1. Introduction

A vehicle's suspension has a primary influence on its ride characteristics. Most of the literature available on this subject concentrates on four-wheel vehicles and in particular on passenger cars. When designing vehicles for other specialised applications, in particular for armoured personnel carriers, many of the standard design parameters may change because of the following considerations:

i. high mass and inertia may arise due to the armour and land mine protection used,
ii. high centre of gravity may occur due to the characteristic V shape hull used,
iii. use of more wheels for increased mobility may be required in order to reduce the average ground pressure to be applied,
iv. the vehicles may be required to operate over a wide range of terrain, from on-road to extreme off-road conditions, and
v. reliability is to be of utmost importance.

A specific example of such a vehicle is the Okapi 6x6 vehicle developed by Armscor, Truckmakers, Reumech Ermetek and Vickers OMC. The vehicle weighs 22 tons, uses three axles, carries electronic equipment and personnel. The vehicle uses a solid axle suspension with leaf springs. Four dampers on each of the front and rear axles and two dampers on the middle axle are used. Figure 1.1 shows the Okapi vehicle.

Figure 1.1: The Okapi vehicle.

During the development of the Okapi axle failures were experienced on the front axle of the vehicle [1, 2]. The axles used on the Okapi vehicle were the same as that used on other vehicles with similar static axle loads. In order to explain the axle failures, experimental evaluations of the Okapi vehicle and two other military vehicles, the Withings and Kwêvoël were performed. Both the Withings and Kwêvoël vehicles use similar axles as the Okapi. These vehicles were driven over the same terrain and at the same speed as the Okapi. Strain gauge measurements indicated that although the Okapi vehicle did not experience the highest axle forces, it was subjected to more severe force reversals and that the axles on the Okapi vehicle failed due to fatigue of the axles. At that stage this phenomenon could only be explained due to the specific axle configuration and suspension used on the Okapi.

An upgrade on the axles of the Okapi vehicle was performed. Higher load rating axles from a different supplier were used. At the same time a suspension upgrade was done based on the results of computer simulations. Computer-aided vehicle dynamic simulations were done with different spring and damper stiffness combinations. In successive simulations the spring stiffnesses in the vehicle were decreased and the damper configurations were changed. Due to time constraints only a limited parametric study could be performed [2].

Vehicle tests were conducted with the upgraded axles and the new suspension. During these tests damper failures were experienced. Investigation of the failed dampers indicated that too high damper deflection rates and forces were experienced during vehicle operation.

Since the start of the Okapi project it was already clear that designing a 6x6 suspension would not be a simple task. Although complete three dimensional vehicle simulation studies had been performed by the Laboratory for Advanced Engineering using GENRIT [3], and by Reumech Ermetek and Vickers OMC using DADS [4], it was realised that to fully understand the 6x6 vehicle suspension and to improve the suspension would be a difficult and cumbersome task. This would be so mainly due to the large number of variables involved and the limit on available computational time.

The need for the present study arose in parallel with experiencing the above problems with the Okapi vehicle suspension. Clearly a simple but yet effective systematic vehicle suspension modelling and optimisation system, enabling improvement of vehicle suspensions was required. The program should be such that it can be used by the project manager and the design team to simulate the vehicle suspension and to determine the parameters to be adjusted so as to improve certain evaluation criteria. In addition the system should be robust, user friendly and easy to implement without the need to simulate the vehicle in the finest detail. For these first order simulations, required during the design stage, it is suggested that a two-dimensional vehicle dynamic simulation program would be appropriate. Although the specific vehicle example is a 6x6 (three axle) vehicle, the same problems occur during the concept design of 4x4 (two axles) and 8x8 (four axles) vehicles.

In a study by Etman [5] a suspension optimisation was performed during the design of a stroke dependent damper for the front axle suspension of a truck. The first optimisation was done using a quarter car model. Due to its simplicity the quarter car model is not highly accurate [5] and a great deal of the dynamic behaviour of the truck is not included. A second optimisation study was done using a full-scale (3-D) DADS model of the truck with a total of 34 degrees of freedom. In his conclusion Etman [5] states that the step from a quarter car model to a full-scale model "has been pretty large" and suggests that possibly a six degree of freedom model should rather be used.

In the remainder of this chapter the basic principles of vehicle suspension are described and a brief literature survey of optimisation of vehicle suspensions is presented. Finally the achievements and shortcomings of the existing studies will be discussed and the specific goal of the present study will be outlined.

1.1 Primary functions of the vehicle suspension

According to Gillespie [6] the primary functions of a vehicle suspension system are:

i. to provide vertical compliance so the wheels can follow the uneven road, isolating the chassis from roughness in the road,
ii. to maintain the wheels in the proper steer and camber attitudes to the road surface,
iii. to react to control forces produced by the tyres,
iv. to resist roll of the chassis, and
v. to keep the tyres in contact with the road with minimal load variations.

The importance of the suspension system in order to provide suitable ride and handling have long been known, for example Sternberg [7] states that: "*The suspension provides a cushioning means between the axle and the frame and therefore is important in providing suitable ride qualities, as well as providing suitable cushioning properties for the cargo. Improved riding qualities are becoming of greater importance and, therefor, the role of suspensions in providing improved cushioning for both the driver and the cargo will receive more attention in the future.*"

Gillespie [6] further states the following regarding the vehicle suspension: "*The properties of a suspension important to the dynamics of the vehicle are primarily seen in the kinematic (motion) behaviour and its response to the forces and moments that it must transmit from the tyres to the chassis. In addition, other characteristics considered in the design process are cost, weight, package space, manufacturability, ease of assembly and others*".

As can be seen from Sternberg and Gillespie the importance of the vehicle suspension in regard to the vehicle quality, ride comfort, handling, driver perception, etc. are of primary consideration during the development and design of vehicles. This is also apparent from the huge amount of research that has been done in this regard (see references).

In general improvement of vehicle suspension will give the following:

i. improvement of vehicle ride quality,
ii. increased vehicle mobility, and
iii. increased component life.

The specific requirements for the vehicle suspension are directly coupled to the mobility requirements and the application of the vehicle. For example in some instances ride comfort may be sacrificed to improve handling or visa vèrsa.

1.2 Vehicle suspension systems

Vehicle suspension systems can be divided into two groups, namely solid axle and independent suspensions [6, 7, 8, 9]. Figures 1.2 and 1.3 show the difference between the two types of suspension systems.

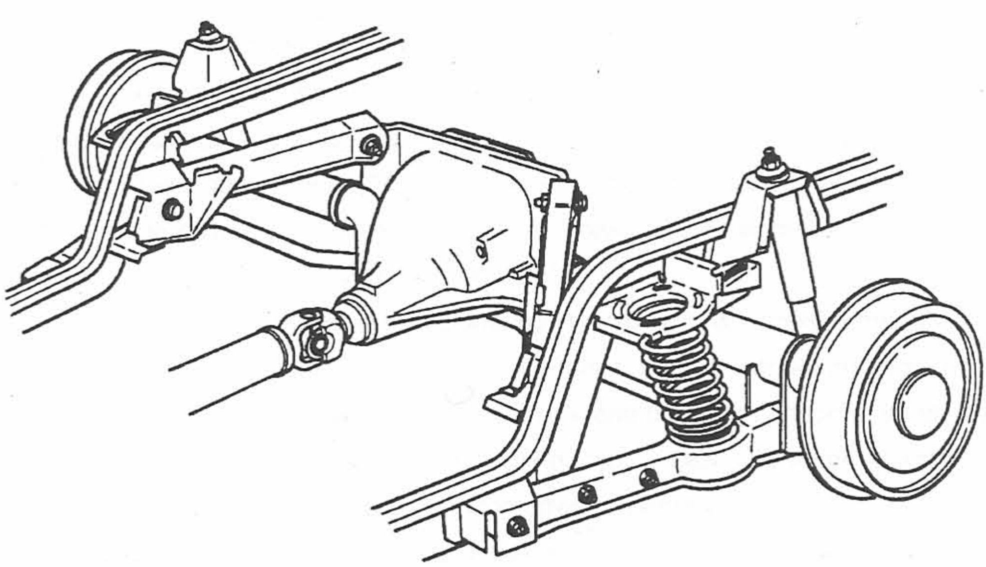


Figure 1.2: Example of a solid axle suspension (after Gillespie [6]).

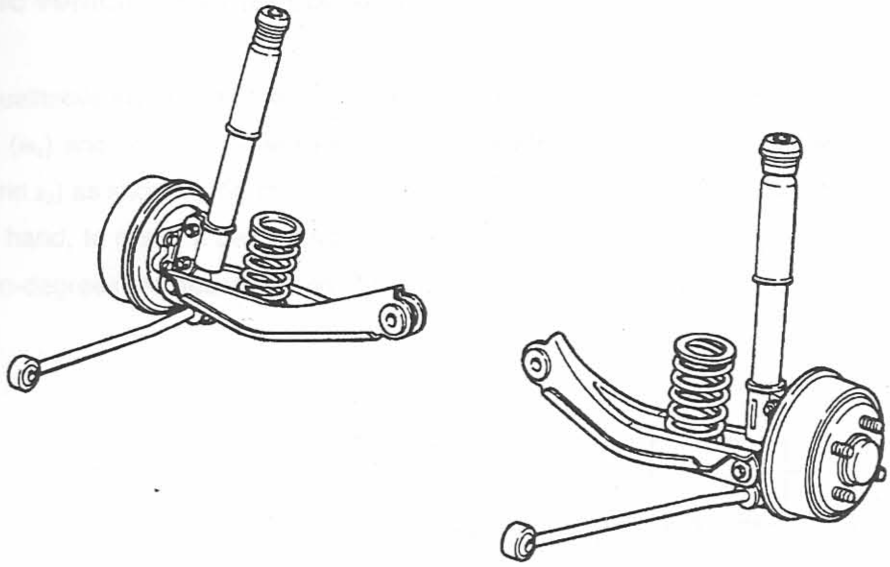


Figure 1.3: Example of an independent suspension system [6].

Many variations of these two suspension systems exist [6, 7, 8, 9]. The main components of a suspension system are the following:

i. Geometrical components. Examples are: connecting rods, swing arms, Panhard rods, links, trailing arms, sliding tubes etc. The functions of these components are to connect the wheels/axles to the vehicle body and to keep the tyres in the proper camber and steer attitude.

ii. Springs. The springs of the suspension can be divided into leaf springs, coil springs, torsion bars, rubber springs and air springs. The function of the springs are explained by Nunney [8]: *"When the road wheels rise and fall over surface irregularities, the springs momentarily act as energy storage devices and thereby greatly reduces the magnitude of loading transmitted by the suspension system to the vehicle structure".*

iii. Dampers. According to Gillespie [6]: *"Damping in suspensions comes primarily from the action of hydraulic shock absorbers. Contrary to their name, they do not absorb the shock from road bumps. Rather the suspension absorbs the shock and the shock absorber's function then is to dissipate the energy put into the system by the bump."* Telescopic shock absorbers have been used almost exclusively for damping in automotive suspensions.

iv. Bump stops. These are in principle secondary springs that limit the total suspension deflection. Generally bump stops consists of a rubber stop.

1.3 Basic vehicle ride models [6, 9]

To obtain a qualitative insight into the functions of the suspension and in particular of the effects of the sprung mass (m_s) and unsprung mass (m_{us}) on vehicle vibration, a linear model with two degrees of freedom (z_1 and z_2) as shown in figure 1.4, may be used. This is the so-called quarter car model [5, 6]. On the other hand, to reach a better understanding of the pitch and bounce of the vehicle body, an alternative two-degrees of freedom (z and θ) model as shown in figure 1.5 may be preferable.

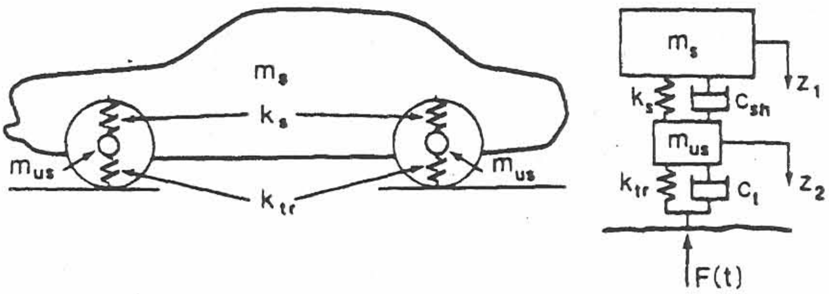


Figure 1.4: A two-degrees of freedom ride model for the sprung and unsprung mass (after Wong [9]).

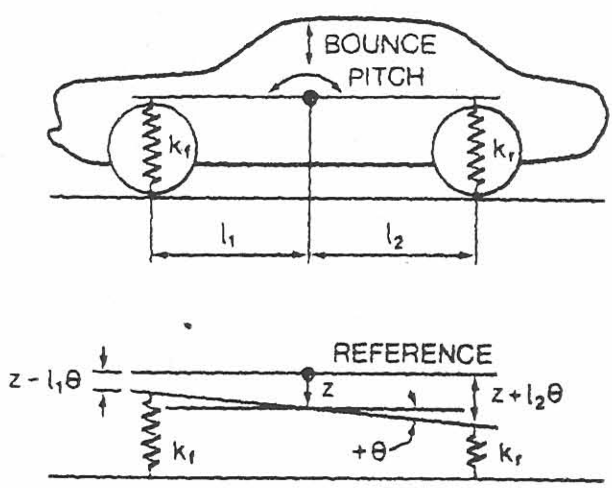


Figure 1.5: A two-degrees of freedom model for pitch and bounce of the sprung mass [9].

1.3.1 The quarter car model for sprung and unsprung masses

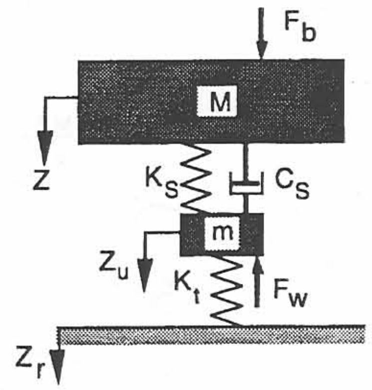


Figure 1.6: Quarter car model (after Gillespie [9]).

The two degrees of freedom quarter car model shown in figure 1.6 includes an unsprung mass m consisting of the wheels and associated components, and a sprung mass M representing the vehicle body. The respective displacements of the masses M and m in the vertical direction are denoted by Z and Z_u, with origins at the static equilibrium positions. Using Newton's second law and neglecting the effect of the tire damping, the equations of motion for the system can be obtained [6,9]:

for the sprung mass:

$$M\ddot{Z} + C_s(\dot{Z} - \dot{Z}_u) + K_s(Z - Z_u) = F_b \tag{1.1}$$

and for the unsprung mass:

$$m\ddot{Z}_u + C_s(\dot{Z}_u - \dot{Z}) + K_s(Z_u - Z) + K_t(Z_u - Z_r) = -F_w \tag{1.2}$$

where

Z = sprung mass displacement [m]
Z_u = unsprung mass displacement [m]
Z_r = road displacement [m]
F_b = external force on the sprung mass [m]
F_w = external force on the unsprung mass [m]
C_s = damping coefficient of the suspension shock absorber [Ns/m]
K_s = spring rate of the suspension spring [N/m]
K_t = spring rate of the tyre [Ns/m]

The above equations can be solved analytically or by numerical methods. The typical response properties are shown in figure 1.7

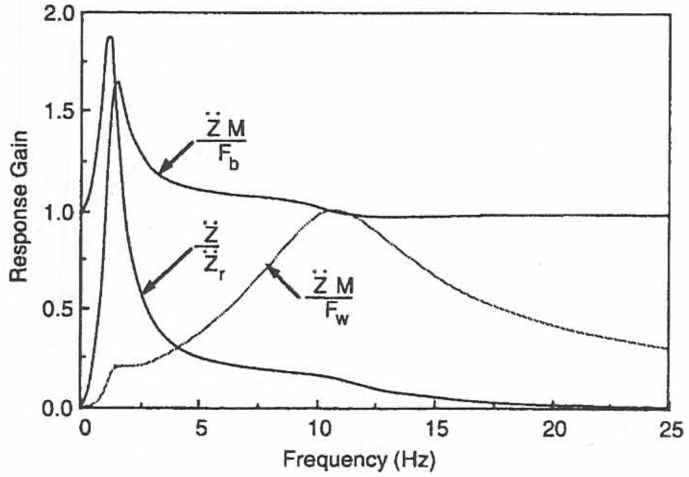


Figure 1.7: Quarter car model response to road, tire/wheel and body inputs [6].

The natural (undamped) frequencies (Hz) of the sprung and unsprung mass are given by [6, 9]:

$$f_{n,s} = \frac{1}{2\pi}\sqrt{\frac{K_s K_t / (K_s + K_t)}{M}} \tag{1.3}$$

and

$$f_{n,us} = \frac{1}{2\pi}\sqrt{\frac{K_s + K_t}{m}} \tag{1.4}$$

When damping is present the damping ratio is given by [6]:

$$\varsigma = \frac{C_s}{\sqrt{4K_s M}} \tag{1.5}$$

and the damped natural frequency for the sprung mass by [6]:

$$f_{d,s} = f_{n,s}\sqrt{1-\varsigma^2} \tag{1.6}$$

For acceptable ride comfort the damping ratio normally falls between 0.2 and 0.4. Within this range the damped natural frequency is close to the undamped natural frequency. The undamped natural frequency is commonly used to characterise the vehicle suspension.

1.3.1.1 Suspension stiffness

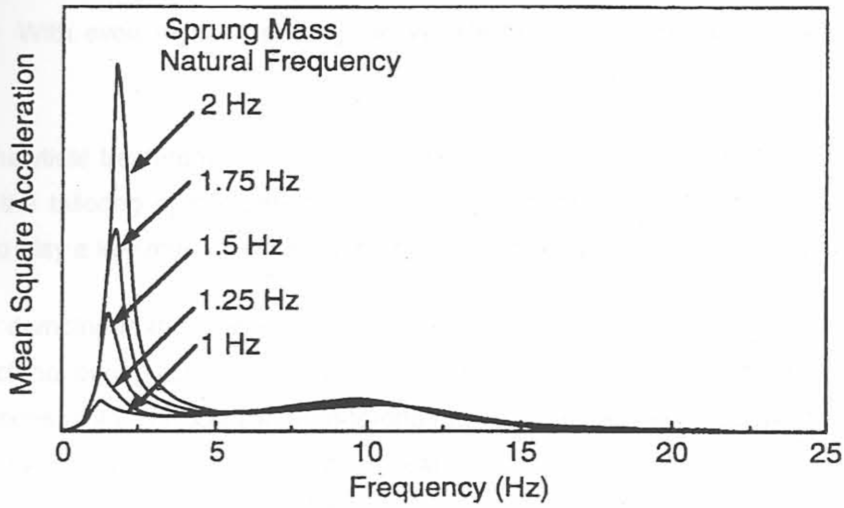


Figure 1.8: Effect of suspension stiffness [6].

The effect of the suspension stiffness (higher stiffness - higher natural frequency) on the mean square acceleration of the sprung mass is shown in figure 1.8. While this analysis clearly shows the benefits of keeping the suspension soft for ride isolation, the practical limits of stroke that can be accommodated within a given vehicle size constrain the natural frequency for most cars to a minimum of 1 to 1.5 Hz. Performance cars, for which ride comfort is sacrificed for handling benefits, have stiff suspensions with natural frequencies of 2 to 2.5 Hz [6].

1.3.1.2 Damping

The nominal effect of damping is shown in figure 1.9.

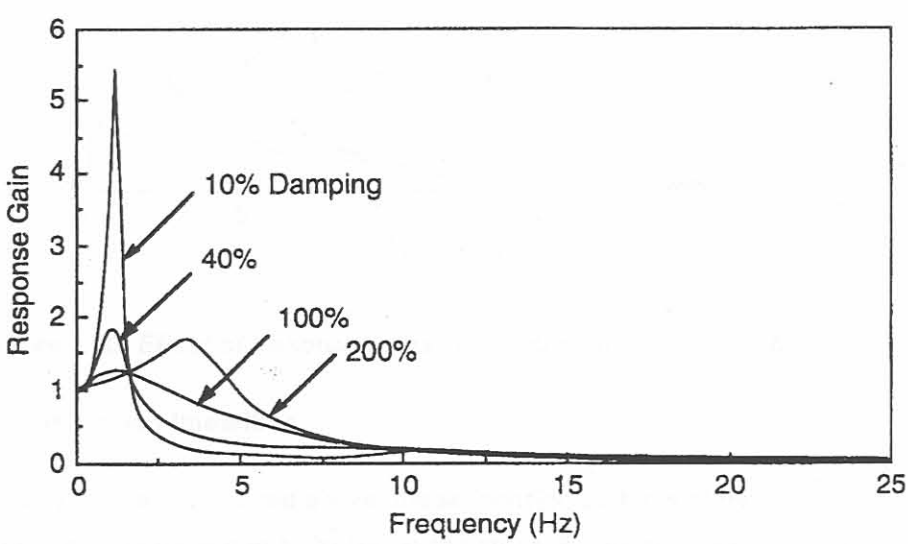


Figure 1.9: Effect of damping on suspension isolation behaviour [6].

At light damping (10%) the response is dominated by a very high response at the natural frequency. The 40% damping (damp ratio = 0.4) is representative of most cars. At 100% damping (critical damping) the natural frequency bounce motion will be controlled but with penalties at higher frequencies. With even higher damping the vehicle bounces on the tires resonating at a higher frequency.

While this analytical treatment provides a simplified illustration of the ride effect of damping in the suspension, the tailoring of dampers to achieve optimum performance is much more complicated. Dampers also play a key role in keeping good tire-to-road contact for road holding and safety.

Normally the damping in the jounce (compression) and rebound (extension) directions are not equal due to the fact that during compression damping aids in the transmission of forces to the sprung mass and is thus undesirable. Typically a three-to-one ratio is used between rebound and jounce damping. The damping for typical dampers is also non-linear.

For a realistic treatment of the dampers in ride analysis the damper must be treated as a non-linear element.

1.3.1.3 Wheel hop

Figure 1.10 indicates the effect of the unsprung mass on the response gain of the sprung mass. As can be seen from figure 1.10 it is best to keep the unsprung mass as light as possible.

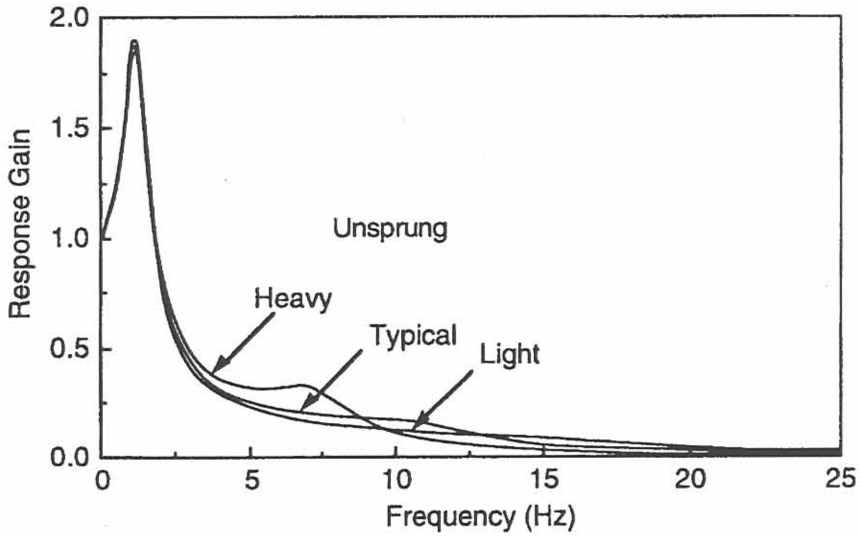


Figure 1.10: Effect of unsprung mass on suspension isolation behaviour [6].

1.3.1.4 Suspension non-linearities

In practice, contrary to that assumed above, the suspension systems of many vehicles are not linear. Non-linearities are introduced due to friction in the struts, bushings, interleaf friction in a leaf spring and other design aspects. In figure 1.11 the solid line indicates the force versus deflection relationship as measured for a particular leaf spring. Due to non-linearities in the spring, especially friction, a hysteresis loop is obtained rather than a single line. The dotted lines indicate the nominal stiffness for the different deflection ranges. From figure 1.11 it can be seen that the effective spring rate for a leaf spring at small deflections can be typically three times the nominal rate.

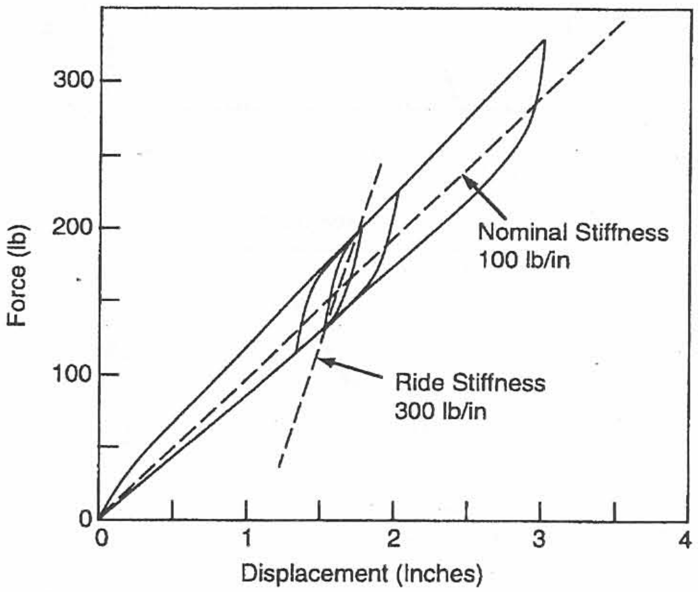


Figure 1.11: Load-deflection characteristics of a hysteretic leaf spring [6].

Figure 1.12 indicates the effect of suspension non-linearities on the response gain for roads with the same frequency content, but at high (rough) and low (smooth) amplitudes. The top figure (for the smooth road) in figure 1.12 shows that the natural frequency of the sprung mass is higher (± 2.3 Hz) than that for the rough road (bottom figure, ± 1.6 Hz). The response gain at the sprung mass natural frequency is also higher for the smooth road than that for the rough road. Both these observations point to the higher nominal stiffness experienced by the leaf spring at smaller deflections on the smooth road. The same tendency can be seen at the unsprung mass natural frequency (± 10 Hz).

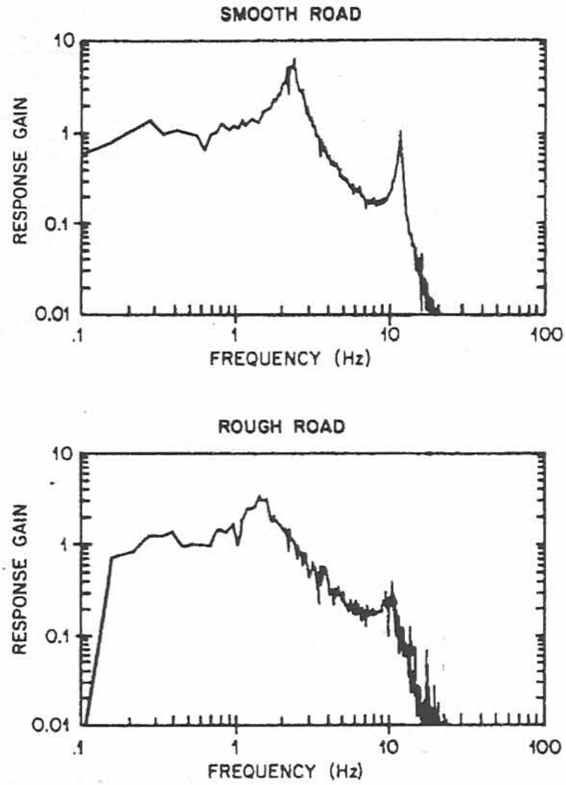


Figure 1.12: Response of a quarter car model with a hysteretic suspension [6].

1.3.2 Two-degrees of freedom model for pitch and bounce

Obviously the simple mechanics of the quarter car model does not fully represent the rigid-body motion that may be experienced by a motor vehicle. Because of the longitudinal distance between the axles, it is a multi-input system that responds with pitch motions as well as vertical bounce. The discussion of the combination of pitch and bounce that follows in this section is also mainly taken from Gillespie [6]. Figure 1.13 indicates this phenomena for different sinusoidal inputs of varying wavelength. As can be seen in the left figure in figure 1.13 pure bounce is induced at integer spatial frequencies and pure pitch in between (right figure in figure 1.13). At any other spatial frequency both pitch and bounce will be induced.

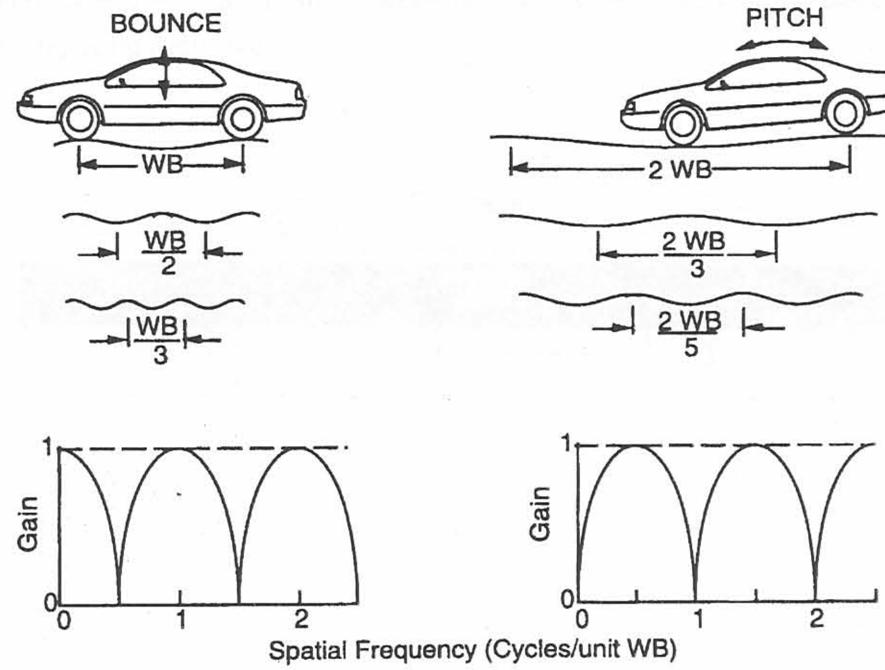


Figure 1.13: The wheelbase filtering mechanism [6].

The combination of pitch and bounce as well as the position on the vehicle will also alter the response gain, as shown in figure 1.14.

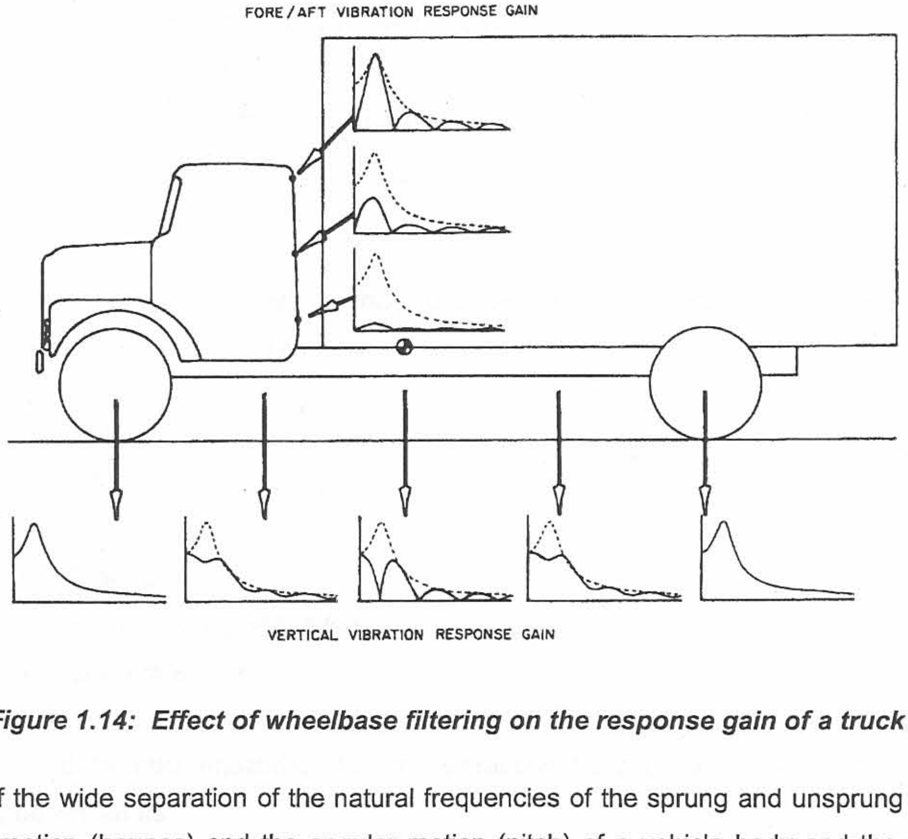


Figure 1.14: Effect of wheelbase filtering on the response gain of a truck [6].

Because of the wide separation of the natural frequencies of the sprung and unsprung mass, the up and down motion (bounce) and the angular motion (pitch) of a vehicle body and the motion of the wheels may be considered to exist almost independently. Figure 1.15 shows the basic model with the effect of damping being neglected.

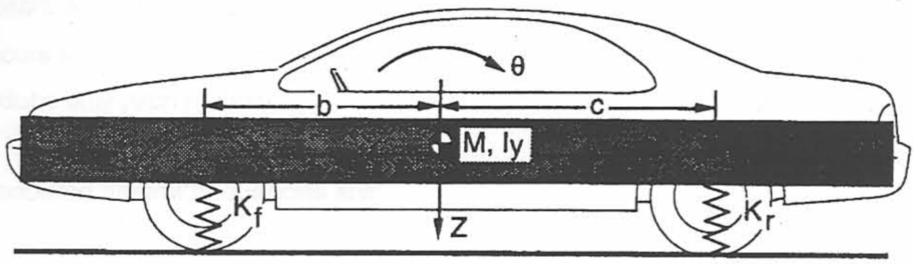


Figure 1.15: Pitch plane model for a motor vehicle [6].

The following parameters are defined:

$$\begin{aligned} \alpha &= (K_f + K_r)/M \\ \beta &= (K_r c - K_f b)/M \\ \gamma &= (K_f b^2 + K_r c^2)/(Mk^2) \end{aligned} \tag{1.7}$$

where

K_f = *front ride rate [N/m]*

K_r = *rear ride rate [N/m]*

b = *distance from the front axle to the CG [m]*

c = *distance from the rear axle to the CG [m]*

I_y = *pitch moment of inertia about CG [kgm²]*

k = *radius of gyration about CG [m]*

M = *mass of vehicle body [kg]*

Neglecting the mass of the wheels the ride rate, i.e. the effective spring stiffness due to the tyre stiffness and spring stiffness clearly is:

$$K = \frac{K_s K_t}{K_s + K_t} \tag{1.8}$$

where

K = *ride rate [N/m]*

K_s = *suspension spring rate [N/m]*

K_t = *tire spring rate [N/m]*

The governing differential equations, for free vibrations for bounce Z and pitch θ motions, can respectively be written as:

$$\begin{aligned} &\ddot{Z} + \alpha Z + \beta\theta = 0 \\ &\ddot{\theta} + \beta Z / k^2 + \gamma\theta = 0 \end{aligned} \tag{1.9}$$

Only β appears in both equations and is therefor called the coupling coefficient. When $\beta = 0$ no coupling occurs and a vertical force at the CG produces only bounce and a pure torque applied to the chassis produce only pitch motion.

The two associated natural frequencies are:

$$\begin{aligned} f_1 &= \frac{1}{2\pi}\sqrt{\frac{(\alpha+\gamma)}{2} + \sqrt{\frac{(\alpha-\gamma)^2}{4} + \frac{\beta^2}{k^2}}} \\ f_2 &= \frac{1}{2\pi}\sqrt{\frac{(\alpha+\gamma)}{2} - \sqrt{\frac{(\alpha-\gamma)^2}{4} + \frac{\beta^2}{k^2}}} \end{aligned} \tag{1.10}$$

The positions of the two oscillation centres from the CG are given by:

$$\begin{aligned} \ell_1 &= \frac{\beta}{(2\pi f_1)^2 - \alpha} \\ \ell_2 &= \frac{\beta}{(2\pi f_2)^2 - \alpha} \end{aligned} \tag{1.11}$$

When the centre distance is positive it is ahead of the CG and if negative behind the CG. When the centre lies outside the wheel base the motion is predominantly bounce and the associated frequency is the bounce frequency. For the centre within the wheel base, the motion will be predominantly pitch and the associated frequency the pitch frequency. These two cases are illustrated in figure 1.16.

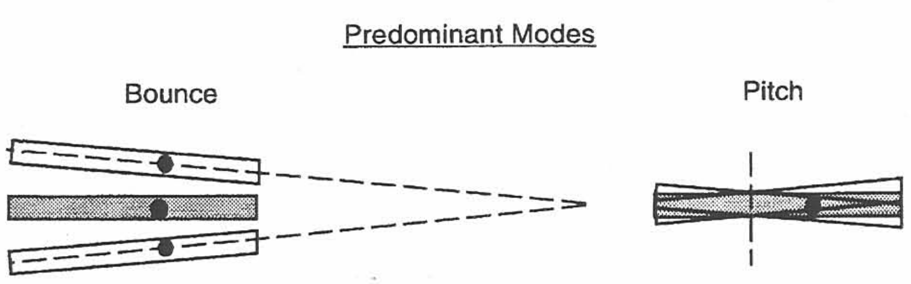


Figure 1.16: The two vibration modes of a vehicle in the pitch plane [6].

For acceptable ride comfort the so-called Olley criteria are suggested:

i. The front suspension should have a lower natural frequency than the rear suspension. Pitching is more annoying than bouncing. The desirable ratio of front frequency to rear-end frequency depends on the wheel base of the vehicle, the average driving speed and the road conditions.
ii. The pitch and bounce frequencies should be close together.
iii. Both frequencies must be smaller than 1.3 Hz.

The Olley dynamic index is defined as:

$$DI = \frac{k^2}{bc} \tag{1.12}$$

The following special cases for the value of the dynamic index and for the other parameters are of particular importance:

i $DI = 1$

In this case the oscillation centres are located at the front and the rear axles. This is desirable for good ride if Olley's criteria are also satisfied. There is no interaction between the front and rear suspensions.

ii $\beta = 0$ (uncoupled motion)

The pitch and bounce oscillations are totally independent. Poor ride results because the motions can be very irregular. Coupling tends to even out ride.

iii $\beta = 0$ and $DI = 1$

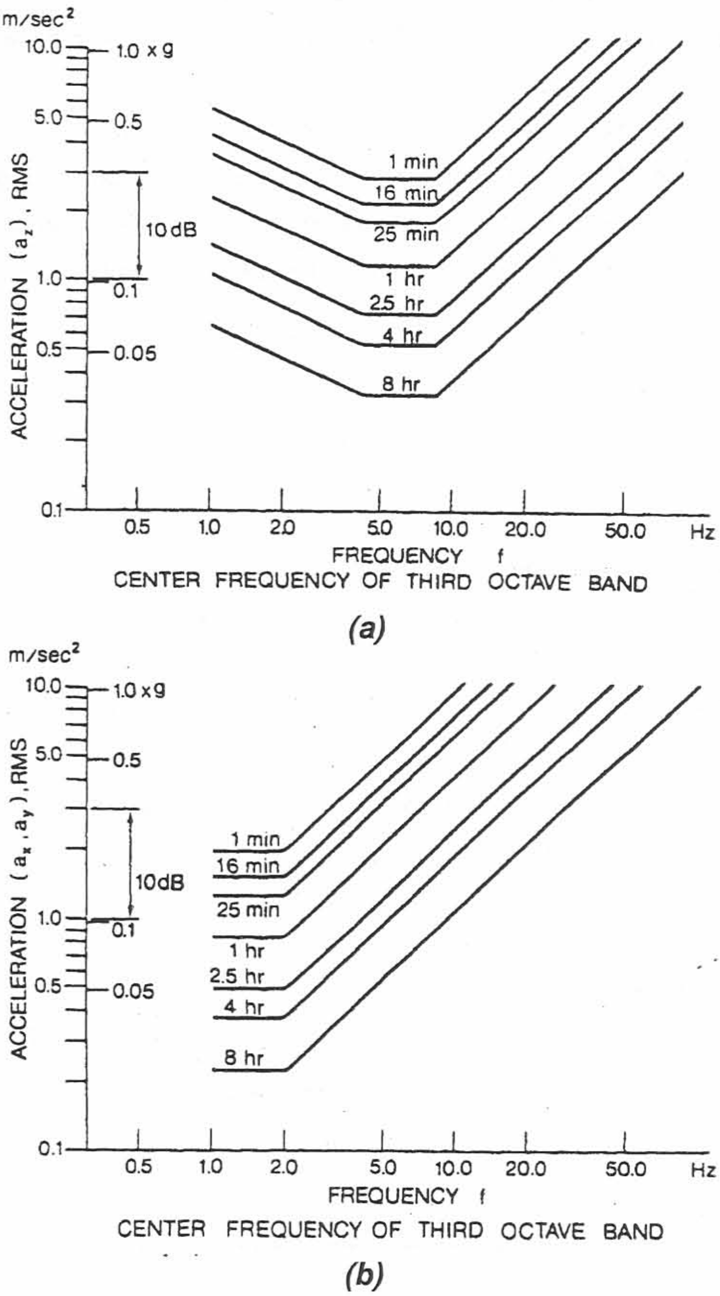
This condition results in equal bounce and pitch frequencies. The ride is inferior because there is essentially no pattern to the road-generated motion; it is quite unpredictable.

1.4 Human response to vibration

According to Gillespie [6]: *"Ride is a subjective perception, normally associated with the level of comfort experienced when travelling in a vehicle"*. Although ride, ride quality or ride comfort is a subjective concept, it is nevertheless necessary to attempt a measurement of this quality so as to be able to evaluate the ride comfort of the vehicle objectively.

A guide to defining the human tolerance to whole-body vibration has been developed by the International Organization for Standardization and published as ISO 2631 [10]. The limits for the vertical and transverse directions are respectively shown in figure 1.17(a) and (b). These limits are for decreased proficiency, which are related to the preservation of working efficiency, and apply to such tasks as driving a road vehicle or tractor. Figure 1.17(a) indicates the proficiency boundaries for <u>vertical</u> vibration, which are defined in terms of root mean square values of acceleration as a function of frequency for various exposure times. It can be seen that as the average daily exposure time increases, the boundary lowers. Figure 1.17(b) indicates the boundaries for the <u>transverse</u> direction, i.e. for side-to-side or chest-to-back vibration.

The tolerance curves shown in figure 1.17 are generally determined from sinusoidal inputs, while the actual ride environment in a motor vehicle contains all frequencies over a broad spectrum. Thus in the application of the above information to obtain objective measurements of ride vibration on the seat of a car or truck, it is necessary to overcome this deficiency. One method is to filter the acceleration data in inverse proportion to the selected tolerance curve, i.e. to accentuate acceleration that occurs at frequencies for which lower tolerance levels exists. The inverse filtering then allows the resultant acceleration spectrum to be viewed as if all frequencies are equally important. In this approach the fore and aft vibration must be evaluated separately.



(a)

(b)

Figure 1.17: ***Limits of whole-body vibration for fatigue or decreased proficiency in:***
(a) ***vertical direction and***
(b) ***transverse direction, as recommended by ISO2631 [10].***

According to the study done by Nell [11], the ISO2631 criterion has certain limitations in measuring the ride comfort of vehicles. One of these limitations is that the limit curves only extend to 1 Hz, whereas many off-road vehicles have suspension resonance frequencies around 1 Hz. Responses below this frequency are therefore also important. A more fundamental measure is the average absorbed power (*AAP*) technique [11, 12]. Here it is assumed that the level of discomfort is related to the vibration power being dissipated in the subject's body, whether vertical, fore/aft, or lateral (side to side) inputs. In this method the tolerance curves are used to weigh accelerations so as to arrive at an absorbed power for each direction and the power quantities are then simply added [11].

Another method used to evaluate ride comfort is to determine the vibration dose value (*VDV*) as calculated according to BS 6841 [11, 13]. For crest factors (defined as the modulus of the ratio of the maximum instantaneous peak value of the frequency weighted acceleration signal to its root mean square (rms) value) of more than 6, the following relationship is used:

$$VDV = \{ \int_0^T [a_w(t)]^4 \, dt \}^{0.25} \qquad (1.13)$$

where

$a_w(t)$ = the instantaneous frequency-weighted acceleration and

T = the duration of the measurement.

The weighted acceleration is obtained by filtering the actual acceleration with a frequency function that is prescribed according to the specific direction of the acceleration. The vibration dose value can also be transformed to an equivalent 4-hour vibration dose value for linear and rotational acceleration by using:

$$VDV = (\frac{t_0}{t_1} VDV_1^{\,4})^{0.25} \qquad (1.14)$$

where

t_0 = the total period of vibration duration (4 hours for the 4-hour *VDV*),

t_1 = the duration of one representative period, and

VDV_1 = the vibration dose value calculated for one representative period.

The 4-hour *VDV* is then equivalent to the *VDV* that would have been obtained if the specific vehicle response was experienced for a duration of 4-hours.

In a local study by Els [14] comparison of measured criteria for ride-comfort were compared with the subjective evaluations of different people. In the experiment a 14 ton, 4x4 mine protected military vehicle (see figure 1.18) was driven over different terrain, at various speeds and tyre pressures. The terrain was chosen to be representative of typical South African operating conditions and such that it excites a significant amount of body roll, pitch and yaw motion. Seven groups, each consisting of 8 or 9 people were used for determining subjective comments while, simultaneously, recording acceleration data required for the objective analysis. People of different postures, age groups and experience were used. Figure 1.19 indicates the subjective response of the different groups to the different tests. A high value in the subjective comment corresponds to poor ride-comfort experienced. Figure 1.20 indicates the correlation for the 4-hour *VDV* value with the subjective value. In the conclusion of his study Els states: "*The conclusion from this investigation is that any of the four methods under consideration, namely ISO2631, BS6841, AAP and VDI2057, can be used to objectively determine ride comfort for the vehicles and terrain of importance similar to those used in this study*". **Accordingly, in this study the vibration dose value (VDV) will be used as an objective indication of the ride-comfort of a vehicle.**

Figure 1.18: Test vehicle for ride comfort evaluation used by Els [14].

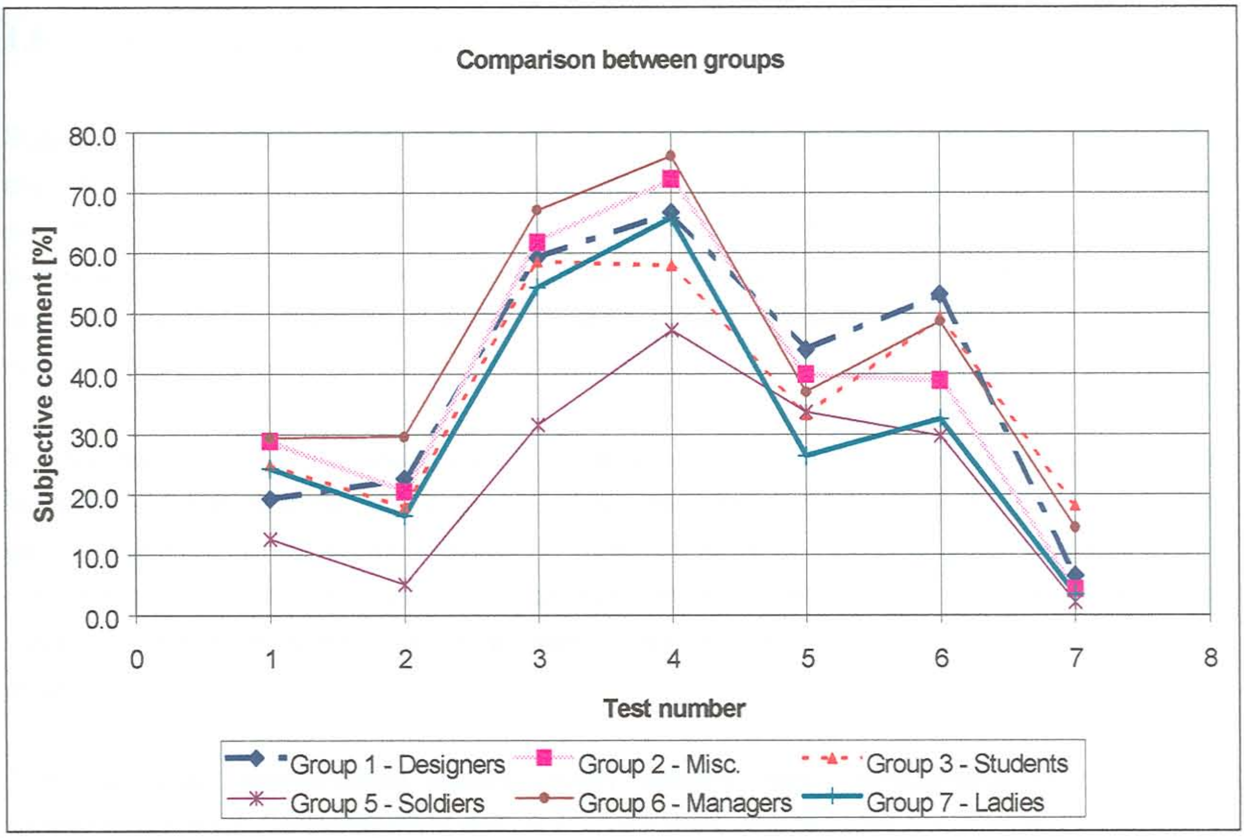


Figure 1.19: Comparison between groups [14].

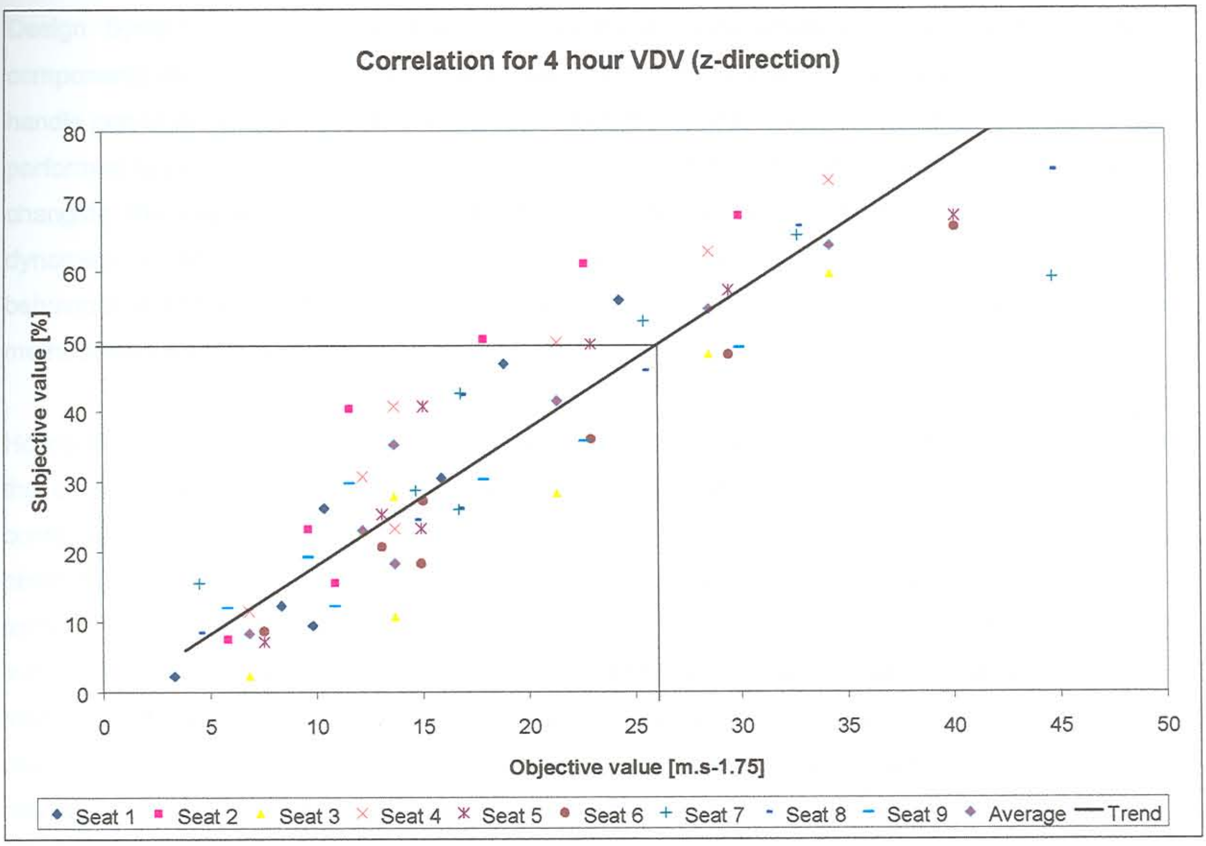


Figure 1.20: Correlation for 4 hour VDV values [14].

1.5 Suspension optimisation

Suspension optimisation is the process through which decisions are made on the specific choices of characteristics for the suspension components such that certain prescribed design objectives or evaluation criteria are fulfilled. These criteria may be the attainment of a certain level ride-comfort over specified terrain and vehicle speed, or the reduction of the maximum accelerations of the cargo to acceptable values, or any other specified criteria, subject to the vehicle complying to prescribed mobility criteria.

The first step in deciding on the suspension criteria to be applied is normally to use basic vehicle models, of the type described in 1.3, to obtain first order computed indications of suitable spring and damper constants for the specific vehicle. A good reference for such criteria can be found in the publication: Magic numbers in design of suspensions for passenger cars [15]. Although these "magic numbers" are specifically prescribed for passenger cars they can be applied to other type of vehicles as well.

The initial basic approximation models usually assume linear characteristics for suspension components such as the springs and dampers. The second step is to perform a more extensive and detailed analysis of the vehicle concerned. This can be done by performing vehicle simulations using vehicle dynamic simulation programs such as, for example, GENRIT [3], DADS (Dynamic Analyses

Design System) [4] or ADAMS/Car [16]. In these more advanced programs the suspension components are normally prescribed using two-dimensional tabular data and these programs can also handle non-linear suspension characteristics. Once the vehicle model is available, simulations can be performed to evaluate the vehicle ride dynamics over different specified route profiles and speeds. By changing the suspension characteristics the influence of these parameters on the vehicle ride dynamics can be assessed. The parameters may then be adjusted accordingly until acceptable ride behaviour is obtained. A brief overview now follows of some reported projects where this design methodology was applied.

Heyns et.al. [17] describe the design of the suspension system for a container carrier. The author of the current study was part of this project team. This carrier is used to carry different types of containers, such as a mobile operating theatre, x-ray equipment, etc. For this type of cargo it is obviously required that low acceleration levels must be maintained when travelling over any kind of terrain. A GENRIT simulation model of the vehicle was built. Using different lay-outs for the suspension system as well as different characteristics for the suspension components, the carrier motion was simulated over different terrain and at various speeds. The suspension lay-out and characteristics varied were those for the suspension at the fifth wheel of the truck-tractor, and for the rear suspension of the trailer. A combination with dual axles at the rear of the trailer was also included in the study. Subjective and objective evaluations of the acceleration experienced by the cargo were then used to determine the "optimum", or rather, the best configuration from those for which simulations were performed. Figure 1.21 depicts the side view of the carrier and figure 1.22 gives a schematic of the GENRIT simulation model used in this study.

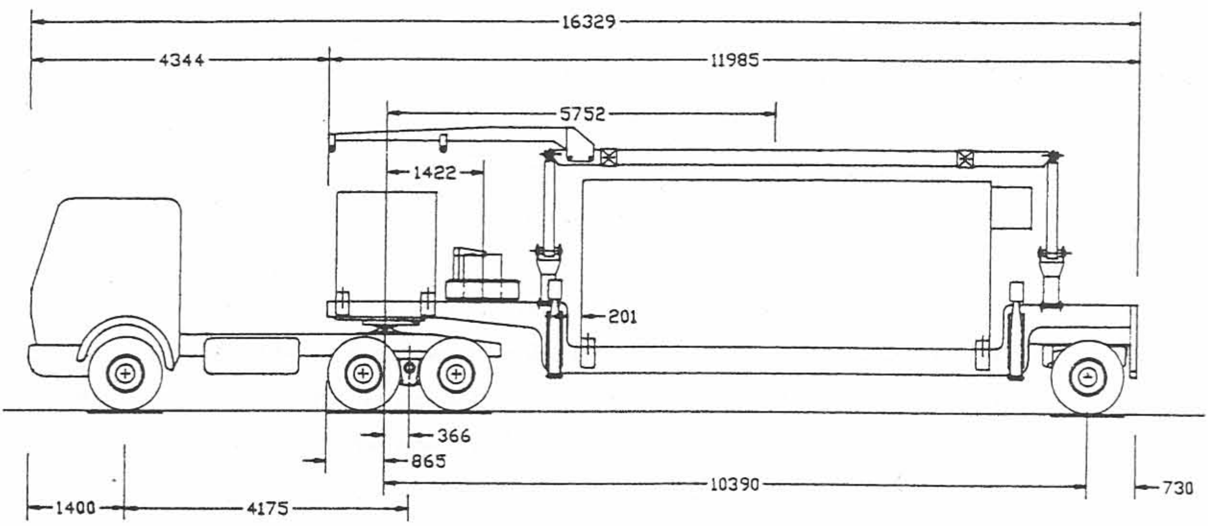


Figure 1.21: Side view of the container carrier for suspension optimisation [17].

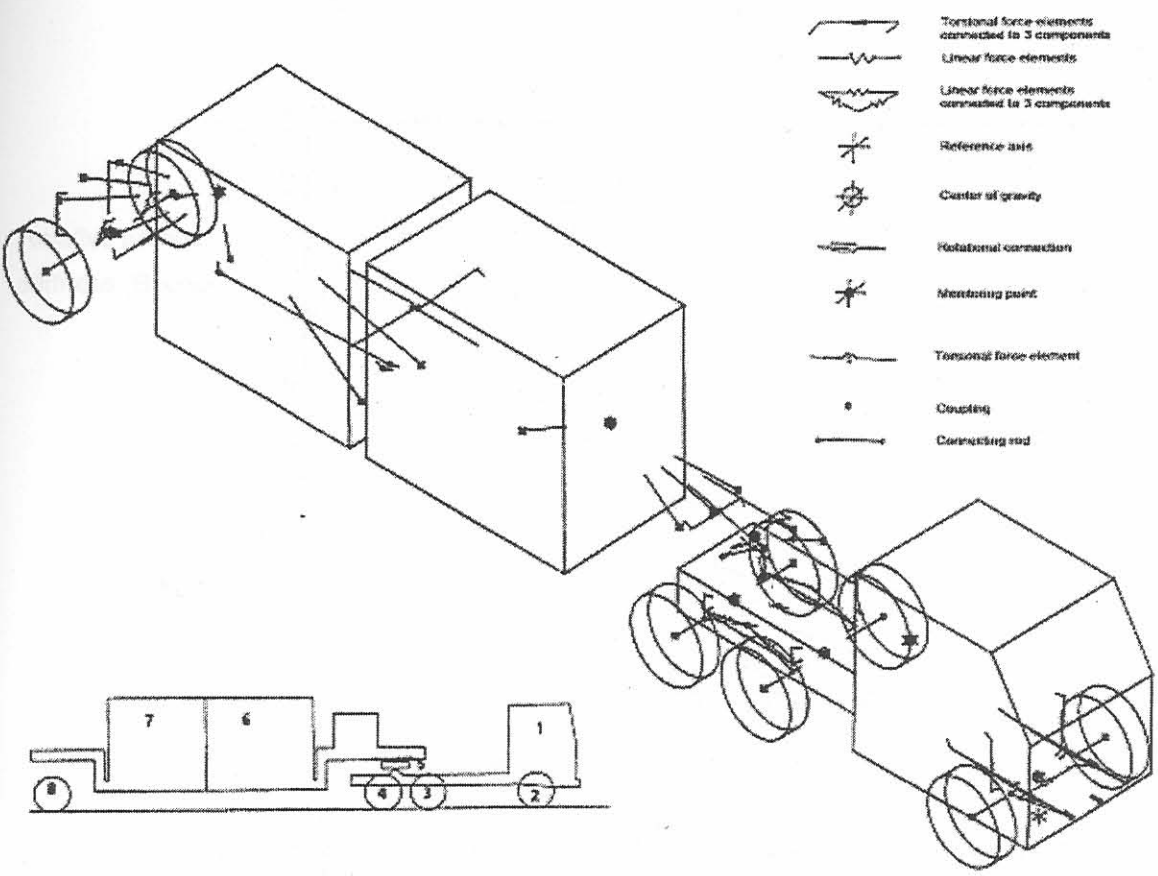


Figure 1.22: Schematic of the GENRIT simulation model used in the suspension optimisation study of a container carrier [17].

The author of the present study (Naudé) was also involved in the optimisation of the suspension characteristics of the Mingwe vehicle [18]. The Mingwe vehicle is a four-wheeled armoured personnel carrier (see figure 1.18). The simulation program GENRIT was used for simulating the dynamics of the Mingwe vehicle. In this study the suspension characteristics of the dampers in the bounce and rebound direction where changed by a certain percentage, the vehicle motion was then simulated over a certain route profile and the vibration dose value at a specific point of interest was calculated. The results were drawn on a two dimensional contour plot which indicates the percentage improvement in the vibration dose value for the different damper stiffness combinations. An example of such a plot, for a specific terrain and speed, can be seen in figure 1.23. A positive percentage indicates an improvement in the ride comfort for the specific terrain and speed. The point (1.0, 1.0) indicates the original design.

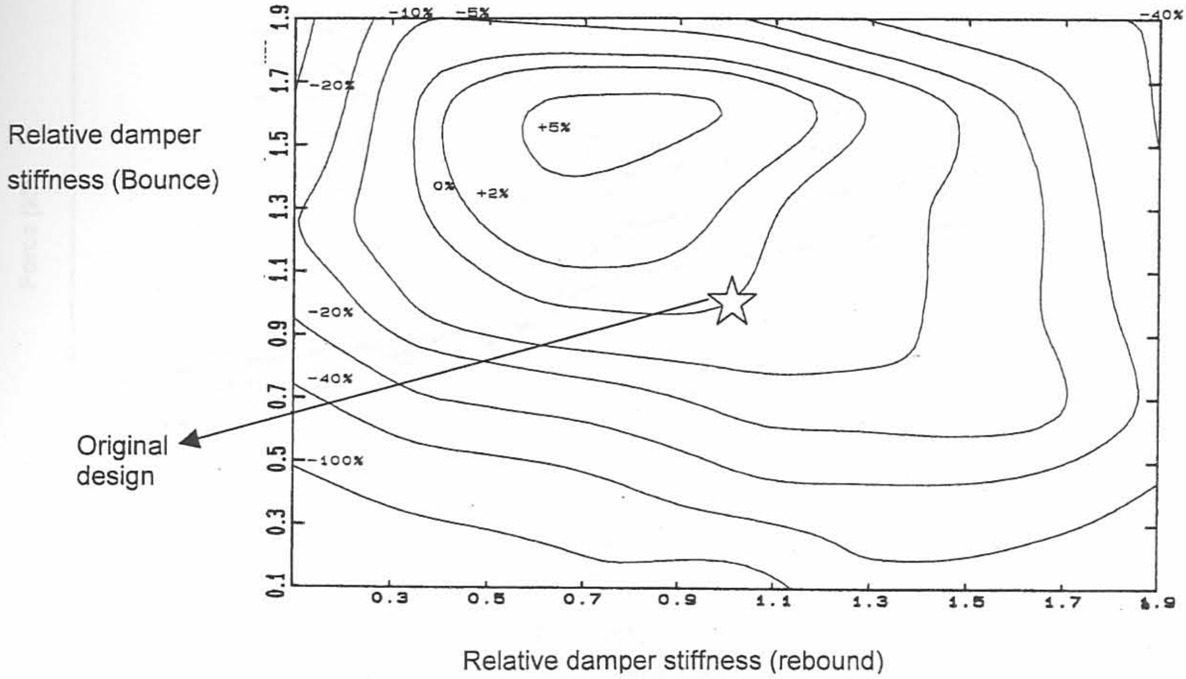


Figure 1.23: Improvement of ride comfort for the Mingwe vehicle as function of the damper stiffness in bounce and rebound [18].

In a study done by Heui-bon Lee et al. [19] a similar approach to that of Naudé [18] was used. Using DADS the ride quality for a medium truck was simulated for different spring stiffness of the front/rear suspension and damping characteristics of the shock absorbers. Examples of the damper characteristics used can be seen in figure 1.24. In figure 1.24 the thick lines indicate the damper characteristics used for the front axles and the thin lines that for the rear axle. As can be seen the configurations used have damper stiffness two and three times the stiffness of the baseline (the solid line). Using power spectrum density (PSD) plots of the truck body acceleration and pitch motion a decision was made regarding the configuration with the "best" characteristics. The configuration chosen was the one giving the lowest PSD plot.

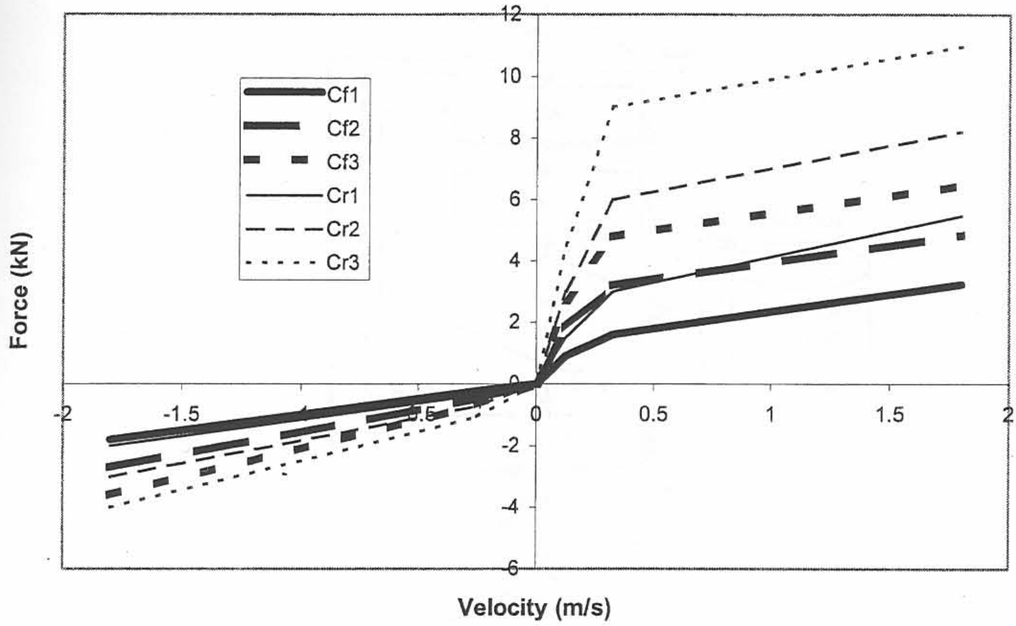


Figure 1.24: Example of discrete changes in damper characteristics used during suspension optimisation [19].

In more recent studies the concept of Multidisciplinary Design Optimisation (MDO) has been proposed in which a more formal mathematical optimisation methodology is adopted in performing the optimisation. Some examples of this approach are presented below.

Motoyama et al. [20] performed an optimal automobile suspension design using a formal optimisation technique. In their study a kinetic analysis of double wishbone independent vehicle suspension system was used in conjunction with a Genetic Algorithm (GA) to determine the optimum values of 18 design variables, specifying the location of the suspension attachment points. The objective function minimised was the change in the toe angle over a stroke of 100 mm of the wheel. ADAMS/Car was used as the simulation program by means of which the objective function was evaluated. Figure 1.25 gives a schematic representation of the "optimisation system" used in the study. Using this optimisation methodology and, for specified sets of starting values for the geometry of the suspension, an optimum configuration was obtained with an average change in the toe angle of 0.06 ° over the 100mm stroke. The original configuration gave a change in the toe angle over the 100mm stroke of –2 ° to +5.6 °. The optimum was obtained after five thousand simulations! To improve the computational economy a Response Surface Methodology (RSM) was implemented to construct approximations to the behaviour of the system. With these approximations the influence of the design variables on the change in the objective function could more economically be evaluated. This method enabled the design engineer to change the design variables, and economically determine the tendency of the toe curve without excessive calling of the simulation program.

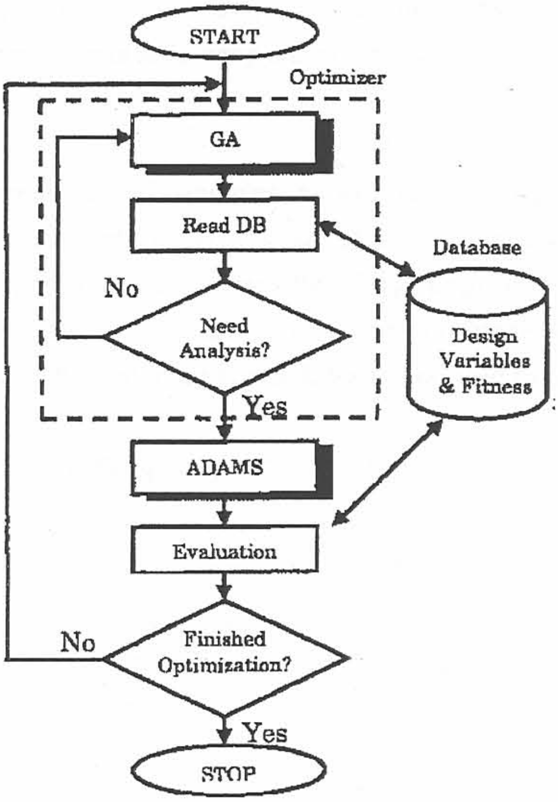


Figure 1.25: The "optimisation system" used by Motoyama et al. [20]

Lee et al. [21] performed a fuzzy multi-objective optimisation of a train suspension also using a response surface model. The dynamic analysis program VAMPIRE was utilised to optimise 58 responses of the suspension with respect to 26 design variables using optimisation applied to a response surface model. A fuzzy decision-making algorithm was used to investigate the engineer's confidence level in the optimisation process. Every first order response surface model of 58 responses was constructed from Taguchi's L_{32} 2 orthogonal array, and these were generated and simulated 32 times by using VAMPIRE. With this system Lee and his co-workers claim that they obtained an optimum design for the suspension of a train.

Although not directly applicable to vehicle suspension systems, Botkin [22] performed a structural optimisation of automotive body components based upon parametric solid modelling. A Convex linearisation technique was used in conjunction with Nastran Solution 200 to perform eight case studies for optimisation of an automotive front structure component. Mass reduction of up to 64% was obtained using composite constructions.

Etman [5] coupled optimisation software to commercial multi-body analysis packages such as MECANO, MADYMO and DADS using approximation concepts. With specific application to vehicle suspensions a stroke dependant damper for the front axle suspension of a truck was optimised. Various optimisation runs were performed for a two degree of freedom model, as well as for a full-scale truck model, using single-point linear approximations. Etman states that: "*This application* *clearly showed that the final optimum design heavily relies on the choices made by the designer."* He also concludes that: *"The ultimate goal is to integrate multibody analysis and optimisation into one general purpose design tool."*

As can be seen from the abovementioned examples, as well as other investigations reported in the literature [23, 24, 25, 39-50], optimisation and analysis of a vehicle suspension can be complex, expensive and time-consuming. These factors force the design engineer to take a more pragmatic approach summarised by the statement of Esat [26]: ***"The important goal of optimisation is improvement and attainment of the optimum is much less important for complex systems".***

1.6 Active / semi-active suspensions

From the reported suspension optimisation studies it is evident that the so-called "optimum" vehicle suspension characteristics are dependant on the particular evaluation criteria set, i.e., on the objective function used, and on the specific route and speed over which the vehicle is to be driven. Thus as the criteria and conditions change, so the "optima" will vary. In general, however, the following tendencies are evident:

i. ride-quality can be improved over moderate, i.e. low amplitude input terrain by lowering both the spring stiffness and the damping,
ii. low spring stiffness and low damping require more suspension deflection, which are limited for practical reasons, and
iii. higher spring stiffness and higher damping are required for improved road holding.

One way of accommodating these conflicting suspension requirements is to use an active suspension. A fully active suspension system incorporates actuators, normally hydraulic actuators, to generate the desired forces in a suspension system to satisfactorily deal with changes in road conditions. External power is required to operate the system. Semi-active suspensions contain spring and damping components, the properties of which can be changed by external control. A signal or external power is supplied to these systems for the purpose of changing the properties. In passive suspension systems conventional springs and dampers are used. These component properties are time-invariant.

Due to the fact that the suspension characteristics in semi-active or active suspensions can be changed according to the terrain requirements, these types of suspension systems can improve ride quality over a wide spectrum of terrain. These improvements come at the penalty of increased complexity and higher cost. A large amount of research has been done on control algorithms for such suspension systems [27-38].

Due to the associated higher cost and increased complexity, semi-active suspension systems and active suspension system are still relatively rare and confined to the more expensive car segments. Some of the newest military vehicles also make use of semi-active suspensions. **This study will, however, be limited to the optimisation of passive suspension systems.**

1.7 Multidisciplinary design optimisation

In search of better designs parametric studies have in the past been performed as part of a general design process. However, a more effective and efficient design method is demanded in order to reduce the time to design and develop products. Design optimisation techniques in which a systematic mathematical approach is adopted is considered to be a better way to search for good designs in comparison to performing parametric studies. Due to the high performance demands and multiple requirements of products, design teams need mathematical optimisation techniques that can reliably optimise complex systems. This need has resulted in the concept of multidisciplinary design optimisation (MDO) [51].

Multidisciplinary design optimisation can be described as the development of <u>general</u> mathematical optimisation algorithms that may <u>universally be applied</u> to design problems arising in various <u>different disciplines</u>. Basic to this approach is the systematic adjustment of the values of the design variables so that they will minimise an objective function subject to constraints. Typical examples of design problems solved in the way include [51]:

i. the minimisation of traffic noise over an irregular wall,
ii. optimal tundish design using CFD with inclusion modelling,
iii. minimum cost design of welded structures,
iv. optimisation of engine mountings,
v. sound and vibration optimisation of carillon bells and MRI scanners,
vi. shape optimisation for crashworthiness, and
vii. optimisation of heat sinks.

Often the problem requires the use of a computer simulation program to enable the determination of the influence of the design variables on the objective function. As stated by Papalambros [52]: *"Design optimisation is now a discipline in high technology product development and a natural extension of the ever-increasing analytical capabilities of computer-aided engineering."*

For many engineering problems, multi-body analysis routines are used to calculate the kinematic and dynamic behaviour of the mechanical design. In these cases the values of the objective function and constraint functions follow from the numerical simulation. Therefor, to solve the optimisation problem, the multi-body code has to be coupled to a mathematical programming algorithm. Such a coupling may be difficult to implement and can lead to high computational cost [23].

Several classes of optimisation algorithms have been developed. The most important classes are [51]:

i. Mathematical programming methods (including gradient based methods such as, for example, Sequential Quadratic Programming [SQP],
ii. Lipschitzian and deterministic optimisation and
iii. Genetic algorithms (GA's).

Engineering optimisation problems where simulation programs are used in computing the objective functions present unique challenges because of

i. *the presence of noise* in the objective and/or constraint functions due to numerical inaccuracies in the simulations that result from discretisation and round-off errors, and the use of not fully converged solutions, and
ii. *the presence of discontinuities* in the objective/constraint functions arising from formulations of the optimisation problem in forms convenient for engineers, e.g., by the use of absolute value objective functions and by using penalty function formulations for constrained problems.

These aspects, presently of great world-wide interest to design engineers, have been addressed by Snyman [53, 54]. Central and essential to his tackling of the above difficulties has been the development of novel optimisation algorithms suitable for engineering problems. This required both the construction of new algorithms, and the testing of these methods on appropriate standard, and new, engineering design problems. A particular successful development has been the "leap-frog" trajectory methods [53,54]. These methods fall within the class of gradient based mathematical programming methods.

The leap-frog unconstrained optimisation algorithms were originally proposed in the early eighties [53, 54]. These algorithms have the unique characteristics, for gradient based methods, that only the gradient of the objective function is used and that no explicit line searches are performed. These algorithms were later refined and extended to constrained problems [55]. The methods were found to be extremely reliable and robust. In particular, the methods are relatively insensitive to problems where discontinuities and noise are present. Since it is expected that the latter problems occur in the numerical simulation of suspension systems, the leap-frog method is the method of choice for this study where reliability and robustness of the proposed Suspension Optimisation System is of prime importance. The most current version of the leap-frog code for constrained optimisation, is called LFOPC [55].

1.8 Summary: purpose of study

The need for a Suspension Optimisation System evolved from difficulties experienced during the design of a 6x6 armoured personnel carrier. Although parametric studies, using complete three dimensional simulation programs, were performed in the design of the vehicle, problems and failures continued to occur on the suspension system. While the need for suspension optimisation was demonstrated for the particular vehicle investigated, the same requirement arises during the design of any other vehicle, especially for vehicles for specialised applications such as armoured personnel carriers. The optimisation methodology needs to be applied during the vehicle concept design stage, at which time little geometrical information regarding the vehicle and its suspension is available. It is proposed that the initial optimal design may be done by coupling a sufficiently representative two-dimensional vehicle dynamic simulation program to a suitable optimisation algorithm.

An outline of a linear quarter car model and of the associated pitch and bounce analysis for vehicle suspensions are given. Although this model can be used in obtaining a first order estimation of the suspension behaviour, it does not, of course, provide a complete description of the non-linear suspension characteristics. Due to the fact that almost all suspension component characteristics, especially that of dampers, are non-linear the model to be used in the optimisation must contain non-linear characteristics.

In an overview of work done in the field of suspension optimisation it is shown that first order optimisation has been done through parametric studies. The latest developments are, however, the application of mathematical optimisation algorithms, in conjunction with computer aided simulation of the vehicle system, to determine values for the design variables that a minimises an objective function related to certain desirable design criteria. The development and application of multidisciplinary design optimisation techniques to almost every engineering field, spurred the application of these techniques in the field of vehicle design. The reported work done in this field all achieve specific design optimisation goals but are not general enough to be directly applicable to the problem at hand. Furthermore the reported methods are numerically extensive and either require a specific code for the particular problem or specific hardware.

By using an active or semi-active suspension it is in principle be possible to incorporate "suspension optimisation" into the suspension system of the vehicle. In this instance the suspension characteristics can be "programmed" to change as required in real-time. The changing of the suspension characteristics is controlled by a control strategy. For the active or semi-active suspension system this control strategy also needs to be optimised. Due to the higher cost and complexity of semi-active suspension systems, **the current study is limited to the optimisation of passive suspension systems**.

The literature reveals several classes of optimisation algorithms. All of these classes have successfully been applied to engineering optimisation problems. One of the mathematical programming algorithms that is singled out here is the LFOPC algorithm [55]. This algorithm is very robust and particularly suited for use in problems where numerical noise and discontinuities may occur, as is typically experienced in the computer aided analysis of mechanical systems. This algorithm is to be used in this study.

The specific overall objective of this study is summarised as **the application of a formal mathematical approach to the optimisation of vehicle suspension characteristics.** The methodology is to be embodied in a Computerised Suspension Optimisation System for which the following requirements are set:

i. The system must be general enough to be applicable to different single body vehicles with up to four axles.
ii. The optimisation must be able to optimise non-linear suspension characteristics for the different suspension components: springs, dampers, bump-stops and tyres. The specific design variables used must be user configurable.
iii. The specific objective function used must be user configurable.
iv. The optimisation system is to be used during the concept design stage of the vehicle. During this stage only limited geometrical information for the specific vehicle and suspension is available. This obviates the need for a full three-dimensional multi-body simulation. A representative two-dimensional simulation is required and will suffice for concept stage evaluation of the objective function.
v. The system must be user friendly and usable by the project manager, who may have limited vehicle dynamic analysis background and experience.

Finally, although the need for this Computerised Suspension Optimisation System arose from problems experienced locally, the same need exists internationally. Etman [5] suggests that in performing vehicle suspension optimisation greater benefit may be obtained by using a simpler vehicle model than a full-scale 3-D model. He also mentions that: *"A serious difficulty is that the designer often does not precisely know how to mathematically formulate the multibody design problem beforehand. It is very likely that during optimization he wants to remove or change* [the] *objective function, constraints and design variables. Therefore, graphical means have to be available, not only for modelling, but for optimization purposes as well."* He thus stresses the importance of **an interactive computer design tool for successful design optimisation**. The development of the latter versatile and flexible tool, specifically for a vehicle suspension system, is therefore indeed the main objective of this study.