Vehicle dynamic analysis of wheel loaders with suspended axles

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Abstract

The wheel loader is a type of engineering vehicle used primarily to move crude material over shorter distances. As the vehicle is designed without wheel suspension, wheel loader drivers are exposed to high levels of whole body vibration which influences ride comfort negatively. The work presented in this thesis has the aim to investigate the potential in adding an axle suspension to a wheel loader in order to reduce vibrations and increase handling quality. While suspended axles have great potential for improving ride comfort and performance, they will also necessarily affect the vehicle dynamic behaviour which is different in many aspects from that of passenger cars or other road vehicles: the wheel loader has a large pitch inertia compared to its mass, the axle loads vary considerably with loading condition, and the vehicle uses an articulated frame steering system rather than wheel steering. These issues must all be considered in the design process for a wheel loader suspension.

The effects of suspended axles on ride vibrations are analysed by simulating a multibody wheel loader model with and without axle suspension. Results from the simulations show that longitudinal and vertical acceleration levels are greatly reduced with axle suspension, but that the decrease in lateral acceleration is smaller. By reducing the roll stiffness lateral accelerations can be further reduced, although this may not be feasible because of requirements on handling stability. The pitching oscillation of the vehicle has also been studied as this is known to have a large influence on ride comfort. An analytical model is used to study the effect of front and rear suspension characteristics on the pitching response of the wheel loader, showing that a stiffer rear suspension is favourable for reducing pitching but also that a similar effect is attainable with a stiffer front suspension. Results are compared to multibody simulations which show the same trend as analytical predictions. By including a linearised representation of a hydropneumatic suspension in the models, it is also shown that favourable dynamic behaviour can be maintained when the vehicle is loaded by utilising the fact that suspension stiffness is increasing with axle load.

Articulated vehicles may exhibit lateral oscillations known as "snaking" when driven at high speed. The effect of suspended axles on these oscillations are analysed using a multibody simulation model of a wheel loader with an equivalent roll stiffness suspension model. It is found that the roll motion of the sprung mass has a slightly destabilising effect on the snaking oscillations. This effect is more pronounced if the body roll frequency is close to the frequency of the snaking motion, although this loss in stability can be compensated for by increasing the equivalent stiffness or damping of the steering system.

Together with existing vehicle dynamic theory and design rules, the studies reported in this work provide an insight into the specific issues related to suspension design for wheel loaders.

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Stockholm, 2008

Adam Rehnberg

Appended Papers

This thesis consists of an introductory overview and three appended papers:

Paper A

Rehnberg, A., Drugge, L., *Ride comfort simulation of a wheel loader with suspended axles*, submitted for publication, 2008.

Rehnberg developed the models and performed the simulations. Drugge assisted with the simulation parameters, test scenarios and conclusions.

A part of the results from this study have been presented as a poster presentation at the 20th symposium of the International Association for Vehicle System Dynamics IAVSD, Berkeley, USA, 2007.

Paper B

Rehnberg, A., Drugge, L., *Pitch comfort optimisation of a front end loader using a hydropneumatic suspension*, SAE Technical Paper 2007-01-4269, 2007.

Rehnberg developed the models and performed the calculations and simulations. Drugge provided input on the theory, test scenarios and models.

The paper was presented by Rehnberg at the SAE 2007 Commercial Vehicle Engineering Congress and Exhibition, Rosemont, USA, October 31 – November 2, 2007.

Paper C

Rehnberg, A., Drugge, L., Stensson Trigell, A., *Snaking stability of articulated frame steer vehicles with axle suspension*, submitted for publication, 2008.

Rehnberg developed the models and test methodology. Drugge assisted with the test scenarios and evaluation criteria. Stensson Trigell assisted with conclusions and writing the paper.

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1 Introduction

1.1 Background

Construction machines, sometimes referred to as engineering vehicles or earth movers, are basically a class of self-propelled machines designed for use in civil engineering. The work in this thesis concerns a special class of construction machines, the wheeled loader. This vehicle is essentially a type of tractor equipped with permanently attached, front mounted lifting arms operated by hydraulic cylinders. Unlike vehicles designed primarily for transport, the wheel loader is built without any axle suspension. The front wheels are attached directly to the vehicle body, and the rear axle is allowed to oscillate around the longitudinal axis, thus allowing all wheels to maintain contact with the ground. An example of a medium sized wheel loader is shown in figure 1.

Figure 1: A wheel loader

Wheel loaders are mainly used in construction or surface mining for moving crude material over shorter distances, but because of the versatility of the machines they are also commonly used for pallet handling, timber loading and similar tasks. Since the advent of the first purpose-built wheel loader about 50 years ago, considerable effort has been made to improve the efficiency, safety and operability of the vehicle. However, less attention has been given to issues like ride vibrations or handling stability. Traditionally, wheel loaders have been seen as machines more than vehicles and so the driving dynamics have been considered less important. However, since the demands for increased task performance and operator comfort are continuously growing, future requirements for vibration isolation and high speed transport capacity will call for suspension systems that allow safe and comfortable travel under a range of operating conditions. This means that vehicle dynamic considerations are becoming increasingly important in the design of wheel loaders.

As a consequence of the primitive vehicle design and the rough surfaces where wheel loaders are usually operated, wheel loader drivers are often exposed to high levels of whole body vibration. In short-term perspective these vibrations will affect operator comfort negatively and cause driver fatigue. Long term exposure to low frequency vibrations has been shown to increase the risk for lower back pain [1], and is also believed to be the cause of various internal organ disorders. Thus, whole-body vibration can be considered a major occupational hazard for earth mover operators and have been the focus of various legislations such as the European Vibration Directive [2].

As the wheel loader does not have any axle suspension, primary isolation from ground induced vibrations is provided by the tyres. Earth mover tyres are typically large and soft, thus filtering out and absorbing some of the vibrations and shocks caused by ground roughness. Theoretically, tyres could be engineered to dampen higher amounts of vibration although this would in reality be highly impractical. According to Lines et al [3], tyres would have to absorb about five to ten times more vibration energy in order to fully protect the operator from ground induced vibrations. This would greatly affect the rolling resistance of the tyre and hence the energy consumption of the vehicle would increase as well. Furthermore, the larger energy absorption would lead to high thermal and mechanical stress in the tyre material, which would accelerate wear. It is obvious that tyre design for increased vibration isolation would soon conflict with economic and environmental considerations.

Cab and seat suspensions also function to reduce operator vibration levels. The cab is usually mounted on rubber elements, sometimes combined with an integrated oil volume, thus providing vibration isolation through flexibility and viscous damping. More refined cab suspensions also exist where the cab is mounted on a suspended subframe [4]. High-end driver seats are commonly designed with hydraulic or pneumatic dampers to cushion the driver from vibrations and shocks. The main limitation of seat and cab suspensions is the small amount of suspension stroke available. Because of these limitations, the stiffness coefficient of the flexible elements needs to be high enough to avoid impacts with suspension ends at large transient loads, and therefore the eigenfrequency of the cab or seat suspension is generally too high to provide vibration isolation in the lower frequency range. Furthermore, cab and seat suspensions mainly reduce vibrations in the vertical direction. This could possibly be alleviated by active systems [5], although the potential of advanced seat technology is

still unclear. It should also be noted that the vibration isolation of the seat is often reduced by inadequate operator posture [4]. In addition to cab and seat suspensions, larger wheel loaders are commonly fitted with a lifting arm suspension system to decouple the mass and inertia of the lifting arms from the vehicle body. The bucket and lifting arms are isolated from the vehicle by means of gas springs incorporated in the lifting hydraulics and thereby pitching and vertical oscillations to some extent, mainly for a loaded vehicle.

Considerably greater vibration isolation can be provided by designing the wheel loader with full wheel suspension. This would mean flexibly attaching the wheels to the vehicle body by means of springs and dampers, as on virtually all road vehicles and on most offroad vehicles. Axle suspensions have greater potential for improved ride comfort as the available suspension travel is larger, and also have the potential to alleviate vibrations in the lateral and longitudinal directions in ways not possible by cab and seat suspensions. Apart from reducing driver exposure to vibrations, axle suspensions could also improve the task performance of the loader. As the efficiency of an earth mover is usually defined by the volume of material that can be transported in a specified time, the working performance is directly related to the maximum practical velocity of the vehicle. This velocity is currently limited to a large extent by the operator's ability to withstand vibrations and therefore shielding the operator from vibrations will in practice increase the attainable vehicle speed. An axle suspension also functions to reduce wheel load fluctuations, thus improving handling quality and manoeuvring stability at high speeds. Therefore suspended axles are not only beneficial for driver comfort but also for the performance of the vehicle.

While axle suspensions are the most potential solution for vibration reduction and improved performance, it is also more complicated than existing systems. Cab, seat and boom suspensions have the benefit of being largely independent of the vehicle body, and can therefore be added or modified on an existing vehicle without altering greatly the vehicle dynamic behaviour since the mass of the cab and operator is relatively small compared to the vehicle body. With the inclusion of suspended wheel axles, a large part of the vehicle mass will be oscillating on the axle suspension rather than simply the tyre flexibility. This will clearly have an impact on the driving dynamics of the vehicle. Naturally, the increased complexity of the vehicle will also add to overall cost and maintenance needs, which is not an unimportant consideration. Suspended axles have been used on certain engineering vehicles for some time, mainly off-highway dump trucks which utilise rubber springs or hydropneumatic struts as suspension elements. On agricultural tractors, suspended front axles are becoming increasingly common and some models feature rear axle suspension as well [6]. Applications to loader vehicles are less common and have mainly been limited to military models that are equipped with suspended axles to enable high speed road travel, thereby allowing rapid deployment and increased combat survivability [7, 8]. Since previous information regarding wheel loader suspension design is limited, a review of the applicability of current design methods, as well as the development of new methodology, is required to provide the necessary knowledge for wheel loader suspension design.

1.2 Objective and research question

The objective of this thesis is to provide a methodology for the development of wheel loaders with suspended axles. This includes analysing the vehicle dynamic behaviour of the vehicle, and how this behaviour will be affected by the addition of a suspension. In summary, the research question can be formulated as:

- What are the potential benefits and problems of adding a suspension to a wheel loader, given the specific vehicle dynamic properties of the vehicle?

The selected research approach is to use simulation models of varying complexity to systematically investigate the dynamic behaviour of the vehicle. Analytical methods are used to predict basic characteristics and to provide input for more refined simulations. The approach requires adequate vehicle dynamic models and also objective methods for evaluation of ride comfort and handling. Thus, an important part of the work is also to investigate what modelling tools are needed for simulation studies of wheel loaders.

1.3 Outline of thesis

The introductory chapter of this thesis establishes the problem background, research goals and selected methodology. Chapter 2 details the specific properties of the wheel loader and provides a brief overview of the historical development of the vehicle. Chapter 3 explains how vehicle dynamic behaviour can be evaluated by objective methods, focusing mainly on ride comfort as this is the primary motivator for introducing axle suspensions. The emphasis is on the standard ISO 2631 which defines much of the demands for ride comfort evaluation and certification. The problems and shortcomings of existing comfort evaluation standards are highlighted and some novel methods are presented that may complement existing standards.

A review of fundamentals of vehicle and surface modelling is presented in chapter 4, focusing on methods and practices that are applicable to the study of wheel loader dynamics. The purpose of this chapter is to give a starting point for the suspension analysis and design and also to provide a background for the studies included in this thesis. In practice, the design and realisation of a vehicle suspension is based on vehicle dynamic theory to some extent while at the same time relying on practical experience. According to Blundell and Harty [9], theoretical methods are usually more important in the beginning of the vehicle design process while the final development and refinement stages tend to rely more on practical tests, expert opinions, and ultimately the concept of common practice or “black magic”. For the wheel loader, practical experience of suspension design is limited since suspended axles have previously not been used on loader vehicles. Therefore it is important that initial design specifications for wheel loader suspensions are built on a solid base of vehicle dynamic theory until a “black magic” applicable to wheel loader suspension design has been established from new experiences or through formalisation of existing knowledge. As the vehicle design process matures and moves forward, the base of experience will grow and practical experience will complement or replace initial theoretical calculations. Until then, vehicle dynamic theory plays an important role in suspension design as the only alternatives are either guesswork or trial and error.

Chapter 5 describes multibody simulations, a method that has been used extensively in the simulations performed in the thesis. The practical use of the method is explained together with an example model, and some of the hazards with multibody simulations are discussed. The performed research is presented in the three appended papers which are reviewed and commented on in chapter 6. Each paper represents a special topic in the study of wheel loader vehicle dynamics and their interconnection and relevance is discussed with respect to the overall outcome of this thesis work. Conclusions are summarised in chapter 7 and some areas deemed relevant for future studies are discussed in chapter 8.

2 The wheel loader

The origin of the wheel loader is the agricultural tractor. Gasoline-powered tractors first came into use at the beginning of the 20th century and eventually found use as industry vehicles as well, equipped with front end lifting attachments to function as loaders. Based on these tractor loaders the rear end loader (figure 2) was developed, basically by turning the tractor backwards and mounting the lifting arms at the rear end. This allowed higher load capacity than the front end loader, but safety problems together with the inherent instability of rear wheel steering led manufacturers to abandon this configuration in favour of articulated frame steering and front mounted lifting arms. The roots of the vehicle, tracing back to the agricultural tractor, can still be seen in the rear mounted engine and rear-facing grill, as well as the rear pivoting axle which was originally the front axle of the tractor. Early articulated loaders also maintained slightly larger front wheels, but for practical reasons these designs have now been superseded by models with equally sized wheels. An example of a contemporary design is seen in figure 3.

Figure 2: Early wheel loader model

Figure 3: Cut-away drawing of a modern wheel loader

The modern wheel loader is basically a rear-engine articulated tractor with a front-mounted lifting implement. As the wheel axles are unsuspended, the rear axle is usually mounted on a pivoting joint to allow wheel contact on uneven ground, although some models accomplish this by a combined centre joint that allows both articulation and roll displacement. When studying the vehicle dynamic behaviour of the wheel loader, some important characteristics can be identified:

Vehicle mass: Modern wheel loaders range in mass, from 2 000 – 3 000 kg compact machines to extremely large mining loaders of more than 200 000 kg total mass. The majority of the vehicle population is found in the range of 10 000 – 50 000 kg. Thus, an average sized wheel loader is about equivalent to a heavy truck by means of gross mass. However, the full weight of the vehicle is carried only by two axles. This means that suspension components must be robust enough to sustain the vehicle axle loads, which are higher than those of a road vehicle of comparable weight. The vehicle wheelbase is relatively short, which especially affects the pitching dynamics. Also, the large vehicle mass and offroad usage means that specialised tyres are used, which may have different characteristics than passenger car tyres.

Axle load variation: The maximum load of a wheel loader is usually about 60% of the unloaded vehicle weight. Unlike a truck or other cargo vehicle, this load is carried ahead of the front axle, meaning that not only the mass is increased but also that the centre of gravity shifts forward. Typically, the load on the front axle will increase about three to four times when the vehicle is loaded, while the rear axle load decreases to about two thirds of the static axle load of the unloaded vehicle. This also affects the pitching inertia of the vehicle. Hence, a suspension for a wheel loader must be able to handle this large range of axle loads. Also, the centre of gravity will shift upwards when the lifting implement is raised. Although this is less relevant for ride and handling as the lifting arms are generally kept low when driving at higher speed, it is still important for static stability considerations and short cycle loading.

Steering: Wheel loaders are generally equipped with articulated frame steering. This means that the vehicle is separated in a front and rear frame which are connected by a vertical revolute joint. The steering angle is attained by changing the relative angle

between the front and rear part in the yaw plane, rather than to steer the wheels as on an Ackermann steered vehicle. The advantage of this steering arrangement is a smaller turning radius and a more robust design, but it also has the drawback of decreased static stability when turning at standstill as the centre of gravity is displaced laterally. Compared to Ackermann steering, articulated frame steering is different in several ways. The mathematical formulation of the in-plane vehicle dynamics becomes considerably more complex and the flexibility of the steering system may lead to instabilities when driving straight at high speed.

Complex dynamics: While a passenger vehicle can be described fairly well by a single rigid body model, the dynamics of a heavy vehicle is more complex. This is mainly because the vehicle consists of several heavy parts and flexible elements, whose respective natural frequencies are all in the same order of magnitude as the rigid body frequencies of the whole vehicle. A similar condition exists for heavy trucks, where the formulation of simple and accurate models is made more difficult as several vehicle parts contribute to the low frequency vehicle vibrations [10]. Although simplified models can be used for initial design considerations, a more refined vibration analysis requires the inclusion of more vehicle parts than simply the vehicle body.

From the listing above, it can be seen that the wheel loader possesses a number of unique properties that sets it apart from the vehicles that have been the subject of previous vehicle dynamic analyses. These properties must all be addressed in order to provide useful input for the design of a wheel loader axle suspension. The characteristics listed above also help to define which methods are suitable for wheel loader modelling and analysis.

3 Evaluation of vehicle dynamic behaviour

3.1 Ride comfort evaluation

Usually, the term “ride quality” is used to describe vehicle vibrations in the frequency range of about 0 – 25 Hz. Higher frequency disturbances are referred to as “noise”. Another distinction is to associate ride quality with tactile and visual vibrations, while noise is representing aural vibrations. Low frequency vibrations in vehicles are usually caused by surface undulations. Vibrations caused by the vehicle itself, such as driveline or engine vibrations, are generally of higher frequency and are hence associated with noise rather than ride comfort. Noise considerations are typically not affected by the axle suspension design to any greater extent and therefore the focus in this thesis is mainly on ride quality.

Besides influencing operator comfort, undesired ride vibrations may also have negative health effects. As health effects are often chronic and related to long exposure times, the quantification of such effects is more difficult than pure operator comfort. Conflicting cases may also exist where improvements in comfort do not necessarily imply improvements in health effects, and vice versa. This further complicates the evaluation of vehicle ride vibrations.

Methods to evaluate ride comfort generally fall into two categories, defined as subjective or objective. Subjective methods include questionnaires, interviews or other tests designed to evaluate driver impressions in a systematic matter. This typically requires trained test drivers, who are able to produce reproducible results and discern between small differences in ride quality caused by minor changes to the vehicle. Objective methods use measurable parameters, most commonly accelerations, to estimate the comfort level. This provides a straightforward way to predict comfort, but it also requires that objective measures are representative of subjective impressions for the estimates to be meaningful. This is especially true for offroad vehicles, where the vibration spectrum may differ considerably from road vehicles. Therefore, some care is needed when interpreting objective comfort measurements. The establishing of robust

and representative objective comfort measures is still a subject of research and no universal correlation exists between objective measures and perceived comfort, although some agreement has been shown in certain test cases. Els [11] showed that objective comfort measures correlate fairly with subjective impressions in an offroad driving scenario. A similar study was performed by Mansfield and Whiting-Lewis [12], using vertical accelerations to estimate ride comfort in a road vehicle on a rough road.

3.1.1 Standards

A number of standards have been proposed for objective evaluation of ride vibrations. While still very much a matter of discussion, these standards have found widespread use and provide a good starting point for ride evaluation.

ISO 2631

The ISO standard 2631 presents a framework for evaluation human response to whole-body vibration in different scenarios. General guidelines for vibration evaluation are found in the first part of the standard, ISO 2631-1 [13], which is currently the most widely used method for vehicle vibration measurements. The standard is said to cover periodical, random and transient vibrations in the range of $0.5-80$ Hz with an extension to 0.1 Hz for motion sickness evaluation. The basic measuring unit used is accelerations, which are to be measured in the interface between the operator and the vehicle, i.e. at the seat surface, floor or seat back. A frequency weighting is applied to the measured accelerations, as vibrations are considered more critical at certain frequencies. The weighting curves for seat vibrations are shown in figure 4. Vertical accelerations are weighted using the W_k curve, while W_d is used for the lateral and longitudinal axes.

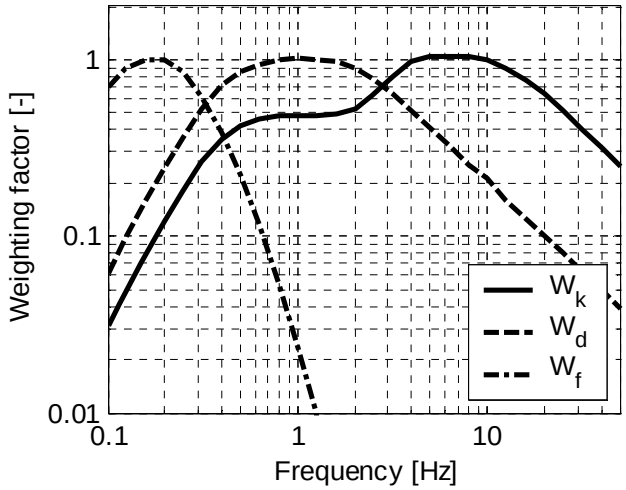

Figure 4: ISO 2631-1 weighting curves

It can be seen that vertical vibrations at about 1 Hz are considered more acceptable than higher frequencies, whereas lateral and longitudinal vibrations are most critical at 1 Hz. The weighting factor W_f is used for motion sickness evaluation, in the vertical direction only. This is less important for vehicle dynamic studies as the ride frequencies of a vehicle are normally higher than the frequencies considered for motion sickness effects. However, it can be noted that motion sickness effects are becoming significant at about 0.3 Hz, meaning that any vehicle suspension should be designed so that ride frequencies are at least higher than that.

The measure of vibration severity is obtained by root mean square (RMS) summation of the weighted acceleration time history $a_w(t)$ to return the single value a_w, defined as

$$a_w = \left(\frac{1}{T} \int_0^T a_w^2(t) dt \right)^{1/2} . \quad (1)$$

Here, T is the duration of the measurement period. Note that ISO 2631-1 uses a_w both to denote the acceleration time history and the RMS value. The unit for a_w is m/s^2.

The guidelines provided by ISO 2631-1 differ in some aspects depending on the purpose of the measurement. For comfort evaluation, accelerations are to be measured in all directions and the vector sum of the weighted accelerations is taken as the final comfort measure. Angular accelerations can also be included and are treated the same way, although this is only valid for seated persons. Translational accelerations may also be applied to standing and recumbent persons. If health effects are to be estimated, the standard recommends instead that each axis is evaluated separately and also includes an additional weighting factor of 1.4 for the lateral and longitudinal directions, since motions in these directions are considered more hazardous. Angular accelerations are recommended not to be included. Furthermore, health effect evaluation is mainly valid for seated persons. As ISO 2631-1 vibration evaluations are used in different ways depending on the purpose of the measurements, the objective results could be interpreted in several ways. This must be taken into consideration if results are to be used as input to the vehicle design process, as improvements in either comfort or health effects may lead to a decrease in the other aspect. For example, if pitch acceleration is reduced at the expense of increased vertical acceleration, this may improve the objective ride comfort but could also result in a less favourable dynamic behaviour with regard to health effects as health evaluation does not consider pitching motions.

One frequently mentioned drawback of the ISO 2631-1 standard is the lack of reliable methods for transients and shocks. As the standard uses RMS values to determine vibration severity, isolated transients will not contribute much to the total value if the measurement period is long. In the case of offroad vehicles, this is a clear limitation as operators of such vehicles may encounter shocks and transient far in excess of road vehicle drivers. For ride comfort evaluation, it has been shown that a running RMS average using a 1 s window length correlates well with the subjective impression from isolated shocks [14]. However, health effects are described less adequately by RMS values only. The latest part of the ISO 2631 standard, ISO 2631-5 [15], has been

developed specifically to address this deficiency. Rather than simply using acceleration values, ISO 2631-5 includes a mechanical model to compute the acceleration of the lumbar spine in response to seat accelerations. To evaluate health effects, accelerations are translated to an equivalent compressive stress dose S_{ed}, expressed in MPa, which is used as a health effect measure. Initial testing shows that this method appears to be able to detect shocks more precisely than ISO 2631-1 [16, 17]. Nevertheless, the application and usability of this standard remains limited for several reasons. The validation data for the mechanical model included is based on a rather narrow selection of test subjects (mainly healthy males in ages 20 to 30 years), meaning that accuracy can not be guaranteed for individuals that do not match this profile. The standard was also developed after legislations were issued. This means that conflicts may occur as current regulations state that older standards should be used for certification purposes although newer methods may actually produce more relevant results.

Other standards

A number of other national standards for vibration evaluation exist. Two examples are the British BS 6841 [11] and the German VDI 2057 [11], which are fairly similar to the ISO 2631-1 standard. Another method is the US Average Absorbed Power (AAP) [11], which uses a different method focusing on the vibration energy received by the human body over a period of time. While these national standards are still used to some extent, ISO 2631-1 is the most commonly used method.

3.1.2 Other comfort evaluation methods

Existing standards for ride comfort and health evaluation have been in use for some time and experiments show that reasonable correlation with subjective impressions can be found. Thus, standardised measurements can be considered a good starting point for engineering purposes even if results need to be treated with caution. The greatest uncertainty is probably in handling the effects of isolated transients and shocks. Based on current research, some alternative approaches have been suggested in order to handle this shortcoming.

Ride diagram

The ride diagram has been proposed by Strandemar and Thorvald [18] as a novel method for truck ride evaluation. The basic idea of the diagram is to separate transient accelerations from stationary vibrations, in order to present a more complete image of the nature as well as the severity of driver vibrations. This is done using an algorithm where the acceleration time history is divided into segments based on sign changes in the time derivative. The segments are then classified as either “transient” or “stationary” segments, depending on whether peak-to-peak changes exceed the RMS value of the entire measuring period. The mean square value of each segment is computed and the values in each category are summed to produce a “transient” and a “stationary” value. This is repeated for a range of velocities to produce a series of data that are plotted against velocity. Figure 5 shows the construction of two data points, as well as a complete diagram. As the ride diagram separates transients and shocks from stationary vibrations in a visual and practical way, the method appears suitable for

evaluation of offroad ride vibrations, although some questions still remain as to how the diagram should be interpreted.

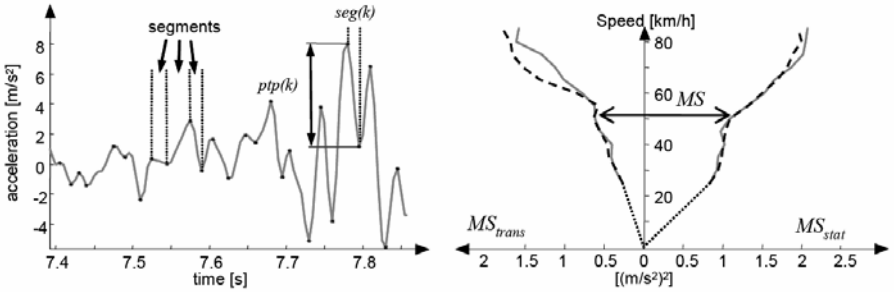

Figure 5: Principle of the Ride Diagram, from [18]

Jerk

The time derivative of acceleration, or ”jerk”, has been suggested by some authors as an indicator on transient vibrations [19, 20]. This has mainly been used to evaluate gear shifting comfort, as gear shifts typically introduce a peak in the longitudinal acceleration which is perceived as a disturbance by the driver. Results from the cited studies indicate that acceleration derivatives do affect the perceived vibration comfort, but that the total acceleration level also influences the operator experience. Hence, the perceived comfort seems to depend on a combined effect of both acceleration and jerk. Previous results therefore do not support any unified comfort measures based on the acceleration time derivative although it is suggesting that it may well be used for comfort evaluations if effects are investigated in further detail.

3.2 Handling evaluation

While established methods exist for ride comfort measurements, the evaluation of vehicle handling is considerably more complex and so far no unified measure of handling quality has been established. This is possibly because favourable handling is to a large extent governed by the actual task at hand, and also because subjective opinions become more important in the evaluation. The consequence of this is that handling evaluation is often left to the opinion of skilled test drivers. This is a useful input to the design process but has the disadvantage of requiring full vehicle tests. Objective handling measures are desirable as they allow early assessments of a given suspension design, by analytical predictions or simulation models.

For a wheel loader, handling quality evaluation is further complicated because usage of the vehicle is more varied and complex than the simple straight-line driving and cornering events encountered by ordinary vehicles. Figure 6 shows a load and carry operation, a typical scenario for a medium wheel loader: the bucket load is taken from a pile of crude material, transported a short distance and deposited on a receiving truck.

This driving scenario includes accelerating, braking and turning as well as driving in reverse and manoeuvring at standstill, all in a relatively short time. The use of the lifting arms to load and unload material also means that the centre of gravity is shifted vertically and longitudinally, thus altering the dynamic behaviour of the vehicle. Clearly, it is not easy to associate a single objective measure with favourable handling in this case.

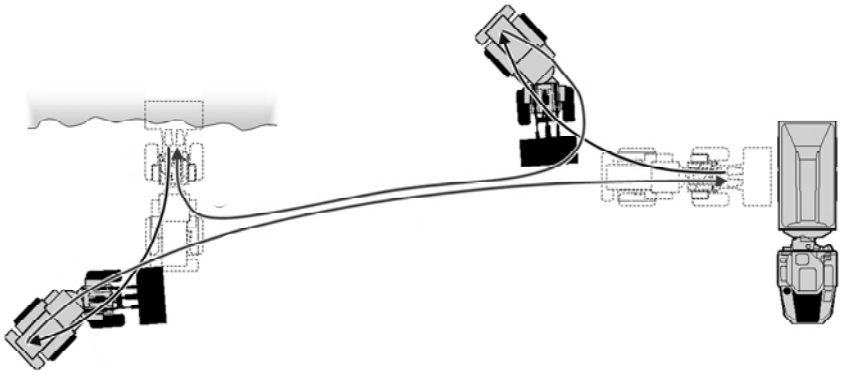

Figure 6: Load and carry operation

To quantify handling objectively, a number of approaches have been taken by researchers. A review by Uys et al [21] lists several of these, suggesting that vehicle roll angle can be used as a simple and robust measure as it is shown to correlate with lateral acceleration in an experiment containing passenger car manoeuvring. Variations in vertical wheel loads may also be used as a handling quality measure, as large fluctuations in wheel loads indicate loss of traction and control which directly relates to poor handling. Different formulations of the rollover stability margin have also been used, focusing on stability in steady state or transient turning events. More specialised studies of automobile handling typically use yaw rate gain, step response or similar measures from control system theory to evaluate handling characteristics. While this may be adequate to describe on-road manoeuvring, it is less likely to be applicable to wheel loader handling as construction machines are operated with larger steering inputs at lower velocities. Also, since the wheel loader is an articulated vehicle the steering dynamics are in some aspects fundamentally different from Ackermann steered vehicles; the steering system acts directly on the inertia of the vehicle body and tyre forces are produced as an effect of this rather than to generate tyre forces at the steered wheel.

One specific characteristic of articulated vehicles is the tendency for lateral instabilities when running straight at high velocities. Because of the flexibility of the hydraulic steering system, the vehicle is not perfectly rigid but maintains a small degree of freedom in the centre joint when driving straight. This means that instabilities may develop where the front and rear body are rotating in the yaw plane, either by an oscillating or a diverging articulation angle. This is roughly similar to unstable modes

occurring in tractor-trailer combinations and is typically aggravated with higher speed [22]. The lateral behaviour of articulated vehicles may be related both to the dynamic behaviour of the vehicle itself and to driver-vehicle interaction [23], and can lead to difficulties with path following or ultimately to loss of control of the vehicle. The tendency for this type of instabilities is a property specifically related to the design of articulated vehicles, and therefore the robustness against snaking or folding is a criterion for handling stability, especially in straight-line, high-speed driving.

4 Vehicle and surface modelling

4.1 Ride models

Vehicle dynamic models for ride analysis typically describe the vertical motion of the vehicle together with body rotations that may influence ride comfort. A commonly used model for vehicle ride dynamics is the quarter vehicle model. This model has been dubbed by Gillespie as a ”work horse tool” of vehicle dynamic studies [10], possibly because the equations of motions are straightforward and easy to use both in time and frequency domain analysis. As shown in figure 7a, the model consists of two rigid bodies, the unsuspended mass m_u representing the axle and wheel of the vehicle while the sprung mass m_s represent the portion of the vehicle body located over the axle. The springs and dampers represent the suspension and tyre stiffness and damping, respectively. The model has the degrees of freedom z_s and z_u, corresponding to the vertical motions of the sprung and unsprung mass. The terrain profile is represented by the displacement function w. By representing the front and rear end of the vehicle by quarter vehicle models as shown in figure 7b, the pitching motion of the vehicle can be computed geometrically from the front and rear end displacements.

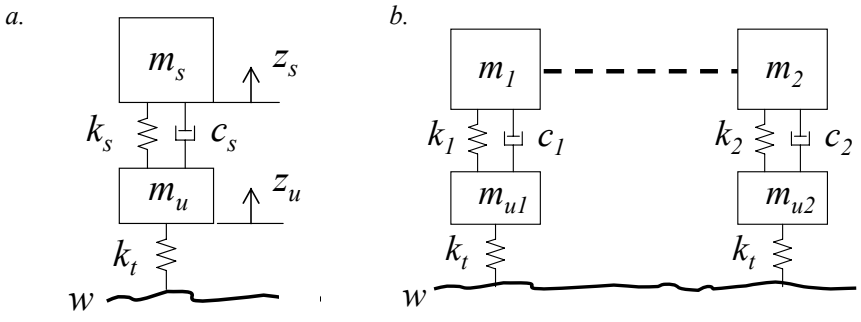

Figure 7: The quarter vehicle model

The approximation shown in figure 7b assumes that the pitch inertia of the suspended masses is matched by the two individual masses, and hence that the front and rear end of the vehicle move independently of each other. A more correct representation of actual pitching dynamics is shown in figure 8, where the vehicle is instead treated as a rigid body characterised by mass m and inertia I_{yy}. This "half vehicle" model has two degrees of freedom, the vertical displacement z and the pitch angle θ. The motion of the unsprung masses is often omitted from the model but can be included as additional degrees of freedom. The half vehicle model includes the possibility of coupled motions and pitching nodes located elsewhere than at the axle positions.

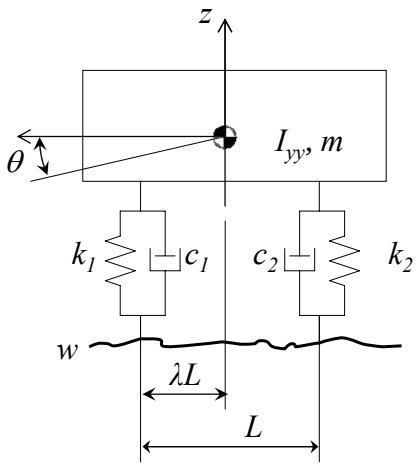

Figure 8: The half vehicle model

Some indication of the validity of the respective approaches can be gained by studying the dynamic index, DI, defined as [24]

$$DI = \frac{I_{yy} / m}{\lambda L \cdot (1-\lambda) L} \ . \tag{2}$$

The parameters L and λL are here the wheelbase and the distance of the centre of gravity from the front axle. If DI is unity, this means that the inertia of the vehicle is precisely represented by the front and rear masses in figure 7b and that the pitch nodes are located at the axle positions. Values of DI differing from 1 indicate that the pitch nodes, and hence the pitch mode shapes, will differ from the simplification in figure 7b and that pitching dynamics are not adequately represented by this model. Table 1 gives an overview of typical DI for some road vehicles compared to parameters representative of a medium sized wheel loader.

Table 1: Dynamic index for different vehicles

Vehicle type	***m***	I_{yy}	***L***	***DI***
Passenger car	1 470 kg	2 560 kgm^2	2.7 m	0.997
Large SUV	2 660 kg	7 280 kgm^2	3.3 m	0.982
Road tractor	13 700 kg	41 550 kgm^2	3.7 m	0.923
Wheel loader (unloaded)	22 000 kg	91 400 kgm^2	3.6 m	1.460
Wheel loader (loaded)	28 000 kg	20 000 kgm^2	3.6 m	2.378

It is seen in table 1 that most road vehicles have a dynamic index close to 1 and hence a model with independent front and rear motion is usually suitable to describe a road vehicle. The DI is considerably higher for the wheel loader which indicates that the pitching dynamics are different than for a road vehicle. This high pitch inertia compared to the mass and wheelbase is a fundamental property of a wheel loader, much owing to large masses being located outside the axles: the lifting arms add considerable weight ahead of the front axle even when unloaded, and the rear mounted engine and counterweight also contribute to the pitch inertia. Hence, a wheel loader is not particularly well described by independent front and rear motions.

The half and quarter vehicle models can be utilised for fundamental vehicle suspension design by means of spring stiffness selection, using the natural frequencies of the vehicle body as a design criterion. The half vehicle model in figure 8 has two degrees of freedom and hence will have two eigenfrequencies corresponding directly to the motion of the vehicle body. Using the front and rear end representation in figure 7b, the model will have four eigenfrequencies although the oscillation of the unsprung masses can usually be ignored when considering fundamental ride behaviour. Hence, the two eigenmodes of the vehicle body can be identified whether the representation in figure 7b or in figure 8 is used. These eigenmodes are commonly denoted “pitch” and “bounce” modes depending on the mode shapes, although both modes will normally contain both pitching and bouncing motion. Design guidelines for the eigenfrequencies of a vehicle body were first presented by Maurice Olley in the 1930:s and were based on ride tests in a passenger car with variable pitch inertia, known as the “k^2 rig” [25]. These design rules have since then remained largely unchanged. As reiterated by Gillespie [24], the basic recommendation for body frequencies state that no frequency should be greater than 1.3 Hz to provide sufficient vibration isolation at low frequencies. Furthermore the frequencies should be close to each other, with a relative difference no greater than 20%. This has been found to optimise ride comfort as the proximity of the frequencies prevents interference between the two eigenmodes, something that would otherwise lead to high oscillation amplitudes. For the same reasons, the body roll frequency should be lower than 1.3 Hz and close to the pitch and bounce frequencies.

While low eigenfrequencies are optimal for isolation of ride vibrations, there is also a lower limit as motion sickness may occur if the natural motion of the vehicle body is too low. Referring to figure 4 in chapter 3, it can be seen that motion sickness effects become apparent at about 0.3 Hz. This imposes a lower limit on the natural frequencies of the vehicle suspension. For passenger cars this is normally not a concern, as the low suspended mass and limited suspension travel provide an effective lower limit for the main ride frequencies. However, for heavy offroad vehicles it may be relevant since the vehicle body mass is higher and the springs are relatively soft.

Another design rule established by Olley is the “flat ride” criterion. Generally, pitching motions are considered more discomforting than pure bouncing motion and it is therefore desirable to tailor the dynamic response of the vehicle to minimise pitching. This can be achieved by having a stiffer rear suspension, as demonstrated by figure 9. When the vehicle strikes an obstacle, the front end will start to oscillate. The rear end of the vehicle will strike the same obstacle a time *t* later, thus starting a similar oscillation. As the graph in figure 9 shows, if the frequency of the rear end oscillation is higher than that of the front end, the front and rear end will soon see a synchronized vertical motion. This will lead to decreased pitching since the pitching motion of the vehicle body becomes smaller as the front and rear end moves more in phase with each other. Thus, a stiffer rear suspension will result in less pitching motion although this will also increase the vertical acceleration. This design rule, originally based on empirical evidence, has been further investigated by later researchers and it has been theoretically shown that the objectives of low vertical acceleration and minimal pitching motions are on opposite ends of the design space [26]. Clearly, the time delay between front and rear excitation is depending on the vehicle velocity and therefore the suspension setup will necessarily be optimised for a certain speed.

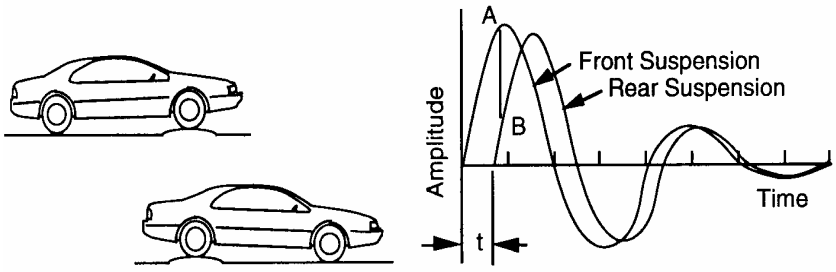

Figure 9: The "flat ride" suspension design principle, from [24]

As the pitch response of a given vehicle is highly depending on mass, inertia and design velocity, rules of thumb for flat ride springing is hard to establish. Generally, the effect of designing for flat ride is more prominent at high speed [27] meaning that the difference in suspension stiffness needs to be greater at lower velocities. The reasoning behind figure 9 is based on the model in figure 7b, since the front and rear end of the vehicle are assumed to oscillate independently of each other. However, the design rule is still valid even if the pitch nodes are located elsewhere than at the axle positions.

4.2 Handling and stability models

The planar, single track model is a common tool for handling analysis of Ackermann steered vehicles. A similar single track model can be used for articulated vehicles as well. This is illustrated by the model shown in figure 10, where the front and rear frames of the vehicle are represented by rigid bodies with specified masses and inertia m_1, I_1 and m_2, I_2, respectively. The frames are connected by a centre joint. Steering is accomplished by altering the angle δ between the frames.

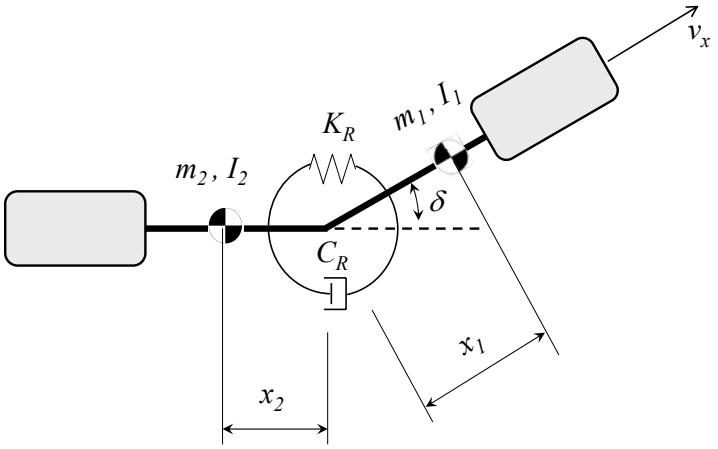

Figure 10: Single track handling model for an articulated vehicle

If the steering system is assumed to be rigid, the steady state turning characteristics can be analysed by means of under- and oversteering characteristics. Depending on the relative tyre stiffnesses and geometry, the vehicle will exhibit understeering or oversteering behaviour in the same manner as an Ackermann steered vehicle. This has been studied analytically by He et al [28] and has also been simulated numerically by Oida [29]. However, of greater interest is to study the possibility of lateral instabilities caused by the flexibility of the steering system. As articulated vehicles typically use hydraulic cylinders to produce a steering torque, some amount of flexibility will always be present due to compressibility of the hydraulic fluid and flexibility of various nonrigid elements. This flexibility around the set steer angle can be modelled by an equivalent torsional stiffness and damping, denoted K_R and C_R in figure 10. By linearising the equations of motion and studying the eigenvalues and eigenvectors of the resulting equation system, the stability properties of the vehicle can be determined. Such an analysis was performed by Crolla and Horton [22], who found that two instabilities are theoretically possible: a collapsing mode where the articulation angle δ grows exponentially and a weaving mode characterised by oscillation of the articulation angle. The stability of these modes, the former denoted “folding” or “jack-knifing” and the latter “snaking” or “weaving”, can be determined by examining the real part of the

eigenvalues related to these modes, as a positive real part indicates an unstable motion. This has been plotted as a function of the velocity in the graph in figure 11, using the parameters for the vehicle studied by Crolla and Horton; an articulated tractor of 3000 kg total mass. Here, the dashed line is representing the oscillatory mode and the solid line the collapsing or folding mode. It can be seen that the vehicle analysed displays a folding instability at about 12 m/s.

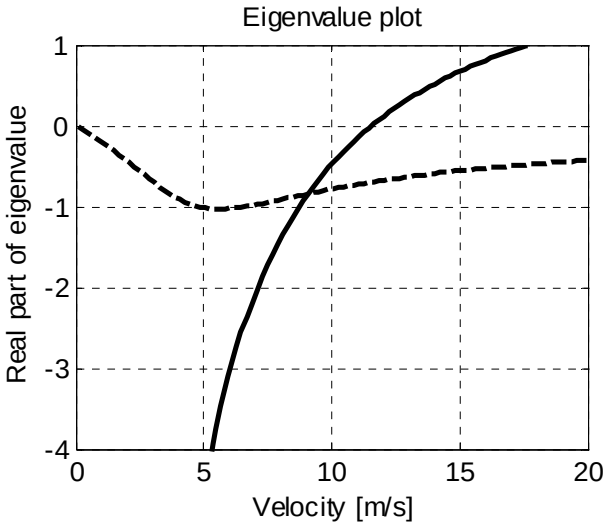

Figure 11: Eigenvalue analysis for single track model with vehicle parameters from [22], showing oscillatory mode (dashed line) and divergent mode (solid lines)

Parameter studies on the linearised single track model showed that high steering stiffness and damping highly augments the lateral stability [22]. Snaking stability is increased if the rear body centre of mass is moved forward, although this also makes the vehicle more sensitive to folding instabilities. Higher velocity generally leads to less stability although this is not always the case, some configurations also exist where stability is independent of or even increasing with velocity. The results from the basic model was later built on by Horton and Crolla [30], using a more refined model of the hydraulic steering system. The more advanced model revealed that an “oversteer” mode is also possible, which is similar to the folding mode but develops slower. Horton and Crolla concluded that the interaction between this oversteering response, and the tendency of the driver to overcompensate for this, is in practice the most likely cause for weaving oscillations. This is supported by later test results reported by Lopatka and Muszynski [23].

Figure 10 also highlights an important difference between the Ackermann steered single track model and the articulated model. For Ackermann steering, the inertia of the steered wheels are usually ignored and it is assumed that the steering angle is applied instantaneously at the steered wheel. For an articulated vehicle, this is clearly not a valid assumption as a step in steering angle would imply an infinite yaw angle

acceleration of the vehicle frames. Hence, the steering angle in the articulated model needs to be considered as a variable rather than an input, and steering should be controlled by means of an applied torque rather than a steering angle.

4.3 Tyre modelling

For any motor vehicle, the tyres are the primary way to transfer forces and moments from the environment to the vehicle. Hence, an accurate description of tyre properties is instrumental for the quality of the vehicle model. For simple, low-order vehicle dynamic models, tyre flexibility may be represented by simple spring elements as the models are accurate only for lower frequencies and therefore do not require higher detail. With increased model complexity and higher frequencies analysed, tyre forces need to be represented in greater detail. Due to its inherent complexity, accurate tyre modelling for vehicle dynamic simulations is an area of considerable research and still very much a work in progress.

Generally, large offroad tyres are characterised by relatively low pressure, a wide range of operating conditions and large axle loads. Offroad tyre forces and deflections are larger than for road vehicles, meaning that tyre nonlinearities become more significant. As the stiffness of the tyre is relatively low compared to the suspension stiffness and mass of the vehicle, the tyre flexibility largely influences the dynamics and natural frequencies of the vehicle body. Tyre models for engineering vehicles therefore need to take this into account. Test results for large tyres are limited, mainly because of the absence of large enough test rigs. Some results for high frequency vertical dynamics of agricultural tyres have been reported by Brinkmann and Schlotter [31]. Static rig measurements by Lehtonen et al [32] on three different types of heavy tyres indicate that dynamic properties, especially damping, varies considerably between different tyre models. Other than these, tyre force measurements for large vehicles are mainly done using full vehicle tests.

A multitude of tyre models currently exist, mostly designed for passenger car tyres and often focusing on lateral and longitudinal dynamics rather than ride. Here, two basic tyre models are described together with some more sophisticated models that are deemed more suitable for large tyres and ride analysis.

4.3.1 The Fiala tyre model

The Fiala tyre model [9] is included in the MSC.ADAMS multibody simulation software [33], and has thereby found widespread use in vehicle dynamic simulations. The main advantage of the model is that only a few parameters are required, all of which represent a physical or geometrical property of the tyre and therefore can be measured or estimated with reasonable accuracy. Lateral and longitudinal forces are computed by the model using a basically linear relationship, taking the lateral and longitudinal stiffness as input parameters. Maximum force is limited by the available friction and the model uses the friction parameters μ_{max} and μ_{min}, representing the available friction at full adhesion and full sliding, respectively. The model does not

adequately represent combined slip conditions, i.e. braking or driving while cornering. An example of the lateral force as function of slip angle is shown in figure 12, computed for a cornering stiffness coefficient of 600 kN/rad, vertical load 30 kN and max/min friction coefficients 0.9 and 0.5, respectively.

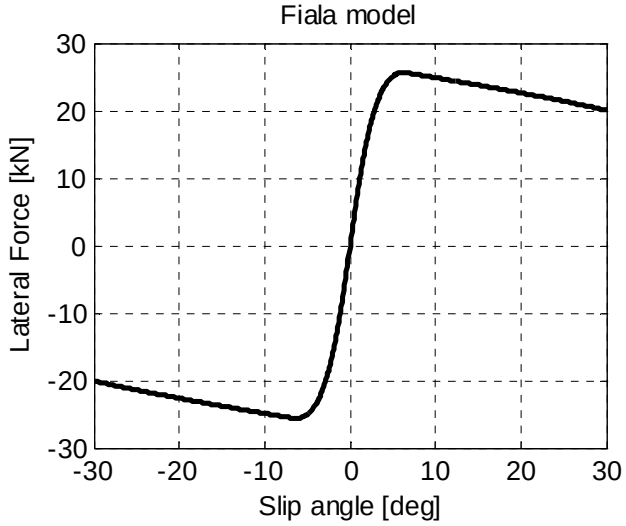

Figure 12: Example of lateral force vs slip angle, illustrating the characteristics of the Fiala tyre model

Vertical force is calculated in the Fiala model using a point follower model combined with a linear spring and damper. The vertical stiffness and damping of the tyre are specified as input parameters, as well as the unloaded radius. The model does not include camber effects, which is an acceptable simplification for earth mover tyres. While the Fiala model may not represent actual tyre physics with high accuracy, the simplicity of the model makes it an adequate choice for initial simulations as the required parameters can be estimated to at least produce reasonable results, which may be sufficient for comparative studies and early design evaluations. The Fiala model is based on a steady state assumption which may decrease precision especially for large and soft tyres that may exhibit considerable hysteresis in the build up of lateral forces.

4.3.2 The Magic Formula tyre model

The Magic Formula tyre model is an empirical model which was introduced by Bakker et al in 1989 [34]. The idea of the magic formula model is to curve fit an analytical expression to experimental data using the generic equation

$$F_q = D\sin\left(C\arctan Bq - E\left(Bq - \arctan Bq\right)\right). \tag{3}$$

To compute the lateral force F_y, q is substituted for lateral slip angle α. It follows from equation 3 that D represents the maximum force, while the product BCD represents the slope of the linear part of the force curve. An example curve is shown in figure 13. It can be seen that the force does not decrease monotonously for large slip angles, but approaches an asymptotic maximum. Although the curve in figure 13 is symmetric, the model allows for vertical and horizontal shifting of the curve to include nonsymmetric effects such as ply steer or tyre conicity.

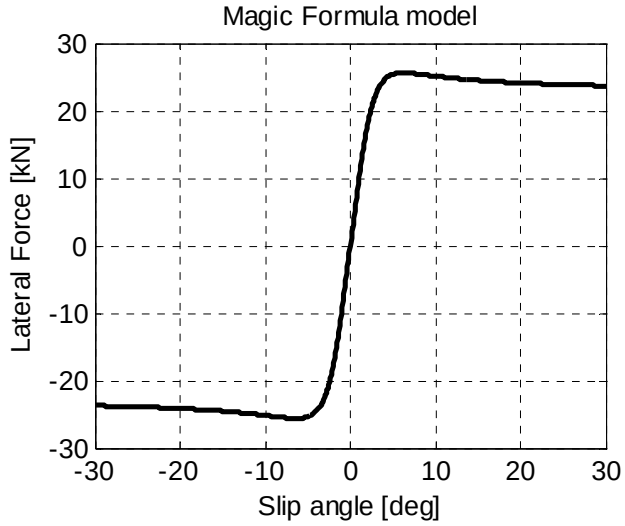

Figure 13: Example of lateral force vs slip angle, illustrating the characteristics of the Magic Formula tyre model

For the longitudinal force F_x, q is substituted for the longitudinal slip rate in a similar way as described above. The Magic Formula uses a point follower model together with a linear spring and damper to compute vertical force. Thus, for studies of vertical dynamics the model will not provide any higher accuracy than the Fiala tyre model. For lateral and longitudinal forces, the model may provide more accurate results if experimental results are available. The original Magic Formula model assumes steady state forces, which is a drawback for large tyre applications. Later updates of the model have attempted to remedy this by using more complex formulae that take transient tyre behaviour into account. This increases the need for experimental data and also requires extensive numerical computations to fit measurement data to the tyre equations.

4.3.3 Other models

The *FTIRE* (flexible ring tire) model [35] is based on a physical representation of the tyre belt, which is discretised as a number of rigid masses interconnected by spring elements as shown in figure 14. Thereby, the dynamic behaviour of the tyre belt mass and tyre-surface interaction is modelled in great detail, allowing high precision in the description of tyre vibrations especially on uneven ground. The drawback of the model

is the high computational effort required. The model is said to be validated up to 120 Hz and requires relatively few parameters, although these may still be hard to obtain for large tyres.

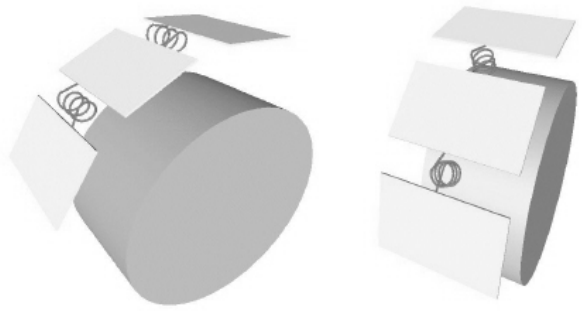

Figure 14: Graphical representation of the FTIRE model tyre belt discretisation [36]

The *SWIFT* model [35] is an empirical model that is originally based on the Magic Formula lateral force model which is combined with a rigid ring model coupled to the wheel rim with springs. This provides a simplified representation of the tyre belt dynamics in the rotational and translational directions, as shown in figure 15. The model is said to represent vertical tyre dynamics up to 80 Hz. A rigid ring approximation is also used for the tyre to ground contact, meaning that enveloping effects of small obstacles is included in the vertical force computation. This gives considerably higher precision than the point follower models used by the Fiala and Magic Formula models. A disadvantage of the SWIFT model is the large number of parameters required. Furthermore, since the Magic Formula model is based on stationary tyre forces it may not be well suited to large tyres.

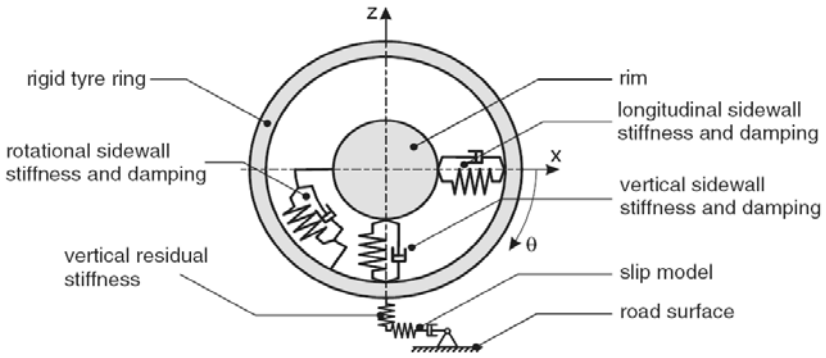

Figure 15: Principle of the SWIFT model tyre belt dynamics [36]

Another model developed specifically for large agricultural tyres is the *Hohenheim tyre model* [37, 38]. This model can be seen as a hybrid between an empirical and a physical model as it uses non-linear spring and damper elements to represent the forces and torques in all directions, but relies on test data to acquire the necessary stiffness and damping parameters to compute forces. The nonlinearity of the model allows for velocity-dependent stiffness and hysteresis, thus representing the tyre forces more accurate than steady state models. Input parameters are limited to physical parameters that can be measured in existing test rigs. The model has been validated for agricultural tyres against rig test results as well as full vehicle test results. However, given the difference in properties between different types of large tyres [32], it is not entirely clear whether the model could be adapted to earthmover tyres.

4.4 Suspension systems and modelling

Most current road vehicle suspensions use steel springs of leaf or coil type. These can be found on offroad and engineering vehicles as well, usually in conjunction with hydraulic dampers. Rubber springs are also utilised, particularly on articulated dump trucks. The main drawback of these springs is the inability to control ride height, or equilibrium suspension position. This means that the loaded suspension will have a different static position than the unloaded, which makes it impractical for a construction machine suspension as the mass and centre of gravity of the loaded configuration differs significantly from the unloaded. Hence, steel and rubber axle springs need to have high stiffness and therefore are subject to the same limitations as seat and cab suspensions, as discussed in chapter 1. For engineering purposes, steel and rubber springs can be modelled using linear or nonlinear springs. Although rubber elements are highly nonlinear, linearised models can be accepted if deflections are small and low frequencies are considered, as is the case for many vehicle dynamic studies. The suspension elements of the cab and seat, if included in the model, can also be modelled in this way as they are not likely to affect the vehicle body dynamics to any larger extent.

The problem with varying axle loads can be solved by using a hydropneumatic suspension, which is currently used on a number of engineering vehicles, military offroad vehicles and some passenger cars. The main advantage of this suspension system is the possibility to maintain constant ride height for varying spring deflection. A hydropneumatic suspension is based on a gas spring and the suspension displacement is carried to the gas spring by a volume of oil, displaced by a piston in the wheel suspension strut. Hence, by varying the total oil volume the equilibrium position of the chassis is altered. For an ideal gas spring, the pressure and volume in the gas container follow the ideal gas law relations

$$pV^{\kappa} = p_0 V_0^{\kappa}, \tag{4}$$

where p and V is the current state of the gas mass and p_0 and V_0 are the initial states. The polytrophic coefficient κ is 1.0 for an isotherm compression, and 1.4 for the

adiabatic case. Equation 4 gives the pressure in the system, which will translate directly to the spring force if piston dimensions are known. As the actual volume V varies directly with the suspension position this gives the force-deflection relation for the suspension. An example of such a curve is shown in figure 16, computed for a 7 litre accumulator, 9 bar pressure and a 75 kN normal load. The piston diameter is 0.125 m.

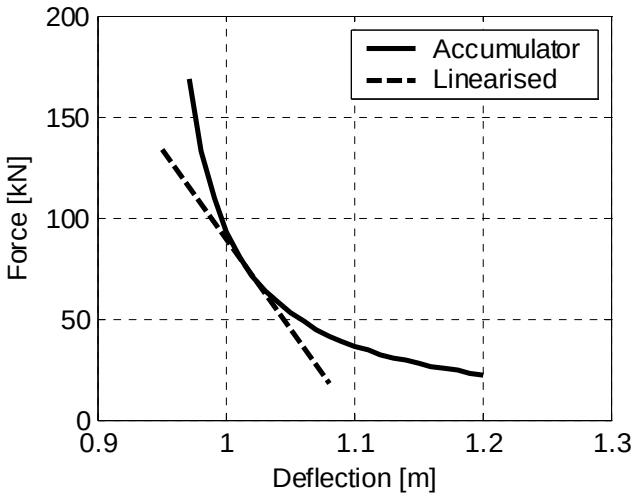

Figure 16: Force-deflection relationship illustrating typical characteristics of a gas spring

Equation 4 can also be used to derive the linearised spring stiffness coefficient k_{lin} for the gas spring. If the total gas mass is constant, this stiffness takes the form

$$k_{lin} = \frac{N^2 \kappa}{p_0 V_0} \ , \tag{5}$$

where N is the normal force acting on the spring. Adjustment of the equilibrium position can be done either by increasing the oil volume in the system (constant mass gas spring), or by increasing the gas pressure p_0 and thereby keeping the gas volume constant (constant volume gas spring). As equation 5 shows, the linearised spring stiffness increases with the square of the normal force on the spring, meaning that a constant mass gas spring in a vehicle suspension will have a tendency to stiffen under load since p_0 and V_0 are maintained constant. This will alter the natural frequencies of the vehicle body and therefore have a major impact on the dynamic behaviour of the vehicle. For a wheel loader, this is an important consideration since the static axle forces change considerably when the vehicle is loaded. The constant volume gas spring is less sensitive to this, as the linearised spring stiffness is approximately constant with load change. The effects of different types of gas springs on ride frequencies have been investigated theoretically by Harrison [39]. Generally, vehicle body eigenfrequencies

are nearly independent of load for the constant volume gas spring, whereas they increase strongly with loading for the constant mass spring. This means that constant volume gas springs provide more control over the vehicle ride behaviour although they also require a more complicated system.

4.5 Terrain modelling

4.5.1 Rigid ground

The most common approach to model surface roughness for vehicle dynamic studies is to assume that the terrain profile follows a random function in space that is stationary, linear and Gaussian. The vertical profile is treated as a function *z(x)* of a coordinate *x*, typically chosen as the longitudinal distance along the road. Using this assumption, the surface profile can be represented using the spatial Power Spectral Density (PSD) of the road profile, which indicates the mean square value of the road profile amplitude per unit spatial frequency bandwidth. The displacement PSD function $G_D(n)$ is computed using equation 6.

$$G_d(n) = \frac{2}{L} Z(n) \cdot Z^*(n) . \tag{6}$$

Here, *Z(n)* is the Fourier transform of the road profile function *z(x)*, with $Z^*(n)$ being the complex conjugate of this Fourier transform. The variable *n* is the spatial frequency, or wave number, equal to the inverse of the spatial wavelength, and *L* is the total length of the profile studied. The unit of surface profile displacement PSD is m^3.

The characterisation of standard road roughness has been defined in the ISO 8608 standard [40]. Using this standard, road profiles may be represented either by the displacement PSD, as defined above, or as acceleration PSD, the acceleration being defined as the rate of change of the road slope per unit distance. The unit for acceleration PSD is m^{-1}. The standard also requires that a "smoothed" PSD is presented along with the original PSD. This smoothing is done by computing the RMS of the PSD in frequency bands, thus avoiding the fluctuations in the curve at high frequencies. Furthermore, ISO 8608 includes a method to fit a straight line PSD approximation to the measured data. This function takes the form

$$G_d(n_0) = G_d(n_0) \cdot \left(\frac{n}{n_0} \right)^{-w} , \tag{7}$$

where n_0 is the reference spatial frequency of 0.1 m^{-1}, and the exponent w is defined by the curve fit. The standard also presents a series of standardised intervals for displacement PSD arranged in order of roughness from A through H. The boundaries for these standardised intervals can be seen in figure 17, where a constant exponent $w = 2$ is used and the change is made in the reference value $G_d(n_0)$. Note that the "A" class road only has an upper bound, whereas the "H" class is only characterised by a lower bound.

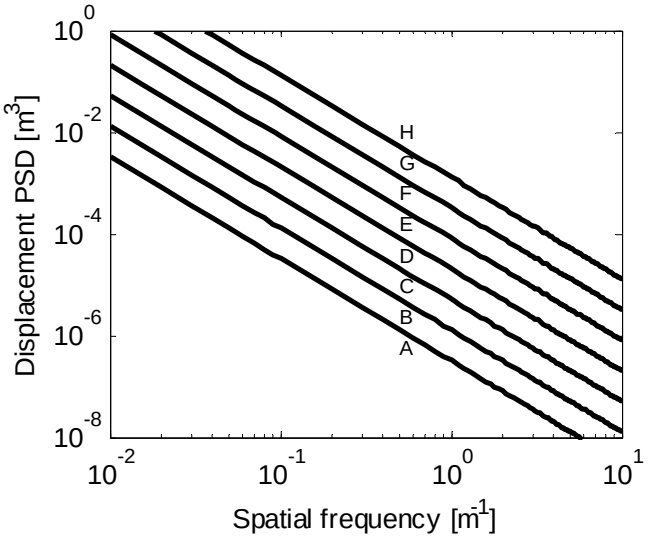

Figure 17: Standardised road roughness according to ISO 8608

The main advantage of this description is that the velocity PSD, where the velocity is defined as the rate of change of the profile displacement per unit length travelled, will be a constant function of the frequency n. This can be seen from the relationship between displacement PSD G_d and velocity PSD G_v, defined by equation 8.

$$G_v(n) = G_d(n) \cdot (2\pi n)^2 . \tag{8}$$

Thus, $w = 2$ means that $G_v(n) = G_v(n0)$ and hence the terrain profile curve can be obtained by integrating white noise. This practice is recommended in ISO 8608 as it considerably simplifies calculations.

A number of PSD estimates for offroad terrain profiles have been published. Generally, these studies indicate that the PSD of uneven terrain is described less accurate by the single line curve fit used in ISO 8608. Hence, more than one line is needed to obtain a representative curve fit. One of the most extensive surveys is published by

Fujimoto [41], who studied measurements on a construction site in Japan using a single wheel profilometer. The results indicate that amplitudes are generally higher at longer wavelengths, as shown in the graph in figure 18 that summarises the measurements results compared to ISO standardised road profiles of the time and also shows the suggested curve fit functions to match the test results. The effect of tyre tread prints, seen between about $n = 2$ and $n = 5$, were deemed by Fujimoto not to affect vibrations because vehicle tyres will filter out these smaller irregularities.

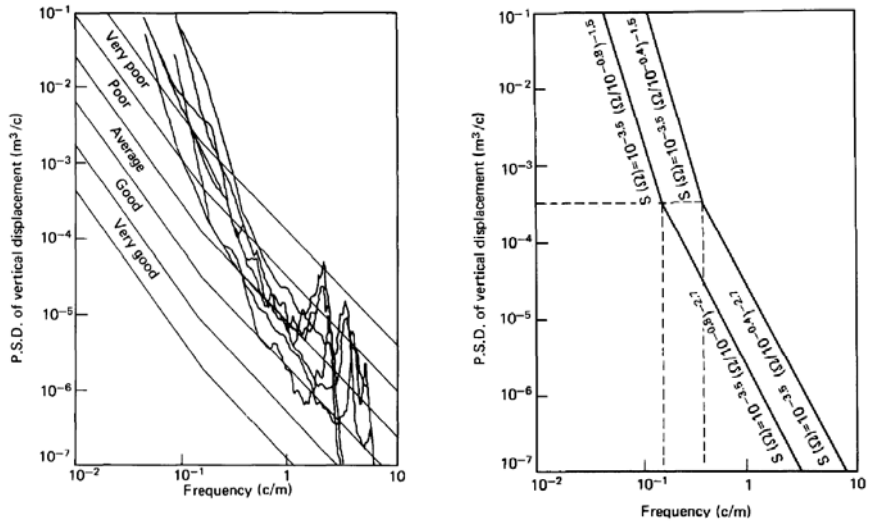

Figure 18: Displacement PSD from construction site and suggested curve fit, from [41]

An analogue investigation by Malmedahl et al [42] shows a similar trend for measurements taken on a military offroad course. Muro [43] also studied a construction site surface for the purpose of tyre wear analysis. PSD curves from this study are basically the inverse of Fujimoto's results, although the study is considerably less extensive and may not be sufficient for general conclusions. Overall, results from offroad terrain profiles indicate that differences exist between individual measurements as well as between the measurements and the standardised road profiles. Hence, care should be taken when selecting a terrain model for construction machine simulation and analysis. Another consideration for offroad terrain profiles is the effect of the vehicle on the terrain by the vehicle. It is likely that obstacles of short wavelengths will be flattened by the tyre and hence will not excite the vehicle. If a soft terrain is represented by a rigid profile model, filtering should be applied to remove the shorter wavelengths.

As previously stated, describing a surface roughness by its PSD assumes a linear, Gaussian and stationary terrain profile. The accuracy of this assumption, especially for offroad terrain, has been discussed by Gorsich et al [44], who performed a statistical analysis of a Belgian block test stretch and a military offroad course. The authors found that the Belgian block terrain was linear and Gaussian, but not stationary. The offroad track was neither linear, Gaussian nor stationary. This indicates that offroad terrain profiles that feature large scale obstacles and undulations are less well described by the

PSD, and that this method is more accurate for repeatable structures of low amplitudes. Alternative modelling techniques for offroad terrain profiles have been proposed by Sun et al [45].

Surface roughness can also be characterised by its effect on vehicle response rather than the actual terrain profile. This approach is the basis for the International Roughness Index, IRI [42]. The index is computed by simulating the response of a quarter vehicle model traversing the surface profile. The suspension deformation velocity of the model is integrated over time and normalised by the distance travelled, so that the final result is total suspension displacement per unit road length. The model parameters are fixed and thereby the quarter vehicle model basically works as a band pass filter for the terrain, as the surface undulations are transferred to the suspension motion of the vehicle model. The IRI provides a direct measure of the amount of suspension deflection caused by the particular surface but the relevance of IRI is also highly depending on the ability of the vehicle model, sometimes referred to as the "Golden Car", to represent the actual vehicle. For a construction machine this is further complicated by the fact that a quarter vehicle model does not represent the actual dynamics of a wheel loader particularly well. Hence, the filtering of the surface profile may not be adequate for the actual effect on the wheel loader. Also, the suspension displacement does not necessarily correspond to ride quality since suspension deflections may not be representative of the perceived comfort; large suspension deflections could actually indicate less body motion which would be beneficial for ride vibrations. For stress and fatigue calculations, the output may be more relevant. One such analysis has been performed by Howe et al [46], who combined results from the IRI quarter vehicle model with a fatigue prediction model to translate IRI values to stress effects. This method could possibly be applied to construction machine studies as well, given that terrain and vehicle models are well calibrated.

4.5.2 Deformable ground

The study of deformable ground mechanics, *terramechanics*, is a complex topic that has been the subject of much research. A comprehensive introduction to the science is given by Wong [47]. Historically, the main focus of terramechanic studies in vehicle engineering has been vehicle longitudinal loads, in order to compute the tractive performance or "drawbar pull" of offroad vehicles such as agricultural tractors. A relevant question is whether a softer suspension designed for improved ride could impair vehicle traction. Studies on dynamic wheel loads on soft soil [48] indicate that some increase in tractive performance may be achieved using wheel suspension, although the study seems too limited for any general conclusions mainly because of the narrow frequency range studied. For lateral loads, it has been seen that the presence of lateral soil displacement or "sidewalling" causes high lateral loads, while the low effective friction limits the longitudinal forces [49]. An experimental study on tyre lateral forces on sand was carried out by Raheman and Singh [50], indicating that the lateral forces can be described quite well by an exponential function of the slip angle. Effects on vertical vibrations from deformable ground seem to be a largely uncharted area, but it could be intuitively suggested that soft ground should not affect ride comfort negatively as the cushioning effect of the ground works as an additional damper,

alleviating surface induced vibrations. Nevertheless, the interaction of ground and suspension damping could possibly affect the ride vibrations negatively and it should not be automatically assumed that a design optimised for good ride on rigid ground is optimal for soft terrain as well.

Because of the complexity of the subject and the lack of relevant research, especially on ride vibrations, effects of deformable surfaces have not been included in this thesis work. As construction machines operate mainly on unpaved ground, it remains an area for future research. A particularly interesting topic is whether conflicts may arise between favourable ride and tractive and/or handling performance on soft ground.

5 Multibody simulations

5.1 Basics

Multibody simulation, MBS, is a method for numerically simulating the dynamics of systems of interconnected rigid bodies. MBS software was first introduced in the 1980:s and has since then become commonplace in the automotive industry as well as in several related areas. To simulate a system of rigid bodies and point masses, MBS algorithms first derive the equations of motion for the system, commonly using Lagrange formalism. The program then uses a set of constraint equations to include the effect of boundary conditions and joints. This gives an equation system that can be integrated numerically to obtain the motion of the system studied. The constraint equations are defined by the joints and boundary conditions present in the model, and works by removing degrees of freedom from the system. For example, a revolute joint allows only rotational displacement between two bodies and thereby removes five degrees of freedom. The number of degrees of freedom n_{DOF} for an arbitrary multibody system can be computed using Gruebler's equation [9],

$$n_{DOF} = 6 \cdot (n-1) - f \,, \tag{9}$$

where n is the number of rigid parts including the ground, and f is the number of constraints.

For vehicle dynamic studies, MBS models enable simulations of full vehicle events in three dimensions with several degrees of freedom, thereby allowing more accurate and complex analyses than simple low-order models. Suspension kinematics may be simulated in high detail, providing designers with input early in the design process and allowing optimisation of detail designs. For a complex vehicle, an MBS model can provide a way to study the interaction of various resonances occurring in the vehicle. This is especially true for heavy vehicles, where low frequency resonances exist in the same order of magnitude as the main ride frequencies. For instance, a wheel loader cab

with eigenmodes in the range of 8 Hz to 20 Hz will clearly have an effect on the ride dynamics and hence needs to be included in the analysis. Commercial MBS software usually also includes graphical functions that enable animation of the simulation results.

While multibody simulations are a potent aid for vehicle dynamic analysis, the use of MBS has also introduced a number of potential pitfalls. Many of these have been pointed out by Blundell and Harty [9]. The most obvious is that the ease of use of modern MBS software may cloud the necessity for basic knowledge of vehicle dynamics that is essential for a full understanding of simulation results. Blundell and Harty states that *"While the freedom from the purgatory of formulating one's own equations of motion is a blessing, it is partly this purgatory that aids the analyst's final understanding of the problem"*. Thus, the use of MBS analysis should not be allowed to offset the sensible practice of basic engineering skills and MBS results should preferably be used in conjunction with results from more fundamental analytical models, as to verify general trends and maintain a basic understanding of the underlying dynamics of the system. Another risk, also pointed out by Blundell, is the "paralysis of analysis", meaning that the possibility to simulate a dynamic system is greater than the ability to analyse it. Hence, post processing of the results may be delayed or hampered because the numerous output variables can not be interpreted in a clear manner.

Furthermore, even a detailed simulation model is no more valuable than the input data. For a preliminary study, limited validation may be accepted as the goal is to provide qualitative results rather than to produce actual data and so a "representative" set of parameters may be sufficient as long as the result are treated with care. However, if results are to be used as an input to the actual design process the model needs to be fully validated within its range of operating conditions. Detailed information about the masses and inertias of the rigid bodies included is also necessary, which is sometimes less easy to obtain as some parts of the model will inevitably consist of several vehicle parts and components. Obviously, the increased complexity of simulation models also increases the need for testing. A number of papers have been published on the topic on MBS model validation. Some of the more interesting are the validation of a heavy truck model done by Anderson et al [51], using a large number of measurements and subsystem validations to obtain simulation accuracy; and the validation of a car model by Rao et al [52], where a statistical method is applied to verify the simulation model against test data.

5.2 Example model

Figure 19 shows a graphical representation of a wheel loader model developed in MSC.ADAMS/View [33]. The figure also shows the included parts. This model consists of a front and rear vehicle frame connected in the centre using a revolute joint. The front wheels are attached directly to the front frame and the rear wheels are connected to the rear axle which is in turn attached to the rear vehicle frame by a revolute joint. This arrangement allows the axle to pivot relative to the frame. All wheels have freedom to revolve around their attachment points. The front lifting attachment is represented by a simplified, single part that is attached to the front part using a revolute joint and a spring/damper that represents the flexibility of the lifting

hydraulics. The vehicle cab is attached to the rear frame using elastic elements and thus has six degrees of freedom relative to the vehicle body.

Using equation 9, the degrees of freedom of the model can be computed for this model. The wheel loader model consists of 10 parts including the ground, hence n = 10. Furthermore, the model includes 7 revolute joints. As a revolute joint allows only one degree of freedom, each revolute joint removes 5 degrees of freedom from the model meaning that f = 35. Hence, equation 9 will return n_{DOF} = 19 for the wheel loader model.

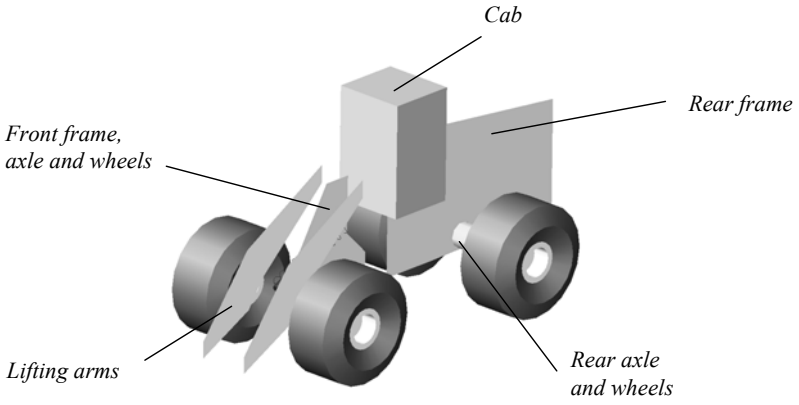

Figure 19: Multibody wheel loader model

Figure 20 shows the frequency response of the wheel loader model. The loader is subjected to a harmonic vertical motion of 0.1 m amplitude applied under each tyre, which are represented here by vertical springs. The resulting vertical displacement and pitching motions have been measured in the cab, which is attached to the rear frame using elastic elements. The resulting transfer functions $H_{\theta w}$ and H_{zw} have been computed using the ADAMS/Vibrations utility, which linearizes the equations of motion around an equilibrium position and uses the linearised state equations to compute the frequency response. Computations have been done with no damping, hence the amplitude gain is theoretically infinite at the natural frequencies and the graphs indicate only the location of the eigenfrequencies. The main body frequencies, where the vehicle is pitching and bouncing on the tyres, can be seen between 2 and 3 Hz. The first eigenmode of the cab is seen at about 8 Hz and it can also be seen that the mode is dominated by pitching motion. The remainder of the cab eigenfrequencies show up as peaks in the range between 20 and 30 Hz. Thus, the multibody representation of the wheel loader can provide more extensive information about the dynamic behaviour than is possible using low-order quarter or half vehicle models.

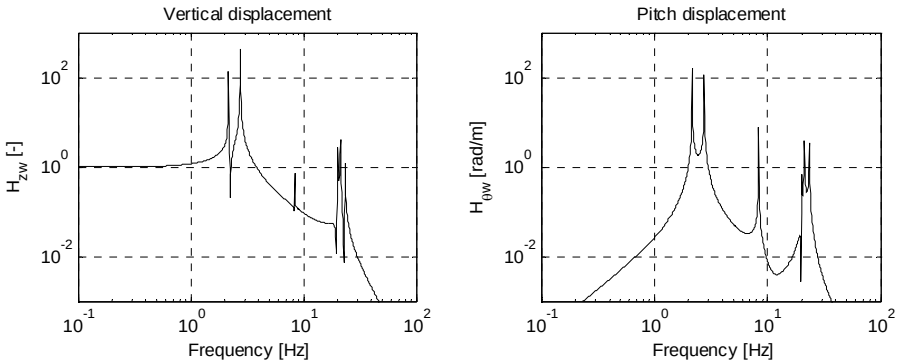

Figure 20: Transfer functions in vertical displacement and pitch for wheel loader model

6 Summary of appended papers

Paper A: Ride comfort simulation of a wheel loader with suspended axles

The objective of this work is to investigate the potential of including axle suspensions on a wheel loader to reduce ride vibrations. A multibody simulation model of a traditionally built wheel loader of 22 000 kg mass is developed. This model is expanded with a three-point rigid axle suspension and linear springs. As a test case, the model is simulated running on a test track consisting of depressions and bumps, modelled after an existing test track designed for testing of construction machines. This provides a repeatable ground excitation and also lays a ground for future field tests. RMS accelerations in three directions are used as a measure of vibration severity. Dynamic wheel loads and vehicle roll angle are used as indicators of handling stability. The unsuspended and suspended vehicles are simulated at different velocities, both loaded and unloaded, and results are compared to show the gains and potential problems when including axle suspensions.

The results show that considerable reduction in vertical and longitudinal vibration magnitude can be attained for the suspended vehicle. The reduction in lateral acceleration is noticeably less. Thus, the simulations indicate that even a simple suspension design will alleviate overall vibration levels but that lateral accelerations are not reduced any greater amount. Parameter studies show that lateral accelerations are decreased if roll stiffness is reduced, although this may also affect the stability of the vehicle. It is also shown that lateral accelerations are reduced if the driver seat is moved downwards. However, this may not be feasible in practice because of visibility requirements.

A basic tyre model and a multibody model with relatively few degrees of freedom are used in the simulations. This means that the ride vibration estimates are simplified and should therefore be seen as a comparative study rather than an absolute prediction. As such, the study gives an indication of the benefits of including an axle suspension.

Paper B: Pitch comfort optimisation of a front end loader using a hydropneumatic suspension

The paper builds on Olley's well known "flat ride" criterion that was first pioneered in the 1930:s. This design concept builds on the idea of a stiffer rear axle suspension to reduce pitching oscillations, usually at the expense of higher vertical accelerations. This original design rule has been theoretically verified in a number of more recent publications. A half vehicle, 2 degree of freedom model of a wheel loader is used to investigate the feasibility of flat ride design for a loader type vehicle. The frequency response of the pitch angle and vertical acceleration is studied, using the ratio between rear and front axle stiffness as a parameter. It is found that pitch angle gain is significantly reduced if the spring stiffness of the rear axle suspension is about 4 times higher than that of the front axle, while maintaining the total suspension stiffness. However, a similar effect is seen if the front axle suspension is made stiffer. In both cases, a moderate increase in vertical acceleration is seen. In order to compare to a more realistic vehicle model, multibody simulations are performed where a wheel loader model with 22 degrees of freedom is analysed in a single bump event and a sine sweep road traverse. Results from multibody simulations are shown to follow the same trend as the analytical predictions.

A linearised model of a hydropneumatic suspension is used to include the effect of load on suspension stiffness. The stiffness of the gas spring element is proportional to the square of the axle load and therefore the front axle stiffness will increase to roughly nine times its original value as the vehicle is loaded. At the same time the stiffness of the rear axle is reduced to about half of its value when unloaded. This means that for most designs, the ratio of rear to front suspension stiffness will be inverted when the vehicle is loaded. As these setups both reduce pitching motion, it is possible to find two optimal setups so that the vehicle will exhibit less pitching tendency in loaded as well as unloaded configuration.

The paper also demonstrates that design rules derived for passenger cars are valid for a loader vehicle as well, even though there are some significant differences between the vehicles. A passenger car typically has a dynamic index close to 1, meaning that the front and rear end are roughly independent of each other and that the respective suspension stiffness directly affects the frequency of the related eigenmode. For the wheel loader the pitch and bounce mode are more complicated as the dynamic index is higher, so the front and rear suspension stiffness values do not relate directly to the natural frequencies of the pitching motion. Nevertheless, the design rules originally established by Olley are shown to be valid for the loader vehicle as well.

Paper C: Snaking stability of articulated frame steer vehicles with axle suspension

Articulated frame steer vehicles such as wheel loaders are known to exhibit weaving instabilities known as "snaking", especially when running straight at high speed, caused by flexibility in the hydraulic steering system. This has previously been investigated using planar models, unsuspended multibody models and field tests using a rigid axle

wheel loader. In the case of a suspended loader, it can be expected that the rolling motion of the vehicle body will have an effect on the lateral stability of the vehicle. For this purpose, a multibody simulation model is developed using parameters representative of a 22 000 kg wheel loader. The suspension is simplified to an equivalent roll stiffness model, meaning that the axles can roll freely but that no pitch or bounce motion is possible. The stiffness of the steering system is represented by an equivalent torsional stiffness applied to the centre joint. An unsuspended vehicle model is developed for comparison, with the front axle locked and the rear axle pivoting freely.

To quantitatively estimate the effects on stability, a mathematical method is used where the rate of growth or decay of the articulation angle is used as a stability indicator. A range of parameter studies are performed on the unsuspended vehicle, to establish the behaviour and form a base for comparison. With suspended axles, roll stiffness and damping are varied to study effects on snaking stability. It is seen that fully suspended axles deteriorate stability to some extent compared to the unsuspended configuration, but that the effect is quite small compared to the influence of centre of gravity location and steering system stiffness. The vehicle is least stable when the frequency of the snaking oscillations coincides with the vehicle body roll frequency. However, if the equivalent torsional stiffness of the steering system is sufficiently high this should not be a problem since a stiff steering system stabilises the vehicle. Thus, the only case where truly destabilising interactions could occur would be a configuration with low steering stiffness as well as low roll stiffness. Such a design would be undesirable for comfort reasons since the roll frequency would be as low as 0.5 Hz and it can therefore be concluded that a well designed suspension should not deteriorate snaking stability to any greater extent.

7 Conclusions

The work presented in this thesis includes a study of the potential to improve ride comfort and handling by adding a wheel suspension to an unsuspended vehicle, the wheel loader. Simulation and analytical predictions have been used to evaluate effects on ride vibrations and vehicle stability. The dynamic behaviour of suspended wheel loader models has been compared to that of unsuspended models, in order to investigate the specific design issues related to a wheel loader axle suspension.

Results from ride comfort simulations indicate that the reduction in vertical and longitudinal acceleration is considerable, even using a simple rigid axle suspension type. However, lateral accelerations remain high. While this can be alleviated by lower roll stiffness, such a modification would also lower the stability of the vehicle and hence is not desirable. It can therefore be predicted that the compromise between low lateral vibration and sufficient roll stability poses the greatest challenge in loader suspension design. These results are in accordance with practical experience from forestry vehicles and even trucks on damaged roads, where large lateral vibrations caused by roll disturbance can be seen despite increasingly sophisticated suspension systems.

The fundamental analysis of the vehicle pitching behaviour also indicates that desirable vehicle ride characteristics by means of pitch and bounce frequencies can be maintained when the vehicle is loaded, if spring stiffnesses are adequately selected. This is in spite of the fact that vehicle inertia properties as well as the stiffness of the hydropneumatic suspension changes considerably under load. Hence, designing a passive suspension system for favourable ride is possible despite the varying axle loads. Furthermore, results show that the “flat ride” criterion may be fulfilled both in loaded and unloaded configuration by selecting sufficiently higher rear suspension stiffness for the unloaded vehicle. This also means that the pronounced tendency for pitching oscillations can be controlled by suspension design. While a similar effect could be attained by altering the wheelbase of the vehicle, this would be highly undesirable due to space requirements and various layout considerations.

Results from snaking stability investigations show that the important lateral stability is slightly deteriorated by adding axle suspensions. The effects are seen to be quite small and problems with low snaking stability appear to be mainly related to low roll

stiffness, which is undesirable for reasons of roll comfort and handling stability. Hence, the selection of suspension roll stiffness for the wheel loader is more likely to be affected by fundamental considerations such as roll stability when turning, than the effects on snaking stability.

The vehicle dynamic behaviour of the wheel loader is in many aspects different from passenger cars and other more traditional vehicles. The papers presented in this thesis show some particular aspects related to suspension design for wheel loaders. Together with existing vehicle dynamic theory and practice, this provides a framework for the continuing development of wheel loaders with axle suspensions.

8 Future studies

This thesis and the appended papers sum up the basics of suspension design applied to wheel loaders and engineering vehicles. Based on previous knowledge and the findings presented here, some areas that deserve further investigation can be identified. Furthermore, some areas have been excluded from this work and hence may be the subject of further investigation.

8.1 Refinement and expansion of simulation models

The results presented in this thesis are based mainly on results from vehicle dynamic simulations. Concerning the simulation models used, a number of possible refinements exist that may increase the accuracy of the simulation results. The most obvious is updates to the tyre model. It is known that large offroad tyres differ in many aspects from smaller tyres and also that the dynamic properties of heavy tyres have been shown to be rather diverse for different types of tyres [32]. This means that the properties of earth mover tyres can be considered largely unknown. As the tyres have a major influence on the dynamic behaviour of the vehicle, increased knowledge of tyre properties is highly desirable for accurate simulation of construction machine vehicle dynamics. For ride comfort analysis, the vertical dynamics of the tyre are most important, and it is likely that tyre belt vibrations will affect ride vibrations due to the large belt mass in heavy offroad tyres. Lateral properties are also important with regard to snaking stability studies, especially tyre nonlinearities in the form of extensive relaxation length and hysteresis that may affect the outcome of simulations as the snaking oscillations include relatively large changes in the tyre lateral force over short time periods. Existing tyre models such as FTIRE or the Hohenheim model are most likely adaptable to construction machine tyres as well. However, the gathering of tyre parameters is still a problem due to the lack of adequately sized test rigs. Unless new test rigs are developed, tyre parameters need to be estimated from full vehicle tests. This can be accomplished with adequate test cases and careful planning of experiments.

Another possible model refinement is the inclusion of vehicle frame flexibility. This has been studied to some extent for heavy road vehicles. For trucks, the bending modes in the longitudinal frame members are of sufficiently low frequency to contribute to

ride vibrations [53]. Studies using finite element methods show that frame bending and torsion affects vehicle ride vibrations and that rigid models will produce different results [54]. A similar case exists for buses, where finite element analysis has shown that vehicle body flexibility affects ride vibrations and also that the ride quality can be optimised by altering the structure of the bus [55]. Although the wheel loader frame is more rigid than a truck or bus structure, it is likely that frame bending may have an influence on the ride vibrations especially in the rear part of the vehicle. This could be incorporated into existing multibody simulations either by finite element simulation of flexible parts, or by discretising the parts into several rigid bodies connected by spring elements. Modal analysis on existing machines can be used to indicate which natural frequencies are related to vehicle frame flexibility.

Naturally, enhanced simulation model detail will also increase the demands for validation. As the requirements for test data grows rapidly as model detail increases, the resources available for testing will most likely be a limiting factor for the development of large and complex simulation models.

8.2 Handling and stability measures

The ride comfort and handling evaluations presented in this thesis are based solely on straight-line driving, which is an important and fundamental operating mode for any vehicle. However, this provides only a narrow view of the vehicle dynamic behaviour. It is well known that vehicle suspension design is a tradeoff between ride comfort and handling. While ride comfort can be reasonably well quantified, no standardised measure for handling exists although a number of estimates have been proposed [21]. This is possibly because “optimal handling” to a large extent is defined by the actual manoeuvring event considered. Commonly used tests include lane changes, avoidance manoeuvres or steady-state turning. As a wheel loader is used in a different manner than a transport vehicle, these tests are not likely to be suitable as evaluations for wheel loader handling stability. Rather, realistic and repeatable test cases should be developed based on the actual use of the loader, which typically includes short driving distances and frequent stops for loading or unloading the bucket. Wheel loader driving scenarios also usually include wide shifts in centre of gravity as the lifting arms are raised and lowered, as well as driving in reverse for a significant part of the operating period. Based on these typical cases, subjective estimates of handling qualities can be investigated and possibly correlated with measurable parameters such as accelerations or wheel loads.

Wheel loaders have been in use for some time and a considerable experience base exists among manufacturers and users, although this has not been formalised to any greater extent. To establish measures of “handling quality” for loader type vehicles, this experience needs to be reviewed and summarised. This task will necessarily include subjective tests and realistic yet repeatable test setups.

8.3 Active spring and damper systems

The inherent compromise between ride comfort and handling involved in vehicle suspension design is even more pronounced for offroad vehicles since the large suspension stroke and soft springs necessary for acceptable offroad ride quality provides poor handling, especially on flat and rigid ground. This has been demonstrated by Els et al [56]. It follows from this that passive suspension designs will necessarily be a compromise between vibration isolation and handling stability.

Active suspensions, allowing adaptive control of spring and damper forces, can provide a way to accommodate both ride and handling in a more efficient way than purely passive systems. For example, active anti-roll systems may suppress excessive roll motions when turning while still maintaining low enough stiffness to reduce lateral accelerations on rough ground. A number of studies have been presented on active suspensions for offroad vehicles, mainly aimed towards military vehicles. These studies are generally applicable for construction machines as well, although some complicating factors exist. The axle loads of a heavy wheel loader are larger than a typical offroad vehicle, meaning that actuators and suspension elements must be able to produce sufficient forces for suspension control. Furthermore the change in axle loads is large, meaning that control algorithms must be tailored to match the change in vehicle dynamics with load configuration. As with any active system, the addition of actuators and sensors will also increase the cost and complexity of the vehicle.

Development of active systems for a wheel loader can be viewed at different stages. On an elementary level, simple on/off algorithms can be developed to activate or deactivate the axle suspension when required by operating conditions. For example, when the lifting arms are elevated to maximum height, the vehicle centre of gravity will be elevated as well and very high roll stiffness will be required to maintain stability. In such a case it will most likely be necessary to lock up the front axle suspension to ensure rollover stability. As this would happen only during a limited period of the operating time, the influence on ride vibrations should be minor. A more refined implementation could be a semi-active system, allocating damper forces to optimise the ride and handling over a range of operating conditions. An example of such an implementation is the modal damping concept, where damper forces are distributed to compensate for the disagreement in relative damping for the different eigenmodes of the vehicle [57]. As the wheel loader has relatively large pitch inertia and offers only limited freedom in suspension layout and damper placement, such an approach may be necessary for the loader suspension as well. The most advanced solution would be a fully active system, using actuators to produce the desired sprung mass response under a full range of operating conditions. This would necessitate development of control algorithms and also the establishment of control objectives representative of the desired dynamic behaviour.

While active systems add considerable more potential for improved ride and handling, it is important that the design is based on a well designed passive suspension. In other terms, basic spring and damper characteristics must be selected so that the natural dynamics of the vehicle are as close to the optimum as possible. If this is not accomplished, an active design will not be an extension of the vehicle suspension but

rather a fix for a fundamentally flawed setup. This will not produce optimal results and hence needs to be avoided.

8.4 Advanced suspension design

Aside from spring and damper characteristics, suspension geometry also has an influence on the vehicle dynamic behaviour by means of roll centre location, squat and dive tendencies and other properties related to suspension kinematics. The vehicle simulation models developed in this thesis use simple suspension layouts based on rigid axles, which provides a robust suspension and also has the benefit of requiring minimal alteration of the existing vehicle frames but is less optimal with regard to ride and handling. More sophisticated, individual wheel suspensions can offer possibility to tailor the suspension kinematics and also reduce the unsuspended mass, thereby improving handling and ride quality. One drawback is that such suspensions are likely to require redesign of the vehicle frame in order to accommodate attachments for linkages and control arms. This may be hard to accomplish due to packing restrictions and it is likely that detailed loader suspension design will always include a number of compromises. By using multibody simulations, suspension designs can be analysed and optimised with high precision, if vehicle and tyre models are well validated.

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