A Rapid Dampening Suspension For Ultra-Class Haulers

R. F. Santos, B.Sc, E.I.T.
M Sc Graduate Student, AEGIS research group, University of Alberta, Edmonton, Canada

T.G. Joseph, Ph D , P.Eng.
Director AEGIS, University of Alberta and Principal Engineer, JPi, Edmonton, Canada

ABSTRACT: In the last decade, the continuously growing demand for production in bulk mining operations such as the Canadian oil sands operations of Northern Alberta has influenced ultra class truck payloads to the extent that 400 tons has been reached, with sights firmly set on the advent of the 500+ ton unit Soft underfoot conditions in unconsolidated deposits such as these have resulted in the transmission of adverse stresses throughout truck frames causing premature failures Poor ground response can ultimately be linked to cyclic fatigue stress and excessive vibration levels affecting both operator and equipment health The focus of this paper is the introduction of a conceptual strut design to mitigate the impact of high loading generated by adverse loading due to soft ground conditions

1 EXECUTIVE SUMMARY

The cyclic response of the oil sand underfoot in response to high g loading contributes to premature failure of existing truck frames and meurs a high rate of operator reported back problems The basic shock absorber or strut design for ultra class trucks was reviewed to facilitate proposal of an alternative solution to mitigate these adverse impacts On-board data was acquired through the original equipment manufacturer (OEM) acquisition system and downloaded for complete round trips to a shovel The pressure changes from these sets were incorporated into a thermodynamic analysis related to the nitrogen gas performance of the shock absorbers

To simplify the analysis, it was shown that negligible impact on the outcome was incurred in treating the nitrogen as an ideal gas, and the compression process as ısentropıc A top clearance inside the shock absorbers was determined for each unit trip, revealing that "topping-up" occurred frequently Vehicle suspension fundamentals, in addition to fluid mechanics and thermodynamic principles were reviewed and applied in a comprehensive shock absorber analysis

Current struts or shock absorbers for ultra class trucks were identified as basically the same design as those used in haulers for the past 40 years, classified as "simple" shock absorbers, containing a fixed orifice and therefore constant damping coefficients for both compression and rebound strokes These were modeled to establish a baseline of current performance A modification was then proposed effecting a semi-active configuration via a variable orifice

A variable orifice allows oil flow through a variable cross sectional area, leading to a variable resistance and variable damping coefficient The cross-section area of flow is a function of the input displacement, proportional to the applied g load on the shock absorber This was modeled to compare the performance of the proposed variable orifice shock absorber against the fixed configuration Several iterations were run at different stroke rates and levels of g loading for both models The results showed that the variable orifice shock absorber creates a dampening proportional to the load and stroke rate applied

2 "TOPPING-UF' PHENOMENON

A simple thermodynamic analysis was performed to determine pressure and volume of the nitrogen gas inside a strut during a given trip cycle Actual pressures were attained via the OEM on-board data acquisition system As a typical example, immediately after shovel loading activity and before initial motion, the rear truck struts of an ultra class unit were evaluated with 3 18 cm of top clearance between the head and casing This provided a base line for the analysis Isentropic compression was assumed, allowing corresponding volumes to be determined at each recorded pressure throughout the duty cycle The result of the analysis showed that there were many points where "topping-up" occurred, illustrated in figure 1 Negative values here are theoretical and actually represent zero top clearance values indicating metal-metal contact, termed "topping -up"

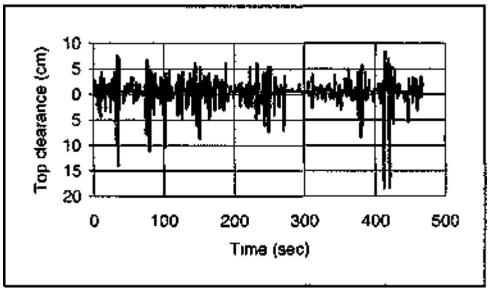

Figure 1 Sample top clearance for strut operation

3 SUSPENSIONS SYSTEMS

Suspension systems may be classified in terms of energy management, such as passive, semi-active or active Passive systems have a constant damping coefficient and stiffness Active systems have the ability to change their characteristics according to the amount of vibration imposed on the system, typically requiring energy input to vary behavior

The principle of active suspension is illustrated in figure 2, where the spring and damper in a passive system are replaced by a force actuator, with operating conditions monitored continuously by a group of sensors The control strategy is to minimize the root mean square (RMS) value of the sprung mass acceleration, suspension travel, and dynamic tire deflection (Wong, 1999)

In order to improve ride characteristics but minimize the complexity that comes with a fully active system, a compromise is the semi active concept illustrated in figure 3 Here the conventional suspension spring is retained, but the damping force in the shock absorber is modulated according to the ride requirements One such simple approach to control the stiffness of the unit is to vary the resistance to oil flow, as adopted by aerospace landing gear By adjusting the orifice area of the shock absorber, the resistance to fluid flow and damping force is varied (Wong, 1999)

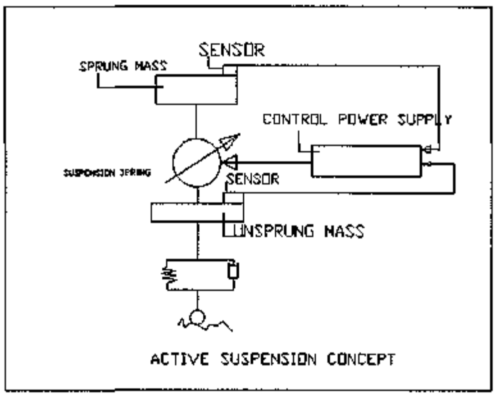

Figure 2 Active suspension concept (after Wong, 1999)

Shock absorbers may also be classified according to the type of spring being used such as solid steel or fluid comprising gas and oil, as m an oleo pneumatic shock absorber, the latter of which has high efficiency under dynamic load conditions m terms of energy absorption and dissipation

The single acting shock absorber is the most common oleo pneumatic design used for ultra class mining trucks, where a fixed diameter orifice effecting a constant damping coefficient is typical of a passive system If the flow cross section area of the orifice is variable commensurate with the applied dynamic load, then the strut could be classified as either an active or semi active suspension The basic components of the simple shock absorber are two telescopic tubes, one functioning as a piston and the other as a cylinder, as illustrated in figure 4 An orifice in the piston head permits fluid to pass from one chamber to the other The fluid flow through the orifice, together with the compression of the gas, absorbs the energy of the load The shock absorber extends when the applied load on it decreases

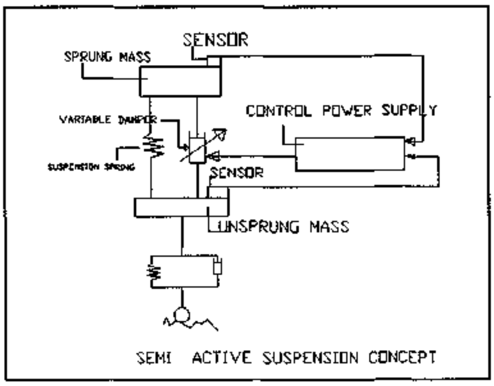

Figure 3 Serra active suspension concept (after Wong 1999)

A complex shock absorber works according to the same principle as the simple system, but has the ability to change the dampening coefficient commensurate with the dynamics of the applied load In order to achieve this goal the orifice size m the shock absorber must change with the load being applied There are many different types of complex shock absorbers such as metering pm, plunger and floating piston, which are descriptors of the variation from the simple system

The metering pm approach, conceptualized in figure 5 and the basis of this investigation, changes the effective size of the orifice and the resulting rate of fluid flow from one chamber to the next The conical shape of the metering pin varies the cross sectional area of the orifice The larger the effective diameter of the pm the greater the resistance to fluid flow, the higher the damping coefficient

3.1 Basic operation of a simple shock absorber

In compression, figure 6, the piston moves up causing gas compression and forcing the oil through both the fixed orifice (Q2) and ball check valve (Ql) to the annular compartment The change in volume for the gas represents the spring action of the shock absorber

The gas spring absorbs the majority of the impact energy Oil flow through the orifice and the ball check valve causes "orifice action" energy dissipation as heat to the surroundings Minor oil flow occurs via the piston-cylinder annular clearance (Q3) causing some additional energy dissipation

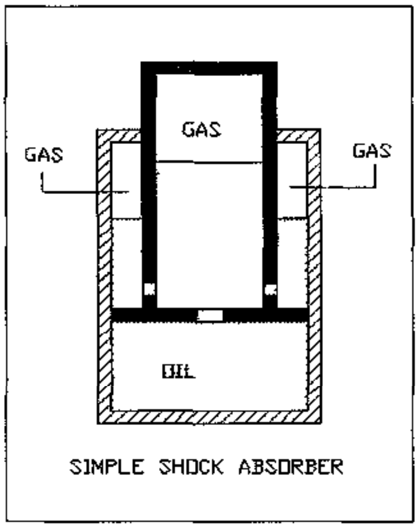

Figure 4 A simple shock absorber

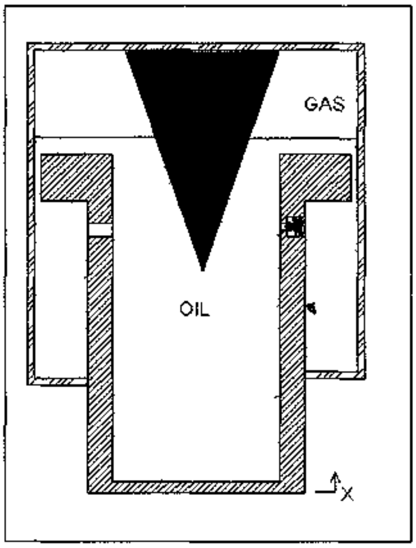

Figure 5 Conceptual metering pm shock absorber

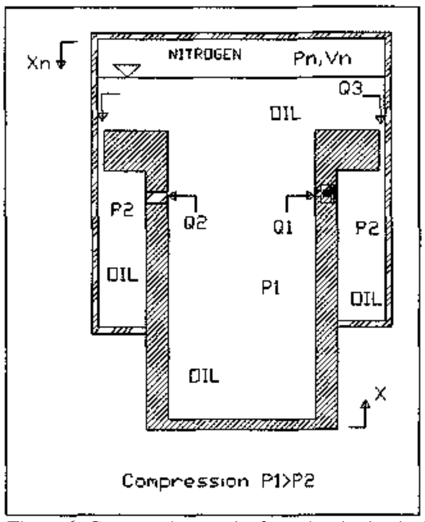

Figure 6 Compression stroke for a simple shock absorber of the type used by ultra class trucks

In rebound, figure 7, the gas tries to regain its original volume and reach equilibrium with the truck weight The gas expands pushing the piston down and the oil through the fixed orifice only, producing the mam damping effect of the shock absorber As the piston moves down, the pressure in the annular compartment increases causing the ball check valve to close leaving the only opening available for oil flow through the fixed orifice

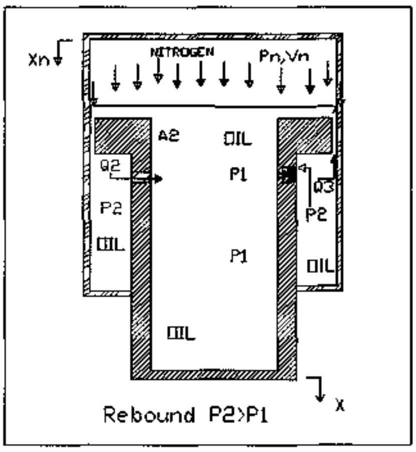

Figure 7 Rebound stroke for a simple shock absorber

4 THERMODYNAMIC CONSIDERATIONS

The gas inside an oleo-pneumatic shock absorber is effectively the system spring The system can be simplified assuming the gas obeys the ideal gas laws under ısentropıc compression, so that

$$P_1 \times V_1^k = P_2 \times V_{21}^k \quad [1]$$

where,

P= Pressure
V= Volume of chamber
k = Specific heat ratio

5 FLUID MECHANICS CONSIDERATIONS

5.1 Flow across an orifice

When fluid flow (Q) passes through an orifice, it loses energy or pressure head The pressure drop is proportional to the orifice dimension and shape Applying Bernoulli's equation for an incompressible fluid, and the principle of flow continuity between any two points, it can be shown that

$$Q_{Actual} = C_d\ Q_{Ideal} = C_d\ A\ v \quad [2]$$

Where the velocity of flow v may be determined via

$$v = [2\,(P_2-P_1)/[\rho\,(1-(D_1/D_2)^4)]^{1/2} \quad [3]$$

where,

A = Cross sectional area of the orifice
Cd = Orifice flow coefficient

5.2 Flow through annular clearance (assuming laminar flow)

Generally, the flow through an annular clearance is given by equation [4]

$Q = AP/R_L$ [4]

Where the hydraulic resistance RL may be determined via equation [5]

$R_L = (24pv\ L)/(7i5^3(D„+D,))$ [5]

Where,

D_0 = Outer diameter of the inner tube
D, = Inner diameter of the outer tube
8 = Annular clearance between the tubes
p = Density
v = Kinematic viscosity

5.3 Damping Force

The damping force, equation 8, of the ultra class truck is a combination of the gas-spring force, equation 6, and the hydraulic damping force, equation 7, caused by the oil.

$$F_{gas-spring} = P_{gas} \times A \quad [6]$$

$$F_{hydraulic} = R_L A^2 (vel) \quad [7]$$

Yielding equation 8:

$$F_{damping} = F_{gas-spring} + F_{hydraulic}$$

6 MODELING AN EXISTING ULTRA CLASS TRUCK SIMPLE SHOCK ABSORBER

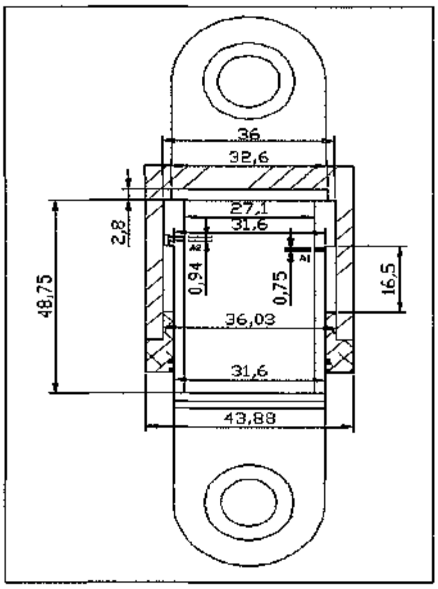

Figure 8 OEM developed shock absorber for an ultra class truck

The analysis of an existing OEM ultra class truck shock absorber was modeled after the example in figure 8. Matlab and Microsoft Excel, along with the formulae above, were used to create the base line model. Figure 9 represents the corresponding SIMULINK model developed in Matlab by El-Sayed (2003) and which was used as the basic strategy for the spreadsheet model used in this analysis.

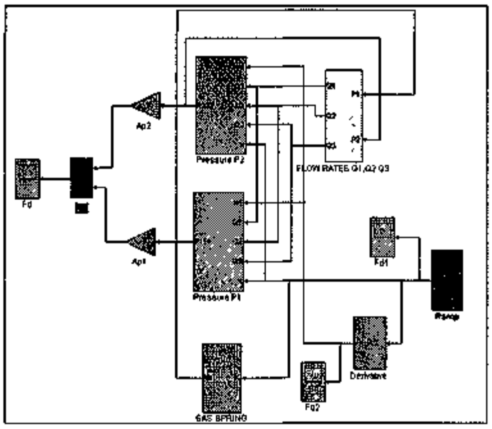

Figure 9 SIMULINK Model of CAT 797B shock absorber, (after El-Sayed, 2003)

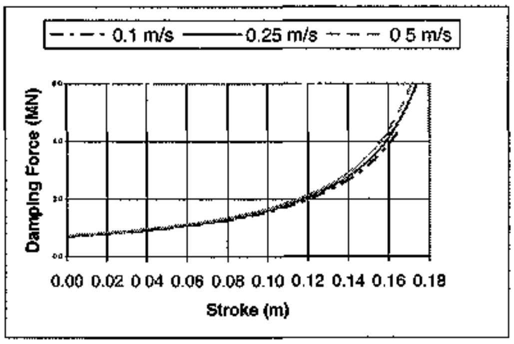

Figure 10 Damping force for a simple shock absorber

Figure 10 shows the base line performance for the OEM shock absorber. The slope of the line shows the stiffness of the unit. For the first 0.09 m stroke the line is slightly sloping up. This is representative of a shock absorber with a fixed orifice. As the stroke exceeds 0.09m the trend increases rapidly due to the damping force of the gas-spring. Since the damping force increases late into the stroke, the chance of topping-up is greatly increased.

7 MODELING THE VARIABLE ORIFICE SHOCK ABSORBER

For the conceptual shock absorber an invented cone was added as shown in figure 5. As the piston moves up, the corresponding opening gets smaller due to cone penetration. The cone increases the flow resistance to the oil, producing damping additional to the original orifice and ball check valve configuration. This additional control of oil flow increases the overall damping characteristics of the shock absorber. The conceptual variable orifice shock absorber was modeled in order to determine the difference in performance to the fixed orifice design currently in use. A sensitivity study has been performed in order to determine the effect of several design parameters on the shock absorber performance.

Modeling was once again performed in Microsft Excel following the interative strategy suggested by El-Sayed (2003) in Matlab Simulink, figure 11.

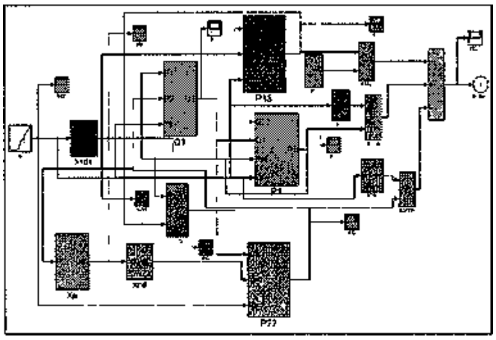

Figure 11 SIMULINK model of the variable orifice shock absorber (after El-Sayed, 2003)

Simulation results are shown in figure 12, as damping force generated inside the shock absorber versus the rate of input displacement. The slope of the curve represents the shock absorber stiffness. The slope steepens as the input velocity increases.

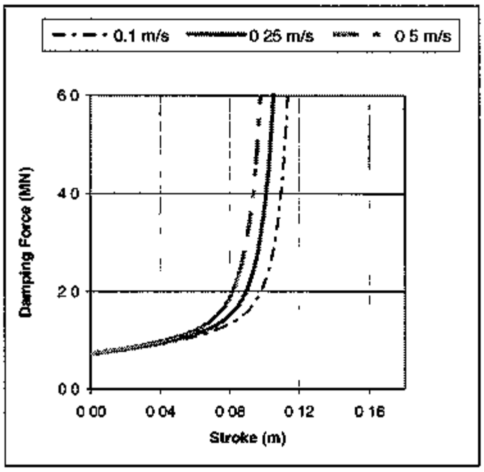

Figure 12 Damping Force versus input velocities for variable orifice shock absorber

8 DISCUSSION AND CONCLUSIONS

The computer simulation showed the capability of a variable orifice shock absorber to provide a rapidly increasing dampmg force to prevent topping-up , as a function of the applied load.

Figures 13 through 15 compare the existing OEM shock absorber to the modified one at various velocity inputs. The output showed that the modified shock absorber generates a greater amount of damping force compared to the existing design. This rapid increase in damping force, will prevent the shock absorber topping up.

As the velocity input increases, the dynamic response of the variable compared to the fixed unit is more sensitive to impact loads and more likely to dampen a corresponding high g effect. This is represented by rapid damping occurring earlier in the stroke with increasing velocity. The stiffness of the variable orifice unit varies dramatically as the load increases.

Physical testing is planned to verify the modeled conclusions on a full scale suspension system during summer 2005. Field verification is also planned to determine the improved ride quality due to the conceptual unit. Further research will investigate how the conceptual shock absorber affects the rebound phase of the cycle.

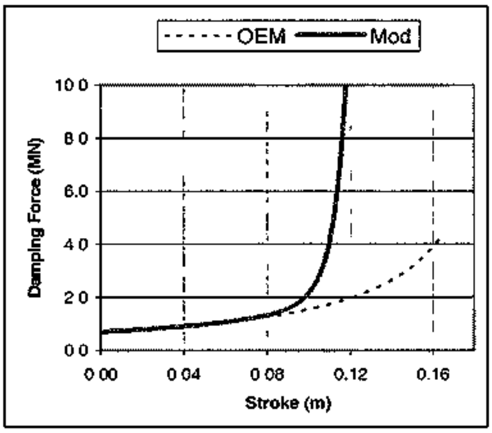

Figure 13 Damping force at 0.1 m/s

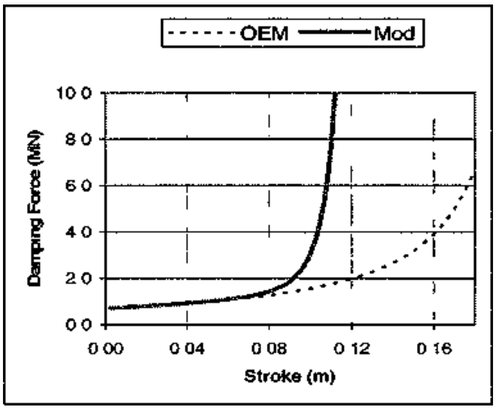

Figure 14 Damping force at 0 25 m/s

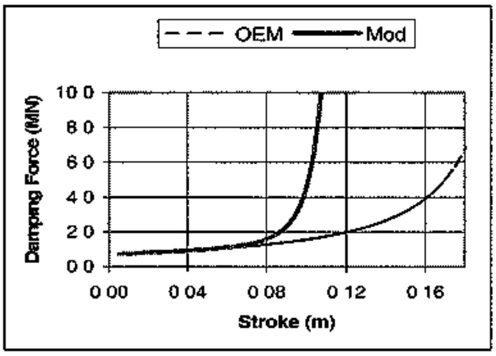

Figure 15 Damping force at 0 5 m/s

REFERENCES

Caterpillar Inc 2002 Specifications Manual from SENR 1540, Specifications, SENR 1455-04 *Caterpillar Service Manual for 793C Truck*

El Sayed, M, 2003 *A new Concept strut design for heavy haulers to combat high g loading* M Eng Thesis, University of Alberta

Kroeell, Dulay, Fueresz, Harkay, & Lukacs 1988, *Fundamentals of hydraulic power tiansmission* Elsevier Science Publishers, Amsterdam

Pinches, M J & Ashby, J G 1990, *Power hydraulics* Prentice Hall,Hertfordshire

Wong, J Y (2^{od})1993, *Theory of Ground Vehicles* John Wiley & Sons, Inc New York