

Reducing Power Demand for Heavy Suspension Tests

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ABSTRACT

Competitive pressures, globalization of markets, and integration of new materials and technologies into heavy vehicle suspension systems have increased demand for durability validation of new designs. Traditional Proving Ground and on-road testing for suspension development have the limitations of extremely long test times, poor repeatability and the corresponding difficulty in getting good engineering level data on failures. This test approach requires a complete vehicle driven continuously over severe Proving Ground events for extended periods. Such tests are not only time consuming but also costly in terms of equipment, maintenance, personnel, and fuel. Ideally multiple samples must be tested to accumulate equivalent millions of kilometers of operation in highly damaging environments. There is a desire to move these tests into the laboratory using servo-hydraulic testing techniques in a fixed reaction configuration similar to those currently applied universally to passenger car and light truck development. The drawback of applying these tests in the laboratory is the extremely high power demanded by full force, real time operation. This paper will discuss some techniques whereby the operating data can be edited to reduce the power demands of the test systems to more practical levels while still providing valid durability evaluation and performance verification for heavy vehicle suspensions.

INTRODUCTION

The heavy (GVW >15 tons) truck market amounted to >850,000 units bringing in revenue of \$130 billion in 2005. If heavy trailers are also considered that puts heavy vehicle axle and suspension sales volume at well over 4 million units per year. With globalization and consolidation the heavy vehicle industry has now reached the same point (in terms of business and market situation) that the passenger car industry reached 20 years ago. In order to stay competitive while maintaining cost goals and achieving performance requirements new suspensions must be developed more

quickly and with increased reliability and performance predictability than in the past.

Additional challenges for heavy vehicle development include the extreme loading environment combined with high mileage accumulation and the intolerance for structural failures, especially in suspensions. Considering that typical vehicles accumulate 200k to 300k km per year with life expectations of 5 to 10 years the structural reliability target quickly exceeds a million km.

Developmental level tests were run in the past on proving grounds and open roads with typical tests taking 3 to 12 months. These tests have poor repeatability and require a complete operating vehicle (with fuel and driver). The cost of fuel, driver, and maintenance are quite high with such an approach and getting useful engineering level data from which to make design decisions can take several years if several design iterations are to be tested with good confidence.

Between globalization and the opening up of Eastern European markets (with higher load limits and lower quality roads) there is high pressure to introduce new suspension designs. There is also a push to reduce damage to roads and provide better ride quality, leading to greater use of pneumatic suspensions with hydraulic dampers, more exotic materials, etc..

This has led the industry to seek quicker, more reliable methods for suspension system development. These new methods involve moving testing from the road to the laboratory where conditions can be better monitored and controlled with reduced cost and quicker results.

Manufacturers have traditionally used quasi-static uni-axial fatigue and durability tests in the lab with some success.[1,2] These tests have not been suitable for complete suspension development, however, due to the multiaxial loading environment encountered in real use. As suspensions have gotten lower and lighter with incorporation of modern air springs and hydraulic dampers the loading environment for individual

suspension components has gotten more complex and uni-axial tests no longer provide good correlation to real life.[3,4]

This drives the heavy vehicle industry to seek more complex, multi-channel road load simulation capability in the laboratory environment. There are several significant challenges to applying this technology to heavy vehicles. The test system requirements include large force capability with high velocity and acceleration. It is not uncommon for a heavy vehicle suspension to respond to road events with vertical responses achieving mid 20g accelerations and 3 to 5 m/s velocity while reacting dynamic loads up to 180 kN or more at each hub. This combination of high force and high velocity creates a very large oil flow demand as shown in the following equation:

$$Q_{(t)} = A\dot{X}_{(t)}$$

$Q_{(t)}$ = Oil flow rate
 A = Piston area
 $\dot{X}_{(t)}$ = Piston velocity

Heavy vehicle road simulation systems can require up to 3500 lpm of hydraulic pump capacity per axle with peak flows up to 5000 lpm during simulation of events such as resonant road (also called washboard or sinus road; see figure 1).

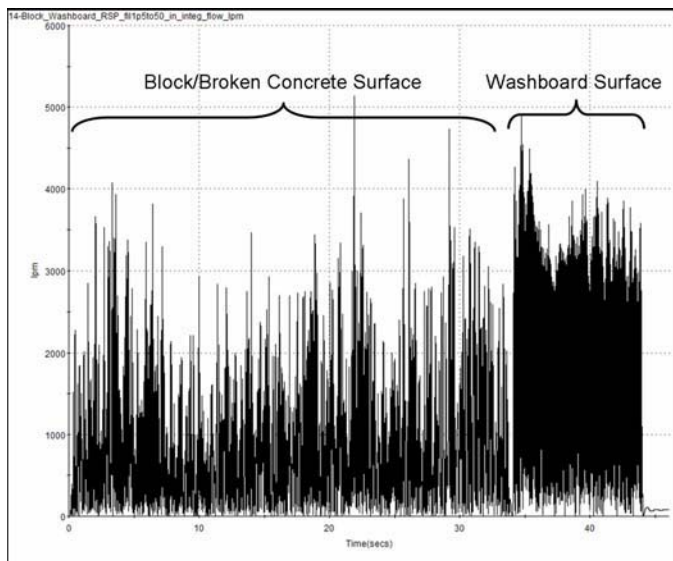


Figure 1: Vertical channel flow demand for one side of an axle on a heavy vehicle road simulator.

Operating costs for such a large hydraulic pumping capacity are substantial, requiring 1.5 megawatts of electrical power and 1500 lpm of cooling water (at 27C). Of course there are additional demands including the space required for such a large pumping system, electrical transformers to handle such high power demands, large diameter hardline, and ongoing maintenance costs, to name a few.

Therefore it is desirable to reduce the overall hydraulic power demand of heavy vehicle suspension testing systems. This can be accomplished in 3 different ways:

- 1) Reduce the system force requirement. This allows for smaller piston area and corresponding lower oil consumption.
- 2) Reduce the operating velocity. This also reduces oil flow demand as mentioned previously.
- 3) Use hydraulic accumulation to provide peak oil flow requirements, thereby reducing the required volume of hydraulic pump.

Low frequency, high load events are common during normal operation of heavy vehicles. Maneuvering into loading docks, positioning near tank fills at tight petrol stations, traversing off-road terrain, sustained cornering, and long hauls with heavy loads all provide significant loading of the vehicle and suspension.

Drive torque can generate significant loads in the suspension members, both creating a mean stress that effects fatigue life but also contributing cyclic loading during shifting and rough road operation. Cornering forces can also be very high, especially on trailer axles during low speed maneuvering.

These maneuvering related forces occur at very low frequency realizing several meters or more of horizontal displacement. Since low frequency high load inputs are so critical to structural durability of the axle assembly and suspension it is preferred to mount the suspension to a grounded reaction fixture for testing. This allows real time application of low frequency maneuvering events in conjunction with high frequency rough road type dynamic loads.

Since body dynamic response is not present using fixed reaction type testing the input frequency, acceleration, velocity, etc. may be changed if these changes do not adversely affect structural durability. Free body type tests must utilize real time data or resulting body motion will not be correct. Using a fixed reaction approach presents an opportunity to modify the test rig input commands to maximize test throughput and minimize overall power requirements while allowing simulation of a broad spectrum of events.

To reduce the system oil flow demand the operating force (piston area) and/or velocity must be reduced. In order to do this there must be an understanding of how the suspension system responds to road disturbances and how this relates to structural durability. The road surface provides a time varying multiaxial displacement input to the vehicle tire contact patch, causing it to move. This motion generates reaction forces within the suspension based on the preload, moving mass and mass distribution, suspension spring, bushing, and component stiffness, as well as damping of the various elements.

The basic force equation is:

$$F_{(t)} = M\ddot{X}_{(t)} + B\dot{X}_{(t)} + KX_{(t)} + \text{Preload}$$

$F_{(t)}$ = Force at location of interest

$X_{(t)}$ = Relative displacement at point of interest

M = Moving mass of specimen and loading fixture

B = Damping coefficient

K = Spring, load train, or component stiffness

DISPLACEMENT EFFECTS ON STRUCTURAL DURABILITY

Axle displacement is resisted by spring elements (suspension spring, bending of axle housing, compression of restraining links and vehicle frame, etc.) which results in forces and corresponding stresses and damage. These spring forces are not dependent on the velocity at which they are applied unless there is significant damping in the load path. Therefore the peak forces and stresses in spring elements are primarily a function of the displacement itself and are independent of changes in velocity, acceleration, or frequency of loading.

It should be noted that many suspension members see simultaneous loading from multiple directions (for example, vertical, lateral, and longitudinal inputs simultaneously create a net bending moment on the axle housing). This means that multiaxial inputs are required in order to generate the proper forces, stresses, and resulting damage in each member with good phase and amplitude integrity compared to real world operation.[3]

Vertical suspension spring rates for heavy vehicles generally range from 300 to 600 N/mm. Typical suspension vertical displacement during rough road driving ranges from 60 to 120 mm, giving maximum spring forces from 20 to 70 kN.

Spring forces are function of displacement/deflection and provide significant contribution to fatigue damage. This means that axle and suspension displacements should closely match those induced during on-road operation

VELOCITY EFFECTS ON STRUCTURAL DURABILITY

Axle velocity creates loads at damping elements within the suspension. Two types of damping occurs in suspensions, viscous and coulomb damping. Viscous damping creates forces in proportion to the relative velocity. Viscous type damping elements include the tire itself, rubber/elastomer bushings, hydraulic dampers (shock absorbers), and airsprings (due to the rubber 'bag', although this damping is very small and is typically neglected). Coulomb damping causes force in proportion to the sign of the velocity and is generally not

rate dependent. Coulomb damping elements include leaf springs and ball or pivot joints.

Typical damping forces range from 4 to 20 kN during normal rough road operation. This is about 5-25% of spring force so not nearly as significant in terms of maximum loading. The shock absorber does not share load paths with the suspension spring. Since the velocity inputs do not generate as significant of forces as other elements (except as they may effect the shock absorber itself and its mounts, which can easily be tested at the component level) they are of little interest in terms overall suspension durability reproduction accuracy.

Therefore the axle velocity can be changed if needed to reduce power demand. If shock absorber forces are critical to the overall test quality then a modified damper or active element (such as a hydraulic actuator) can be substituted to achieve the same force levels as with on-road operation even while operating at lower velocity. Another option is to use a dedicated test for shock absorber mounts. This type of component level test can be run at an accelerated rate compared to the full suspension test, increasing test efficiency.[1]

ACCELERATION EFFECTS ON STRUCTURAL DURABILITY

Axle acceleration generates inertial loads proportional to the amount of mass being accelerated. Accelerating the parts themselves does not create significant internal damage, rather the damage occurs on parts that make up the load path between the input and the moved mass. Therefore the effect of this inertial loading is dependent on the mass distribution and maximum acceleration.

Drive axles can weight up to 1.5 tons and axle accelerations can be 25g or more so the resulting inertial forces can be significant, and in fact often result in the highest forces the hub transmits to the axle (note that 25 g is the combined acceleration from heave and tramp. When separated the maximum accelerations induced by either condition alone is typically <20g). Lighter axles such a truck front steering axles or trailer axles may achieve significantly more acceleration (to 35 g or more) but have less moving mass so do not generate as high a level of inertial forces as drive axles.

Drive Axle Acceleration

Take an example of a 1.2 ton drive axle (600 kg/side not including tires and wheels) accelerating to 25g. This generates inertial loading of 150 kN, significantly more than the spring forces noted previously. The effect of this force is dependent on the mass and strength distribution of the axle assembly. See the example mass distribution shown in figure 2: