4 EXPERIMENTAL WORK

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4.1 Preamble

In this chapter, the experimental work is presented. The experimental work can be divided into two stages, namely characterisation tests and single degree of freedom tests. The test setup, test equipment and characterisation procedures for both these stages are discussed in this chapter. Where deemed necessary, some background information is supplied, in order to elucidate the characterisation process. The test results are presented in graphical and tabular format.

In the following paragraphs, firstly the test setup is discussed, with reference to the hardware, software and test equipment. Secondly, the characterisation of the spring, damper and valves are discussed. After that, the single degree of freedom tests are discussed and finally some closing remarks are made. All the test results are supplied in Appendix D.

4.2 Test setup

4.2.1 Experimental setup

Two experimental setups were used, one for the component characterisations (springs, damper and valves) and another one for the single degree of freedom tests. In both cases, a $160kN$ Schenck hydraulic actuator was used to supply the desired input. Figure 4-1 shows schematically the characterisation test setup.

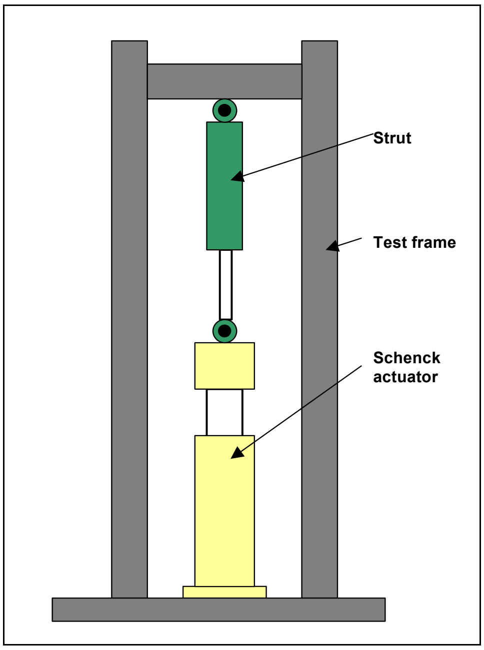

Figure 4-1: Characterisation test setup

The top of the strut was fixed to the rigid test frame with a locating pin, while the bottom mounting was fixed to the hydraulic actuator. Spherical rod ends were used, in order to eliminate any bending moments on the strut. In this setup, the required relative strut displacement is generated by vertical actuator motion.

For the single degree of freedom tests, a separate test frame was build. Figure 4-2 shows a photograph of the single degree of freedom test setup.

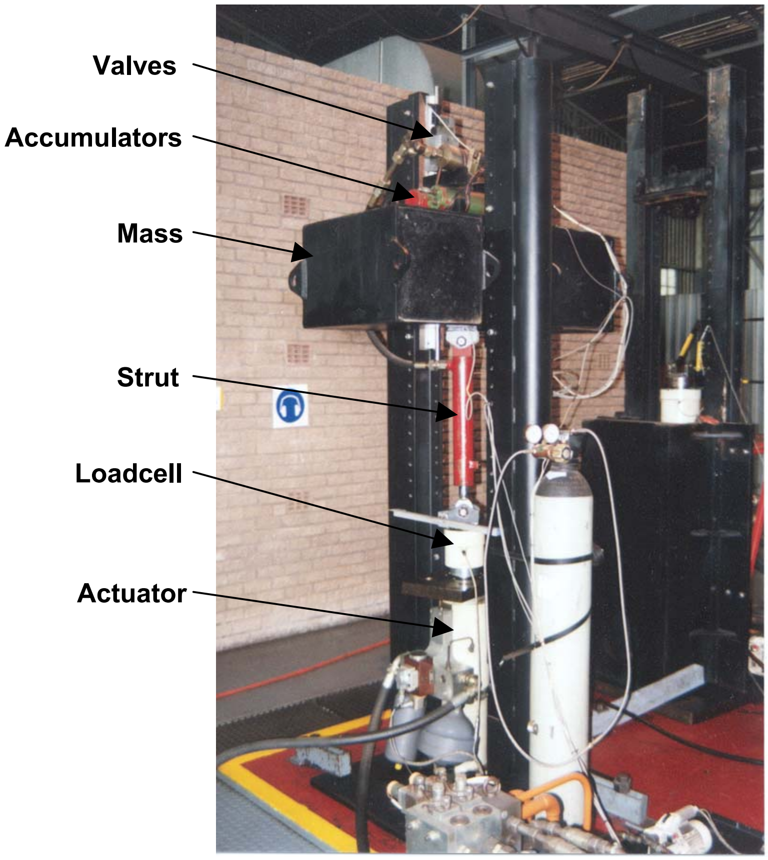

Figure 4-2: Single degree of freedom test setup

A lead mass of approximately 3 tons was used to simulate the sprung mass of a vehicle, since a static wheel load of between 2,5 tons and 3 tons are common for military off-road vehicles. The test frame was equipped with a set of linear bearings, guiding the sprung mass, which was supported by the lower mounting on the Schenck actuator. Also visible in the photograph is the nitrogen cylinder used to fill the accumulators. The accumulators, damper and valves were secured on top of the lead mass (see Figure 4-3). The test frame was securely fixed to the test floor, to ensure that the SDOF setup does not fall over.

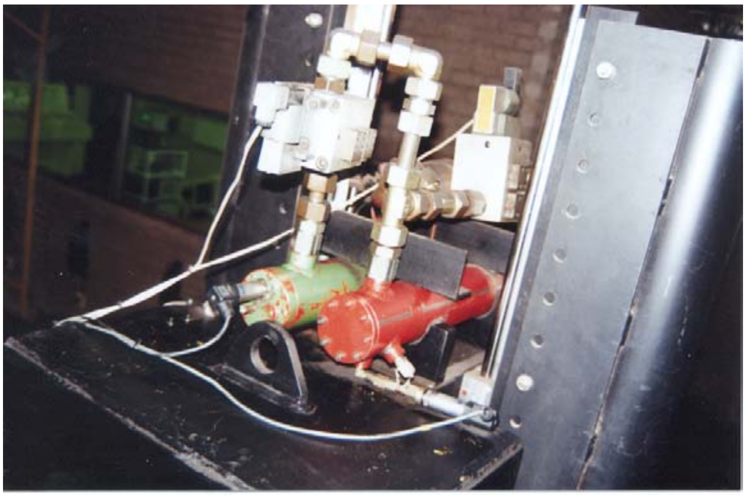

Figure 4-3: Accumulators, damper and valves secured on top of sprung mass

4.2.2 Instrumentation, control and data acquisition

The Schenck actuator has a PID controller, which controls the actuator to follow a desired input signal. A built in Schenck signal generator can be used to supply sinusoidal, square and triangular signals. A sinusoidal input signal was use to characterise the spring and the damper. An external control signal can also be supplied to the Schenck controller. This feature was used to determine the valve response times, where it was necessary to construct a custom displacement signal, which is synchronised with the valve switching signal. A 486 personal computer with a Burr-Brown D/A card was used to supply the desired input signal.

A separate 486 personal computer with a Burr-Brown A/D card was used to record the signals from the test equipment. Table 4-1 indicates the parameters, as well as the measuring equipment.

Table 4-1: Test equipment (component characterisation)

No.	Parameter	Instrument
1	Actuator displacement	Schenck PFM LVDT
2	Actuator force	Schenck 160kN PM-Rn loadcell
3	Pressure in 0,3*l* accumulator (see Figure 4-5)	Wika 40*MPa* pressure sensor
4	Pressure in 0,7*l* accumulator (see Figure 4-5)	Wika 40*MPa* pressure sensor

For the single degree of freedom tests, some additional parameters were measured. Table 4-2 supplies a list of parameters and measuring equipment used for the single degree of freedom tests.

Table 4-2: Test equipment (single degree of freedom tests)

No.	Parameter	Instrument
1	Actuator displacement	Schenck PFM LVDT
2	Actuator force	Schenck 160*kN* PM-Rn loadcell
3	Relative displacement (see Figure 4-4)	Penny & Giles linear potentiometer
4	Sprung mass acceleration	VTI Hamlin A050AA 5*g* accelerometer
5	Pressure in 0,3*l* accumulator (see Figure 4-5)	Wika 40*MPa* pressure sensor
6	Pressure in 0,7*l* accumulator (see Figure 4-5)	Wika 40*MPa* pressure sensor

Figure 4-4: Linear potentiometer measuring relative strut displacement

Figure 4-5: 40*MPa* pressure sensor measuring accumulator pressure

4.3 Hydro-pneumatic spring characterisation

In this section the characteristics of the semi-active hydro-pneumatic spring is discussed. The physical attributes and characterisation procedures are also outlined.

4.3.1 Physical attributes

There are many different types of hydro-pneumatic springs, but the basic difference lies in the way the gas and the oil is separated. Some hydro-pneumatic springs have a rubber bladder separating the gas and the working fluid, while others have a floating piston. Both the hydro-pneumatic springs considered in this study are of the floating piston type. Figure 4-6 shows a schematic drawing of the floating piston accumulator.

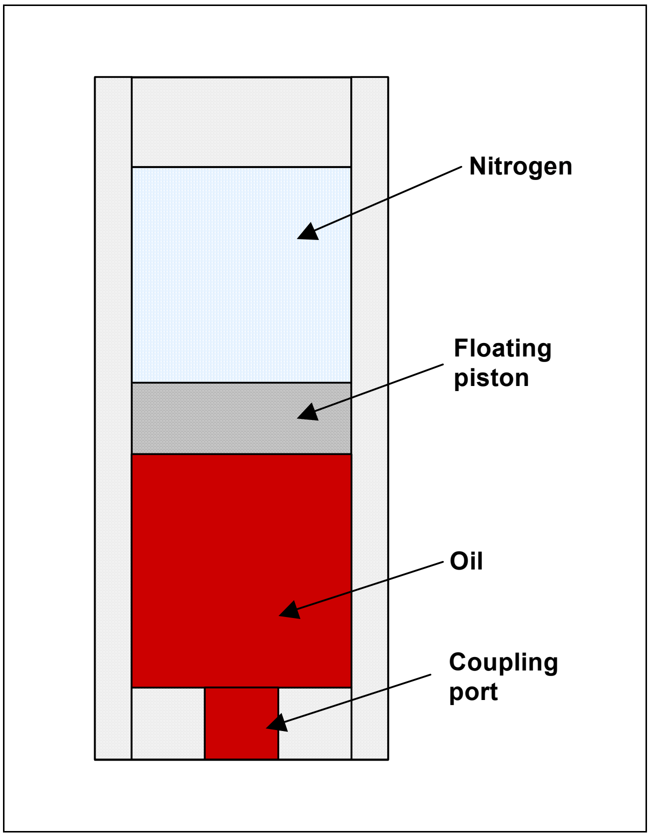

Figure 4-6: Floating piston hydraulic accumulator

The static volumes of the two accumulators are 0,3l and 0,7l respectively. When the valve is open the combined volume of the two accumulators is 1,0l and when the valve is closed only the 0,3l accumulator is connected with the single acting cylinder.

Nitrogen gas was chosen as springing medium. The reasons for using Nitrogen are:

- Nitrogen is widely used in accumulators and hydro-pneumatic springs.
- Properties of Nitrogen are well documented.
- Nitrogen is an inert gas.

4.3.2 Characterisation procedure

For the hydro-pneumatic spring characterisation the first experimental setup described in paragraph 4.2.1 was used. The two stages of the semi-active hydro-pneumatic spring were characterised by subjecting the strut to a sinusoidal displacement, of varying frequency. The excitation speed is defined as the piston speed when moving through the static position. The excitation speed is therefor a function of the excitation frequency and the excitation amplitude. The amplitude of the signal was approximately 100*mm*. Figure 4-7 indicates the two spring characteristics for an excitation speed of 0.01*m/s*. From this figure, it can be seen that two very different spring characteristics were achieved.

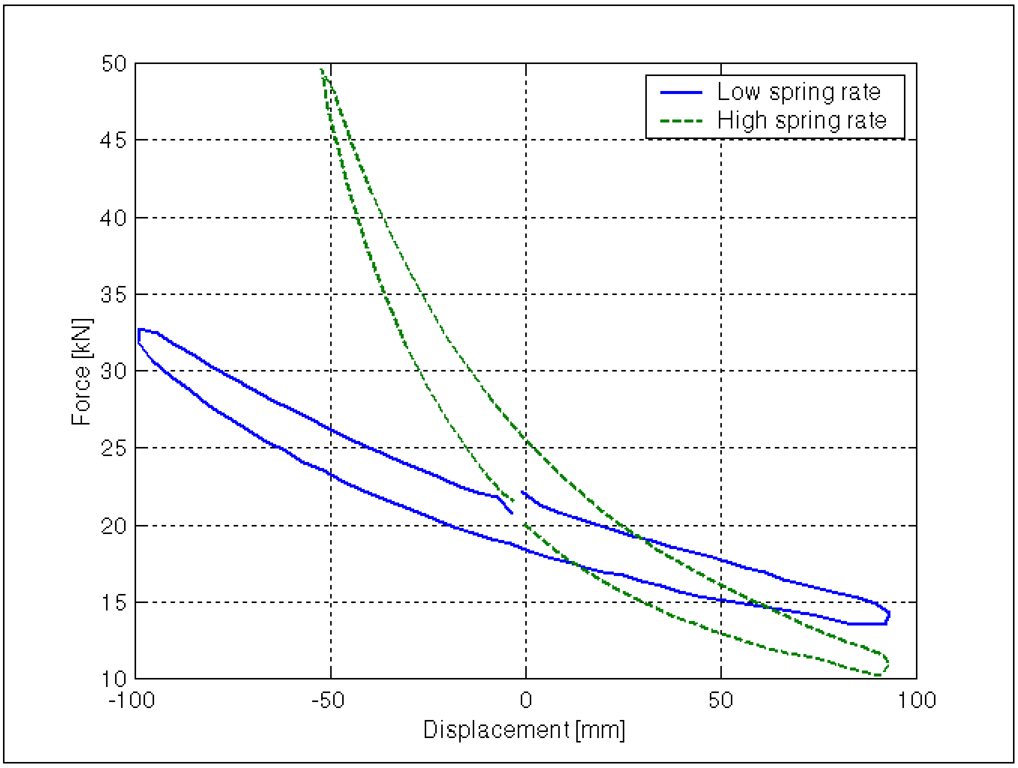

Figure 4-7: Semi-active spring characteristics (0.01*m/s*)

4.4 Hydraulic damper characterisation

In this paragraph, the methodology for determining the damping force versus velocity relationship of the semi-active damper is discussed. A description of the hardware, as well as the characterisation procedure is supplied.

4.4.1 Physical attributes

As explained previously, the low damper state is achieved by short circuiting the damper with a bypass valve. The characteristics of the valve will be discussed in more detail in paragraph 4.5. The damper pack used in the semi-active damper assembly is the same as found in a Ratel damper. No modifications were made to the damper characteristics and the damper pack was built into a custom made damper housing. The Ratel damper has non-linear damping characteristic, which is achieved by a system of orifices, sealing washers and Belville springs.

Unlike a conventional translational damper, which acts directly on relative suspension motion, this damper was mounted statically between the strut and the hydro-pneumatic springs (see Figure 1-10 in Chapter 1). The damping force is therefor supplied by resistance to fluid flow through the damper pack.

4.4.2 Characterisation procedure

The damper characteristics of the two-state damper were determined by subjecting the strut to a sinusoidal displacement input. The actuator force was recorded when the strut travels through the static position, where the velocity is almost constant and a maximum. In order to determine only the damper force, the spring force was subtracted form the total force (actuator force). The spring force was calculated from the accumulator pressure and the cylinder area. From these measurements, the force versus velocity relationship of the damper could be determined.

Figure 4-8 shows the measured damper characteristics for both the "on" and "off" states. From this figure it can be seen that the "off" characteristic for velocities above 0.25*m/s* is approximately half that of the "on" state, in the compression direction (negative velocity). It can also be seen that the damper force, in the rebound direction, shows an almost constant damping force at high velocities. This is because during rebound motion the driving force behind the hydraulic fluid is the accumulator pressure and in the compression direction, the single acting strut cylinder. Rebound damping can therefore not be made too high, since cavitation may occur at high velocities.

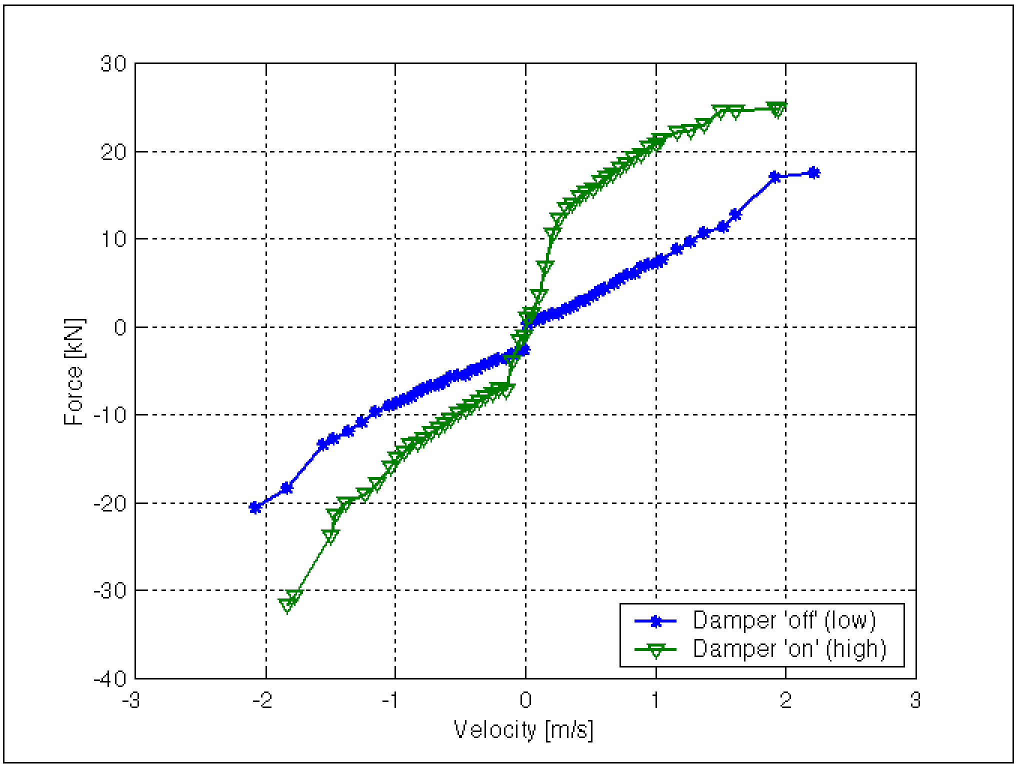

Figure 4-8: Two stage damper characteristics

4.5 Hydraulic valve

In this section, the type of hydraulic valve used in the experimental spring/damper unit is discussed. The working principle, characterisation procedure and response times are supplied in the following paragraphs.

4.5.1 Valve type and working principle

Regular solenoid valves are limited by the amount of flow they can handle. Semi-active dampers for large off-road vehicles have very large flow requirements, because of the large size of the dampers. In order to accommodate the high flow rates (up to 1000*l/min*), a logic element type valve was used.

According to DIN24342 the correct name for this type of valve is a "2-Way Cartridge Valve", but for simplicity the valve will be referred to as a hydraulic valve. The basic element is a 2/2 way valve i.e. a valve with 2 service ports and 2 operating positions, namely "open" and "closed". The logic element status is determined by the force balance on the valve poppet. From Figure 4-9, it can be seen that the pressure from the two service ports A & B, the control port X and the closing spring (3) acts on the valve poppet (2).

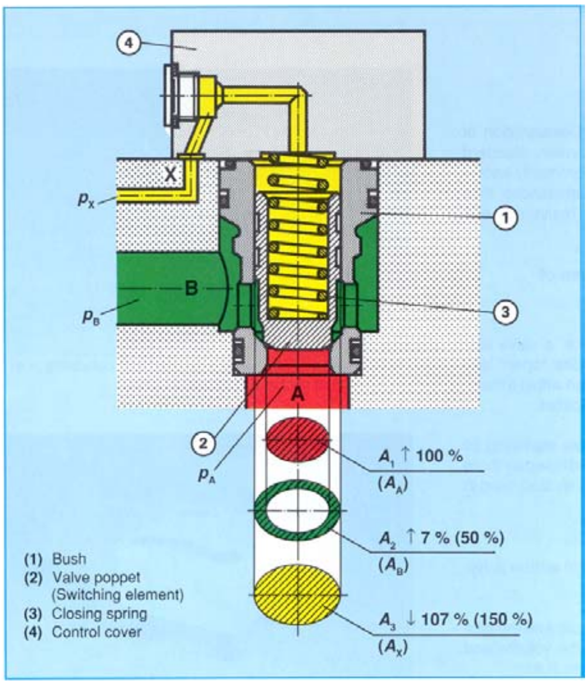

Figure 4-9: 2-Way Cartridge Valve sectional drawing

The control port pressure is switched by a Mannesman Rexroth WSE3, 3-way solenoid spool valve. The solenoid valve connects either a high- or a low-pressure line, supplied by an arrangement of one-way valves, to the control port. The control pressure determines the direction of the resultant force on the valve poppet, causing it to either open or close.

4.5.2 Valve response times

Valve response times were determined by subjecting the strut to a triangular displacement input resulting in constant velocity regions between turning points. The valves were then switched on and off in both the compression and rebound directions at different speeds. Valve response time is defined as the difference between the trigger signal and the time at which the force reaches 95*%* of its final value. This includes the initial delay and the rise time of the hydraulic valve, as described in by Nell (1993).

As was mentioned in paragraph 4.5.1, the valve state is dependant on the pressure balance on the valve poppet and therefore the valve switching times are a function of the pressure difference between the two service ports.

Figure 4-10 indicates the response times of the semi-active spring valve, as a function of force difference. The force difference is based on the pressure difference over the valve. The valve response times for the semi-active spring varies between 170*ms* at low difference in force (or pressure) to 70*ms* at high difference in force (or pressure). It can also be seen that valve response times decrease as the pressure difference increase.

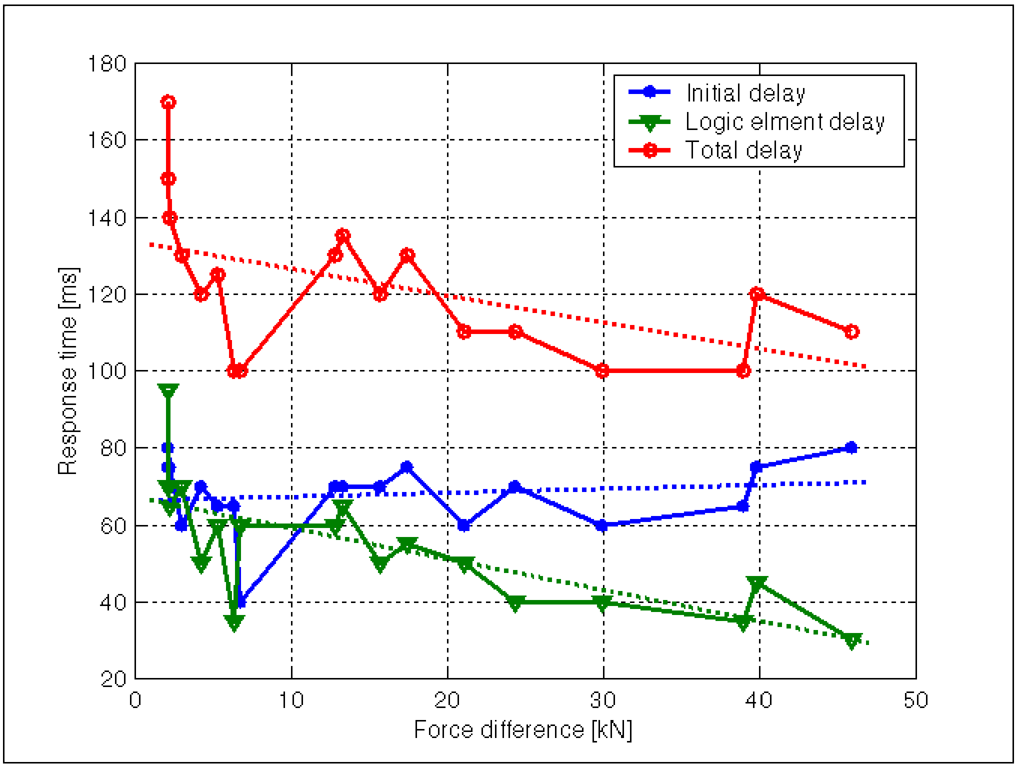

Figure 4-10: Valve response times for semi-active spring valve (on to off)

The valve response times for the semi-active damper valve decreases as the relative velocity increases. This is because an increase in velocity results in a higher differential pressure, which causes the valve to open or close quicker. Valve response times ranges from 145*ms* at low relative velocities to 40*ms* at high relative velocities. Figure 4-11 shows the valve response times for the semi-active damper valve, as a function of relative strut speed. The values indicated in this figure represent the average response time from the "off" condition to the "on" condition for both tension and compression motion.

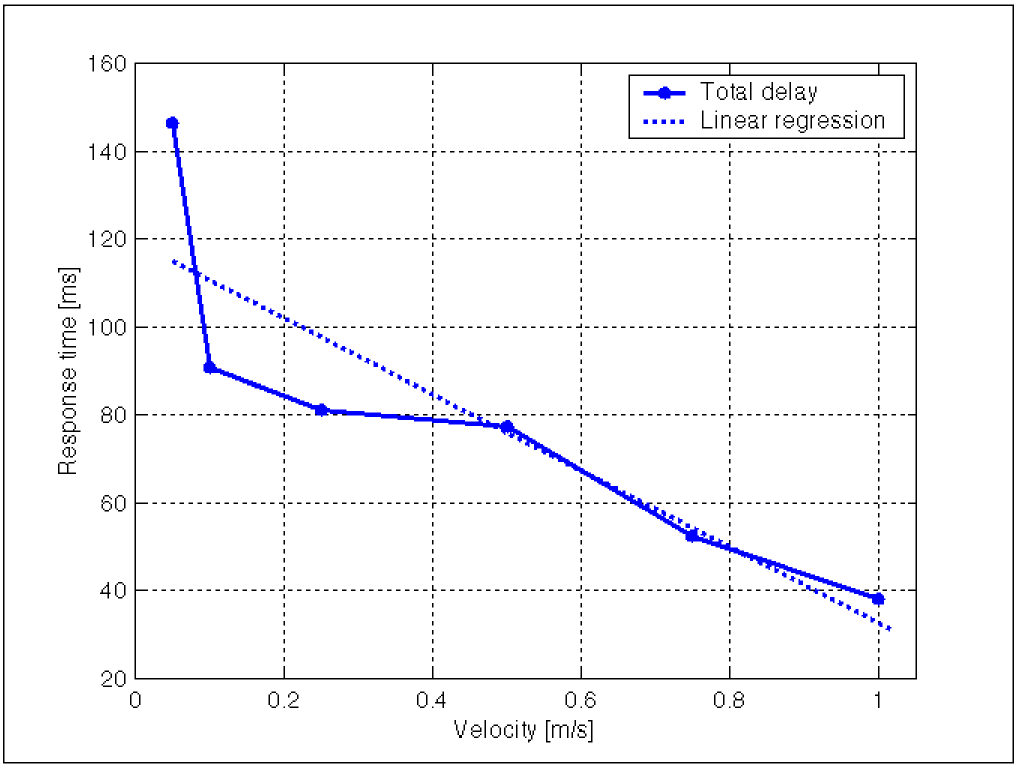

Figure 4-11: Valve response times for semi-active damper valve (off to on)

Although the valve response times are slow at low relative velocities, it is fast enough at higher velocities where fast response times are needed. Over severe off-road terrains where the wheel travel is large, the valve response times are faster.

4.6 Single degree of freedom testing

Several single degree of freedom tests were performed to evaluate the characteristics and performance of the semi-active spring/damper system. Step response, random input response, sine sweeps, as well as a novel ride height adjustment feature were evaluated. The single degree of freedom test setup, as explained in paragraph 4.2.1, was used for the tests described in the following paragraphs.

4.6.1 Step response

The purpose of the step response tests is to evaluate the spring and damper performance potential. The tests were done by subjecting the system to a step displacement input of 43*mm*. The sprung mass displacement, for different spring and damper combinations, are shown in Figure 4-12. From this figure, it can be seen that for the spring "on" condition an effective natural frequency of approximately 1,5*Hz* is achieved. For the spring "off" state, the natural frequency is in the region of 1*Hz*.

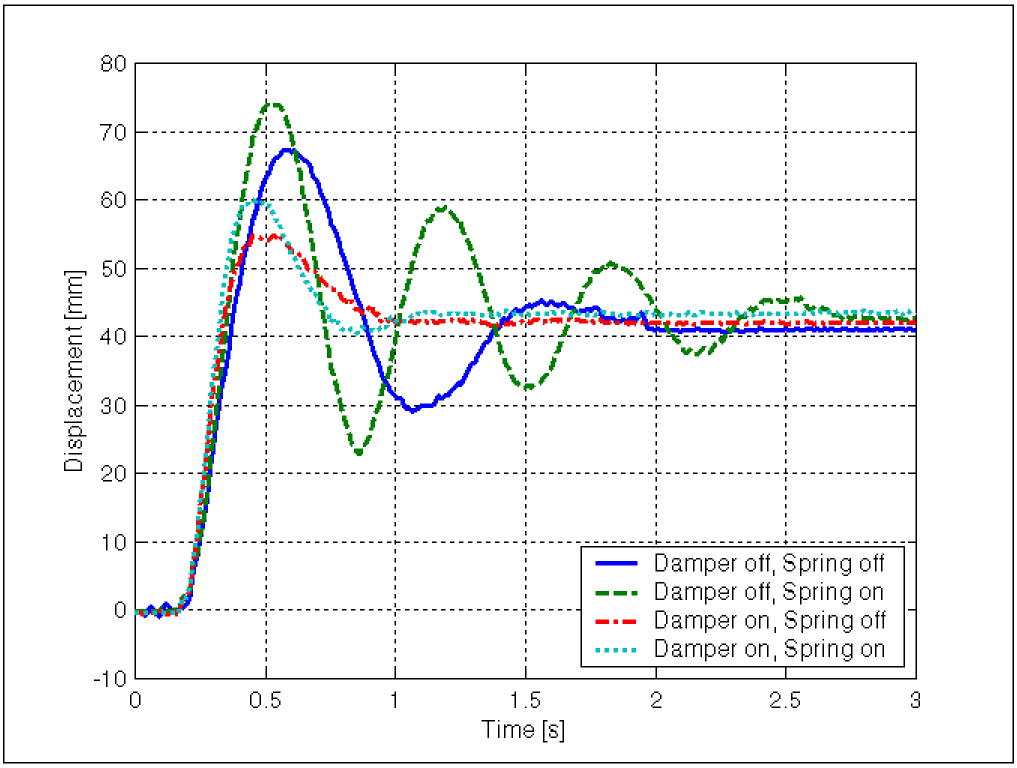

Figure 4-12: Step response of the sprung mass for different spring and damper combinations

The response for the two cases where the damper is in the "off" state indicates that the system is under damped. For the damper "on" state, the motion is damped out within one cycle.

The single degree of freedom test results indicate that good on-road, as well as off-road behavior is possible by selecting the appropriate spring and damper settings. A more detailed set of step response test results are supplied in section D.1 in Appendix D.

4.6.2 Random input response (Belgian paving)

Random input response tests were performed to evaluate some existing semi-active damper control strategies, as well as to quantify the difference in sprung mass acceleration for the two spring settings. The same control strategy used for the semi-active damper was tested for a semi-active spring strategy. The reason for applying the damper strategies directly to the spring is that the same principle holds, namely the spring setting that would minimize sprung mass acceleration is chosen.

The Belgian paving track at Gerotek Vehicle test facility was chosen as a representative random road input. The track is 100*m* long and consists of randomly packed cobblestones. The Belgian paving has a roughness coefficient of $2*10^{-5}$ and a terrain index of 2. No provision was made for including the effect of the tyre, since the purpose of the tests was only to do relative performance comparisons.

Figure 4-13 displays the sprung mass acceleration for different spring settings over the Belgian paving track. In both cases, the damper was in the passive "on" state. From the figure, it can be seen that the vibration levels for the spring "off" state is much lower than for the spring "on" state.

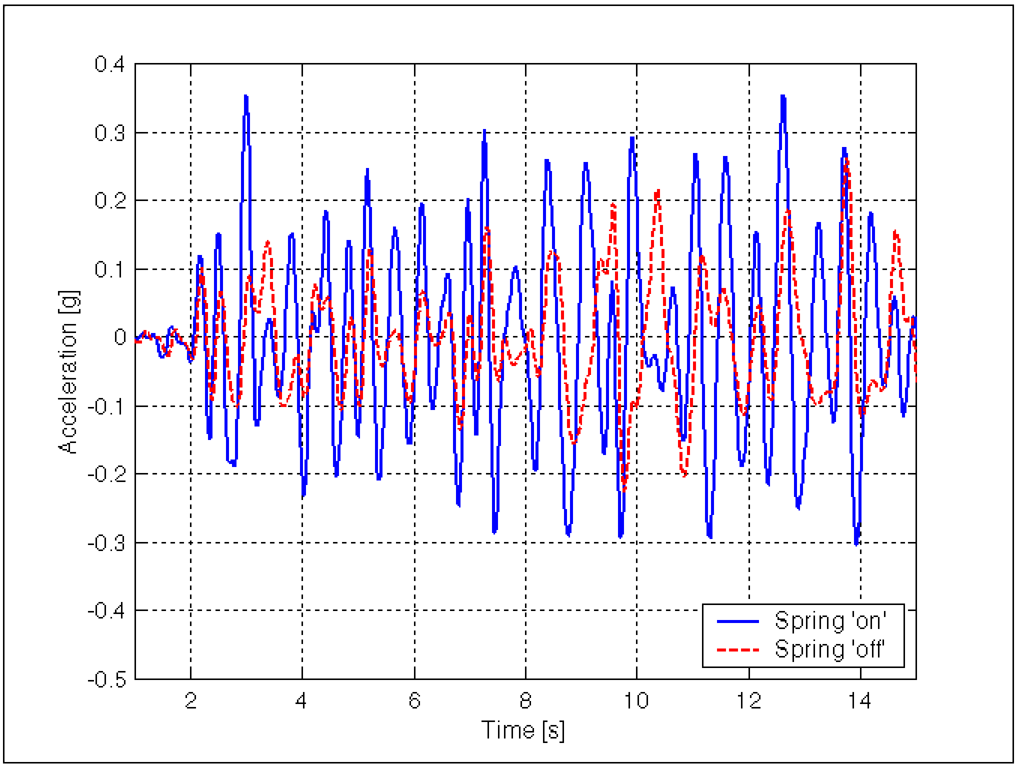

Figure 4-13: Sprung mass acceleration for different spring settings over the Belgian paving

The RMS acceleration for the spring "off" state is 0,72m/s^2 and for the "on" state 1,36m/s^2. An improvement of 47% was therefore achieved by a using different spring characteristics.

In Figure 4-14, the sprung mass acceleration for the damper "on" and the damper semi-active is displayed. In both cases, the spring was set to the passive "off" state. The semi-active control strategy of Karnopp (Barak 1989) was implemented for controlling the damper. From the figure, it is evident that much lower acceleration levels were experienced when the damper is controlled semi-actively.

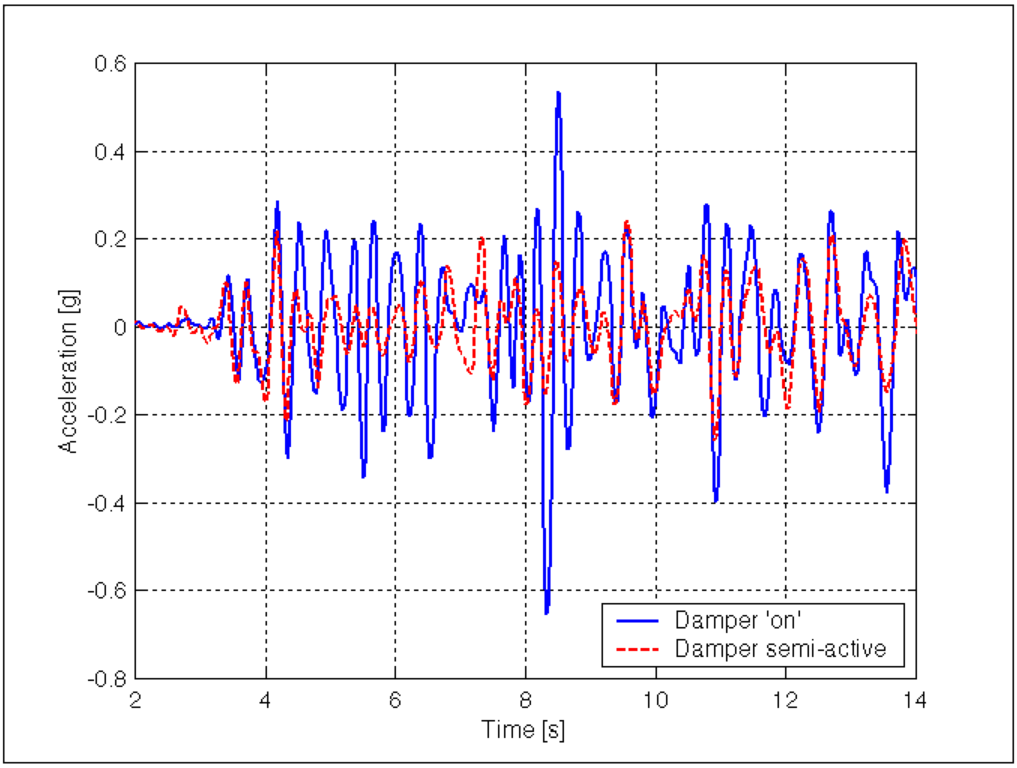

Figure 4-14: Sprung mass acceleration for the damper "on" setting and damper semi-active

The RMS acceleration for the passive "on" state is 1,48m/s^2 and for the semi-active case 0,83m/s^2. An improvement of 44%, in vertical acceleration levels was therefore achieved by controlling the damper semi-actively.

The results for different spring and damper combinations are supplied in section D.2 in Appendix D.

4.6.3 Sine sweep

A sine sweep experiment was performed to determine the properties of the system in the frequency domain. The results of these tests are supplied in section D.3 in Appendix D. Figure 4-15 shows the transmissibility of the spring/damper system for different configurations. The definition of the configurations can be found in Appendix D. From this figure, it can be seen that all the configurations show a resonance between 1Hz and 2Hz. It is also clear that the spring "off" damper "off" configuration offers the best vibration isolation at higher frequencies, while spring "on" and damper "off" resulted in severe resonance, which caused the test to abort due to high actuator force (refer to Figure D-24).

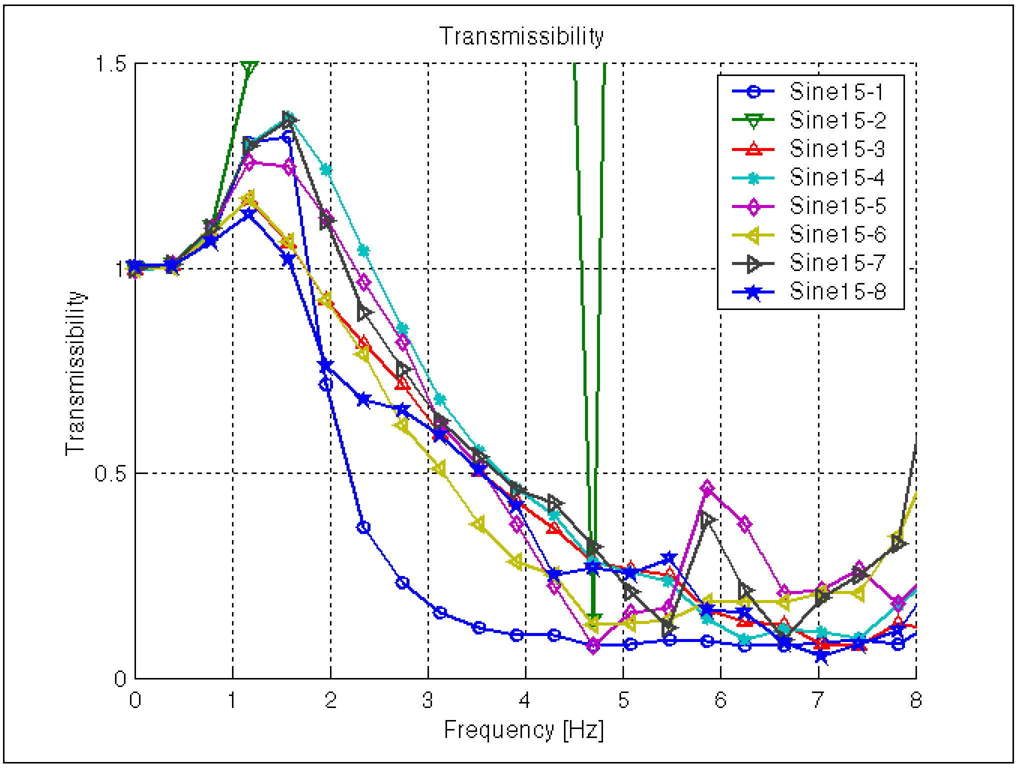

Figure 4-15: Transmissibility of spring/damper for different configurations

Although the spring/damper system is non-linear and FRF's (Frequency Response Functions) are theoretically meaningless, the transmissibility graph from the sine sweep test report on an actual operating condition, which gives some insight into the behavior of the system.

4.6.4 Ride height adjustment

The ride height of vehicles fitted with hydro-pneumatic suspensions can be easily adjusted by adding or removing hydraulic fluid from the system. A lower ride height results in reduced body roll, lower center of gravity, a more stable firing platform and a lower silhouette. The ride height can also be increased when a greater ground clearance is needed.

Ride height adjustments are usually achieved by making use of an external power source, such as an engine driven hydraulic pump. A control system then regulates the amount of hydraulic fluid in the system with a network of pipes and valves. The Swiss Mowag Piranah III is an example of a vehicle fitted with such a system (refer to Chapter 2).

The semi-active hydro-pneumatic spring/damper system investigated in this study has the added advantage of being able to adjust the ride height without using an external pump. Referring to Figure 1-10 in Chapter 1, the ride height adjustment works as follows:

When valve (6) is closed, the hydraulic fluid in the accumulator (2) is effectively removed from the system. By opening and closing the valve at the right moment an amount of hydraulic fluid can be stored in accumulator (2), thus varying the ride height. To decrease ride height, valve (6) is kept closed and only opened when the pressure in accumulator (3) is higher than in accumulator (2). This is done until the pressure in accumulator (2) reaches a predetermined value corresponding to a specific reduction in ride height. To increase the ride height, valve (6) is only opened when the pressure in accumulator (2) is higher than in accumulator (3).

The control strategy for changing ride height works well for big suspension inputs, but not very well when accumulator pressure fluctuates rapidly (high frequency suspension inputs). This problem can be solved by specifying a dead band for which the valve is not allowed to switch keeping the valve from oscillating between the "on" and the "off" state.

Figure 4-16 shows the result of the ride height adjustment control algorithm. The vehicle ride height was increased by 25*mm* within a period of 20 seconds while driving over the Belgian paving track at 20*km/h*.

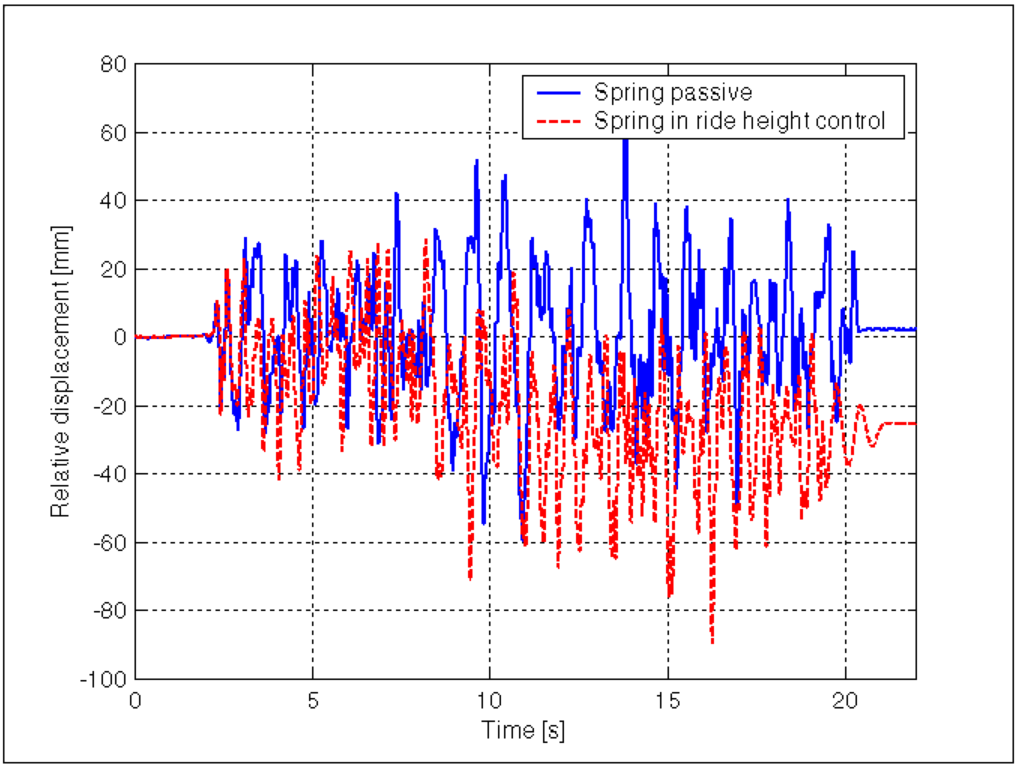

Figure 4-16: Ride height adjustment for driving over the Belgian paving track

4.7 Closing

The characteristics discussed in this chapter are first order test results to quantify the properties and performance potential of the semi-active spring/damper system. More detailed characterisation of the different elements is possible, but for the purpose of this study, the data presented in this chapter is sufficient to adequately characterise the system.

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