ANALYSIS OF A TRUCK SUSPENSION DATABASE

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ABSTRACT

Experimental data from 61 heavy vehicle suspensions are analysed in this paper to investigate the trends in performance of contemporary suspension designs. Two suspension databases and a mathematical vehicle model are used to calculate the ride and roll performance of heavy vehicles, based on the measured suspension parameters. The suspensions are divided into eight categories according to their spring and axle types. The contribution of suspension springs to rollover stability is evaluated, and the conflict between vehicle ride and roll stability is analysed. Methods for resolving this conflict are discussed from the viewpoint of systematic design principles. A comparison of measured total roll stiffness and roll centre height with rollover threshold is conducted to search for the possible patterns of performance.

Keywords: suspension design, ride comfort, dynamic tyre forces, rollover stability

1 INTRODUCTION

In the UK, 80% of goods are transported by road [Anon, 1992 #225], mainly by heavy vehicles. In terms of freight distance travelled (tonne-km), road transport accounts for 66% of all transportation modes, including air, sea, rail, etc. Heavy vehicles are important component of the national economy, but are also responsible for a significant part of the damage to roads and bridges [Cebon, 1999 #264]. Suspension design is a key factor in the dynamic tyre forces which are thought to contribute to this damage [Mitchell, 1991 #132; Cole, 1996 #129; Woodrooffe, 1996 #162; Cebon, 1999 #264], as well as ride comfort [Woodrooffe, 1996 #162], roll stability [Winkler, 2000 #266], handling [Anon., 1989 #206], and braking [Hardy, 1993 #2].

There is a fundamental trade-off between the ride vibration and roll performance of heavy vehicle suspensions. In general, in order to increase roll stability it is necessary to increase vertical stiffness [Cole, 1996 #129]. The use of anti-roll bars alleviates this trade-off to some extent, since they can allow the suspension to be soft in the vertical direction, but stiff in roll [Cole, 1996 #129].

Key factors that influence roll stability are the height of the centre of gravity (CG) and track width; the compliance (and any back-lash) of the suspensions and tyres; movement of the payload (e.g. sloshing of fluid in a tanker); and the torsional flexibility of the vehicle frame (see review by Winkler [Winkler, 2000 #266]).

Ride and dynamic tyre forces (road damage) are strongly influenced by the stiffness and damping of the suspension [Cole, 1996 #129],. Softer suspensions generally have lower 'natural frequencies' and generate lower dynamic tyre forces than stiffer suspensions [Cebon, 1999 #264; Gyenes, 1992 #267] .

In this paper, measured suspension data from the Transportation Research Institute of the University of Michigan (UMTRI) are analysed [Fancher, 1980 #210; Fancher, 1986 #231; Winkler, 1992 #215]. Analytical models and assumptions are employed to fill the gaps in the data. Analyses presented here aim to elucidate trends in the stiffness characteristics, natural frequencies and roll performance achieved by current suspensions.

2 THE VEHICLE MODEL

Rollover is a specific problem in heavy vehicle operations due to their large payload and high centre of gravity (CG). The problem can occur while the vehicle is in steady turning or a transient handling manoeuvre, such as a lane change. The main parameters involved are the CG height, payload shifting (e.g. liquid sloshing in tanks), suspension and tyre compliance. The propensity to rollover during steady turning can be characterised by the static rollover threshold. This is the steady lateral acceleration which causes rollover of the vehicle [Fancher, 1985 #135].

A simple static roll plane model was used to predict the rollover threshold of vehicles from measured suspension parameters. As illustrated in Figure 1, the model has two degrees of freedom: sprung mass and unsprung mass roll angles (θ_r and θ_t). The effect of spring stiffness on roll motion is combined with auxiliary roll stiffness (K_{raux}) into the total roll stiffness (K_{rtot}) of the suspension according to the following relation:

$$K_{rtot} = K_{raux} + 2 * k_s * p^2 . \quad (1)$$

In this equation, $2 * k_s * p^2$ is the part of roll stiffness which is contributed by vertical stiffness of suspension springs (k_S is the vertical suspension spring stiffness, and $2p$ is the spring centre separation). K_{raux} is the component of the total roll stiffness which is derived from suspension components other than the springs, for instance, the trailing arm systems in air suspensions or anti-roll bars in steel steering axle suspensions, etc. Symbols used in Figure 1 are defined as follows:

- m_s Sprung mass.
- h_s Sprung mass CG height.
- m_u Unsprung mass.
- h_u Unsprung mass CG height.
- h_{rc} Roll centre height.
- K_{rt} Roll stiffness due to the vertical stiffness of tyres

Taking moments about each roll centre in Figure 1, and assuming small angles throughout, two equations of motion can be derived:

$$m_s A_y h_s + m_u A_y h_u + m_s g(h_s - h_{rc})\theta_r = (K_{rt} - m_s g h_{rc} - m_u g h_u)\theta_t \quad (2)$$

$$m_s(h_s - h_{rc})A_y + (m_s g(h_s - h_{rc}) + K_{rtot})\theta_r = -K_{rtot}\theta_t \quad (3)$$

Where θ_r and θ_t are the roll angles of the sprung and unsprung masses. Arranging these into a matrix form and using rollover threshold A_y and sprung mass roll angle θ_r as the state variables gives:

$$\begin{bmatrix} m_s h_s + m_u h_u & m_s g(h_s - h_{rc}) \\ m_s(h_s - h_{rc}) & m_s g(h_s - h_{rc}) - K_{rtot} \end{bmatrix} \begin{bmatrix} A_y \\ \theta_r \end{bmatrix} = \begin{bmatrix} (K_{rt} - m_s g h_{rc} - m_u g h_u)\theta_t \\ -K_{rtot}\theta_t \end{bmatrix} \tag{4}$$

Inverting the left hand side to eliminate the sprung mass roll angle θ_r, from (4) gives:

$$\frac{A_y}{g} = \frac{m_s g(h_s - h_{rc})(K_{rt} - m_s g h_{rc} - m_u g h_u) + K_{rtot}(m_s g h_s + m_u g h_u - K_{rt})}{m_s g(h_s - h_{rc})(m_s g h_{rc} + m_u g h_u) - K_{rtot}(m_s g h_s + m_u g h_u)} \theta_t \tag{5}$$

The rollover threshold is defined as the lateral acceleration when one wheel lifts off the ground. In this condition, the unsprung mass roll angle is:

$$\theta_t = \frac{(m_s + m_u)g}{2K_t t}, \tag{6}$$

where $2t$ is the track width, and K_t is the tyre stiffness. (Note that $K_{rt} = 2K_t t^2$.) Substituting (6) into (5) gives the rollover threshold.

Note that this simplified model was chosen specifically for comparing the measured suspension data. It is not dynamically similar to any particular heavy goods vehicle, because: (i) it only has one axle, and therefore does not account the effects of multiple axles, including roll/torsion and bounce/pitch coupling; (ii) its inertia is all concentrated above the suspension. Nevertheless the model is representative of the broad class of heavy vehicles to which the measured suspensions are normally attached.

3 SOURCES OF SUSPENSION DATA

3.1 UMTRI test rig and experiments

Winkler et al at UMTRI built a test rig in 1979 to measure parameters of heavy vehicle suspension systems [Winkler, 1980 #254]. Over the years, they have collected experimental data for many commercially-available suspensions. The test rig is capable of testing both single-axle and tandem-axle suspensions while they are mounted on the vehicle frame. The measurements are made in the steady state or quasi-steady state.

The rig can be used to measure nine physical parameters for single-axle suspensions, and various other specific parameters for tandem and steering axle suspensions. In this project, only those parameters which are related to the ride and roll performance of heavy vehicles are analysed. The parameters investigated here are:

(i) Roll stiffness: (total roll stiffness and auxiliary roll stiffness);

(ii) Roll centre height;

(iii) Vertical spring stiffness.

Two databases of parameters measured on the UMTRI test rig were analysed. The first is from [Winkler, 1992 #215]. It comprises 94 suspensions including air, steel, walking-beam, torsion-bar and coil springs. The suspensions are from steering, drive and trailer axle groups. Suspension parameters available in [Winkler, 1992 #215] are total roll stiffness, auxiliary roll stiffness and roll centre height. The second databases was obtained directly from UMTRI [Winkler, 1997 #265]. It consists of 111 suspensions, for which the vertical force-deflection characteristics are documented. Although based on the same test facility, these two databases only have 54 suspensions in common.

3.2 The baseline vehicle

In order to use the static roll plane model to simulate vehicle performance, values of parameters which are not provided in the suspension databases were estimated. A typical articulated heavy vehicle was chosen as the common basis for comparing suspensions. This vehicle was a tractor / semi-trailer with four axles. Dual radial ply tyres were assumed to be installed on the drive axle and trailer axles. For the steering axle, single radial ply tyres were assumed.

Details of the dimensions of the baseline vehicle are provided in [Lin, 1994 #234; Fu, 1998 #260]. Table 1 provides some physical properties of the baseline vehicle. The sprung mass CG height of trailer axle group was taken to be the same as the overall CG height of the trailer. For steering and drive axles, the CG heights were derived from the CG height of tractor and trailer sprung masses, using weighting factors which depended on the static load carried by each axle. The unsprung mass CG height and half wheel track were set to $0.53m$ and $0.93m$, respectively, for all three axle groups.

3.3 Selection of the working database

A selection procedure was employed to construct a complete suspension database for analysis in this project. It was necessary to use various assumptions, mainly based on the static roll plane model and physical parameters of the baseline vehicle, to estimate missing items in the original databases. The aim of this procedure was to make the derived working database as complete as possible. It was also necessary to account for the fact that the static loads used in the suspension tests do not always match the axle loads of the baseline vehicle. The selection procedure included:

(i) Removal of suspensions which lacked information on total roll stiffness or roll centre height;

(ii) Elimination of suspensions with more than 25% difference between their test loads and standard axle loads;

(iii) For the suspensions without known spring vertical stiffness, the distance between spring centres was assumed to be equal to 980 mm. The vertical stiffness was then calculated from the roll stiffnesses and the spring centre distance using Equation 1.

The results of the selection procedure are presented in Figure 2. It contains 61 suspensions in eight categories. They constitute the working database, used for further investigations.

In the working database, 75% of the suspensions have steel leaf springs. These include three different axle types (steer, drive and trailer group). 16% of the suspensions in the database have air springs. These exist in drive and trailer axle groups. There are also a few tandem drive axle walking-beam and torsion-bar suspensions.

The origins of the suspensions and markets in which they are used are shown in Figure 3. The majority (77%) of suspensions analysed in this paper are used in the North American commercial market.

Typical contents of the working database are depicted in Tables 2 and 3, for 19 of the 61 suspensions in the working database. Table 2 shows the original experimental data from the UMTRI databases. It includes descriptive information which gives the types of axle and spring for the suspensions, their origins and applications, etc. It also includes quantitative information: the rated load, total and auxiliary suspension roll stiffness, vertical spring stiffness and suspension roll centre height.

Table 3 shows measured and calculated data corresponding to the suspensions in Table 2. In this table, the rollover thresholds (A_y/g) were calculated using equations 5 and 6, with parameters for the baseline vehicle.

The natural frequency (f_n, in the unit of Hz) shown in Table 3 was calculated from

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_e}{m_{testing}}} \tag{7}$$

where, k_e is the effective vertical stiffness for the combination of spring and tyres (N/m) and $m_{testing}$ is the equivalent mass (kg) derived from the suspension test load.

4 ANALYSIS OF THE SUSPENSION DATABASE

Various analyses were performed on the database. Parametric analyses were performed to explore relationships between various suspension stiffnesses, in order to elucidate their contributions to the total roll stiffness. The conflict between vehicle ride and roll performance was also investigated from the suspension design perspective. This conflict is demonstrated by the interrelationship between natural frequency and rollover threshold. The third type of analysis examined the joint effect of total roll stiffness and roll centre height on rollover threshold, in order to understand trends in suspension design.

4.1 Parametric analysis of total roll stiffness

There are two basic elements that constitute the total roll stiffness of a suspension system: the springs and the auxiliary roll stiffening effect from the other suspension components, including the anti-roll bars. Their relation is defined as Equation 1.

Steel suspensions obtain their auxiliary roll stiffening effect mainly from the twist of spring leaves about a longitudinal axis and the anti-roll bar, if there is one. On the other hand, the auxiliary roll stiffness of trailing arm air suspensions comes from the rigid axle and trailing arm assemblies which act together like an anti-roll bar. This means that trailing–arm air suspensions, as used in semi-trailers, are normally quite stiff in roll (see [Anon., 1989 #206] for details).

Figure 4 shows that there is a clear dependence of total roll stiffness on auxiliary roll stiffness for all suspensions, with an overall ‘coefficient of determination’ about 0.66 (This coefficient is a better measure of dependence than the regression slope or correlation coefficient, because it takes account of both effects [Gujarati, 1978 #232].). Much stronger positive dependence can be seen in air suspensions. Their coefficient of determination reaches a level as high as 0.91. On the other hand, the dependence of steel suspensions is much lower and the coefficient is equal to 0.33. This is probably due to wide variations in the number of steel leaves and the geometry of leaf spring components in steel suspensions. By contrast, the component designs of air suspensions are more uniform, which makes them more correlated in roll stiffness. (The exception is an independent air suspension, which has a particularly low roll stiffness: the lowest point in the trailer air suspensions in Figure 4).

Steel trailer suspensions have similar levels of total roll stiffness. This is because trailer suspensions are normally required to support heavy payloads with high centres of gravity. In order to have satisfactory roll safety on the road, trailer suspensions are designed to be very stiff in roll. On the other hand, the load and CG height for steering axle suspensions and the CG height for drive-axle suspensions are relatively low when compared with trailer suspensions. The demand for their total roll stiffness is correspondingly less. There is one exceptional drive air suspension, which has a very high total roll stiffness. It is a European tandem axle design with a similar layout to typical trailer suspensions.

A solid line of equal total roll stiffness and auxiliary roll stiffness is drawn in Figure 4. As expected, for all suspensions, the total roll stiffness is greater than the auxiliary stiffness alone. Trailer air suspensions and steel steering axle suspensions are all closely distributed and near to the line. Trailer and drive axle steel suspensions are scattered away from the line of equality. This demonstrates that auxiliary roll stiffness contributes most to total roll stiffness in trailer air and steel steering axle suspensions, but does not make much contribution to the total roll stiffness of trailer and drive axle steel suspensions. This point can be emphasised by calculating the percentage contribution of auxiliary roll stiffness in the total roll stiffness, as shown in Figure 5.

In Figure 5, air suspensions have the highest percentage of auxiliary roll stiffness among all axle groups (mostly above 70%). This is expected, since air suspensions are designed to be soft in the vertical direction, but the trailing arm suspensions have very high auxiliary roll stiffness and hence high total roll stiffness. For drive axle air suspensions, it is not desirable to twist the axle and differential too much, so the auxiliary roll stiffness is often provided by an anti-roll bar. This generally keeps the auxiliary roll stiffness lower than that of trailer air suspensions. By contrast, auxiliary roll stiffness has a wide range of importance in the total roll stiffness of steel suspensions, from as low as 10% in drive axles up to 70% in steering axles. This is again due to wide variations in steel spring designs and different design considerations for the different axle types.

Less correlation is found between total roll stiffness and vertical stiffness, as shown in Figure 6. Except for steering axle suspensions, steel suspensions tend to have higher vertical spring stiffness than air suspensions. Again, if all suspensions are separated into steel and air groups, positive dependence can be found in steel suspensions with a coefficient of determination being equal to 0.87 and negative dependence in air suspensions with a coefficient of 0.20. This trend, together with the trend in Figure 4, confirms that vertical stiffness plays an important role in total roll stiffness of steel suspensions, whereas auxiliary roll stiffness plays the most important role in that of air suspensions.

Steel steering axle suspensions obtain their auxiliary roll stiffness by mounting the axle off the centre of the steel leaf span, and by twisting the leaf springs about the longitudinal axis. Since they have lower static load and more design emphasis on ride comfort, their vertical spring stiffness tends to be significantly lower than that of drive suspensions. Both reasons make the percentage of auxiliary roll stiffness in the total roll stiffness of steering axle suspensions higher than that for other axle types of steel suspensions. Therefore, as can be seen in Figures 4 and 6, the stiffness characteristics of steel steering axle suspensions are more similar to that of air suspensions than the other steel suspensions.

According to [Anon., 1989 #206], steel suspensions normally do not have additional roll stiffening mechanisms deliberately incorporated in their designs. This is why they generally have a low proportion of auxiliary roll stiffness. However, the anti-roll bar effect of rigid axle and trailing arm assemblies provides very stiff roll resistance for air suspensions. Their auxiliary roll stiffness is not only high in proportion to total roll stiffness (because their vertical stiffness is normally low) but also high in absolute terms.

4.2 Combined analysis of vehicle ride and roll performance

In the following analysis, rollover threshold and natural frequency, are used as indices to represent vehicle roll and ride performance. High values of rollover threshold and low values of natural frequency generally mean better performance. Both indices were calculated from the measured data in the working database.

The relationship between rollover threshold and natural frequency is depicted in Figure 7. The best performance is towards the top left of the chart, where rollover threshold is high and natural frequency is low. Trailer and drive axle suspensions are divided into two distinct groups by their suspension types. The air suspensions have low natural frequencies and the steel suspensions have high natural frequencies. This is due to the significant difference in vertical stiffness of these two types of suspensions as described in previous section. The rollover thresholds of the trailer axle suspensions are nearly constant, independent of natural frequency. Conversely, the rollover threshold of steel-suspended drive axle suspension increases with natural frequency. This is expected since a large proportion of their roll stiffness comes from the springs and little from auxiliary effects (Figure 5).

Steering axle suspensions also show insensitivity of rollover threshold to natural frequency, but they are more scattered than for the other axle types. The design of steering axle suspensions is governed by the need to provide appropriate steering geometry through a wide range of deflections, as well as good ride comfort for the driver; rather than providing roll stability for the vehicle. The high rollover thresholds of the steering axle suspensions

are essentially the result of their low payload and low effective CG height. Essentially the steering axle suspensions do not 'pull their weight' - they are generally capable of providing significantly more stabilising moment than they are called upon to provide in normal operation.

Calculated natural frequencies of steering axle suspensions are between 1.0 and 1.5 Hz. For drive and trailer steel suspensions, the natural frequencies are 1.5 to 2.5 Hz, with the frequencies of trailer suspensions being higher than those of the drive suspensions. For drive axle and trailer air suspensions, they are around 1.2 to 1.5 Hz. The calculated rollover thresholds of the drive axle and trailer suspensions generally vary from 0.4 to 0.5g. This would appear to indicate that suspension designers aim for a specific level of rollover performance.

Steel suspensions, which derive most of their roll stiffness from the vertical stiffness of the spring elements (figure 5), generally have a high dependence of rollover threshold on natural frequency. If the vertical stiffness of a steel suspension is reduced in order to improve its ride performance, there will be an adverse effect on roll stability. This makes it difficult for steel suspensions to achieve both good ride comfort and high roll stability at the same time.

Conversely, air suspensions, whose rollover thresholds have little dependence on the vertical ride natural frequency, possess more flexibility for improving both roll and ride performance simultaneously. This is because the function of providing roll stiffness is mostly taken away from the air springs by other auxiliary mechanisms (i.e. the assembly of the axle and trailing arms, etc.). The function of the air spring is mainly provision of vibration isolation, and designers are able to improve ride and roll performance simultaneously. Adding anti-roll bars to leaf spring suspensions has a similar effect of relieving the need of the springs to provide roll stiffness. This separation of functions (ride isolation and roll stiffness) is considered to be good design practice [Pahl, 1996 #6].

In addition to suspension parameters, rollover threshold is also significantly affected by effective payload CG height. It is calculated that a 10% change in effective load height will give 12% to 14% change in rollover threshold of trailer suspensions [Fu, 1998 #260].

4.3 Combined analysis of suspension parameters and rollover threshold

Roll centre height and total roll stiffness are the most important suspension parameters affecting rollover threshold [Winkler, 2000 #266]. They are both available as measured quantities in the working database. In this section, the relationship between these two parameters and the calculated rollover threshold of the suspensions is investigated.

Figures 8 and 9 show the roll centre height plotted against total roll stiffness for the suspensions in the database, together with contours of constant rollover threshold A_y/g, plotted using equations 5 and 6. Figure 8 shows drive and trailer suspensions, while figure 9 shows steering axle suspensions. The values of rollover threshold are indicated by numbers plotted on the contours. The contours span the range of from zero to an upper limit $A_y / g\big|_{\max}$, which was calculated from Equations 5 and 6, assuming infinite suspension roll stiffness $K_{rtot} \to \infty$. This gives:

$$\left.\frac{A_y}{g}\right|_{\max} = \left[\frac{K_{rt}}{m_s g h_s + m_u g h_u} - 1\right]\frac{(m_s + m_u)g}{2K_t t} \tag{8}$$

This limit is the highest rollover threshold that suspensions of a specific axle type can achieve. For the drive axles and trailer axles in figure 8, using the parameters of the baseline vehicle, the maximum rollover threshold is $A_y / g\big|_{\max} = 0.5574$. For the steering axle suspensions shown in figure 9, the limit is $A_y / g\big|_{\max} = 0.8009$.

Note that Equation 8 does not contain the roll stiffness K_{rtot} or roll centre height h_{rc}. So the limits can be determined simply from the relevant geometry, masses and tyre stiffnesses.

In order to make a comparison between drive and trailer suspensions, the same 'baseline' axle loads and CG heights were assumed in the calculations of the rollover threshold for all suspensions.

There is no strong relationship between roll centre height and total roll stiffness for the various suspensions in figure 8. Some correlation exists between these parameters for steel suspensions, because the roll centre heights are determined by the geometry of leaf springs. Stiffer multi-leaf steel suspensions normally require more spring leaves, which pushes the roll centre to a higher location. On the other hand, because the roll centres of air suspensions are determined by the geometric configurations of components which are not normally influenced by the stiffness of the air bags, the location of roll centre height and the suspension roll stiffness are independent. One particular trailer air suspension has the lowest roll centre height, at about 100 mm. It is an independent air suspension for which the roll mechanism is different from the other trailer air suspensions of the rigid axle type.

The results for steering axle suspensions are plotted separately in Figure 9 because their loading conditions are quite different from those of the other two axle types. Their roll centre heights are essentially constant, and independent of roll stiffness. This is because their values of vertical spring stiffness are very similar to one another, as demonstrated in Figure 6. All of these suspensions have similar leaf spring configurations, which keep their roll centre locations at approximately the same level. The constant roll centre height is also a consequence of the requirement to minimise roll-steer effects generated by the steering linkage. As shown in this figure, the roll centre heights are all about 450 mm. Data points for the tandem walking-beam suspensions lie outside the range of data for steel suspensions. The walking-beam steering axle suspensions have an entirely different duty from other suspensions (particularly for military usage), with a different steering mechanism, and a different mechanism defining the position the roll centre. This leads to their significantly different performance, as shown in figure 9.

5 CONCLUSIONS

(i) Analyses in this paper have explored the characteristics of various existing suspensions. Both experimental data and analytical methods were applied to generate a complete suspension database. Levels of theoretical performance were investigated for different suspension designs.

(ii) The vertical stiffness of an air spring does not contribute much to its total roll stiffness in a typical air suspension system. Therefore, it has little effect on the rollover threshold. Steel leaf spring suspensions have more influence on the rollover performance.

(iii) One of the design trends of a suspension system is to separate the functions of ride and roll performance into different components. In particular, the duty of spring elements in providing roll stiffness can be released and the conflict in achieving both good ride comfort and rollover performance can the be partially resolved. Typical examples are the introduction of anti-roll bars in steel suspensions and the auxiliary roll stiffening mechanisms in air suspensions.

(iv) The design of suspensions for steering axles is governed by factors other than rollover performance, so the rollover performance normally exceeds practical operation requirements.

(v) Designers of drive axle and trailer axle suspensions appear to aim for a rollover threshold of 0.4 to 0.5g, and achieve this with a wide range of design parameter variants.

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7 REFERENCES

8 TABLES

Item	Steering axle	Drive axle	Trailer axles
Sprung mass (kg) (axle loading)	6062	9179	8750
Sprung mass CG height (m)	1.17	1.68	1.79
Unsprung mass per axle (kg)	706	1107	792
Vertical stiffness of tyre (N/m)	8.76×10^5	1.75×10^6	1.75×10^6

Table 1 Properties of the baseline vehicle

Axle Type	Axle no.	Susp. Type	Spring Type	Type of Leaf/Clamp	origin	Rated L. kN	Tot. R. Stiff. kN-m/deg	Aux. R. Stiff. kN-m/deg	Vert. Stiff. N/m	R C H mm
drive	single		steel	flat (9)	NA	71.2	10.73	1.16	8.06E+05	780
drive	single		steel	flat (14)	NA	93.4	9.69	2.26	1.15E+06	719
drive	tandem	4-spring	steel	flat (7)	NA	80.05	6.17	1.13	8.18E+05	677
drive	tandem	2-spring	steel	flat (9)	Eur	77.85	9.19	1.24	9.57E+05	657
drive	tandem	2-spring	steel	taper (2)	NA	75	10.8	0.8	1.19E+06	863
drive	tandem	2-spring	steel	taper (3)	Eur	77.85	7.22	1.36	6.82E+05	334
drive	tandem		walking-beam	flat	NA, M	71.15	5.12	2.03	3.69E+05	930
drive	tandem		torsion-bar	torsion-bar	NA	71.15	8.11	0.79	8.73E+05	638
steer	single		steel	flat (6)	Eur	62.3	2.52	1.24	2.13E+05	515
steer	single		steel	flat (9)	NA	53.4	2.77	1.47	2.45E+05	473
steer	single		steel	flat (10)	NA	64.9	2.88	1.07	3.04E+05	473
steer	tandem		walking-beam	flat	NA, M	62.3	4.87	1.69	3.79E+05	927
trailer	single	trailing-arm	air	overslung	NA	80.1	12.32	10.73	4.06E+05	557
trailer	single	trailing-arm	air (iso)	overslung	NA	89	17.74	13.56	3.62E+05	571
trailer	single	trailing-arm	air	ind	Eur	89	14.24	10.17	4.86E+05	100
trailer	tandem	trailing-arm	air	underslung	NA	71.15	8.93	6.08	3.40E+05	615
trailer	tandem	4-spring	steel	flat	NA	71.15	13.56	1.13	1.41E+06	795
trailer	tandem	4-spring	steel	flat	Eur	84.5	16.16	2.26	1.47E+06	736
trailer	tandem	4-spring	steel	taper (3)	NA	71.15	11.72	2.95	1.05E+06	592

Table 2 Suspension experimental data from [Winkler, 1992 #215; Winkler, 1997 #265]

Rated L. : Rated load of the suspension. It is also the testing load applied in the measurements.

Tot. R. Stiff. : Total roll stiffness of the suspension (K_{rtot}).

Aux. R. Stiff. : Auxiliary roll stiffness of the suspension (K_{raux}).

Ver. Stiff. : Vertical stiffness of the spring (k_s).

R C H : Roll centre height of the suspension (h_{rc}).

Axle Type	Axle no.	Susp. Type	Spring Type	ms Kg	hs m	mu Kg	hu m	kt N/m	t m	p mm	fn Hz	Ay/g g
drive	single		steel	9179	1.68	1107	0.53	1.75E+06	0.93	583.3	1.963	0.524
drive	single		steel	9179	1.68	1107	0.53	1.75E+06	0.93	430.7	1.921	0.512
drive	tandem	4-spring	steel	9179	1.68	1107	0.53	1.75E+06	0.93	420.0	1.861	0.469
drive	tandem	2-spring	steel	9179	1.68	1107	0.53	1.75E+06	0.93	487.9	1.988	0.501
drive	tandem	2-spring	steel	9179	1.68	1107	0.53	1.75E+06	0.93	490.0	2.169	0.532
drive	tandem	2-spring	steel	9179	1.68	1107	0.53	1.75E+06	0.93	496.3	1.770	0.423
drive	tandem		walking-beam	9179	1.68	1107	0.53	1.75E+06	0.93	490.0	1.459	0.499
drive	tandem		torsion-bar	9179	1.68	1107	0.53	1.75E+06	0.93	490.0	2.018	0.489
steer	single		steel	6062	1.17	706	0.53	8.76E+05	0.93	415.4	1.168	0.662
steer	single		steel	6062	1.17	706	0.53	8.76E+05	0.93	390.1	1.334	0.660
steer	single		steel	6062	1.17	706	0.53	8.76E+05	0.93	412.9	1.315	0.666
steer	tandem		walking-beam	6062	1.17	706	0.53	8.76E+05	0.93	490.0	1.453	0.792
trailer	single	trailing-arm	air	8750	1.79	792	0.53	1.75E+06	0.93	334.9	1.430	0.465
trailer	single	trailing-arm	air (iso)	8750	1.79	792	0.53	1.75E+06	0.93	575.5	1.294	0.484
trailer	single	trailing-arm	air	8750	1.79	792	0.53	1.75E+06	0.93	490.0	1.457	0.432
trailer	tandem	trailing-arm	air	8750	1.79	792	0.53	1.75E+06	0.93	490.0	1.410	0.448
trailer	tandem	4-spring	steel	8750	1.79	792	0.53	1.75E+06	0.93	501.9	2.338	0.488
trailer	tandem	4-spring	steel	8750	1.79	792	0.53	1.75E+06	0.93	519.8	2.170	0.491
trailer	tandem	4-spring	steel	8750	1.79	792	0.53	1.75E+06	0.93	490.0	2.139	0.465

Table 3 Samples of calculated data for the suspensions listed in Table 2. See Figure 1 for definition of the variables. Additional parameters are: k_t - tyre stiffness, and t - half wheel track.

9 FIGURES

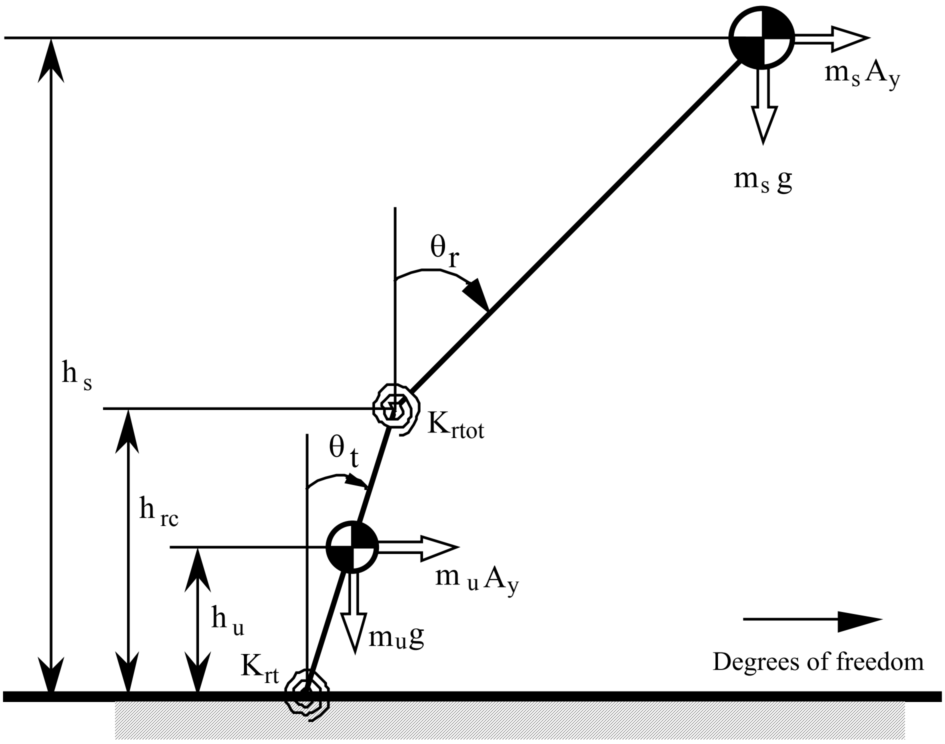

Figure 1 The static roll plane model

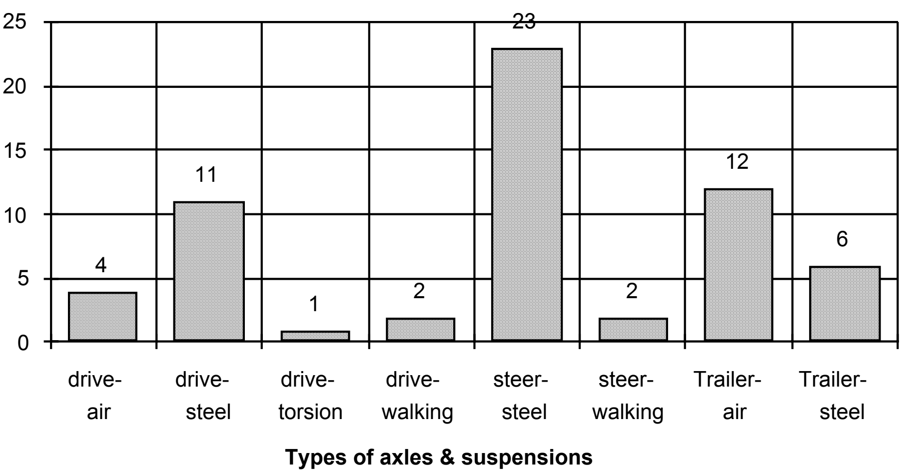

Figure 2 Population of the working database

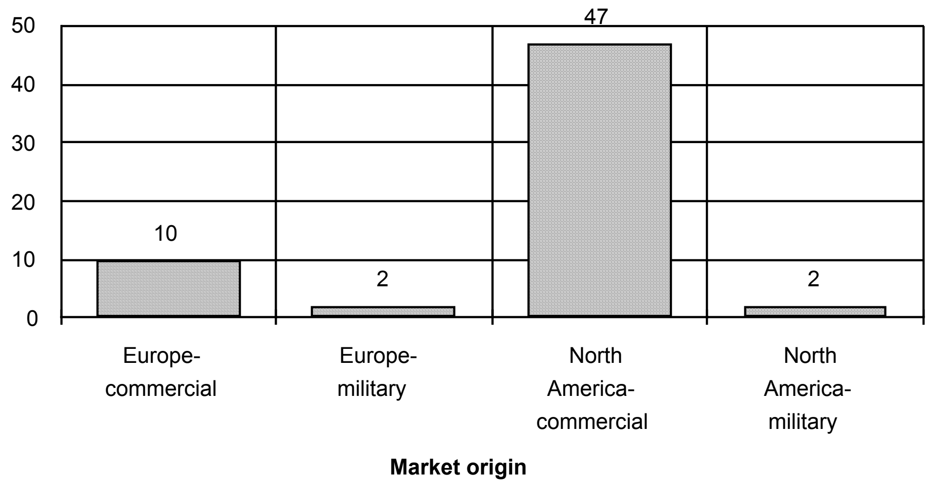

Figure 3 Market origin of the working database

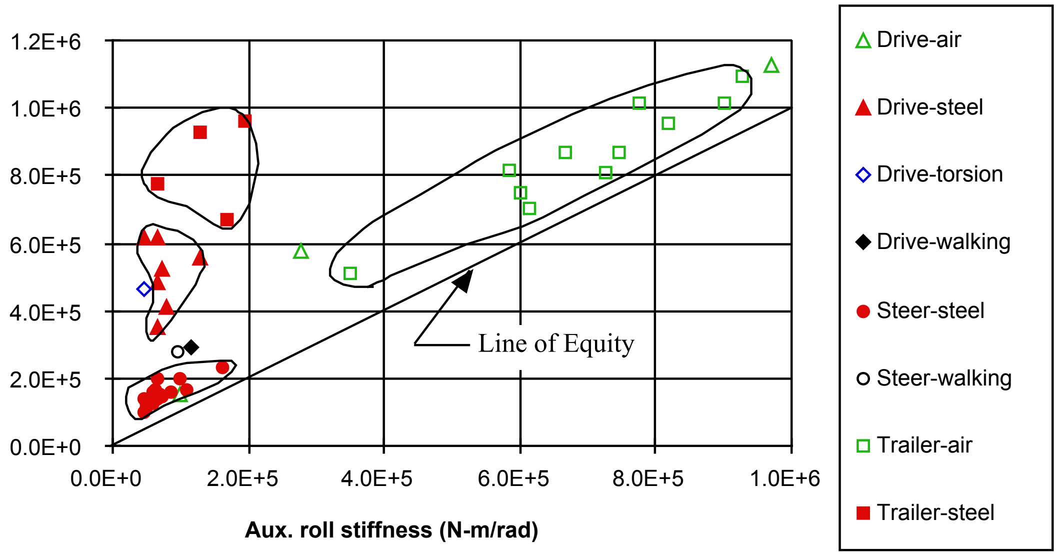

Figure 4 Relationship between total and auxiliary roll stiffness

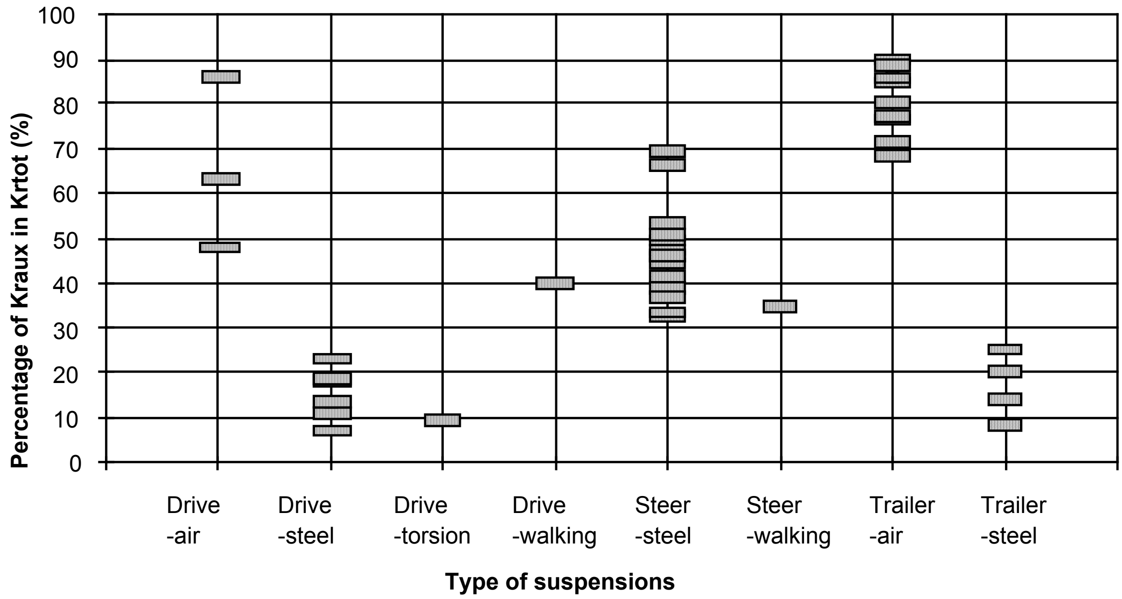

Figure 5 The percentage of auxiliary roll stiffness in total roll stiffness

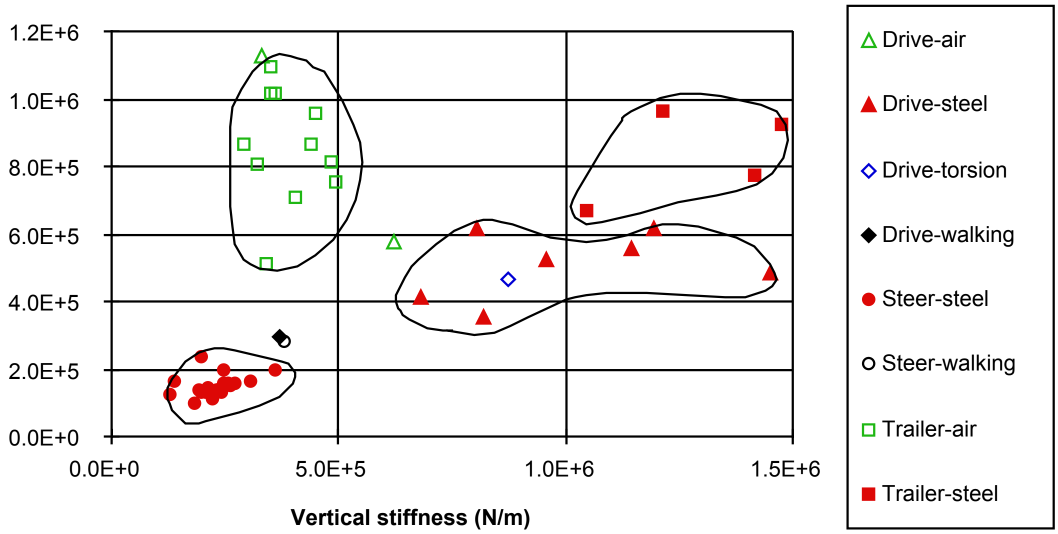

Figure 6 Relationship between total roll stiffness and vertical stiffness

Figure 7 The relationship between rollover threshold and natural frequency

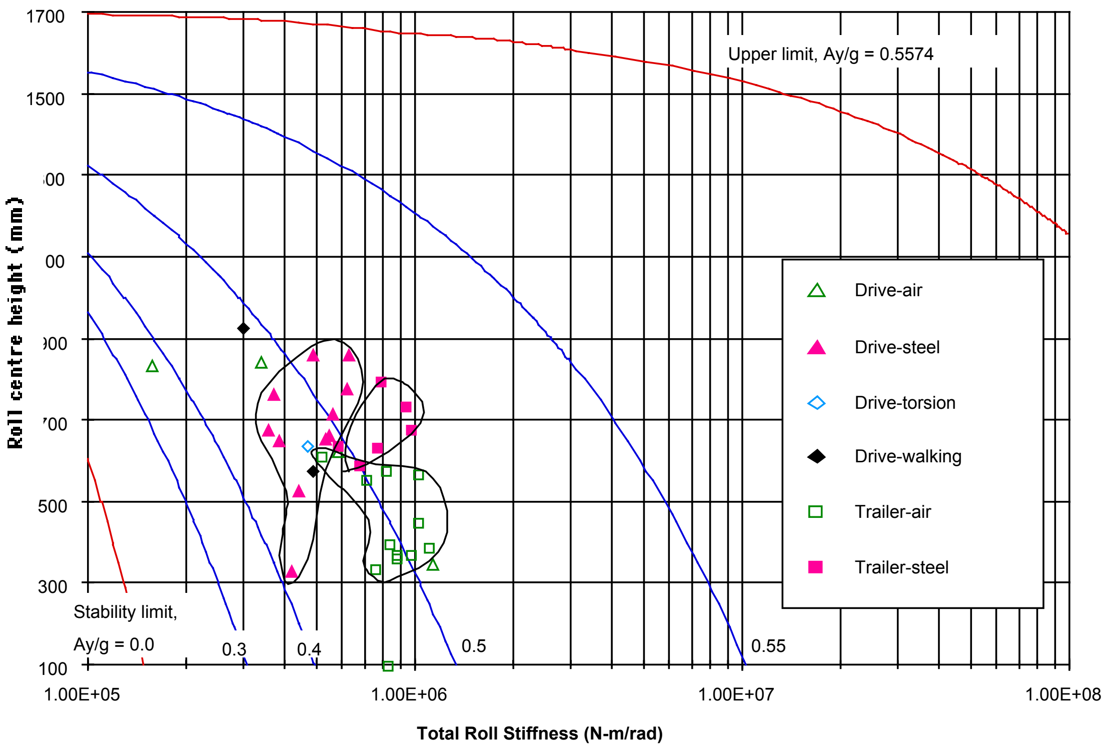

Figure 8 Combined analysis for drive and trailer axle suspensions

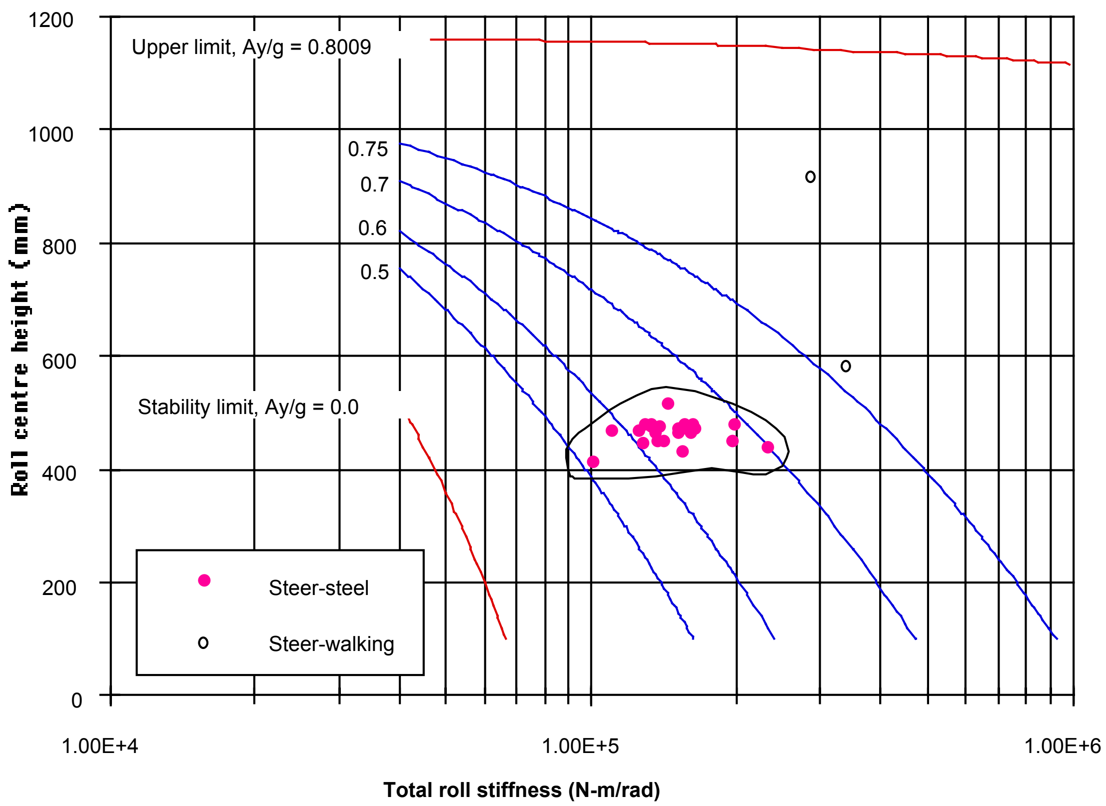

Figure 9 Combined analysis for steering axle suspensions