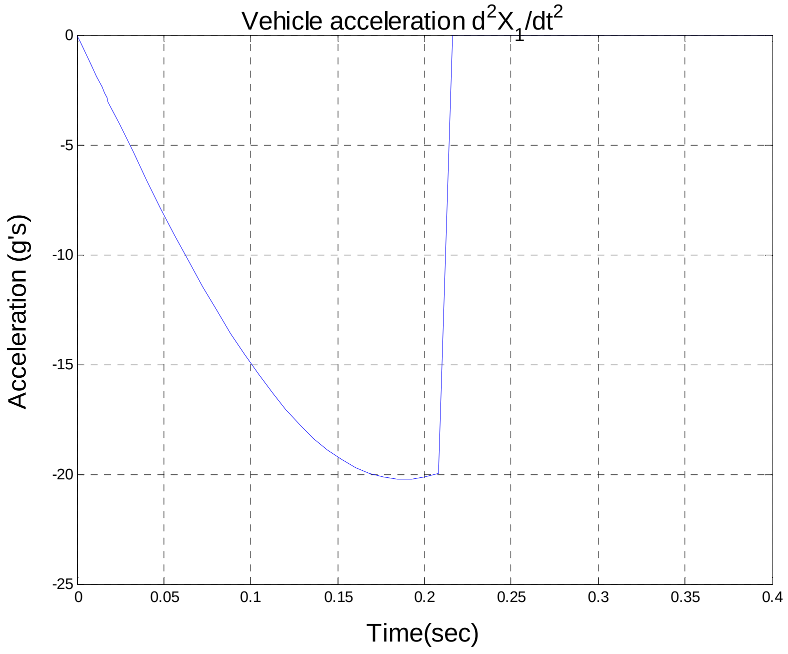
Simulation results

To maintain -20g acceleration, K =3,100,000,000 N/ km = 210,000 lb/ft is chosen

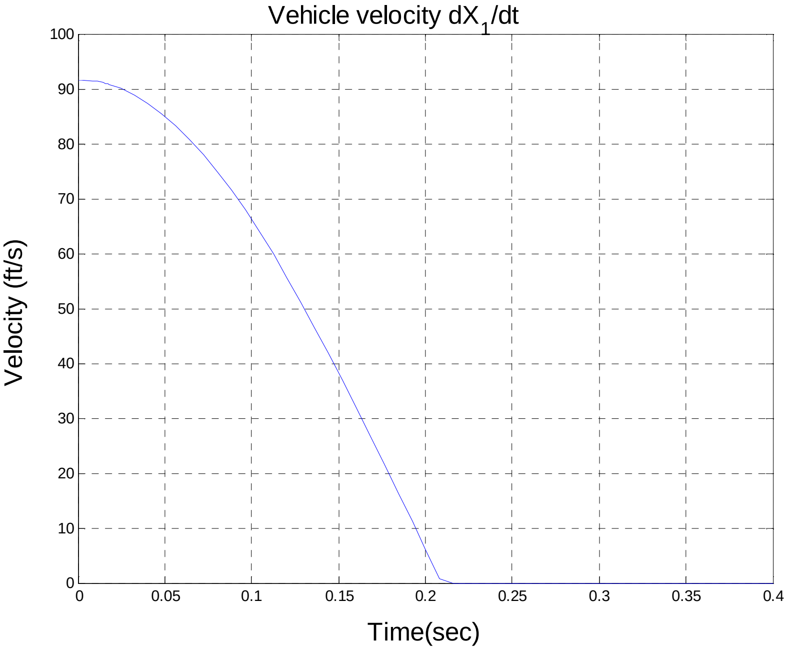
Velocity:
$$V_1 = \dot{X}_1 = -\overline{X}_1 \omega \sin \omega t$$
$$V_2 = \dot{X}_2 = -\overline{X}_2 \omega \sin \omega t$$

; Acceleration:
$$A_1 = \ddot{X}_1 = -\overline{X}_1 \omega^2 \cos \omega t$$
$$A_2 = \ddot{X}_2 = -\overline{X}_2 \omega^2 \cos \omega t$$

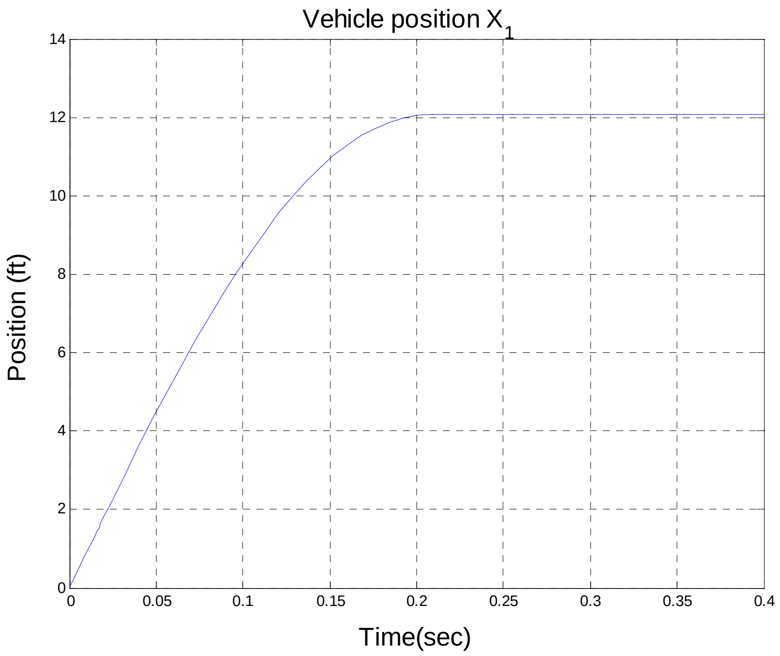
The results for vehicle M_1 follow:


Vehicle acceleration d^2X_1/dt^2
Acceleration (g's)
0
-5
-10
-15
-20
-25
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
Time(sec)

(a)


Vehicle velocity dX_1/dt
Velocity (ft/s)
100
90
80
70
60
50
40
30
20
10
0
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
Time(sec)

(b)

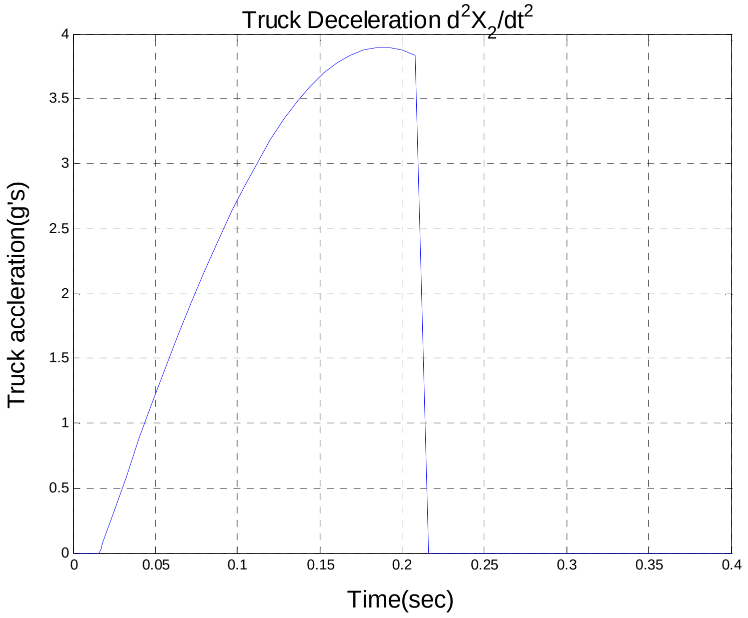


(c)

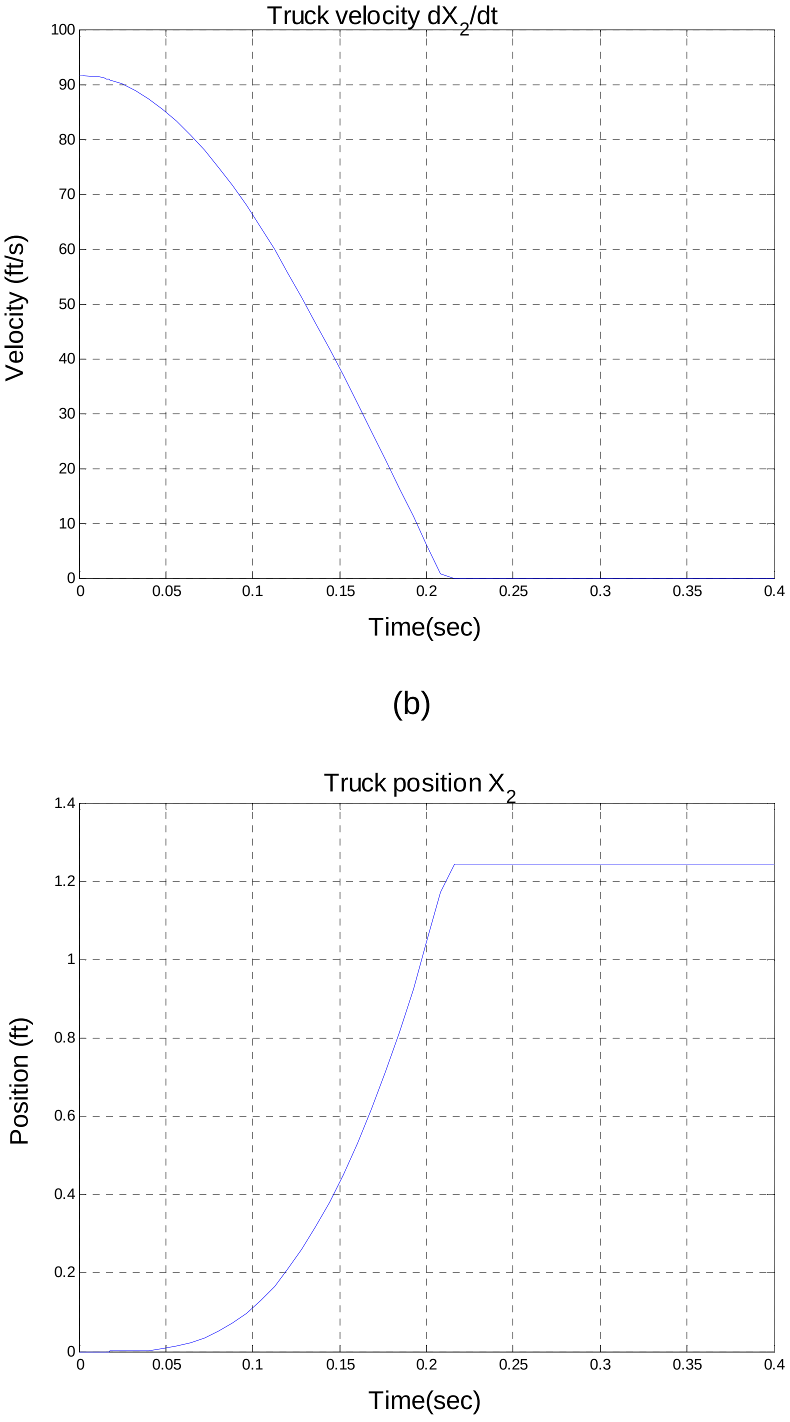
Figure 4. Simulation results of vehicle M_1.

As the results of the simulation, the maximum deceleration of the vehicle was 20g. From 0.21 second, the spring was fully compressed. Also from around 0.22 second, the deceleration of the vehicle became constant after the vehicle stopped. It stopped at 0.21s.

The results for truck M_2 follow:



(a)



(b)

(c)

Figure 5. Simulation results of truck M_2.

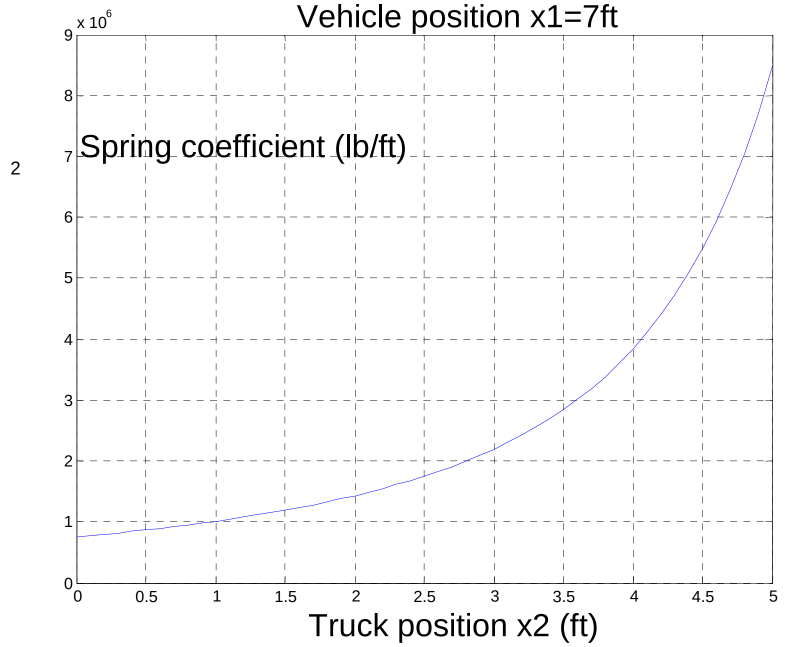
The truck displacement during the impact is shown in figure 5. In the results, the truck stayed from 0.0 to 0.04 second. Because the mass of the truck was heavier than that of the vehicle, it required more energy to push the heavier mass. Moreover, during the impact, the friction force of the truck tires existed. From 0.04 second, the truck slides slowly. Finally the truck stopped at 0.21 second.

Energy Conservation Method

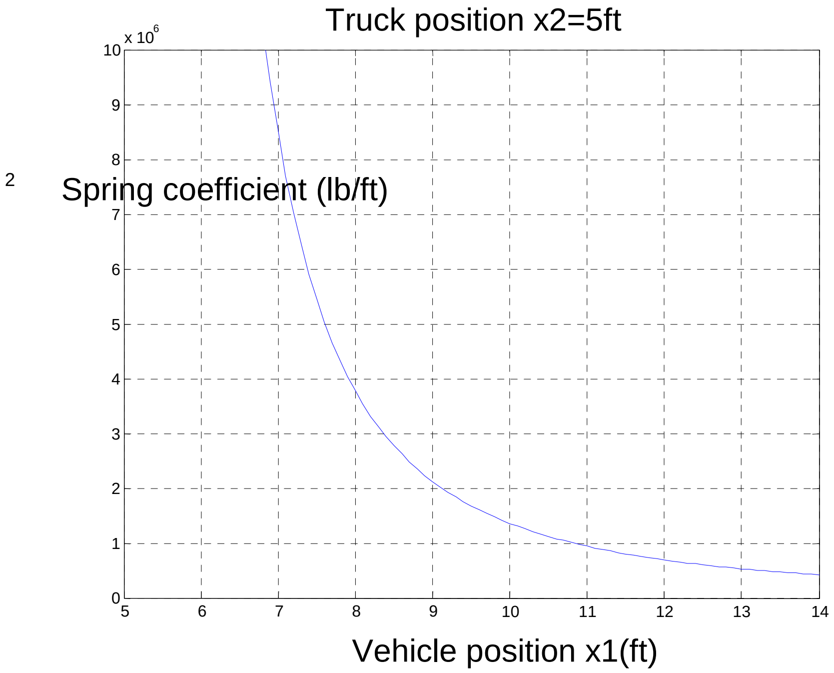
$$\frac{1}{2}M_1V_V^2 = \frac{1}{2}K(X_1 - X_2)^2 + \mu M_2 g X_2$$

where M_1 and M_2 are the masses of the vehicle and truck, μ is the friction coefficient, g is the gravity, V_V is the velocity of the vehicle , X_1 and X_2 are the positions of the vehicle and truck.

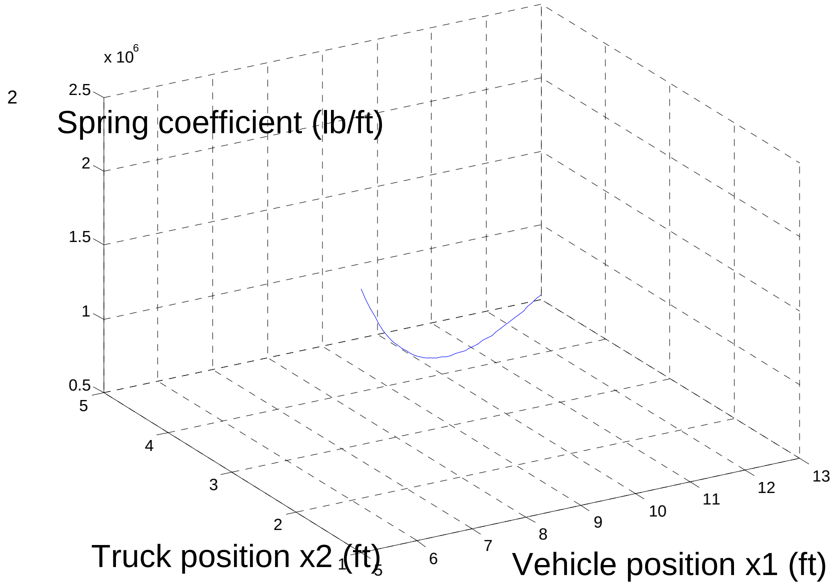
In the following simulation results, Figure 6 (a) and (b), show the spring coefficient varies when we define the different desired positions of the vehicle and truck, respectively.



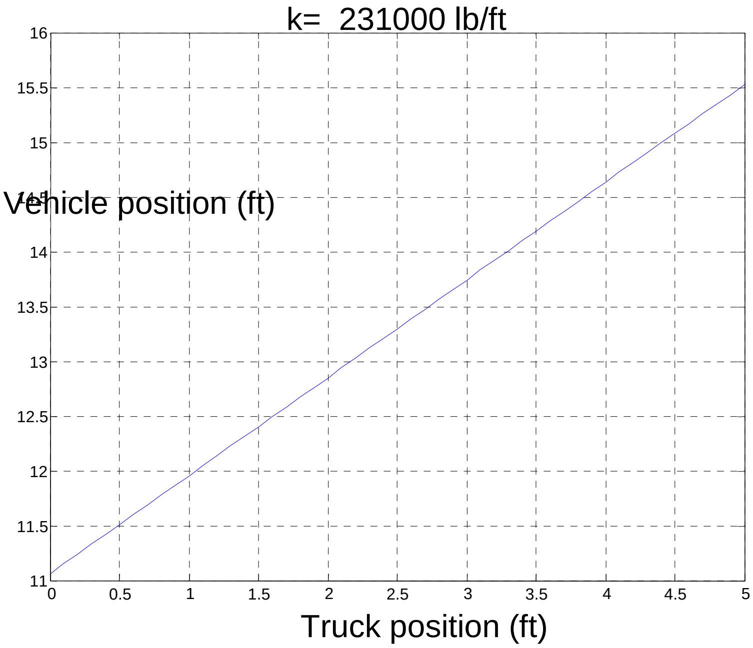
(a)


Truck position x2=5ft
x 10^6
10
9
8
7
6
5
4
3
2
1
0
2
Spring coefficient (lb/ft)
5
6
7
8
9
10
11
12
13
14
Vehicle position x1(ft)

(b)


x 10^6
2.5
2
1.5
1
0.5
2
Spring coefficient (lb/ft)
5
4
3
2
1
Truck position x2 (ft)
5
6
7
8
9
10
11
12
13
Vehicle position x1 (ft)

(c)



(d)

Figure 6. Simulation results by energy conservation.

Figure 6 (a) shows the relationship between the spring coefficients and the truck positions when we have the corresponding desired position of the vehicle (7 feet).

Therefore when we look for the desired position of the vehicle or truck, we can pick the most suitable spring coefficient from Figure 6 (a) or (b) respectively. Conversely, if we desire the spring coefficient, we can also find the desired positions of the truck or vehicle in the figure 6 (c).

Additionally, Figure 6 (d) shows the position of the truck and the vehicle when we choose the spring coefficient 231,000 lb/ft to maintain the 20g. In this case, increasing the stopping distance of the vehicle can increase the sliding distance of the truck.

Finally, as the results of the simulation, when vehicle's stopping distance was around 12.2 ft, truck's stopping distance was 1.25 ft as shown in Figure 6 (d).

APPENDIX E ANALYSIS & SIMULATION OF HSA IMPACT DYNAMICS: QUARTER SCALE

Vehicle Impacting Immovable Truck

In the first case, M_{small} can be moved along x-axis only. (This has a scaled down force and energy in the simple simulation.)

The TMA, as shown in Figure 1, has the characteristics of hydraulic bumpers. The TMA is extended from the wall. The vehicle M_{small} is supposed to impact it at the desired speed (62.5 mph). In this section, we scale down the energy and the impact force by a quarter.

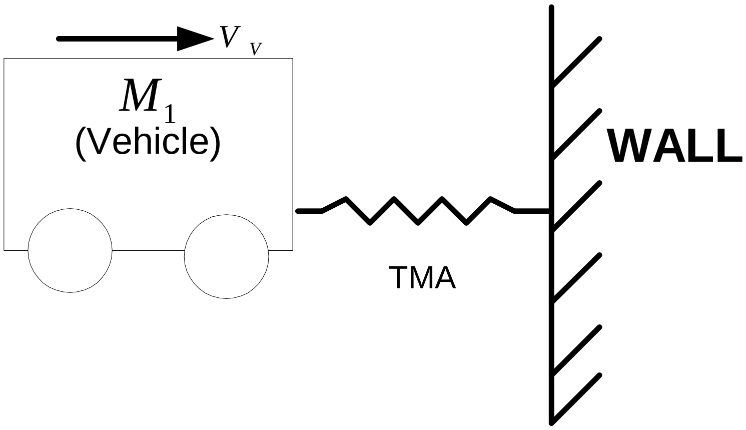


Figure 1. TMA connection between a vehicle and a wall.

Dynamic Model

We scale down the force by one quarter. So the relationship is derived in the following

$F_{new} = \frac{1}{4} F_o = M_{small} \times \ddot{X}_o$

where F_{new} is the scaled down force acting against the scaled down vehicle in the following simulation. F_o is the real force acting against the vehicle in the TMA application. $F_o = M_1 \times \ddot{X}_o$ where M_1 is the real mass of the impacting vehicle (2000 kg, 4402 lb). $\ddot{X}_o$ is the DOT prescribed safety deceleration of the impacting vehicle (20g) M_{small} is the mass of the scaled down impacting vehicle.

From the above equation, the M_{small} is obtained.

From the energy conservation point of view, the ¼ energy can be found in the following.

Total Energy in scale down case= $\frac{1}{2}M_{small}\dot{X}_1^2 = \frac{1}{4}$ (Total Energy in real case) = $\frac{1}{4}E = \frac{1}{4}\left(\frac{1}{2}M_1\dot{X}_1\right)$ where E is the total energy in real case.

When $\dot{X}_1$ is defined as 62.5 mph = 100 km/hr = 91.7 ft/sec,

$M_{small} = \frac{1}{4}M_1$ is obtained.

Then, we have the following dynamics

$$\ddot{X} + \frac{F_{bumper}\ (t)}{M_{small}} = 0$$

where

F_{bumper} is the force, a function of time, acting against the scaled down vehicle and obtained from the known data of the hydraulic bumper as shown in Figure 1.

Constraints

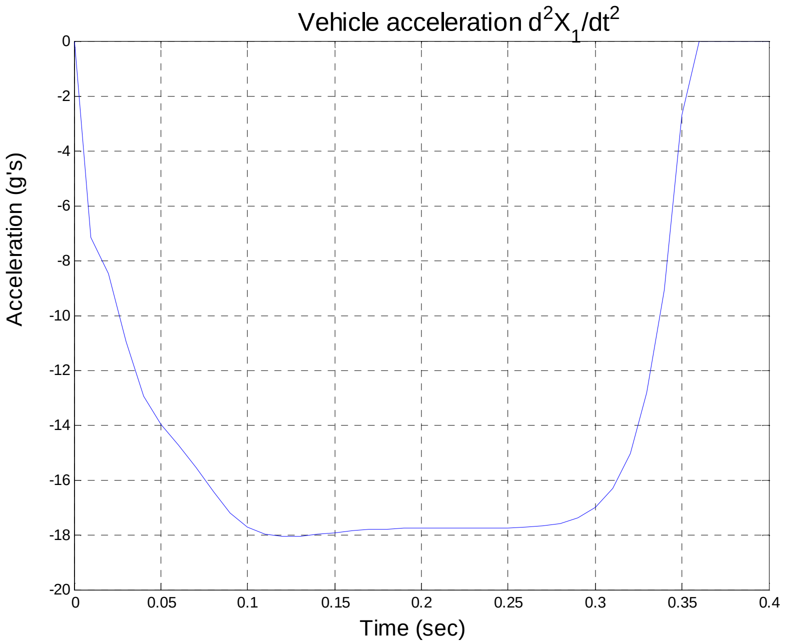
- Vehicle Mass: M_1 = 500 kg = 1,101 lb
- Initial Velocity: $V(t=0) = 100\ \ km/hr = 62.5\ \ mph = 91.7\ \ ft/\sec$
- Initial Position: $X(t=0) = 0$ ft
- Displacement: X $\geq$ 0 and Velocity: $V \geq 0$

(Impacting vehicle always has a positive motion, i.e. it doesn't move backwards, and the velocity is never negative)

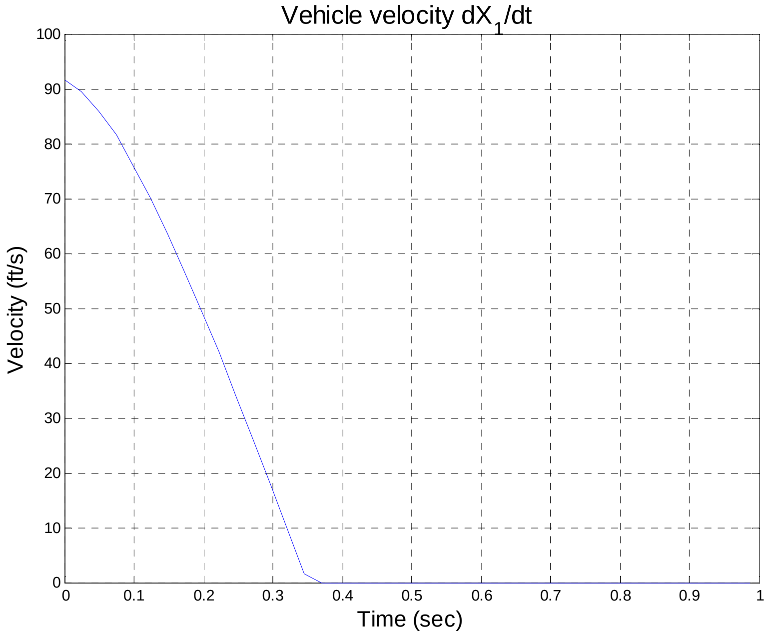
- The acceleration of impacting vehicle $\geq$ -20g

Simulation Results

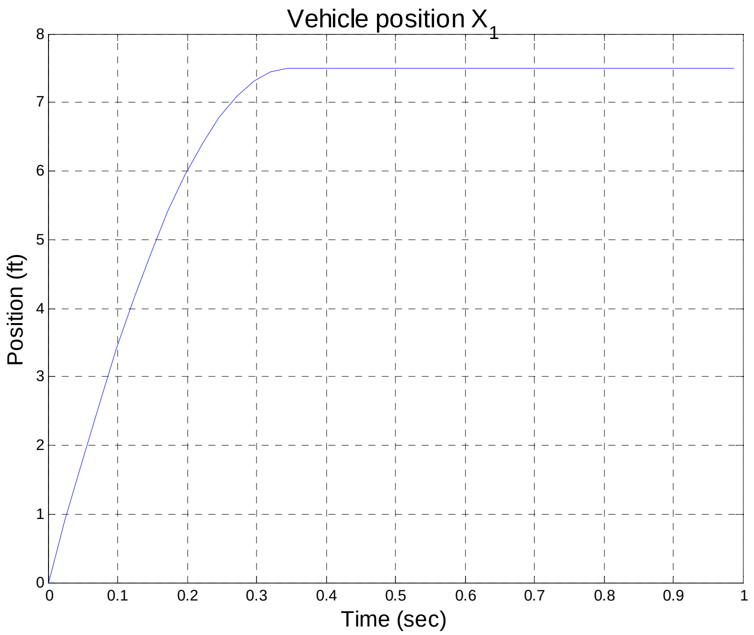
To maintain less than 20g deceleration, the TMA performance data as shown in Figure 2 (a) is assumed below.


Vehicle acceleration d^2X_1/dt^2
Acceleration (g's)
0
-2
-4
-6
-8
-10
-12
-14
-16
-18
-20
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
Time (sec)

(a)


Vehicle velocity dX_1/dt
Velocity (ft/s)
100
90
80
70
60
50
40
30
20
10
0
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
Time (sec)

(b)



(c)

Figure 2. Simulation results of vehicle M_1.

As a result of the simulation, the maximum deceleration of vehicle is less than 20g. The initial impacting speed of the vehicle is 91.7 feet/sec. The stopping distance of the vehicle is 7.5 feet. The length of time is around 0.37 sec.

Vehicle Impacting Movable Truck

In this case, we allow the truck to slide during the impact. (in the previous case, we assumed the truck was equivalent to a rigid wall). We use a scaled down force and energy in the simple simulation.

We consider two masses of a vehicle and truck. The initial velocity of the vehicle is 62.5 mph. In order to dissipate the impact energy, we place the proposed TMA in between the vehicle and truck with the brake on. Each tire of the truck has friction acting against the ground. The coefficient of the friction (0.72) is found in the research paper [1].

So the results of simulation based on two dynamic models are shown in the following. In this simulation, the force and energy are scaled down by one quarter. The simulation is performed when the truck with the brake on is assumed to be slided during an impact. Therefore, a friction of those tires put on the truck is produced and used to slow down the truck's speed.

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[1]: Glenn Elert; Friction; 1998-2004 The Physic Hypertextbook^{TM}; http://hypertextbook.com/physics/mechanics/friction/

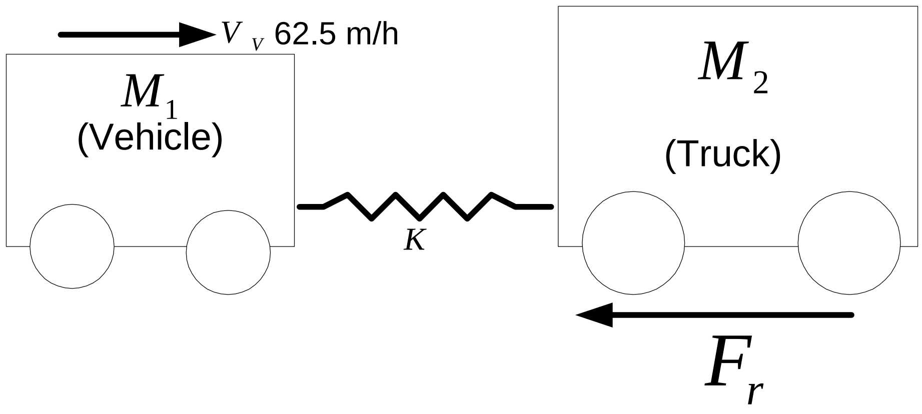


Figure 3. Spring connection between two masses.

Dynamic Model

In case 2, the mass M_1 can be allowed to be moved along x axis.

$M_1\ddot{X}_1 = F_r(t)$ ……………………………(1)

$M_2\ddot{X}_2 = -F_r(t) - F_f$ ………………………(2)

where F_f the sliding truck’s tire friction acting against the road

F_r the relative force produced by the TMA acting against the vehicle

Constants

- M_1: Vehicle mass = 500 kg = 1,101 lb
- M_2: Truck mass = 2500 kg = 3,198 lb
- F_r: Friction force of truck
- Initial Velocity: $V_1(t=0) = 100 \quad km/hr = 62.5 \quad mph = 91.7 \quad ft/\sec$
- Initial Velocity: $V_2(t=0) = 0 \quad mph$
- Initial Position: $X_1(t=0) = 0$ ft, $X_2(t=0) = 0$ ft
- Displacement: X_1, $X_2 \geq 0$ and Velocity: $V_1, V_2 \geq 0$

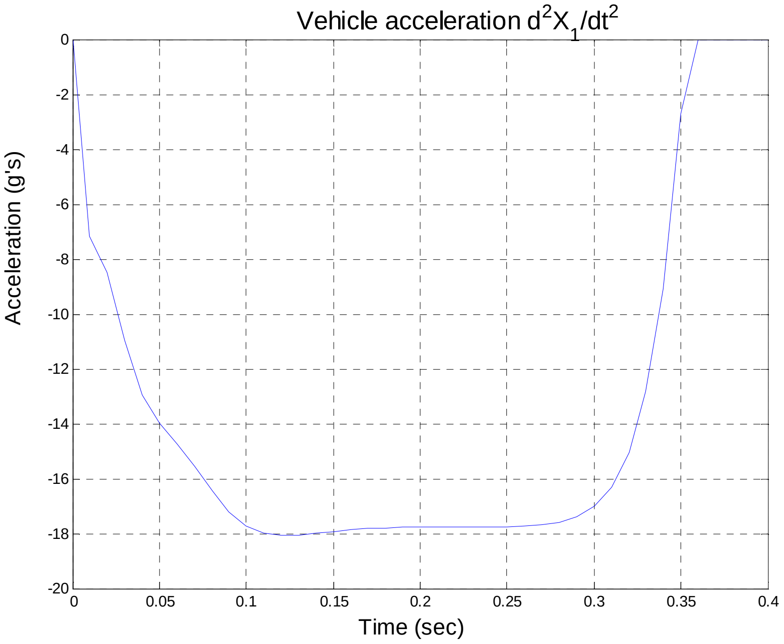
- (Impacting vehicle always has a positive motion, i.e. it doesn't move backwards, and the velocity is never negative)

- The acceleration of impacting vehicle ≥ -20g

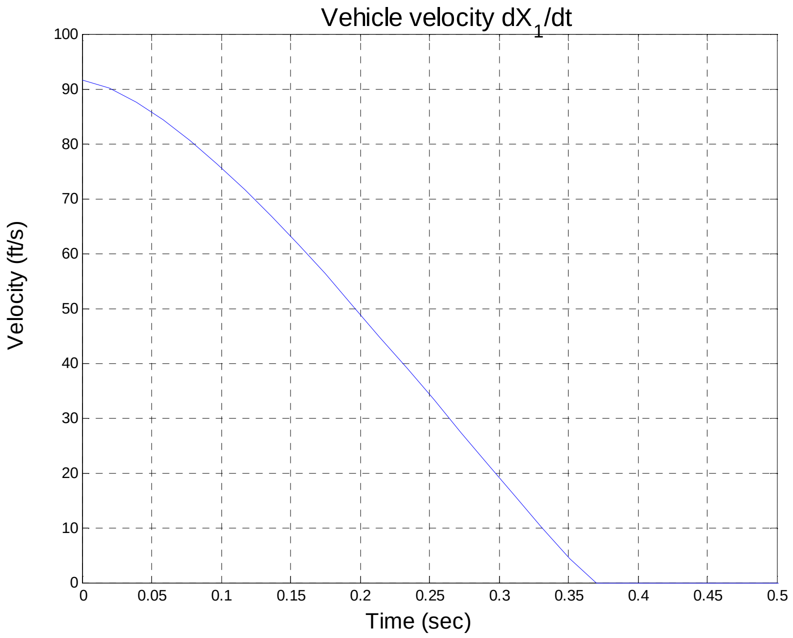
Simulation Results

To maintain less than 20g deceleration, the TMA performance data as shown in Figure 4 (a) is assumed below.

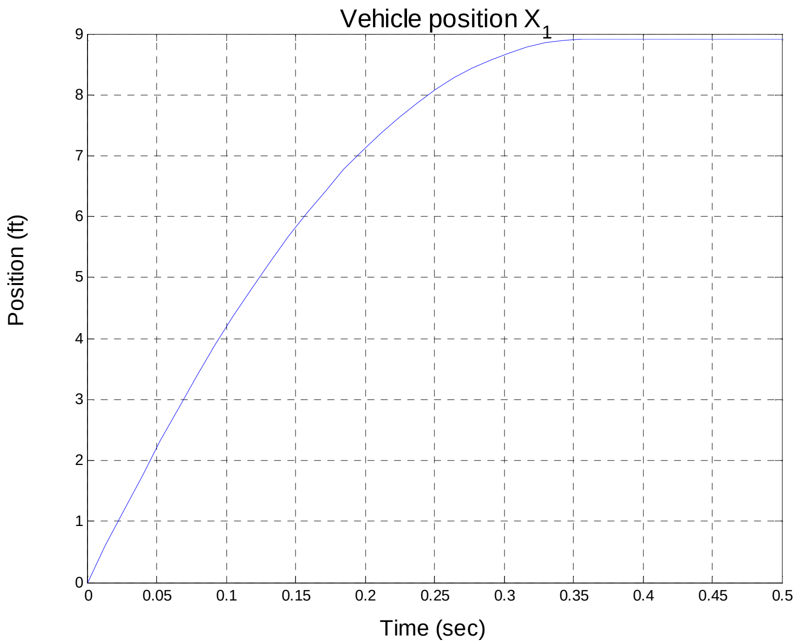
Results for vehicle M_1 follow:



(a)



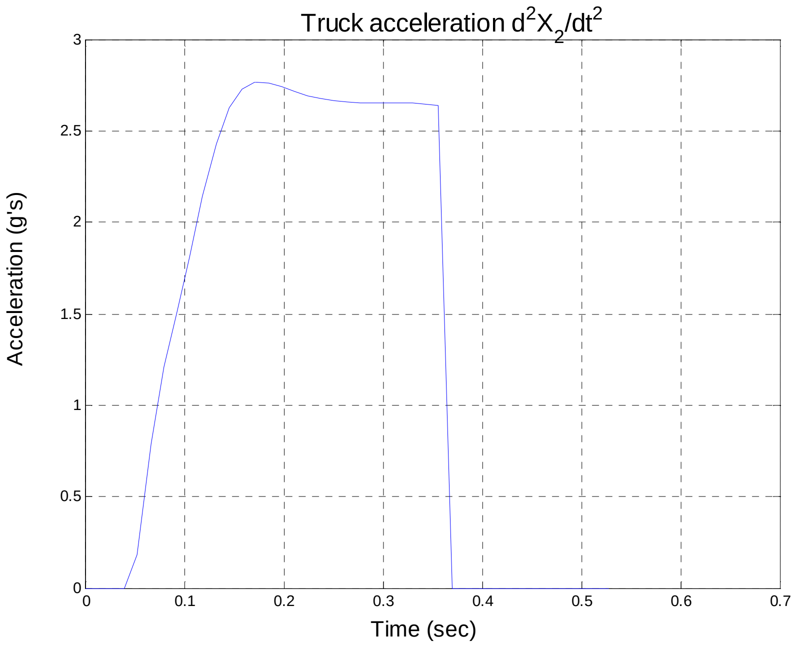
(b)



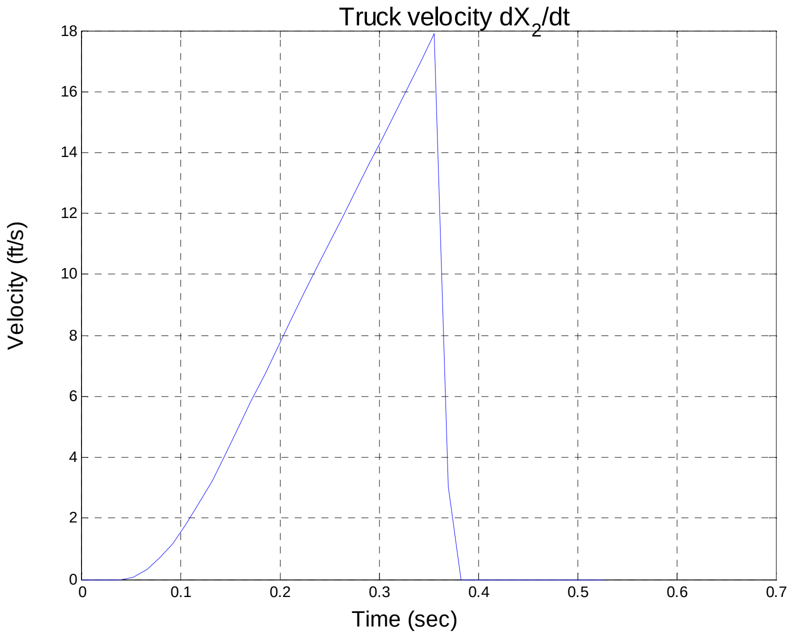
(c)

Figure 4. Simulation results for vehicle M_1.

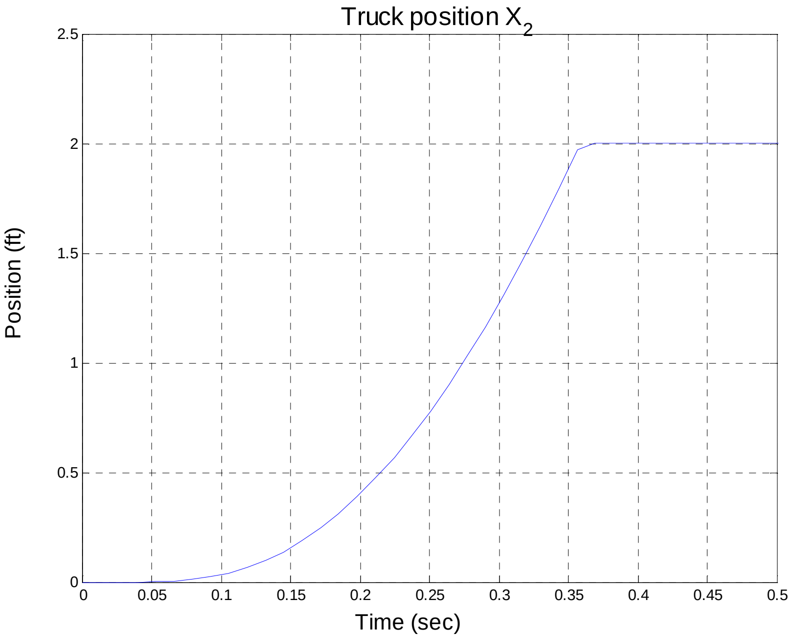
Results for truck M_2 follow:



(a)



(b)



(c)

Figure 5. Simulation results for truck M_2.

Figure 5 shows the truck displacement during the impact. The static friction force of the truck tires, which has the same magnitude but in the opposite direction, stops the truck within the certain distance. In the results of the simulation, the maximum deceleration of the vehicle is 20g.

The sliding distance measured from the starting point is 9 feet. The retraction stroke of the TMA is 7 feet. (i.e. the sliding distance of the vehicle minus that of the truck during the impact). Therefore according to both simulation results, the stroke of the TMA in case 2 is shorter than case1 because when the truck slides during the impact, the inertia mass of the truck and the friction force of the tires absorb the impacting energy.

Besides, in terms of manufacturing devices, the small TMA can be more easily implemented than the full scale one. It does not need to absorb that great amount of energy and break a real vehicle, which is very costly and not necessary. Moreover, through the small scale impact, the impact mechanism can be simulated. The smaller impact can be modeled easily in current US facilities.

APPENDIX F FEM ANALYSIS OF SCALED TMA DESIGN

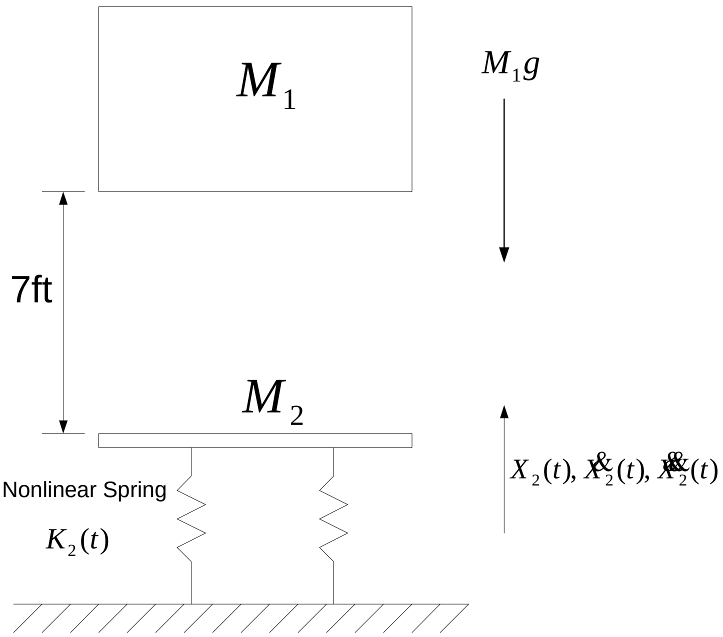
TMA Dynamic Modeling

This report presents FEM simulation of a 1/4 scale model of a TMA. The simulation was performed using COSMOSWORKS in SOLIDWORKS.

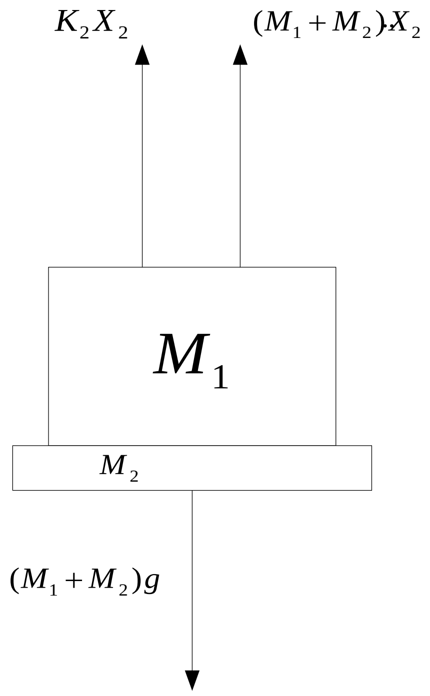
For the 1/4 scale model, a weight of 4,000 lb was dropped onto the end of the TMA structure from the height of 7 ft so that the resulting energy was 1/16 of the full scale model.

The truck supporting TMA was replaced by the rigid wall fixed on the ground as shown in Figure 1. The quarter scale model is illustrated with a free body diagram in Figures 1 and 3 which consists of two hydraulic cylinders and a dead weight 4000 lb.

Figure 4 shows detail drawings of the quarter scale model assembly design developed by Duane Bennett. In this simulation, the impacting vehicle deceleration was kept at 20g, but that had to do with the nonlinear stiffness of the cylinders. The characteristics of the cylinder in terms of its compliance was adopted from the simulated data of the commercial hydraulic cylinder in Enidine's product catalog, as shown in Figures 2(a) and (b).



(a)



(b)

Figure 1. (a) Drop test of TMA, (b) Free-body diagram.

As shown in Figure 1, a vertically guided drop is simulated, which assumes that the truck dump body and chassis guide structure is essential in absorbing all side loads and, therefore, the cylinders only see pure compressive loads. The constraints of the quarter scale model are as follows:

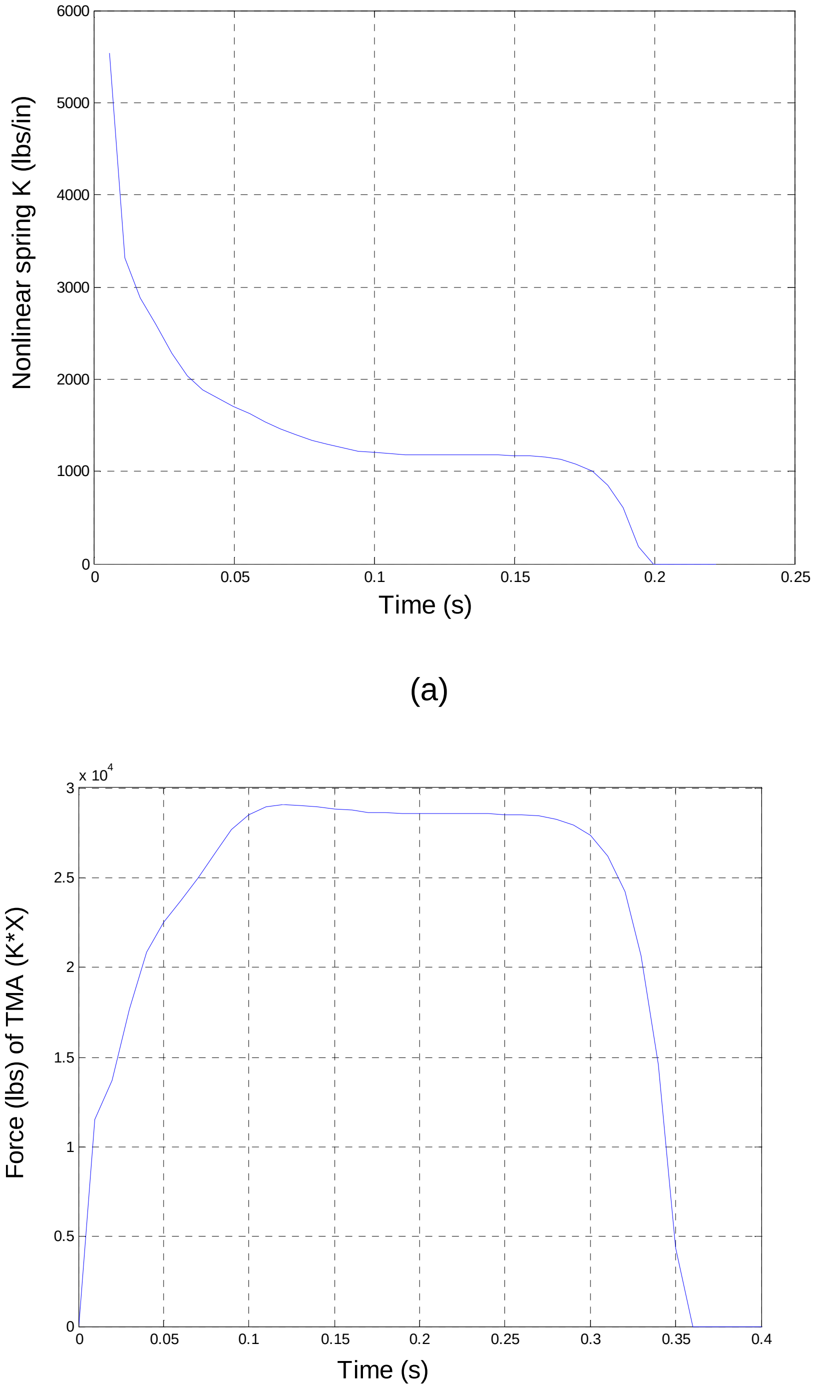
- Initial Position: 7 feet
- W_1= 4,000 lb where $W_1 = M_1 g$
- The deceleration of impacting vehicle $\leq$ 20g

In Figure 1(a) and (b), $K_2(t)$ is the nonlinear stiffness function for the scaled model TMA, which is equivalent to the combined stiffness of two nonlinear springs as shown in Figure 2 (a). $X_2(t), \dot{X}_2(t), \ddot{X}_2(t)$ are the position, velocity, and acceleration of the system.

The equation of motion of the system can be expressed as

$$\ddot{X}_2 + \frac{K_2(t)}{M_1 + M_2} X_2 = 0 \qquad (1)$$

where M_1 and M_2 are the masses of the impacting object and TMA, respectively. It was assumed that the impacting object is subject to downward motion only.



(a)

(b)

Figure 2. (a) Nonlinear stiffness, (b) Reaction force.

TMA Design Model

In Figure 3, the moving portion includes the dump body sliding longitudinally along the truck chassis. The truck dump body and chassis guide structure is essential to absorb all side loads from an impact. The cylinders are subjected to pure compressive loads. (Cylinders can not be subjected to side forces or bending moments.)

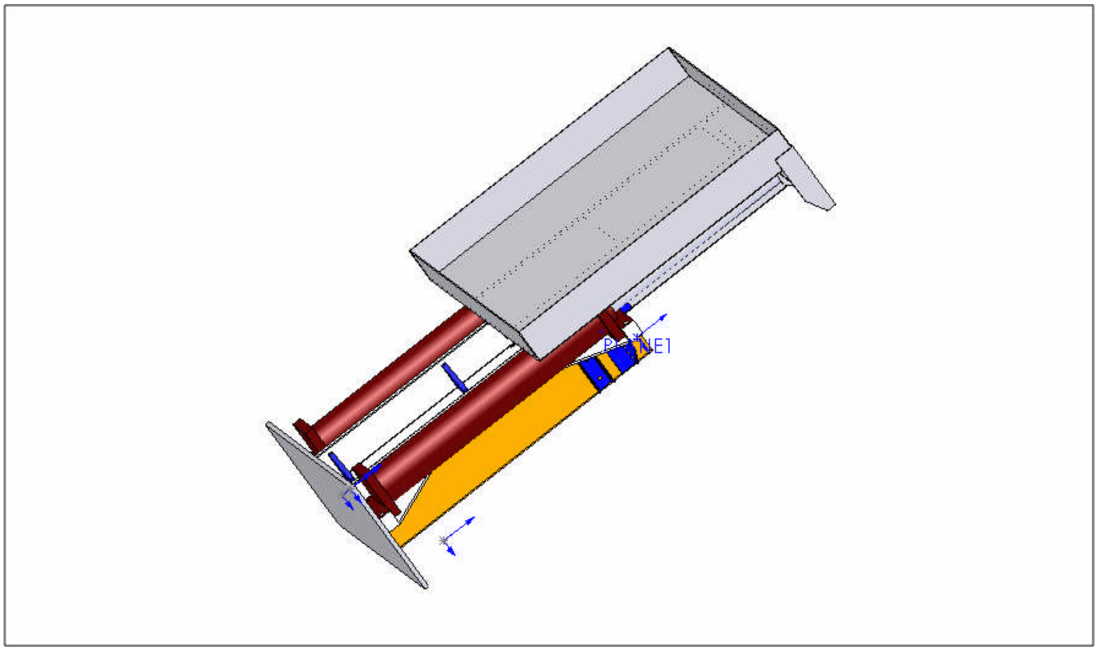


Figure 3. Prototype of TMA.

Figure 4 shows the dimensions of the scaled down TMA. The units are IPS (Inch, Pound, Second). The following section shows the FEM simulation of the TMA.

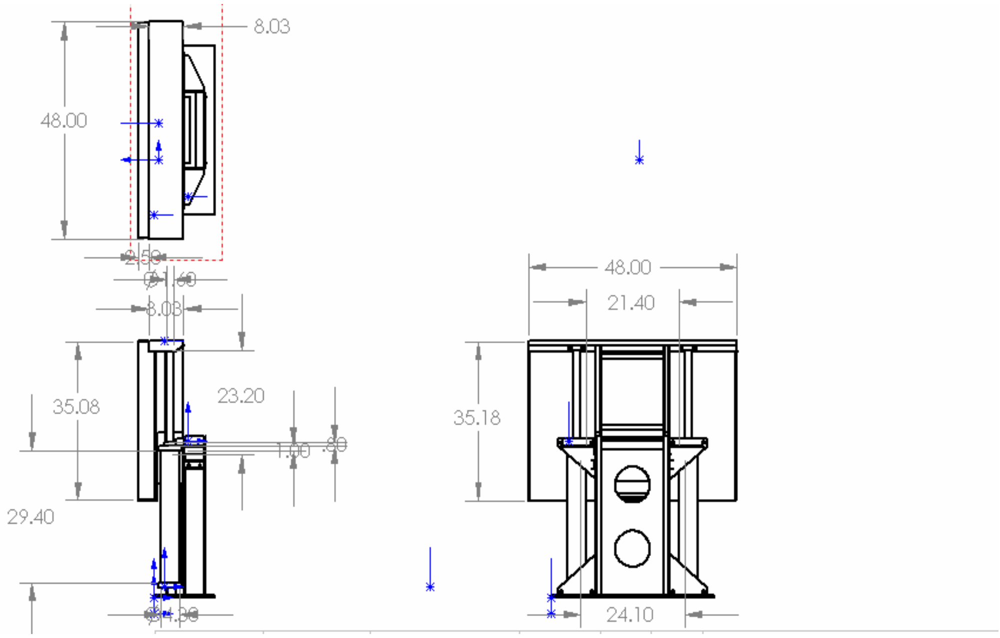


Figure 4. Drawing of prototype TMA.

Finite Element Method (FEM) Simulation

In the FEM simulation, the following conditions were defined in order to mesh the scaled model TMA:

Mesher Used: Standard
Automatic Transition: Off
Include Mesh Auto Loops: Off
Smooth Surface: On
Jacobian Check: 4
Element size: 1.10877 in
Tolerance: 0.0554386 in
Mesh quality: High
Total nodes: 37638
Total elements: 18604

Material source : COSMOS library

1	EX	2.9013E+007	psi
2	NUXY	0.29	
3	GXY	1.117E+007	psi
4	ALPX	8.3333E-006	/Fahrenheit
5	DENS	0.28541	lb/in^3
6	KX	0.0006288	BTU/(in.s.F)
7	C	0.10033	Btu/(lb.F)
8	SIGYLD	51000	psi
9	SIGXT	61000	psi
10	SIGXC	0	psi

A finite element mesh was constructed as shown in Figure 5.

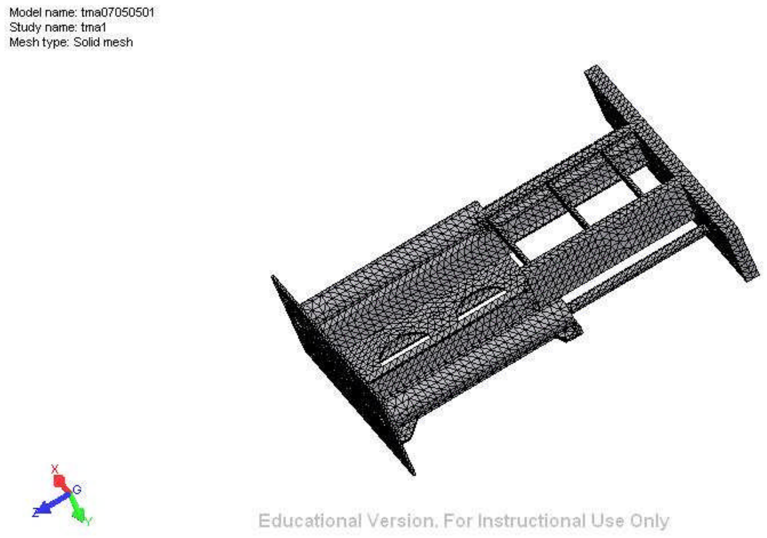


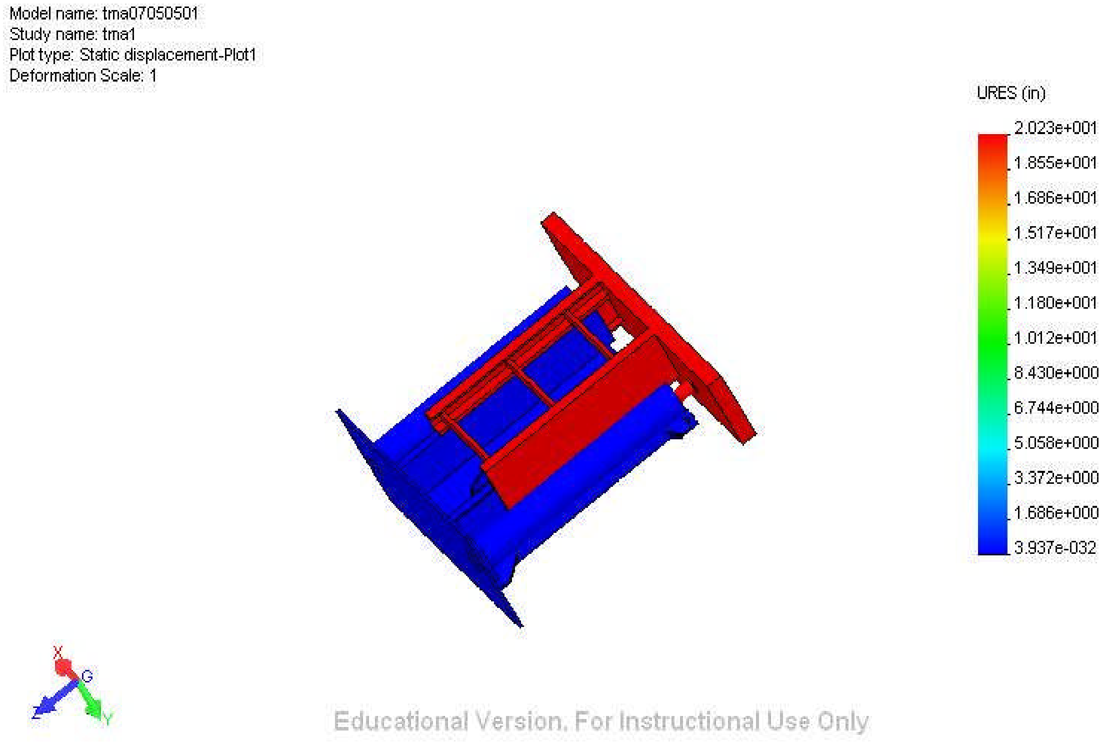
Figure 5. Finite element mesh of TMA.

Through the 3-D mesh, the material property of TMA was put into the model.

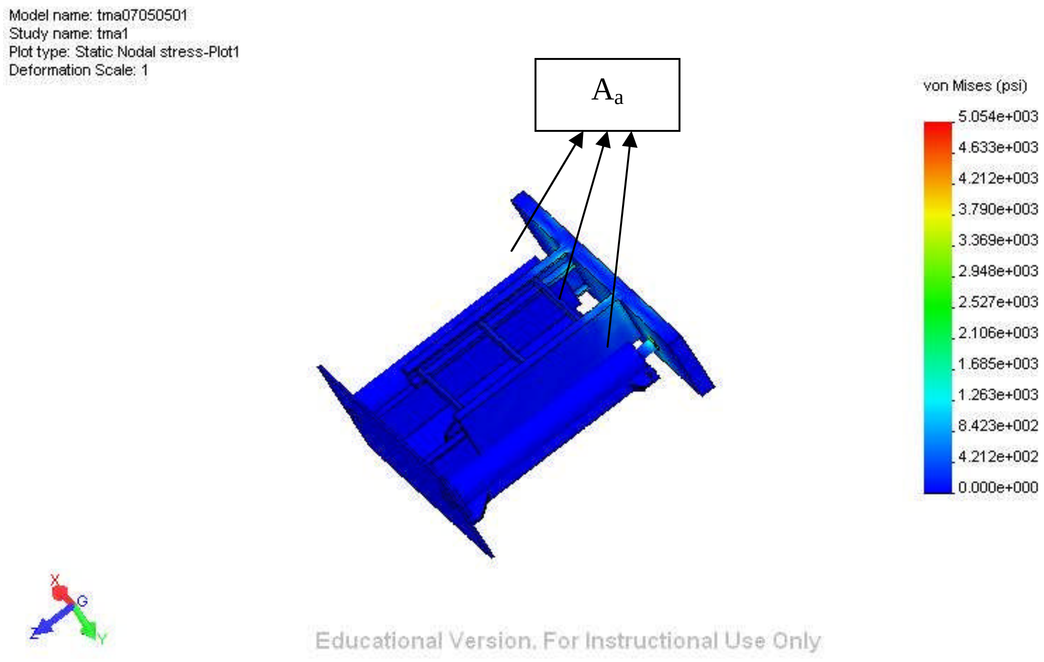
The TMA performance property was set up with that of the commercial hydraulic cylinder. Then, the impact force, restraint, and the resistance force of TMA were put into the FEM simulation.

The data, such as forces, mass, and etc, were reduced by 1/4 as stated in the previous section. The impact force was calculated based on the time varying deceleration of the impacting object provided by the cylinder manufacturer.

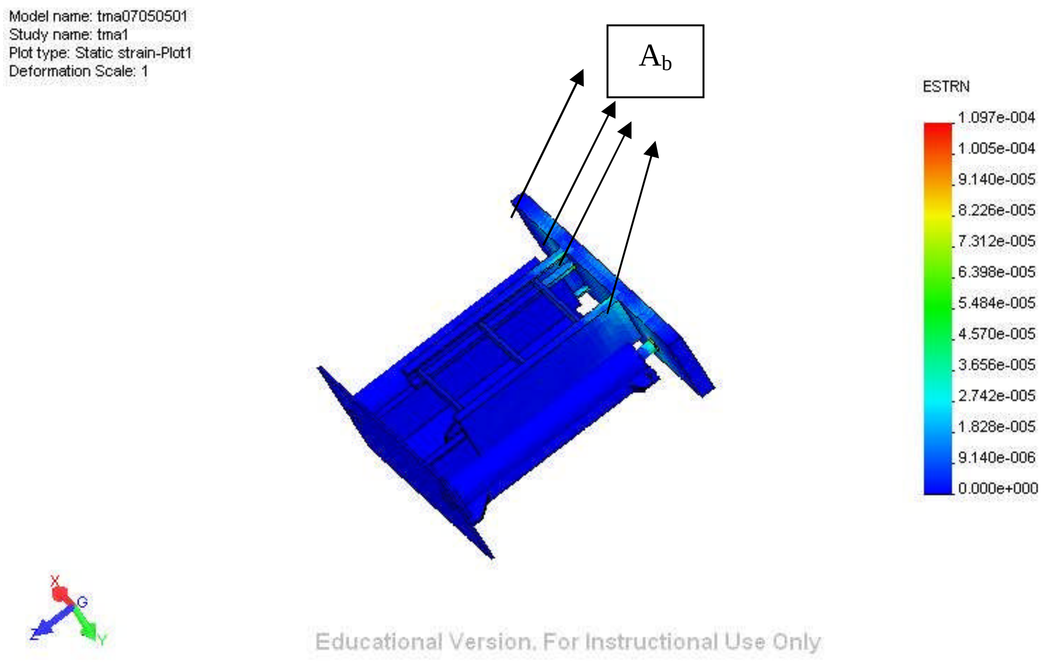
Therefore, the reduced impact force should be equal to the multiplication of the 1/4 mass and the deceleration. Figures 6 and 7 show the results of FEM simulations.



(a) FEM rod traveling distance.



(b) Stress of FEM.



(c) Strain of FEM.

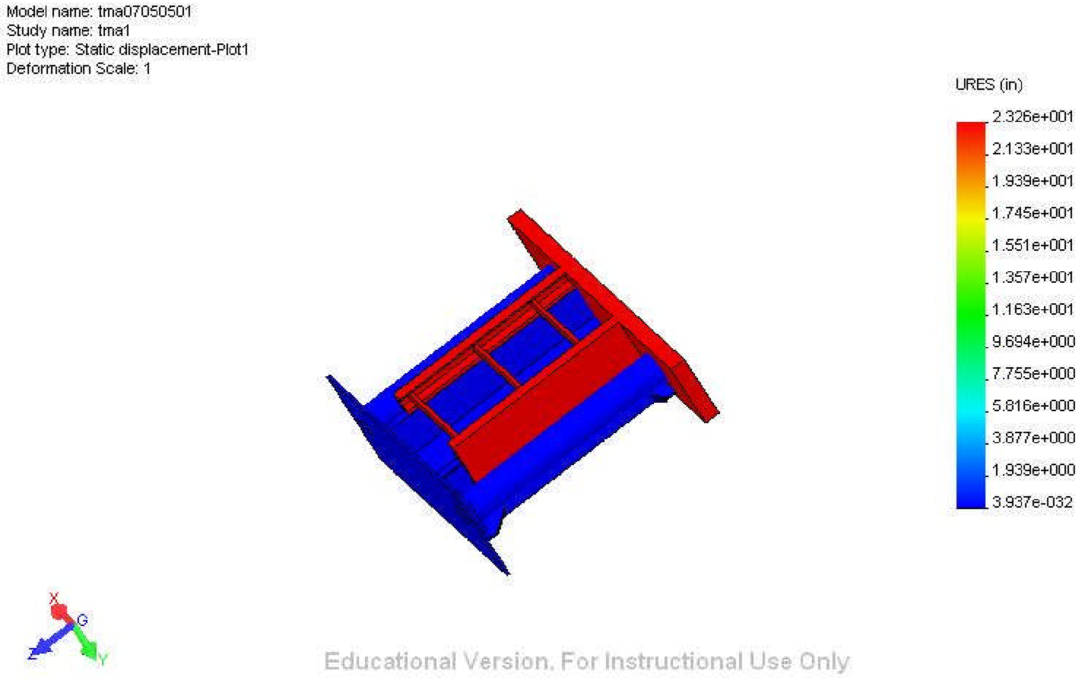
Figure 6. FEM anaylsis for the impact force of 25,000 lb.

In Figure 6(a), the rod traveling stroke is shown. In this case, when the impact force is 25,000 lb, the length of the retraction is 23 in. The maximum stress is 5,054 psi at A_a, as shown in Figure 6(b). The maximum strain is 0.00011 in/in at A_b as shown in Figure 6(c).

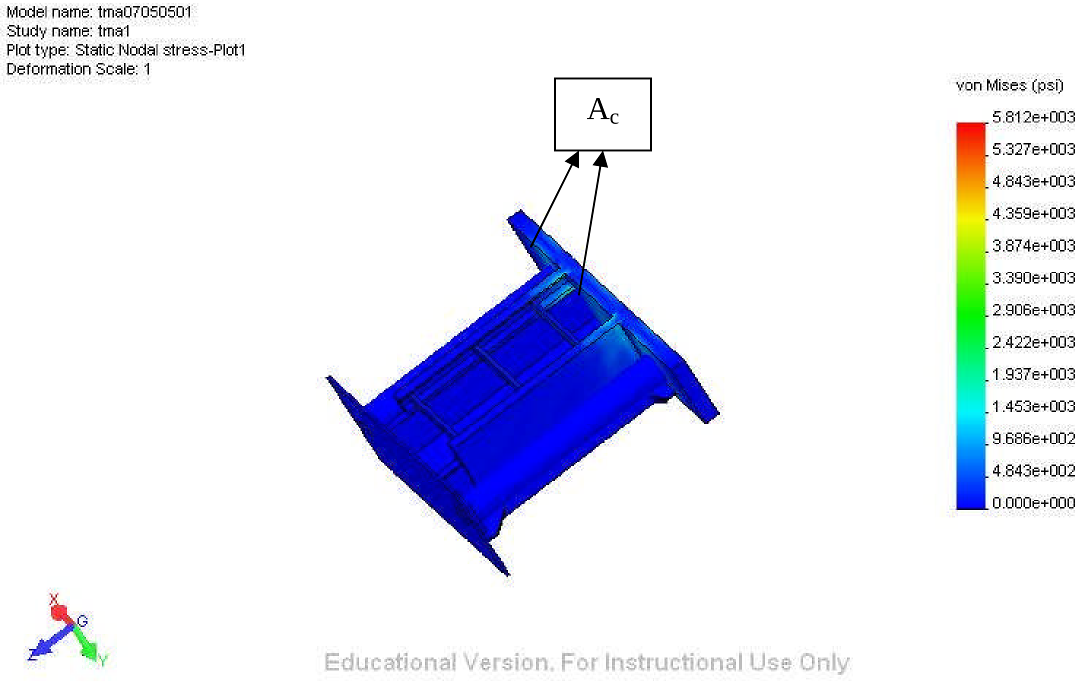
In addition, the effect of the maximum impact force was investigated.

Figure 7 demonstrates the results of FEM simulation for the maximum force of 30,000 lb. It can be seen from Figure 7(a), the length of the retraction is 24 in. The maximum stress is 5,812 psi at A_c, as shown in Figure 7(b). The maximum strain is 0.00013 in/in at A_d, as shown in Figure 7(c).

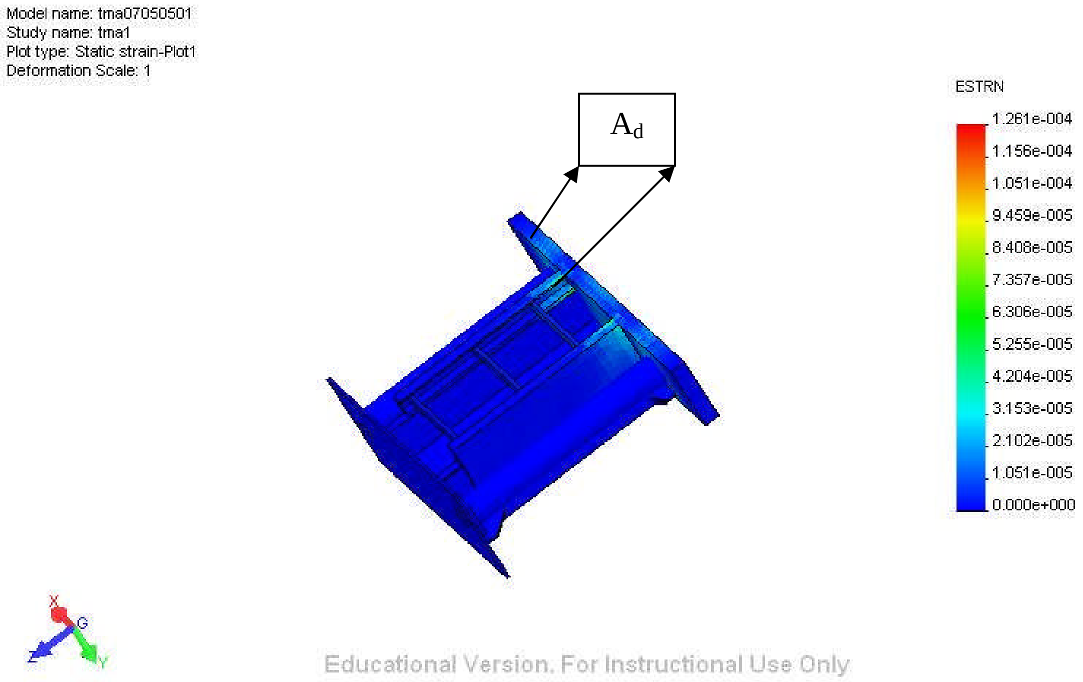
Through these studies, we recognized that the maximum impact force the TMA absorbed was 30,000 lb.



(a) Rod traveling distance



(b) Stress



(C) Strain

Figure 7. FEM analysis for the impact force of 30,000 lb.

Stress Analysis

In these FEM analyses, we could see where “the weakest parts” were and how much “allowable stress” could be sustained by our designed TMA. Due to the highest impact force of 30,000 lb, the factor of safety, $F.S.$ = 2.5, was used in the simulations. The maximum allowable stress is:

$$\sigma_{allow} = \frac{\sigma_{fail}}{F.S.} \tag{2}$$

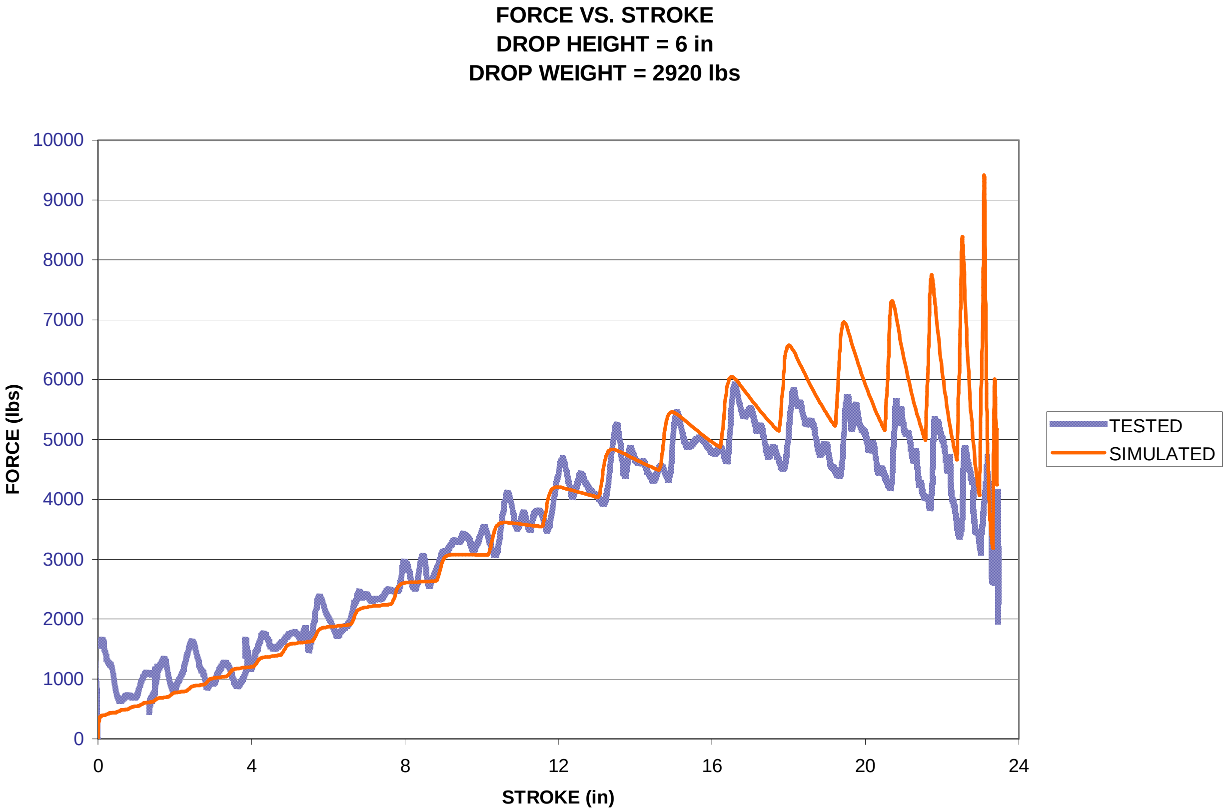
where σ_{allow} is the allowable stress, σ_{fail} is the failure stress, $F.S.$ is the factor of safety.

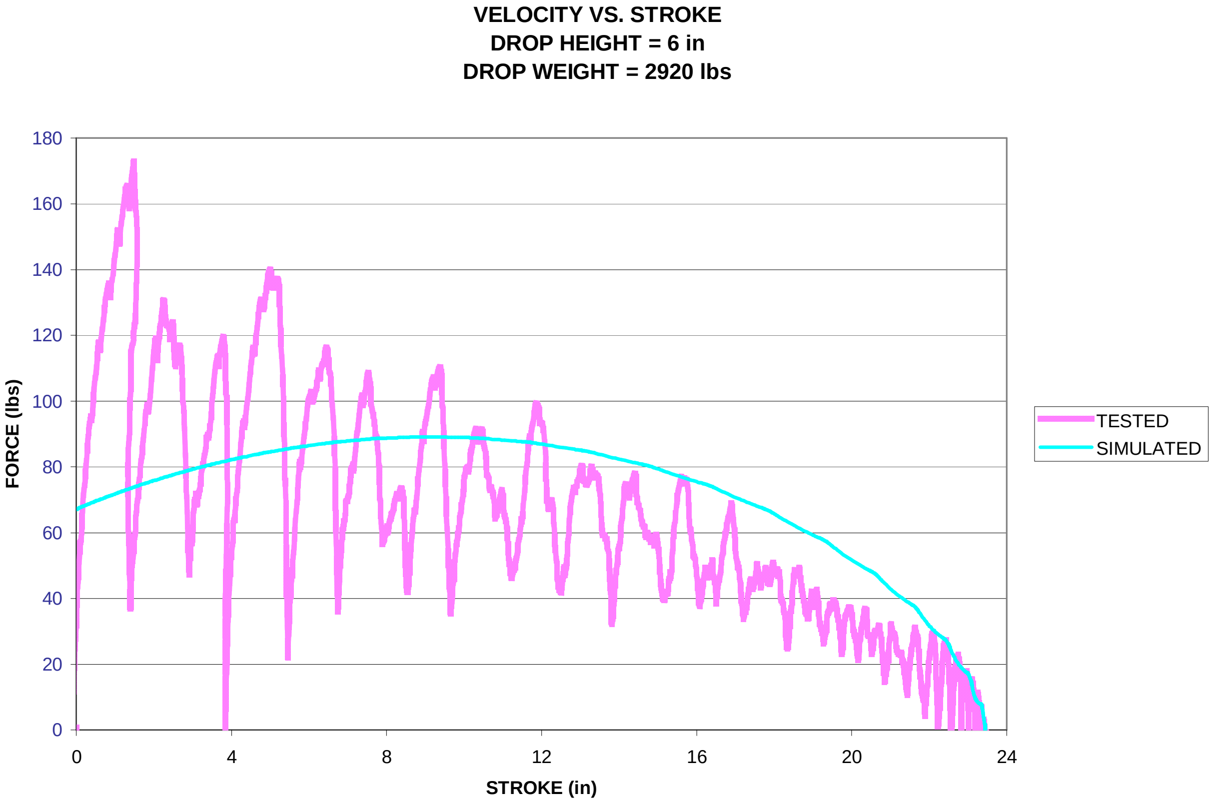
Therefore, according to Eq. (2), the maximum allowable stress was calculated as 11.6 x 10^6 psi. The highest stresses and deformations were located at A_a, A_b, A_c, and A_b in Figure 7(b), 7(c), 6(b), and 6(c), respectively. As soon as the highest stresses in the Figure 6 (b) and 7(b) go beyond the yield strength of 11.6 x 10^6 psi., permanent deformation may occur.

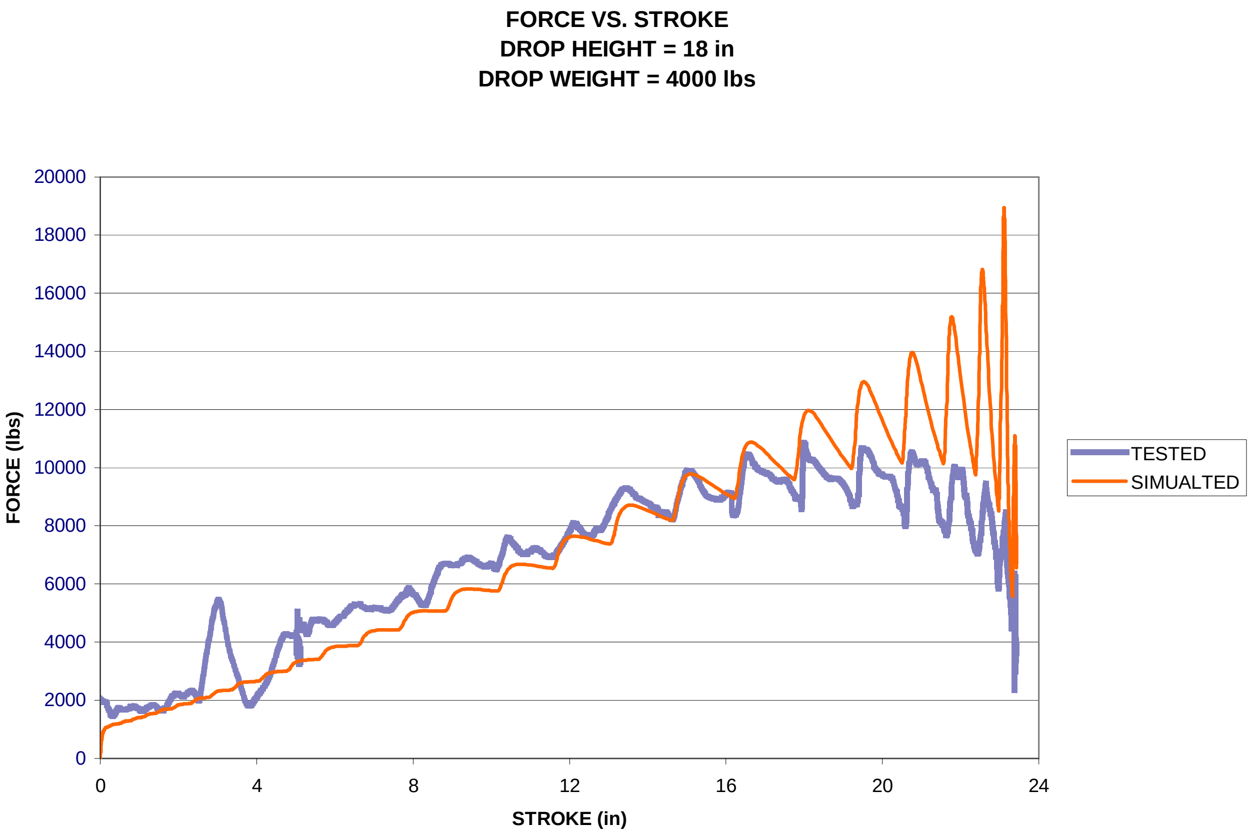
Conclusion

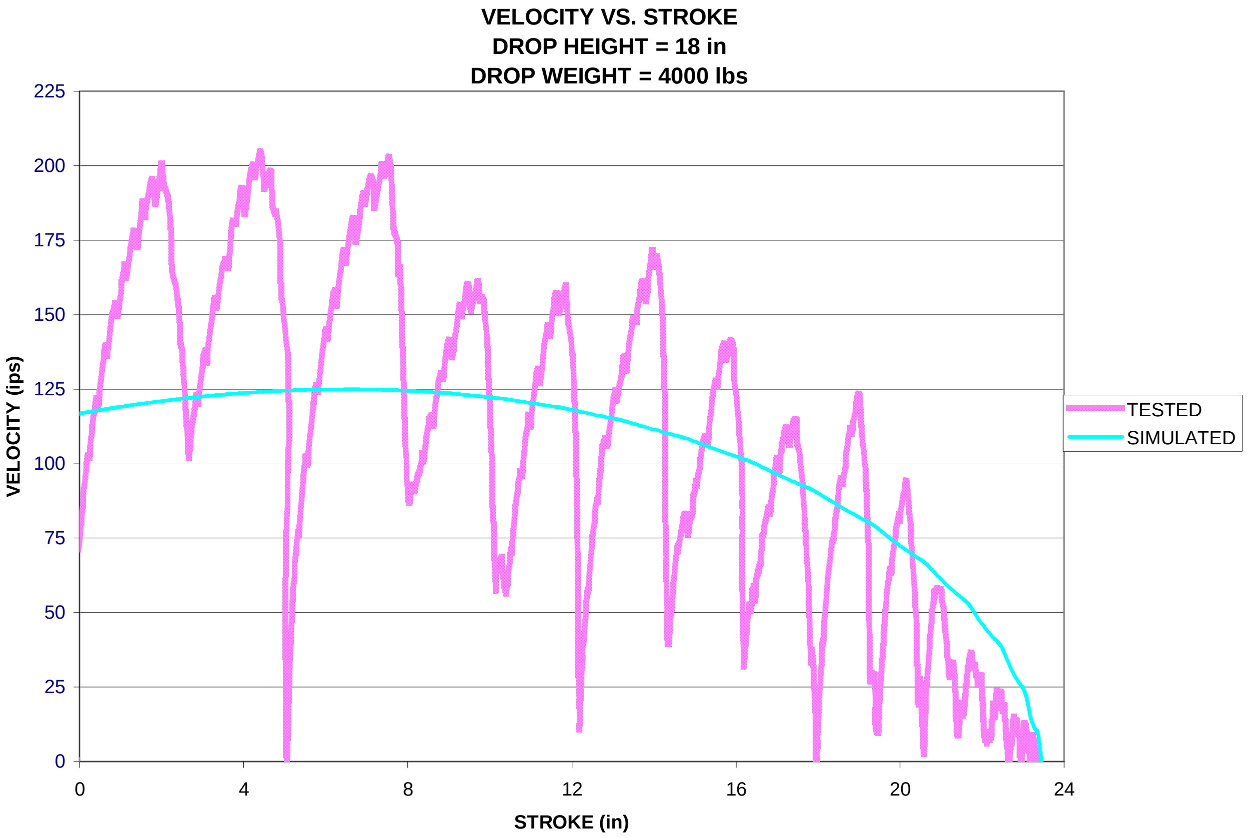
The FEM simulation study of the scaled model TMA was performed and demonstrated that the system could absorb the impact energy prescribed by NCHRP 350 report. . The impact was mathematically modeled, and the resulting stresses and displacements determined. The net result was that, for the scale unit, there were no anticipated weak points or expected failure locations.

APPENDIX G TEST RESULTS & SIMULATION DATA FOR 6", 18", AND 47" DROP HEIGHTS

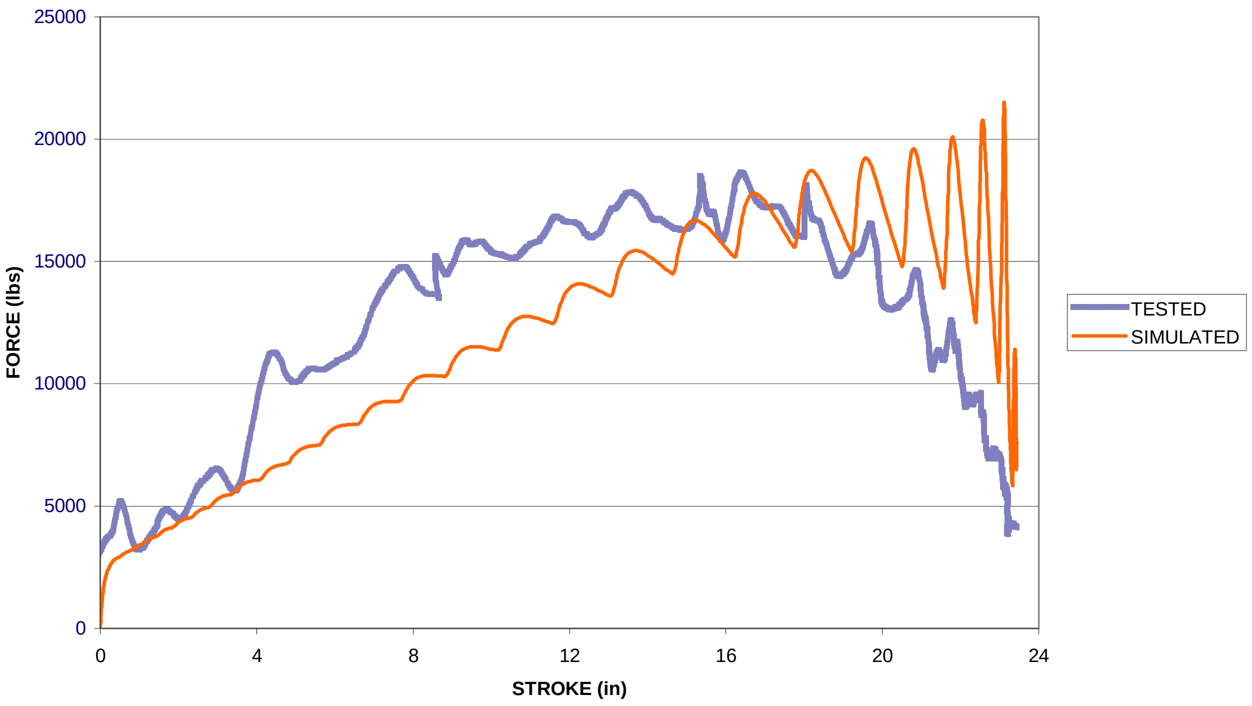





FORCE VS. STROKE
DROP HEIGHT = 18 in
DROP WEIGHT = 4000 lbs


VELOCITY VS. STROKE
DROP HEIGHT = 18 in
DROP WEIGHT = 4000 lbs

FORCE VS. STROKE
DROP HEIGHT = 47 in
DROP WEIGHT = 4000 lbs



VELOCITY VS. STROKE
DROP HEIGHT = 47 in
DROP WEIGHT = 4000 lbs