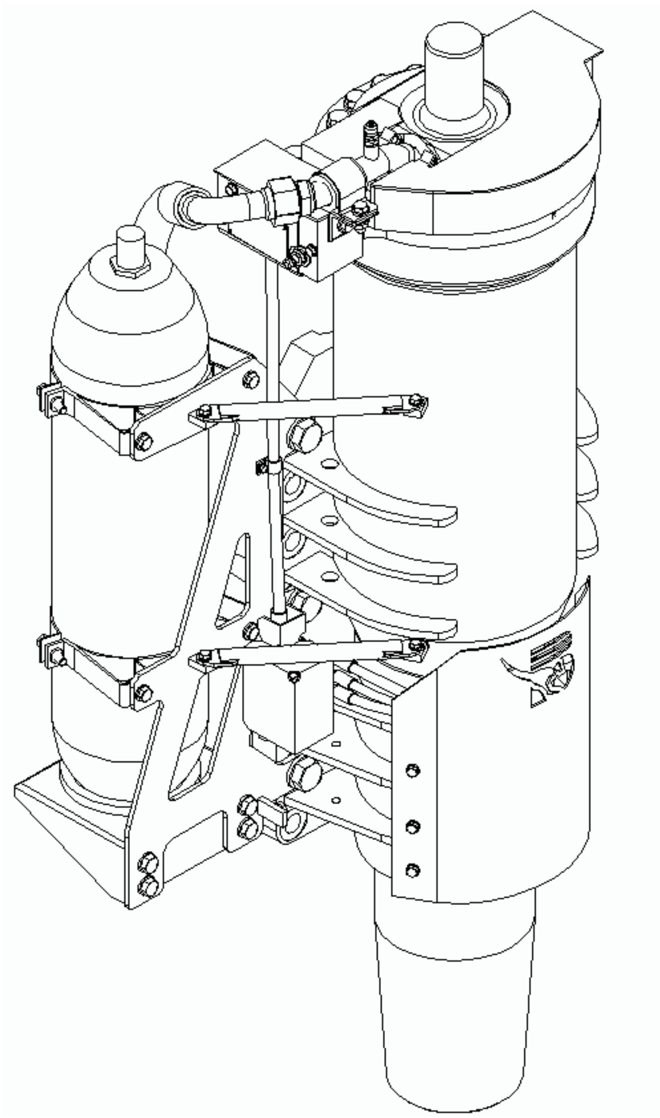
BIRRANA SMART STRUT

TECHNICAL PAPER

The Development of the Smart Strut Improved Sliding Pillar Front Active Suspension System for Mining Trucks



Author: Paul van de Loo, BE (Hons), MSME (Stanford)

Birrana Engineering Pty Ltd

Innovative drive line solutions for the mining industry

The development of the Smart Strut™ improved sliding pillar front active suspension system for mining trucks

1. Abstract

The ride performance of mining dump trucks and haulers has long been recognised as problematic with regard to occupational health and safety. The rough ride, even on relatively good haul roads, can lead to spinal injuries to vehicle operators. Poor suspension performance is also a major factor in the deterioration of haul roads and the fatigue of truck chassis. It also results in increased tyre loading and reduced tyre life – a significant factor, as tyres represent a large proportion of operating costs, second only to fuel.

The development of the active Smart Strut™ system, which provides a major improvement in dumper and hauler “sliding pillar” style front suspension performance, is described. It is a cost effective replacement strut for vehicles in service, or ideally would be fitted to new trucks by the manufacturer.

2. Introduction

The predominant front suspension design for large mining trucks is the sliding pillar strut, and whilst this is cheap to manufacture and durable in service, it has shortcomings in providing an effective suspension system. These shortcomings manifest themselves in rough ride, and high tyre and chassis loadings. The high tyre loadings lead to accelerated tyre wear and degradation of mine roads. The high chassis loadings lead to chassis fatigue and reduced vehicle life. This problem and the potential to improve the sliding pillar design were recognised by the Australian mining engineering equipment company, Birrana Engineering. Birrana specialise in retrofit engineering improvements, primarily to Caterpillar* mining machinery. In partnership with design engineers Applidyne, Birrana set out to investigate potential design improvements to these suspension systems. The program to develop an improved dumper and hauler suspension system commenced in 1997. The early parts of the program were assisted by funding from the Australian Coal Association Research Program (ACARP).

The development program focussed on the CAT* 776 hauler and 777 dumper and commenced with testing to identify the deficiencies of the sliding pillar strut. The test program involved instrumentation of a strut with accelerometers and a displacement transducer and sought to identify the cause of the characteristic “loping” motion of the vehicle as well as identify the response and performance of the suspension under various conditions.

It was suspected that the influence of strut friction, coupled with the moment associated with the wheel offset from the sliding pillar axis, was a key factor in the performance deficiencies. It was thought that high friction levels would lead to "friction locking" of the strut allowing the relatively undamped tyre bounce vibration mode to become dominant, and would result in the pitching or "loping" motion commonly observed.

Testing on an instrumented dumper strut indicated that strut locking did occur in practice. Experiments with low friction strut bearing materials followed, but the results were not as promising as expected. A quarter vehicle computer model was then developed in an effort to better understand the suspension dynamics. The model confirmed that friction was the major problem, but indicated that it needed to be reduced to well below that provided by low friction bushes to obtain significant performance improvements.

The computer model was then extended to investigate a number of other modifications, including semi active and active systems. Modelling resulted in a preferred design being identified. The analysis, design, development and testing of this slow-active (sometimes termed a “fast load leveller”) Smart Strut suspension system is covered in this paper.

3. Standard suspension

3.1 Suspension configuration

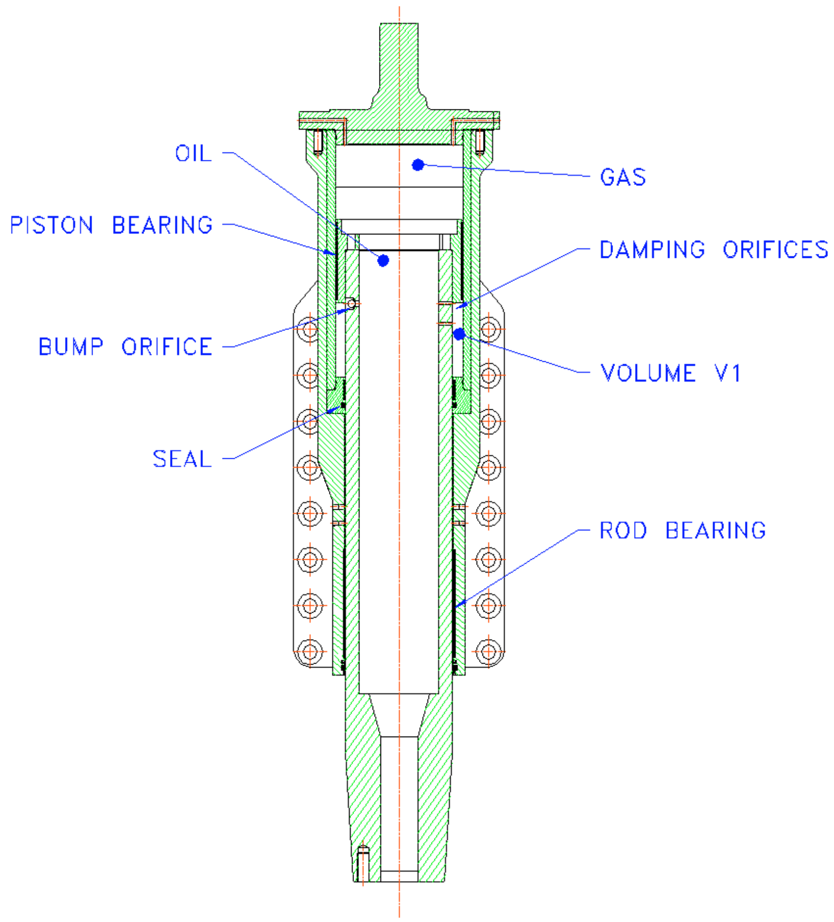
The suspension on dumpers and haulers typically consists of oleopneumatic sliding pillar struts at the front and suspension struts at the rear with the live rear axle located by a Panhard rod and a V-shaped trailing arm mounted to the chassis by a ball joint.

Whilst the focus of this study was on the CAT* 776 hauler and 777 dumper, the suspension on these vehicles is identical in configuration to most other dumpers and haulers, including the larger CAT* trucks and many of the popular Komatsu models. The main factors resulting in poor suspension performance is the sensitivity to friction of the sliding pillar configuration, and the large variations between laden and unladen weight. When unladen the CAT* 777 weight distribution is 47% front and 53% rear. When laden the weight distribution is 33% front and 67% rear. As the rear has dual tyres at each side this results in uniform tyre loading. When laden, the weight on each front tyre increases from 13.7 tonnes to 24 tonnes.

The oleopneumatic sliding pillar design, as used on the CAT* 777 and 776 is shown in Figure 1. Fundamentally this consists of a hydraulic cylinder with a nitrogen "spring" reservoir over oil which resides above the strut piston, and a stub axle protruding from the cylinder rod to carry the wheel. The oil flow is constrained by damping orifices, the size of which varies between bump and rebound due to the incorporation of a check valve on one orifice. This provides rebound damping which is greater than bump damping. It is important to note that the strut configuration can result in cavitation in the bump orifice, due to oil flow into the expanding volume below the piston bearing (volume marked V1 in Figure 1) being limited by the damping orifice connecting this to the volume above the bearing. The use of special antifoaming oils helps counteract this problem but cavitation significantly degrades damping performance. The rod and piston bearings carry the radial forces required to counteract the moment of the wheel load offset from the strut axis. The steering axis is provided by rotation of the strut rod within the strut housing or cylinder. On the 776 and 777 the strut is inclined at a castor angle of

3 degrees and a camber of 3.5 degrees. The nitrogen volume is relatively small at approximately 7.5 moles, to ensure that the strut does not bottom or "top out" over bumps over the entire load range, given the limited total strut travel of 320mm. This results in a fairly high spring rate which, it should be noted, is not constant. As the spring is a gas volume, the rate rises with pressure (ie strut load). As will be appreciated the front suspension ride height varies between laden and unladen states. Due to the simple geometry of the steering drag link which ties the struts together, there is an appreciable variation of front wheel toe angle from laden to unladen. This obviously involves a compromise; it is not possible to maintain the optimal toe angle for handling response and tyre wear at both load conditions.

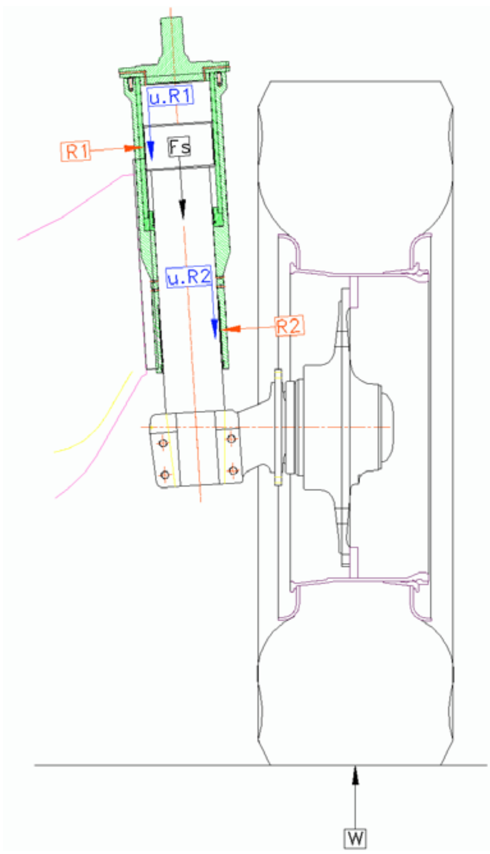
Figure 1 – The oleopneumatic sliding pillar strut



3.2 Performance

Figure 2 shows the force vectors in the strut. The friction forces associated with the strut bearing reaction forces R1 and R2 tend to resist strut motion and if large enough can result in "strut locking". Testing of the strut indicated that it is common for the struts to remain locked over bumps. Figure 3 shows a trace of sprung and unsprung mass accelerations and strut displacement measured on a CAT* 777 dumper which shows extended periods of locking. This results in the tyres becoming the main suspension element. Tyres have a low damping component to their compliance, and this results in the front of the truck oscillating in a pitching mode termed the "tyre bounce" or "loping" mode. In the 777 this resonant frequency is approximately 1.6 Hz when laden and 2.4 Hz when unladen [7]. Birrana Engineering identified this fundamental problem with strut friction and collected suspension strut test data which showed that the strut could lock for a large percentage of the time. Early trials with modified struts using low friction bearing materials offered little improvement. These are discussed in reference [6].

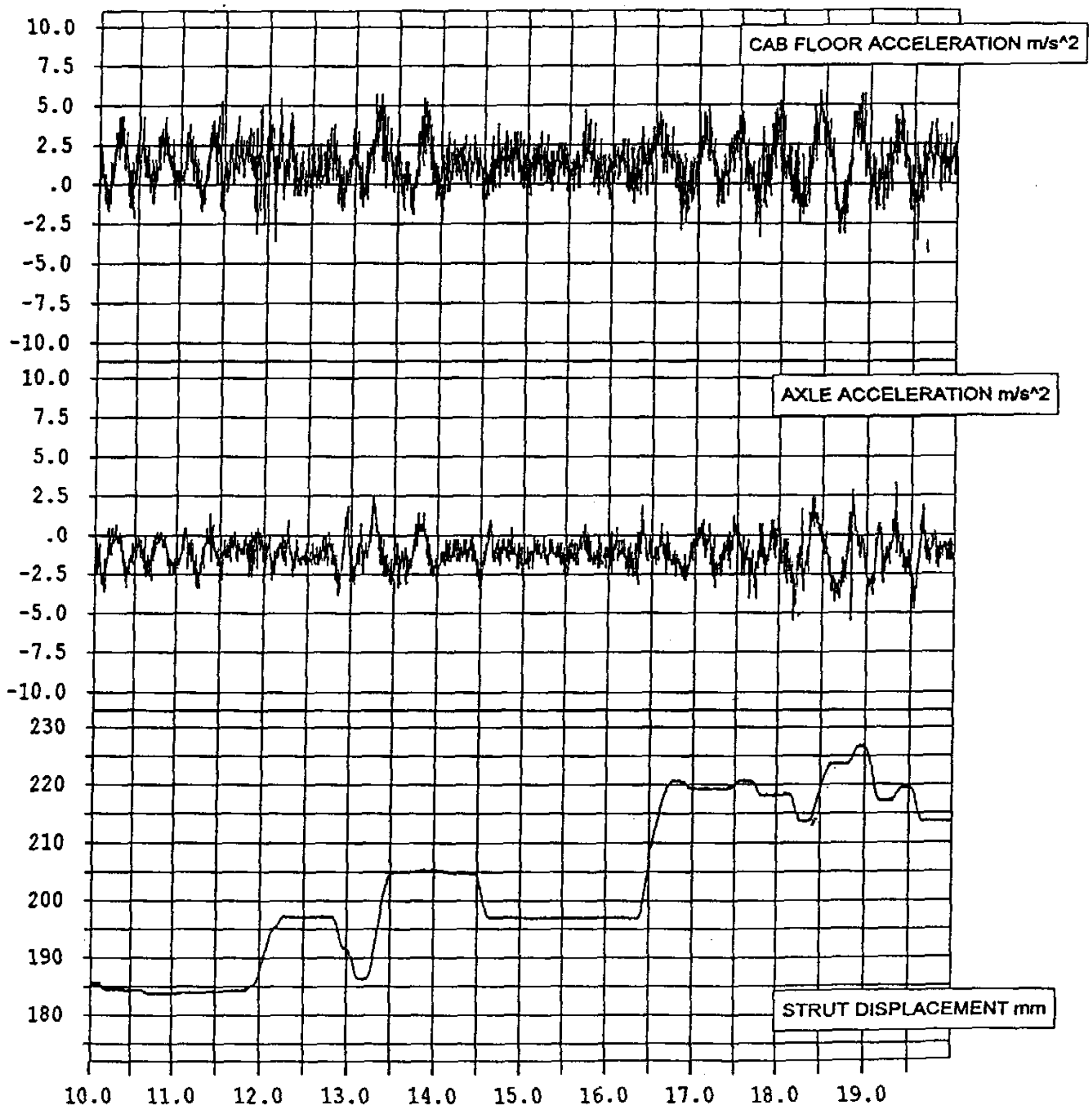
Figure 2 – Strut force vectors



A further problem with the standard suspension configuration is that the front suspension roll stiffness is greater than that at the rear. This results in a strong understeering handling characteristic, and also results in very high loadings on the outside front tyre during cornering. The front tyres are already loaded equally to the rear tyres when laden, and this load share is increased further by the roll stiffness disparity when cornering. Whilst the rear struts have a high spring rate, they are positioned close together relative to the front struts, thus resulting in low roll stiffness. The front suspension thus provides a disproportionate amount of roll stiffness, and this results in very high outside front tyre loading during cornering whilst laden.

The driver is roughly in line with the front axle line, and hence the ride experienced by the driver is influenced strongly by the front suspension performance. To maximise the benefit to the driver, and to address the perceived deficiencies in the sliding pillar design and the high peak front tyre loads, the front suspension was the focus of the suspension improvement investigation.

Figure 3 – test data on a CAT* 777 dump truck showing strut locking phenomenon
(source, reference [6])



3.3 Health issues

Whole Body Vibration (WBV) associated with the typically rough ride of dumpers and haulers used in the mining industry has long been recognised as a cause of spinal injuries. The understanding of the characteristics, level and duration of vibration necessary to cause injury has advanced considerably in recent years. The international standard dealing with this issue, ISO 2631, was comprehensively updated based on new research in 1997. In addition to vibration limits for health, it also specifies limits with regard to comfort and motion sickness. It indicates that health problems resulting from WBV are predominantly to the lumbar spine and the connected nervous system of the segments affected but can also affect the digestive system, the genital/urinary system and the female reproductive organs.

The responsibilities of mine operators are increasingly being defined and enforced, as evidenced by the European Parliament which adopted a joint text for a Physical Agents (Vibration) Directive regarding exposure of workers to the risks arising from vibration (including whole body vibration) on 21 May 2002. The Directive was published in the Official Journal of the European Communities on 6 July 2002 (L177 Vol 45, p12) as Directive 2002/44/EC on the minimum health and safety requirements regarding the exposure of workers to the risks arising from vibration, and has now come into force. Member States have three years to implement the Directive from 6 July 2002.

A typical approach to improving ride comfort is the fitment of suspension seats for vehicle operators. Whilst these seats are effective at reducing shock amplitudes and high frequency motions they amplify rather than attenuate low frequency motions. Commonly the resonant frequency of seat suspension systems is of the order of 1.5Hz. The human body is particularly sensitive to frequencies in the range of 0.5 to 10Hz, as discussed in AS 2670.1-2001 (ISO2631-1:1997). Suspension seats will thus worsen the ride comfort, particularly if the vehicle natural frequencies tend to the lower end of this range, as is generally the case. The performance deficiencies of suspension seats are discussed further in [8], [9] and [10].

3.4 Economic factors

The most significant cost, both human and financial, associated with poor suspension performance is arguably the cost associated with resulting health problems. Health issues are becoming increasingly expensive for mine operators with increasing compensation claims, costs of injury prevention by road and vehicle maintenance, and reduced operator shift times to reduce vibration exposure time.

There are other significant costs however. The behaviour of suspensions, in particular wheel hop and body motions, has a significant effect on the cyclic component of tyre loading and hence the rate of mine road degradation and tyre life [1]. Mine road maintenance is a major cost to mine operators and tyre replacement is a major vehicle operating cost. The roll stiffness disparity between front and rear suspensions prevalent on dumpers and haulers leads to the overloading of the outside front tyre during cornering. This also has an adverse impact on tyre wear and road degradation. Poor suspension performance also increases dynamic loads on the vehicle chassis leading to cracking and premature failure.

3.5 Computer simulation

In an effort to better understand the dynamics of the front suspension, and due to the limited success achieved with low friction bearing trials, a comprehensive quarter vehicle dynamic model was developed. This modelled the suspension on one front wheel and the vehicle mass riding on it. This is simpler than half vehicle and full vehicle models as it does not include the many additional degrees of freedom such as vehicle rolling and pitching. It is however adequate to understand the dynamics of the front suspension struts, and allows detailed simulations of different road and suspension characteristics at different load conditions. The struts are substantially non linear in a number of respects:

- The gas spring rate varies with deflection. The spring characteristics also vary with deflection rate, slow movements being isothermal whilst faster ones are isentropic.
- The "stick-slip" characteristic of the strut friction as it changes from static to dynamic friction is inherently non linear.
- The damping force generated by the damping orifices is governed by the quadratic relationship between flow rate and pressure, at least for turbulent flow.
- The magnitude of strut bearing friction force varies with strut deflection due to the varying distance between the piston and rod bearings as the strut deflects.
- The sliding seal friction varies with strut pressure and velocity.
- Gross non linearities such as impact with bump stops and the tyre losing contact with the roadway can also significantly affect the suspension behaviour under severe conditions.

To gain a good understanding of the strut behaviour it was considered essential to incorporate all of these non linearities in the model; preliminary calculations indicated that the classical approach of "linearising" these behaviours for an eigenvalue analysis would lead to substantial errors. A numerical analysis based on the 4^{th} order Runge Kutta numerical analysis technique was thus selected. This model was implemented using the Matlab mathematical analysis software tool.

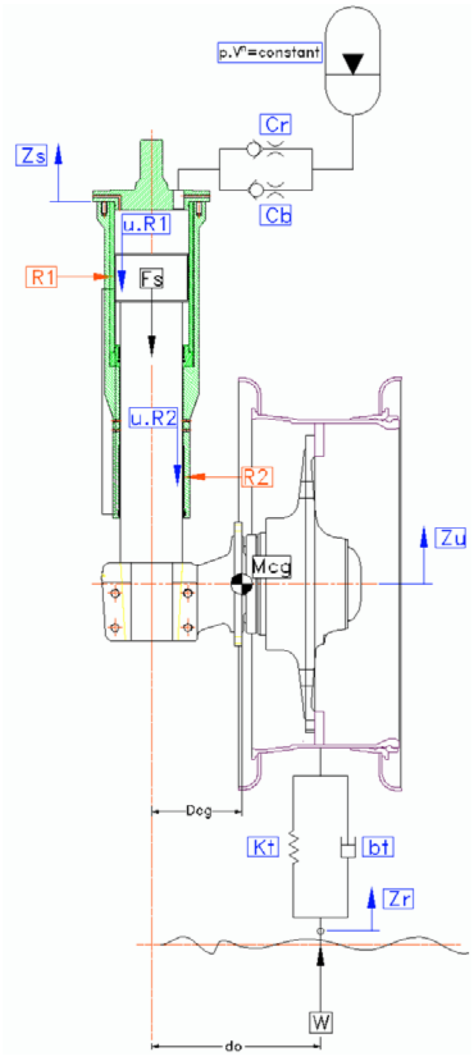
A physical model for the tyre was developed. The tyre spring rate and damping coefficient was based on tyre data available and a simple first order tyre filter model [2], [3] was used to filter the road profile. The tyre model also included tyre eccentricity, which can be significant in dumper and hauler tyres.

Following a review of the literature, the white noise road profile model of Sayers [4] and Bulman [5] was used as input to the model. It was found that a roughness coefficient of $C=0.25x10^{-5}$m together with a tyre eccentricity of 5mm (offset between tyre centre and wheel axis, ie TIR of 10mm) gave realistic results when compared with test data for a truck with standard suspension on a typical mine haul road. This value was used for the modelling described below. Due to the significant variation observed in tyre eccentricity in field measurements, it was decided to base the modelling on the ideal situation of zero eccentricity.

A standard vehicle speed of 40km/h was applied in the modelling, although the suspension behaviour at other speeds was also investigated.

A diagram of the model standard strut is shown in Figure 4 below. To facilitate the modelling, this differs from the actual standard strut in that the damping orifices have been repositioned to avoid oil cavitation on bump travel. Critical to the results of the modelling, were the values used for coefficient of friction of the piston and rod bearings. Testing on an actual strut was performed to measure these coefficients (refer ref [6]) and these were used in the model. Testing of the rod bearing indicated static frictions in the range 0.11 to 0.62 depending on rod roughness and type of grease used. Dynamic friction varied from 0.05 to around 0.2. Seal friction forces were also measured and found to be 7700N static and 6000N dynamic at zero pressure and increasing linearly with pressure at 0.45 and 0.36N/kPa respectively.

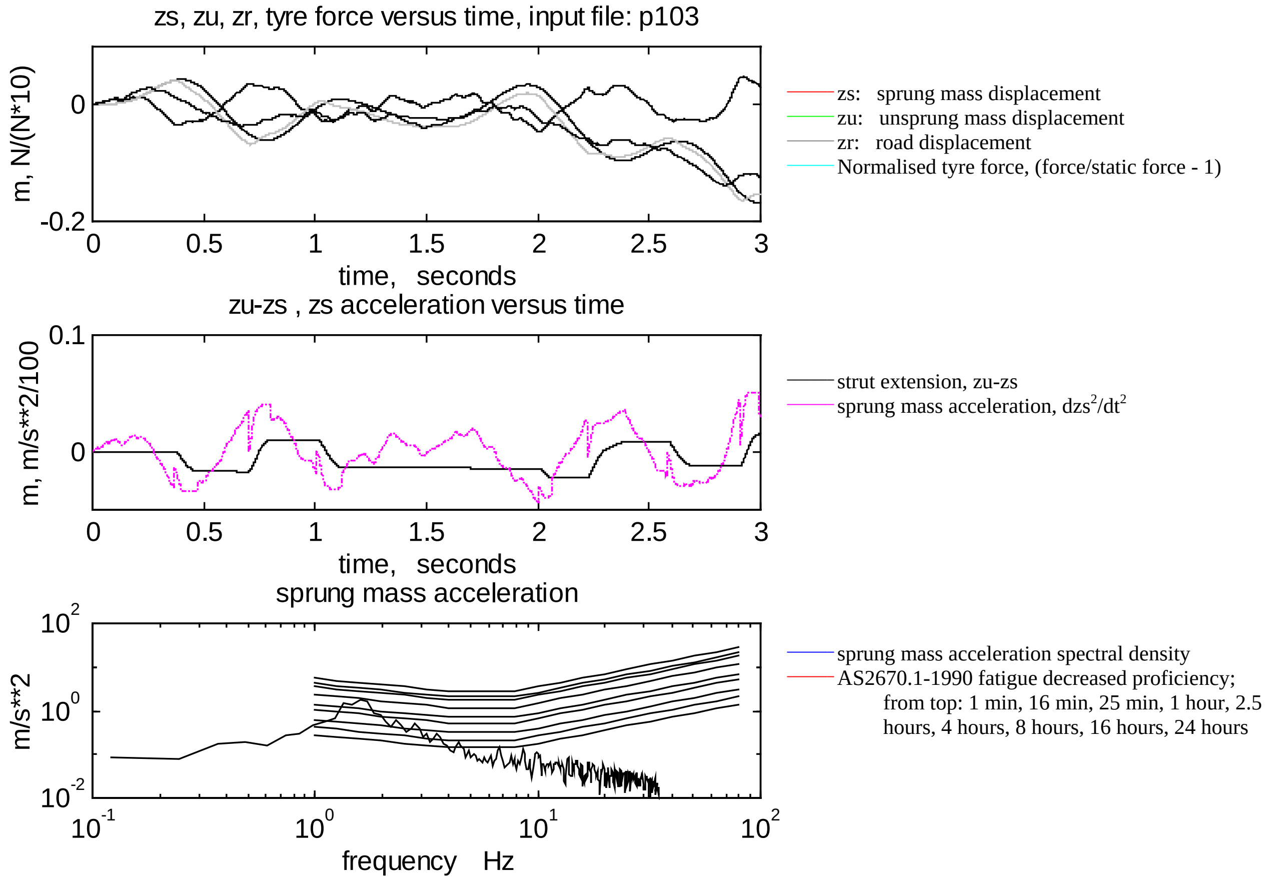
Figure 4 – Standard strut model



The results for a standard strut under laden condition are shown in Figure 5. As can be seen from the strut displacement trace, the strut remains locked for a large proportion of the time. The frequency spectrum shows the distinct resonant motion peak at 1.6Hz. By variation of tyre stiffness and damping parameters this was confirmed to correspond to the strut locked tyre bounce vibration mode.

The ride comfort was assessed in accordance with AS 2670.1-1990 (equivalent to ISO 2631/1-1985). This categorises ride quality in terms of FDP (fatigue decreased proficiency) times. This represents the time for which an operator can endure the ride without loss of performance due to fatigue.

Figure 5 – Modelling results for standard strut, laden

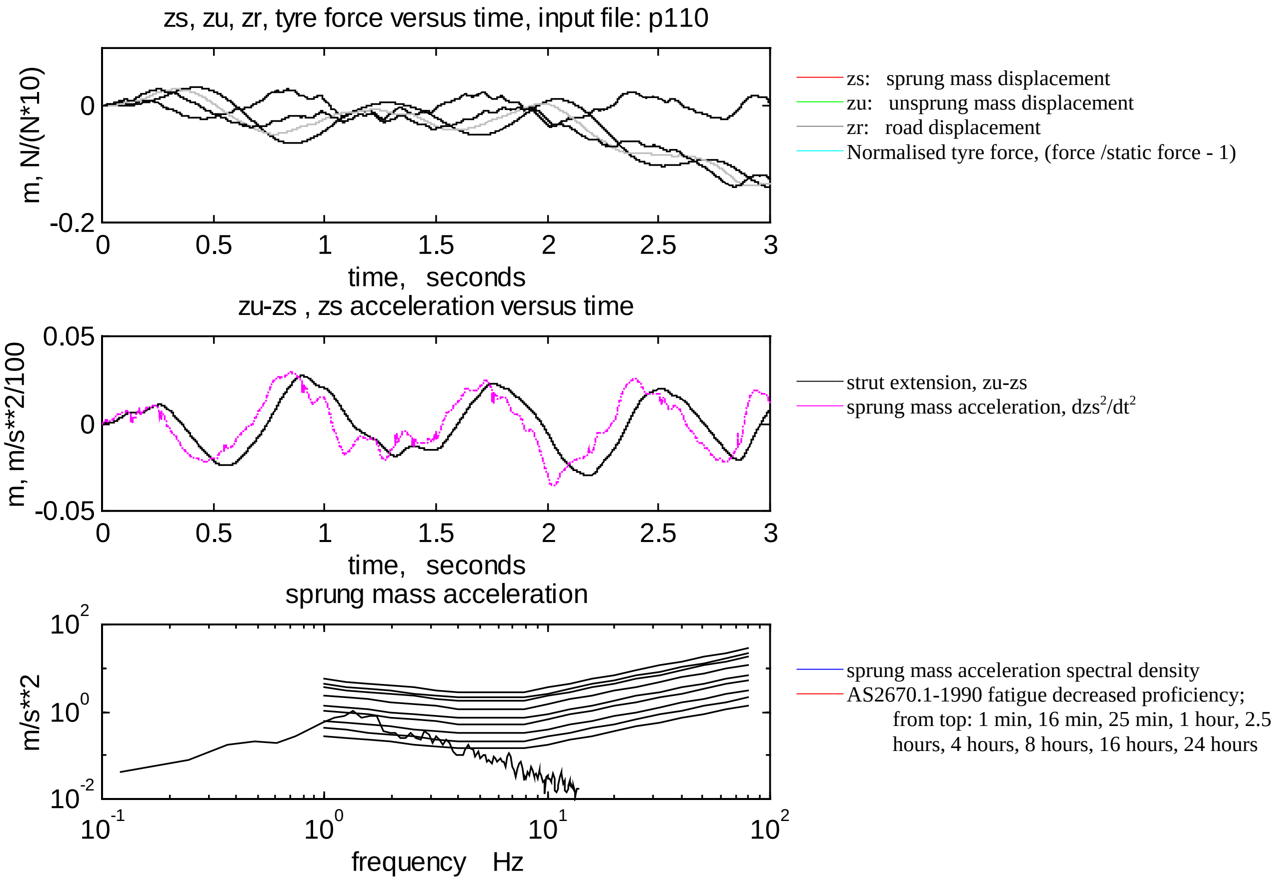


4 Investigation of strut design improvements

4.1 Reduced friction strut

The standard strut computer model was run with reduced bearing friction coefficients. Static and dynamic coefficients of 0.0012 and 0.0008 respectively were used whilst seal friction was based on data for a low friction seal. This indicated that a significant improvement in ride comfort could be obtained. Also, different damping characteristics based on varying orifice diameters were modelled to determine if improvement could be achieved. The results for the reduced friction strut when laden are shown in Figure 6. By comparison with Figure 5 it can be seen that a significant improvement has been achieved. The FDP time has improved from 1 hour to 3.5 hours. The strut displacement trace now shows almost constant motion of the strut, indicating an absence of friction induced strut locking.

Figure 6 – Modelling results for standard strut with reduced friction, laden



4.2 Active solutions

The effective suspension travel is significantly less than the actual range of strut movement, as the strut must operate without running out of travel at the widely differing laden and unladen conditions. The suspension uses only a portion of its travel towards the fully compressed limit when laden, and a portion of total travel towards fully extended when unladen. The total travel available is only 320 mm on the 777, and the effective range available at each load condition is significantly less again. To avoid impact with strut travel stops a high spring rate is used.

The solution to this poor utilisation of strut travel is to provide a ride height adjustment system, which adds oil when the truck is laden, and removes it when the truck dumps its load. This maintains a fixed mean ride height, allowing more of the strut travel to be utilised, and a lower spring rate (larger gas volume) to be used. This type of system requires some sort of load or mean ride height sensing and the means to add and extract oil, and is commonly termed a slow-active suspension system. The logical extension to this is to use rapid oil movements into and out of the strut to actively control suspension forces, which is termed an active suspension system.

A review of the literature on active and slow-active suspension systems was conducted. A number of architectures and control strategies were identified which could be readily applied to the oleopneumatic strut architecture, without substantial modification.

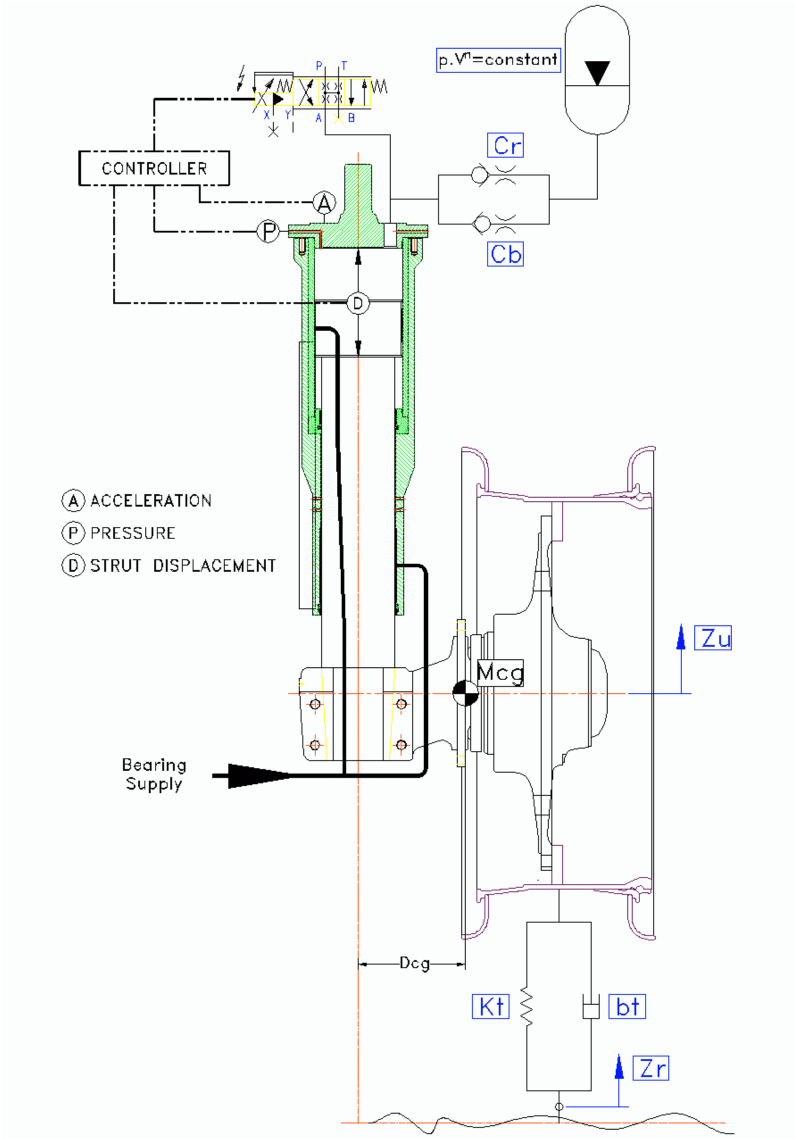
These solutions offered ride height control, but did not offer a solution to the remaining problem of strut friction. In an effort to identify a solution to this problem a significant design breakthrough was made. Due to the lack of success with the use of low friction bearing materials, other concepts for friction reduction were identified. One of these was the incorporation of hydrostatic bearings. These consist of bearings which contain grooves in their surfaces which are fed with oil from a high pressure pump. This maintains an oil film between the bearing surfaces, even at very high loads, thus offering very low friction. The problem however was to deal with this flow of oil into the strut at the piston bearing. The rod bearing is located below the rod seal, so the bearing oil can be drained off. However, oil flow to the piston bearing feeds into the oil volume, and must thus somehow be removed. The breakthrough was to allow the valve, which is necessary for any active or slow-active system, to drain this oil from the strut, controlled so as to maintain the desired mean ride height. This approach has been protected by Birrana Engineering in US patent 6,416,061.

Various configurations for the bump and rebound damping and oil control valve position were identified and analysed. The control valve controlling oil supply into and out of the strut effectively makes it an actuator. Fundamentally two configuration options are possible, those with the damper in parallel to the actuator and those in series.

The quarter vehicle computer model was modified to incorporate these various configurations and also various control strategies. The model used for the servovalve was a second order model, with the essential non linearities of spool slew rate and travel limits. The model was validated against the servovalve manufacturer's data.

Extensive modelling and optimisation was performed to identify the suspension performance improvements that could be obtained. The key parameters that were monitored were ride comfort (FDP time), tyre forces (peak and rms) and power consumption (peak and mean). Active control requires substantial amounts of power, and this is a real cost of providing superior suspension performance. The results showed that the most promising configuration was a series configuration as shown schematically in Figure 7. It uses a servovalve (MOOG D634 chosen for much of the modelling) controlled by a controller which receives inputs from accelerometers on the sprung mass at each strut and a displacement transducer in each strut. It implements a skyhook damping control algorithm within the constraints such as servovalve and hydraulic supply capacity and strut travel limits. This configuration was then modelled extensively with varying levels of damping, controller gains, servovalve characteristics, and gas volumes.

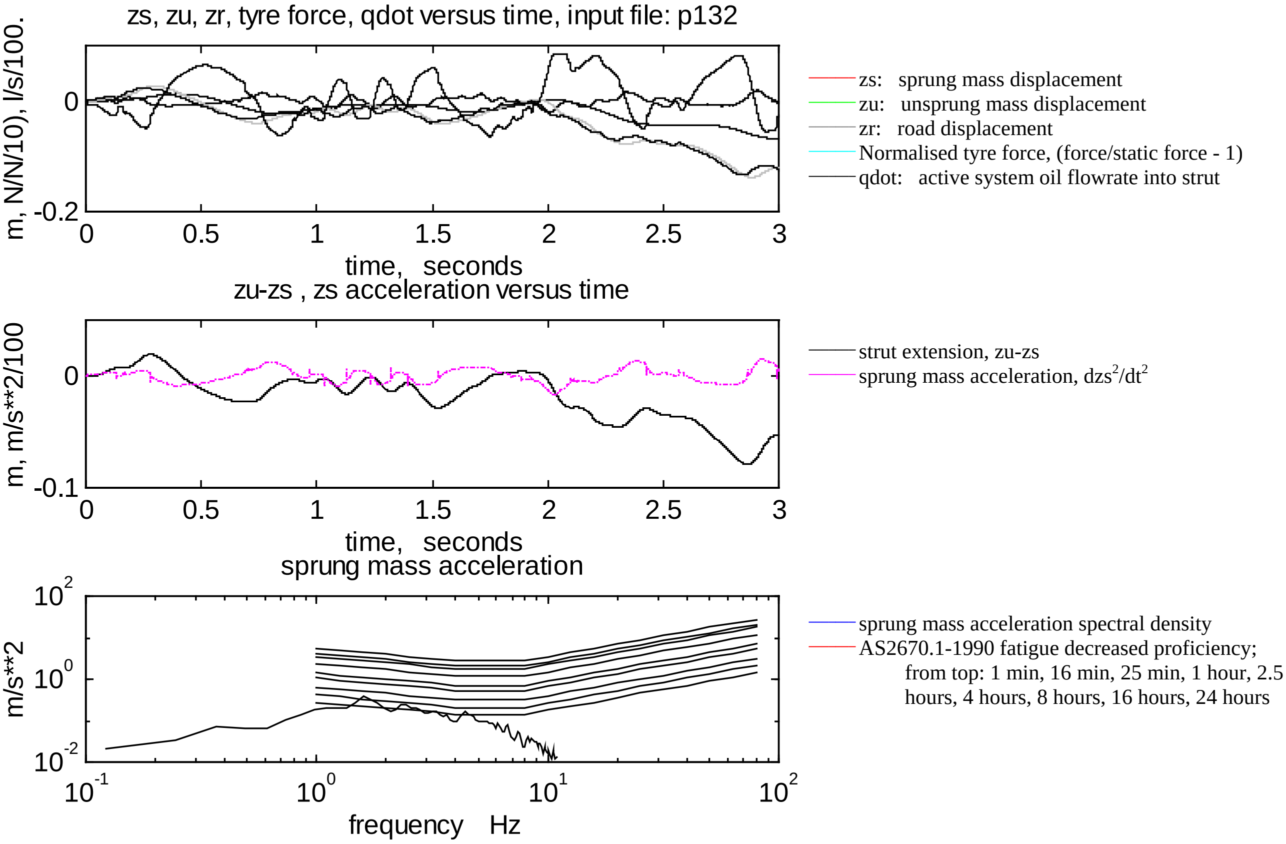
Figure 7 – Active system with hydrostatic bearings



The results were used to guide the design towards an optimal solution. This solution is represented by the active suspension column in Table 1 and its performance is shown graphically in Figure 8. As can be seen from the table and comparison of the figure to Figure 5 and Figure 6, the active system delivers significantly superior suspension performance. The FDP time has been increased from the 3.5 hours of the low friction passive strut to 14 hours. Tyre forces have been significantly reduced. The rms and peak amplitudes about the mean have been reduced to less than half those of the standard strut. The average power requirement (one strut only) was 15.5kW. This increases markedly as road roughness increases.

A key finding of the analyses was that even a sophisticated active system is severely affected by high bearing friction and that its performance improves progressively as friction is reduced.

Figure 8 – Modelling results for active suspension, laden



Following the conclusion of the modelling work a prototype system was constructed. This was fitted to a CAT* 777 dumper and was instrumented with accelerometers to evaluate suspension performance. A standard truck was also instrumented and comparison testing conducted on a mine road at the Brocks Creek mine in the Northern Territory. Despite some teething problems with the hydraulic pump powering the system and with the hydrostatic bearings, the test results indicated that a very substantial improvement in ride performance could be achieved. This testing, and the modelling of the active system is described in [7].

Whilst the sophisticated active systems such as that described above provide the best performance, it is achieved at the expense of much increased complexity and capital cost. Operating costs of the truck will also be significantly increased, as the hydraulic power required to operate the system is quite substantial, especially on rough surfaces, and is reflected in increased fuel consumption. Maintainability and reliability are also likely to be impacted, the expensive high frequency response servovalves required are sensitive to oil contaminants and require very fine filtering.

4.3 Slow active Smart Strut™

The reduced control bandwidth of slow active systems compared to active systems offers benefits in reduced cost and complexity. Slow active systems do not attempt to actively control the suspension forces and wheel hop resonance, leaving this to the passive spring and damper elements. The oil added to and removed from the strut acts merely as a fast load levelling system. Ride height control allows lower spring rates and more freedom in damping coefficients. A wide disparity between bump and rebound damping is even possible, due to the ability of the ride height control to counteract the tendency of the suspension to “pump down” in the case where the bump damping is substantially less than the rebound damping. The slow active system offers greatly improved performance over the standard strut, but with only a small increase in complexity of the suspension system. In the slow active system shown in Figure 9 the only oil introduction is via the hydrostatic bearings and a simple 2 way control valve removes oil under the control of a low bandwidth ride height controller. This system, which is particularly compatible with typical sliding pillar suspension configurations, was named Smart Strut. The cost of this system is low enough to be viable as a retrofit package, and is even more cost effective if fitted as OEM equipment.

Extensive modelling of the Smart Strut system was conducted to optimise damping, gas volume and controller gains. A model was developed and validated for the simpler (compared with the active system servo valve) valve used in the slow active model.

The results for the optimum design are shown in Figure 10. The comparison in Table 1 shows that whilst it falls well short of the performance of the active system, it is a distinct improvement over the low friction passive system. The Smart Strut was also found to have significantly superior performance with step inputs which approximately simulates driving over a single large bump. In this instance the greater compliance and greater effective strut travel provide a substantial performance increase. The modelling also indicated that the Smart Strut greatly reduces the effect of tyre eccentricity on ride quality and tyre forces compared to the standard strut.

Figure 9 – Slow active Smart Strut

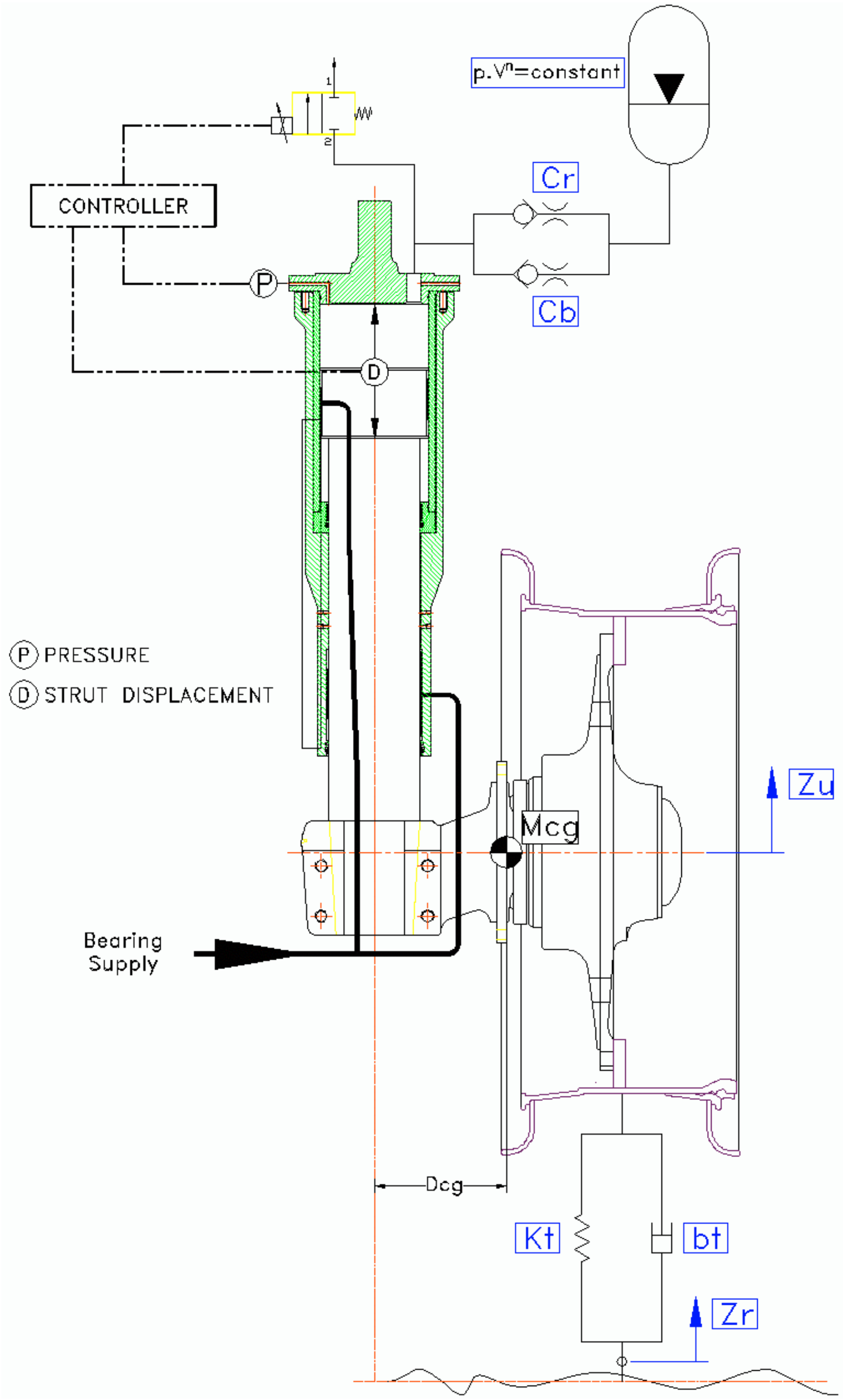


Figure 10 – Modelling results for Smart Strut, laden

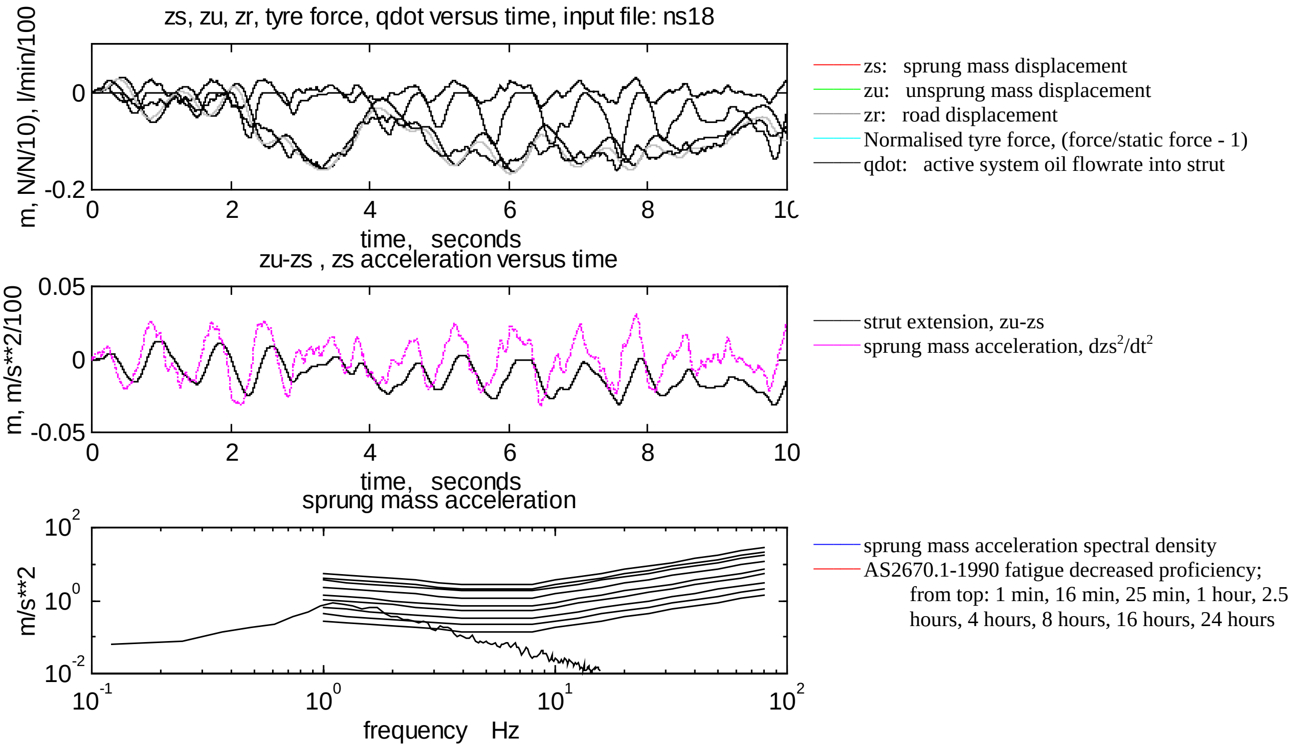


Table 1 - Comparison of suspension configurations - vehicle laden, white noise road profile, 40km/h

	Standard Passive* (refer Figure 4)	**Low Friction Passive*** (refer Figure 5)	**Active** (refer Figure 7)	**Slow Active (Smart Strut)** (refer Figure 9)
Maximum acceleration (ms^{-2} @ Hz)	1.8 ms^{-2} @ 1.6Hz	0.72 ms^{-2} @ 1.1Hz	0.38m s^{-2} @ 1.6Hz	0.62m s^{-2} @ 1.6Hz
Fatigue decreased proficiency (fdp) AS2670 - Hours	1	3.5	14	4
Normalised tyre force - Max.	0.49	0.36	0.21	0.41
Normalised tyre force - Min.	-0.62	-0.41	-0.24	-0.46
Normalised tyre force - RMS	0.21	0.12	0.08	0.13
Power (kW) - Maximum	0	0	87.8**	0**
Power (kW) - Mean	0	0	15.5**	0**

* damping orifice configuration differs from standard suspension
** neglecting the small power requirement for the hydrostatic bearings

5 Slow-active "Smart Strut" suspension development

Following the identification of the preferred “Smart Strut” slow active suspension architecture, it was decided to build a prototype system. The first stage in this process was the further development and testing of the hydrostatic bearings used in the active suspension prototype. Following a lengthy optimisation process, which involved extensive analysis, testing and fine tuning, an extremely low coefficient of friction in the range 0.005 to 0.01 was consistently achieved, an impressive result compared to the 0.1 to 0.2 measured with the standard bearings.

Following successful testing of the bearings, a revised strut head incorporating the control valve and strut displacement transducer was developed. This allowed for a much increased gas volume to achieve the desired spring rate, as well as the provision of externally adjustable damping orifices. The standard CAT* pressure transducer was retained to monitor strut pressure. The pressure data is used for roll stiffness control and for control of large strut motions such as during tipping when the load on the front struts reduces rapidly. The control valve controls flow out of the strut only, and is far simpler than the servovalve used in the active system.

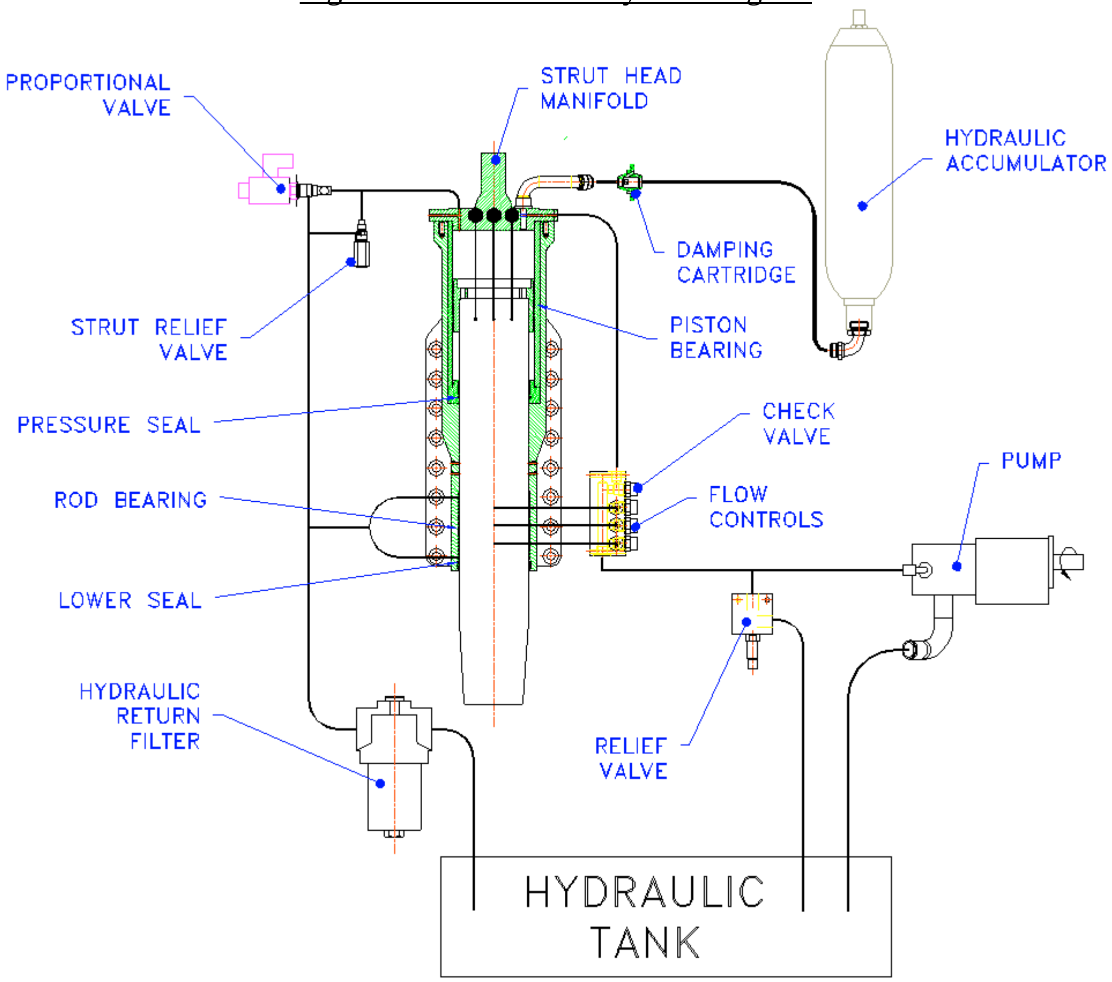
A microprocessor based control system was developed to control the system. This system implements a PID ride height controller. It also incorporates features such as a comprehensive diagnostic system and a novel start up and shut down ride height adjustment. At shutdown the struts are allowed to settle to their full retract stops. This provides easy operator access and egress to the truck as the front step is positioned just above ground level. On start up the struts "pump up" to normal ride height. This also avoids safety issues with struts settling when the vehicle is parked, which could become an issue if the control valve were to develop a leak.

Implementing ride height control on the front suspension exacerbates the roll stiffness disparity between front and rear of the standard suspension. On a long sweeping corner, or on a cambered road, the ride levelling will tend to equalise front suspension heights effectively producing infinite roll stiffness. The ride height control system has thus been modified to address this problem. The pressure in each front struts is monitored and any sustained difference triggers ride height adjustments to be made to provide the desired level of steady state roll compliance. A limit is imposed on the magnitude of these equal and opposite ride height adjustments to prevent excessive body roll.

The system can also be readily reprogrammed in the field with the aid of a laptop computer, permitting control over key parameters such as ride height, roll compliance, large strut motion control parameters and error checking parameters.

A diagram of the Smart Strut system is shown in Figure 11.

Figure 11 – Smart Strut system diagram



5.1 Testing

The Smart Strut semi active system was built and tested, first on a test rig and then on vehicles. Rig testing consisted of mounting one strut on a purpose built hydraulic test rig which simulates its operation on the vehicle. The photograph in Figure 12 shows a strut mounted on the rig undergoing testing. The rig is powered by a 75kW hydraulic power pack and incorporates a large accumulator allowing high peak "bump power" for short periods, allowing rough road surfaces to be simulated.

Following successful rig testing, the system was installed and trialled on both a 777 dumper and a 776 hauler. The system is shown in Figure 13 and Figure 14. The system was trialled extensively to measure performance over a range of conditions and to gain operator feedback. The positive results and enthusiastic feedback led to an extended mine site trial on a hauler during which approximately 3,500 hours of operation were completed.

Figure 12 – Smart Strut on the test rig

Figure 13 – Smart Strut being installed

Figure 14 – Operator interface on dashboard

The hauler was instrumented with accelerometers to measure the ride performance. A standard vehicle was also instrumented and data recorded for both on the same road under identical controlled conditions. The accelerometer data included triaxial accelerations on the suspended seat pad, accelerations at the base of the seat and on the wheel stub axle on the driver’s side. The accelerometers were sampled at 100Hz and the data recorded on a laptop computer. This data was then analysed in accordance with AS

2670.1-2001 (ISO2631-1:1997) and the results are summarised for various different test roads in Figure 15. Two sets of data are shown for each suspension on each test road.

The results show that the Smart Strut provides a very significant improvement over the standard suspension, with health limit times increasing by a factor of 2 or more for the laden case and significantly better when unladen. Modelling indicates that the laden performance may be improved further at the expense of unladen performance by increasing the damping. This is yet to be confirmed by testing.

The spectral data in Figure 16 and Figure 17 shows that the "loping" or "tyre bounce" mode has been virtually eliminated with Smart Strut, whilst the higher frequency wheel hop mode is of similar amplitude to that of the standard strut. It can also be observed by comparison of the seat accelerations with the sprung mass accelerations in these figures that the seat is amplifying rather than attenuating the low frequency motions up to around 5Hz. Further improvement in ride comfort could thus be obtained by improved seat design.

The eccentricities of the front tyres on the test vehicles were measured to eliminate tyre eccentricity as a cause of the difference in performance. The tyre eccentricities were very similar as shown in Table 2. In both cases, the eccentricity was worst on the left side, which was also the side where the accelerometers were mounted.

Table 2 – Tyre eccentricity on test haulers, TIR in mm

	Smart Strut	Standard
Left	9	7
Right	5	6

Figure 15 – Performance comparison with standard suspension

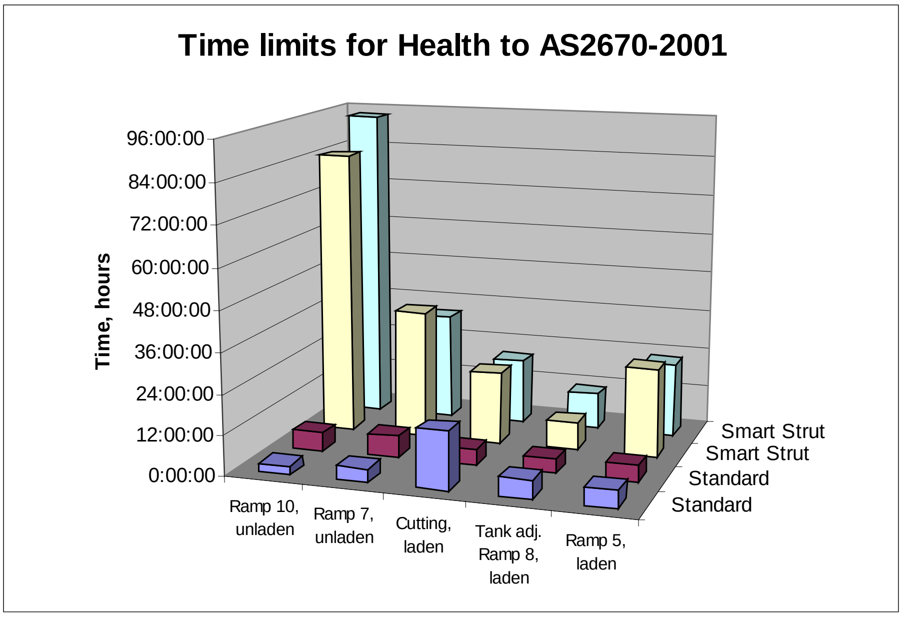
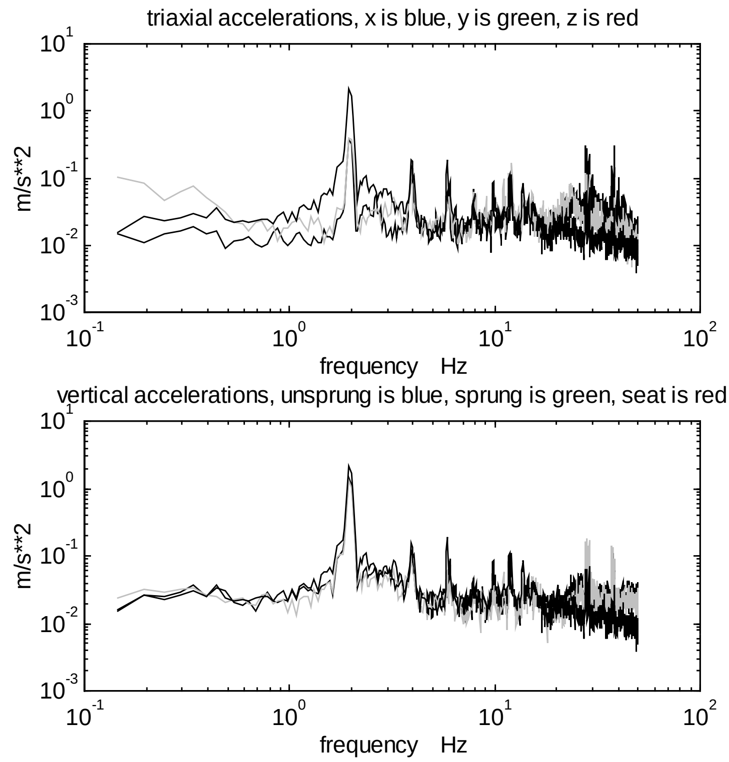
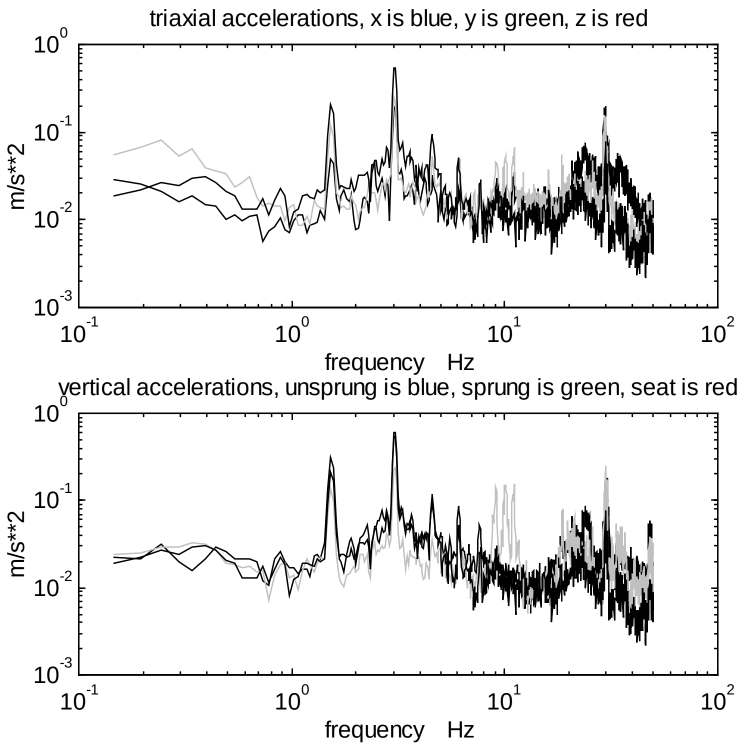


Figure 16 - Standard Suspension acceleration spectra at seat, unladen, ramp 10



Directions, at seat
x is fore/aft
y is lateral
z is vertical

Figure 17 - Smart Strut™ acceleration spectra, unladen, ramp 10



Directions, at seat
x is fore/aft
y is lateral
z is vertical

References

1. Prem, H. and Dickerson A. SAE 921735 "A Study of the Steady State Roll-Response of a Large Rear-Dump Mining Truck"
2. Thompson A.G. "The Effect of Tyre Damping on the Performance of Vibration Absorbers in an Active Suspension", Journal of Sound and Vibration (1989) 133(3) pp457-465
3. Sharp R.S., Crolla D.A., "Road Vehicle Suspension Design – A Review", Vehicle System Dynamics 16 (1987) pp355-380.
4. Sayers M., ASME Symposium on Simulation and Control of Ground Vehicles and Transportation Systems AMD Vol 80, DSC Vol 2 1986.
5. Bulman D.N., "Cross Country Vehicle Ride Simulation", I Mech E Feb/Mar 1979.
6. French P.J., "Intelligent Dumper and Hauler Suspension System (IDHSS), Final report, ACARP Project no. C4013, 8 February 1997.
7. Gordon D.L., "Intelligent Dumper and Hauler Suspension System (IDHSS) – Stage 3, Final report, ACARP Project no. C6007, 26 November 1998.
8. Johanning E et-al, "Whole body vibration exposure study in US railroad locomotives – An ergonomic risk assessment" AIHA Journal (63) July/August 2002
9. Gunston T and Griffin M.J., "The isolation performance of a suspension seat over a range of vibration magnitudes tested with an anthropodynamic dummy and human subjects" Paper presented to Inter_Noise '99, Florida, USA
10. McDonald G.L., "Damage to people from ride disturbance", unpublished, Geoff McDonald and Associates Pty Ltd, 36 Judith Street, Crestmead, Queensland, Australia 4132.