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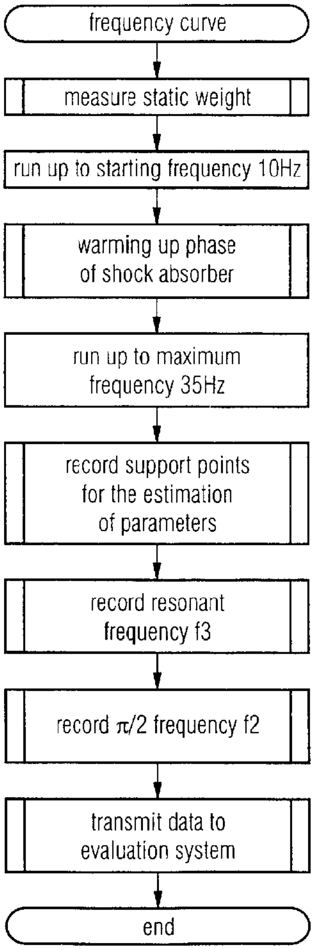
(54) METHODE ET APPAREIL POUR L'EXAMEN D'AMORTISSEURS MONTES (II)

(54) A METHOD AND AN APPARATUS FOR THE EXAMINATION OF INSTALLED SHOCK ABSORBERS (II)



(57) In the method and the apparatus for the examination of shock absorbers installed on a vehicle, a vibratory platform, on which one wheel of the vehicle is stood and which may be reciprocated with a suitable amplitude and variable frequency in a vertical, is employed to subject the wheel to an oscillation. The damping of the shock absorber is checked from the force response of the running gear to the oscillations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the motion of the vibratory platform. The data signal,

which represents the variations with time of the force response at a target frequency, is split by a tension-compression stage means into a compression stage signal, which is characteristic for the compression stage of the shock absorber, and a tension stage signal, which is characteristic for the tension stage of the shock absorber. The compression stage signal and the tension stage signal are separately supplied to further preparing means. As a basis of assessment for the quality of the installed shock absorber the degree of axle damping as related to the quotient of the spring-suspended mass to the non-spring-suspended mass is calculated. The measured values of the degree of axle damping are related to a characteristic line of a critical or border value degree of axle damping, which delimits the range of non-acceptable degrees of axle damping.

ABSTRACT

In the method and the apparatus for the examination of shock absorbers installed on a vehicle, a vibratory platform, on which one wheel of the vehicle is stood and which may be reciprocated with a suitable amplitude and variable frequency in a vertical, is employed to subject the wheel to an oscillation. The damping of the shock absorber is checked from the force response of the running gear to the oscillations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the motion of the vibratory platform. The data signal, which represents the variations with time of the force response at a target frequency, is split by a tension-compression stage means into a compression stage signal, which is characteristic for the compression stage of the shock absorber, and a tension stage signal, which is characteristic for the tension stage of the shock absorber. The compression stage signal and the tension stage signal are separately supplied to further preparing means. As a basis of assessment for the quality of the installed shock absorber the degree of axle damping as related to the quotient of the spring-suspended mass to the non-spring-suspended mass is calculated. The measured values of the degree of axle damping are related to a characteristic line of a critical or border value degree of axle damping, which delimits the range of non-acceptable degrees of axle damping.

FIELD OF THE INVENTION

The present invention relates to a method and an apparatus for the examination of shock absorbers installed on vehicles.

BACKGROUND OF THE INVENTION

Apparatus for the measurement of the efficiency of running gear of a motor vehicle without having to remove the shock absorber from the vehicle, have already been proposed. There are various different methods for this purpose, as for example using a vibratory platform, on which one wheel of the motor vehicle is stood and which is reciprocated with a suitable amplitude and a variable frequency in a vertical direction in order to subject the wheel to oscillations, the reading for the force exerted by the running gear on the vibratory platform being evaluated.

The Method of the European Shock Absorber Manufacturers' Association (EUSAMA), the largest organization of its kind, involves a uniform testing method for shock absorbers in the installed state. The examination is in this case performed using a vibratory platform, which with the aid of an eccentric drive is caused to oscillate with a stroke of exactly 6 mm. During examination the measuring setup is excited to run at approximately 23 Hz and freely vibrates down to 0 Hz. The respective sinusoidal dynamic wheel load is measured by sensors and stored. prior to producing the vibrations the static wheel load F_s is measured. The stored, dynamic wheel loads are investigated to find their minimum F_{min}. From the data the so-called relative road adherence A is calculated as a percentage: $A = F_s/ F_{min}/F_s$.

In a real application the road adherence value is corrected, since particularly light vehicles have only poor road adherence figures. For evaluation the following table is employed:

100% - 60%	good
59% - 40%	fair
39% - 20%	poor
19% - 0%	insufficient.

The EUSAMA Method is widely employed but the data provided by it depend not only on the condition of the shock absorber but also on design features, which are characteristics of the wheel suspension, as for example the ratio between the spring-suspended mass and the non-spring-suspended mass, the type of tires, the tire pressure and the type of suspension and furthermore on the conditions of measurement, as for example the loading of the vehicle or the temperature of the surroundings or the operating temperature of the shock absorber. The dependence on a large number of parameters means that in practice the value defined by the EUSAMA Method is prevented from being representative for the condition of the shock absorber itself. Accordingly the EUSAMA method does not lead to any reproducible and reliable results.

Similar problems have occurred with other known measuring systems for shock absorbers, wherein the dependency of the measurement on a plurality of physical parameters of the running gear means that the measured data must be interpreted using table or the like. This is something leading to high degree of approximation and large ranges of error and for different types of vehicle different tables are necessary, dependent on which shock absorbers are employed.

The European patent publication 0476746 A1 discloses an apparatus for testing running gear of a vehicle, which like the EUSAMA Method operates with a vibratory platform means. A computer processes the data and from such data produces a transfer function for the wheel suspension, which is looked upon as a system with a plurality of system elements. On the basis of a model of the system the computer derives the values of the system elements from the transfer function and with the calculated values of the system elements calculates the values of at least one measurement variable and compares such value with a stored value of the dimensional variables. The computer assesses the wheel suspension from such comparison and displays the result of the comparison. Here as well suitable tables are necessary in a data base, which must hold the values for all current types of vehicles and tires. Furthermore the evaluation of the running gear by way of the individual system variables is inaccurate.

The patent publication WO 96/07882 discloses an apparatus for the examination of shock absorbers of motor vehicles, in the case of which the wheel on a vibratory platform is caused to vibrate and on the basis of the amplitude of vibrations at different frequencies the results of measurement are derived. In the case of this apparatus the resonance of the suspension is sensed and then the frequency of the stroke and accordingly the energy supplied is varied. The damping force of the shock absorber is plotted against speed. In the case of this model some important effects, as for example internal friction, as will be described below, are not taken into account.

The European patent publication 0647843 A2 discloses a system for measuring the damping coefficient of a shock absorber mounted on a motor vehicle, as is recited in the preamble of claim 1 herein. The damping coefficient is calculated as a function of the derivation of the frequency of the relative phase function between the force transmitted to the vibratory platform and the movement of the vibratory platform. In this apparatus an imperfect vibration model is taken as a basis, the value termed the "degrees axle damping path" only applying for the particular model employed so that in practice there are substantial departures between the results of this method and the actual degrees of damping of the shock absorber being examined. In particular the effect of external friction is not taken into account.

Methods in existence so far, in which evaluation takes place on a basis including a vibration model, suffer from deficits in theory. This is for instance an inappropriate dissection of the simple vibration model into different model parts for different characteristic frequency ranges; the inappropriate mathematical description of the resonant frequencies. In part methods on the basis of free undamped elastic vibration systems are utilized. The basis is however forced and heavily damped elastic vibration systems; the invalid simplification or omission of parts of terms of the mathematically described model; leaving out external friction within the simple one quarter vehicle model and infringing against the linear force law by progressive suspension springs. Finally, the distortion of the appropriate sinusoidal signal form by the different compression and tension stage behavior is not taken into account in the known methods, or, in other words, the measurement signals are not brought back or compared to the sine-shaped signals upon evaluation.

One object of the invention is to provide a method and an apparatus for the examination of shock absorbers mounted on vehicles, which from the characteristic curve of the force response using an oscillation model produces reproducible measurement results for the quality of the shock absorber for forced oscillations.

In order to achieve this object the method of the invention for the examination of shock absorbers installed on a vehicle using a vibratory platform, on which one wheel of the vehicle is stood and which is able to be reciprocated in a vertical direction at a suitable amplitude and with a variable frequency, an oscillating action is exerted of the wheel, the damping of the shock absorber being derived from the force response of the running gear to the oscillations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the movement of the vibratory platform, the data signal which represents the force response of the shock absorber being subjected to the evaluation, is characterized in that, as a basis for assessment of the quality of the installed shock absorber, the degree of axle damping as related to the quotient: suspended mass divided by non-suspended mass, is calculated, the measured values of the degree of axle damping being related to a characteristic curve of a critical degree of axle damping, which delimits the range of non-acceptable degrees of axle damping. The assessment of the shock absorber is performed with the classification into one of the following quality categories: "very good", "medium" and "failed", with which a recommendation of replacement of the installed shock absorber is coupled. The assessment takes place in a manner dependent on the position of the degree of axle damping found in relation to the characteristic curve of the critical degree of damping and not in a manner dependent on the individual parameters of the one quarter running gear, i.e. of the suspension of one wheel. Since the data signals have been processed in the above mentioned manner, it is also possible to obtain reproducible, i. e. satisfactory results of measurement from the evaluation of these signals.

SUMMARY OF THE INVENTION

In accordance with a further advantageous development of the method of the invention calculation of parameters is performed, as same is recited in claim 7, and from the parameters the degree of axle damping of ascertained, the parameters utilized here being a result of model iteration. In tests it has been shown that with this one quarter vehicle model satisfactory results may be obtained in the testing and classification of installed shock absorbers. More particularly, the problematical influence of so-called external friction is taken into account by the model iteration.

The advantageous development of the method of the invention as defined in claim 8 means that the amount of computation for the evaluation of the measurements is substantially reduced, since the number of variables to be processed can be reduced to six.

The advantageous development of the method of the invention as defined in claim 14 leads to a still further reduction in computation work without any reduction in the measurements processed.

A further advantageous development of the method of the invention is characterized in that after calculation of the degree of axle damping and the plotting thereof in a graph of degree of axle damping to the suspended/non-suspend mass quotient a recommendation can be made as regards the need for replacement of the shock absorber.

A further advantageous development of the method of the invention is characterized in that the data signal, which represents the changes in the force response with time at a target frequency, is split up in a compression-tension stage splitting means into a compression stage signal, which is characteristic for the compression stage of the shock absorber, and a tension stage signal, which is characteristic for the tension stage of the shock absorber, and in that the compression stage signal and the tension stage signal are separately supplied to further processing means. This means that a reproducible basis is provided for later evaluation of the measurements.

A further advantageous development of the method of the invention is characterized the tension stage-compression stage split is performed on the basis of frequency splitting, the basic frequency being kept and the measured values during the time $\pi/2$ of the positive half wave of the being associated with the compression stage whereas the measured values of the negative half wave during the time $\pi/2$ are associated with the tension stage

A further advantageous development of the method of the invention is characterized in that the tension-compression stage split is performed on the basis of the amplitude split, the higher basic frequency of the compression stage and the lower frequency of the tension stage being employed for the calculation of the amplitude by displacement of the narrower, positive compression stage half wave by d phi to the right, and displacement of the wider, negative tension stage half wave by d phi to the left.

A further advantageous development of the method of the invention is characterized in that the signals obtained by the frequency split are subjected to digital filtering, preferably a Fourier transformation and more particularly a fast Fourier transformation. This means that corruption of information as possible with conventional filtering of the entire signal is avoided and the characteristic features of the respective stage (tension and, respectively, compression stage) are preserved.

Owing to the advantageous form of the method of the invention as claimed in claim 26, the measured data curve is smoothed so that the further processing is facilitated.

In an advantageous form of the method of the invention after such digital filtering a quality assessment of the processed signal is carried out as to the sine-conformity, wherein for each measured value the relative departure from the sine-signal, which represents the ideal filtered value, is calculated, the transverse sum of all departures is found and those measurement points are rejected, whose mean departure from the ideally filtered value exceeds a predetermined value, such predetermined value preferably being 5%. Thereby it can be achieved that two sine-shaped signals are derived which are then feed to the further processing. Only by means of the sine-shaped course of the signals the correctitude of the calculation method with respect to the parameter calculation and to the parameter estimation is guaranteed. Furthermore, the measurement curve is smoothed such that the further processing is facilitated.

According to one of the advantageous developments of the method of the invention the effect of the vibratory platform on the result of measurement is compensated for, because in a dynamic calibration run the frequency response of the vibratory platform is recorded and the amplitude spectrum is interpolated by a parabolic curve. This means that a significant interfering factor is cut out.

In accordance with an advantageous development of the method in accordance with the invention prior to the actual measuring run a warm-up phase is provided for the vibration absorber in order to heat the shock absorber during measurement up to a predetermined temperature and accordingly to cut out the effect of temperature variations on the result of measurement.

In accordance with an advantageous development of the method in accordance with the invention for recording the support points for estimation of parameters a quality test is performed, the departure of the actual signal from the required sinusoidal signal is detected and, if the departure is too large, the reading is rejected and the next frequency aimed at will then lie below the actual frequency by a predetermined value, as for example 1 Hz, while, if the quality of the signal is sufficient, scanning is continued in small steps of, for instance, 0.5 Hz. Accordingly the pitch of the steps is automatically adapted to the circumstances, something which leads to substantially improved results of measurement.

A further advantageous development of the method of the invention is characterized in that on determining the frequency response the following steps are performed in the order given:

- Measurement of the static weight;
- Running up to starting frequency of for example 10 Hz;
- Warming up phase of shock absorber;
- Run up of exciting vibration of the vibratory platform to the maximum frequency of for example 35 Hz;
- Recording of support points for the estimation of parameters;
- Recording of resonant frequency f_3;
- Recording $\pi/2$ frequency f_2 and
- Transmission of data to evaluation system.

A further advantageous form of the method of the invention is characterized in that for recording the frequency points the following steps are performed in the order indicated:

- Run up to target frequency;
- Check to see if frequency becomes stable;
- Recording readings;
- Separation of tension from compression stage signal;
- Digital filtering, preferably Fourier transformation, of the separated signals;
- Quality assessment of the signals;
- Check of road adherence;
- Check to see whether measurements have been recorded for a given number of revolutions;
- Check to see whether measurements conform to the quality standard for at least a fraction of the revolutions;

- Extraction of mean value for the measurements classified as being good;
- Compensation of the effect of the vibratory platform.

A further advantageous form of the method of the invention is characterized in that for evaluation of the results of measurement the following steps are performed in the order indicated:

- Check to see whether the road adherence is within predetermined tolerance limits;
- Extraction of the characteristic frequencies;
- Parameter estimation as regards the calculation of k_R, the spring constant of the tire;
- Check of air pressure in tire;
- Check of signal form of the amplitude spectrum;
- Check on whether the $\pi/2$ phase state is reached;

  Check on whether the equation $f_2 = f_3$ is fulfilled;
- Parameter calculation
- Check to see whether parameters can be calculated;
- Calculation of degree of axle damping;
- Assessment of degree of axle damping and issue of recommendation concerning the need for replacement of shock absorber.

The various different embodiments of the method of the invention possess the following advantages: The preparation of the measured values assesses the quality of a point of measurement, makes it available for extraction of the mean value and separates the upper and the lower half wave into two sinusoidal signals.

After digital filtering the corrupting effect of the vibratory platform on the shear force sensors is compensated for.

The determination of parameters is based on mathematical modeling of the running gear, which in its complexity can only be solved by so-called computer algebra. The equation systems in this case involve several hundred individual summands with complex 18th order polynomials. This method, which is also known as parameter calculation, is in addition combined with parameter estimation.

The assessment of the running gear is performed not by using the isolated assessment of the model parameters, since for each vehicle type, including its different forms of suspension, the transmission ratios and the characteristics of the installed shock absorbers and suspension springs must be held in the data base. In view of the variety of possible forms such a data base would not be practicable for real use on the customer's premises. Here recourse is had to the EUSAMA Method, which provides a value for road adherence which is unsatisfactory, but is not limited to vehicle types. The so-called degree of axle damping, which is evaluated on the basis of the ratio between suspended and non-suspended mass, is to overcome the known disadvantages of the road adherence value. On the basis of this assessment item a recommendation for replacement is produced for the installed shock absorber.

The advantages of this method are more particularly valuable in the case of modern axle systems, as for example employing pneumatic suspension, freely floating axles etc., in the case of which the classical methods of assessment fail more or less completely and as a rule classify such suspensions unfavorably.

An apparatus for the performance of the method for the examination of shock absorbers installed on a vehicle comprises a vibratory platform, on which one wheel of the vehicle is placed, means for causing the vibratory platform to vibrate in a vertical direction with a suitable amplitude and variable frequency and to cause the wheel to vibrate, the damping of the shock absorber being derived from the force response of the running gear to the vibrations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the movement of the vibratory platform, is characterized by shear force sensors on the vibratory platform and a pulse source for the angularly equidistant scanning, by a means, by which the data signal is split into a compression stage signal and a tension stage signal and by a means in order to separately supply the compression stage signal and the tension stage signal to the means for the further processing of the signals.

An advantageous design of the apparatus of the invention is characterized in that the assessment means assesses the quality of the installed shock absorber using the degree of axle damping as related to the quotient of the suspended mass and the non-suspended, the readings for the degree of axle damping being related to a characteristic curve of a critical of the degree of axle damping, which delimits the range of non-acceptable degrees of axle damping.

The apparatus of the invention has the advantage that the vibrations caused by the vibratory platform are true sine-wave vibrations as desired for being able to obtain also a sine-shaped response of the system.

Further advantageous developments of the invention will be seen from the remaining dependent claims.

A further understanding of the nature and advantages of the present invention and embodiments of the invention may be realized by reference to the remaining portions of the specification and the drawings.

Other advantages, features and characteristics of the present invention, as well as methods of operation and functions of the related elements of the structure, and the combination of parts and economies of manufacture, will become more apparent upon consideration of the following detailed description and the appended claims with reference to the accompanying drawings, the latter of which is briefly described hereinbelow.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 of the drawings appended hereto is a flow chart of the recording of readings.

Figure 2 of the drawings is a flow chart of the preparation of readings.

Figures 3A and 3B of the drawings are graphs indicating the principle of frequency splitting.

Figures 4A and 4B of the drawings are graphs of the principle of amplitude splitting.

Figures 5A and 5B of the drawings are flow charts of evaluation using the method of the invention.

Figure 6 of the drawings is a graph to indicate the action of disruption of road adherence on the frequency curve.

Figure 7 of the drawings is a graph to indicate the action of good shock absorber performance on the frequency curve.

Figure 8 of the drawings is a graph to indicate the action of poor shock absorber performance on the frequency curve.

Figure 9 of the drawings is a graph indicating the target line k_R for the pre-check of tire pressure.

Figure 10 of the drawings is a graph, which represents the k_R line of a tire (Uniroyal™ 155/70 R13).

Figure 11 of the drawings is a graph, which shows the signal form of a satisfactory frequency curve.

Figure 12 of the drawings is a graph to show the signal form applying for a defective shock absorber.

Figure 13 of the drawings is a graph indicating the signal form applying for a suspension with track jitter.

Figure 14 of the drawings shows the theoretical curve form in the m_U-k range.

Figure 15 of the drawings is a graph of the practical curve form in the m_U-k range.

Figure 16 of the drawings is a graph of the scan range of model iteration in the m_U-k range.

Figure 17 of the drawings is a graph of the pole position and of the asymptote of the characteristic curve of the degree of critical value damping.

Figure 18 of the drawings is a graph of the degrees of axle damping of different compression stages.

Figure 19 of the drawings is a graph of the degrees of axle damping of different tension stages.

Figure 20 of the drawings is a graph of dependency of the degree of axle damping on tire pressure.

Figure 21 of the drawings is a graph of dependency of the degree of axle damping on the loading of the vehicle.

Figure 22 of the drawings is a graph of dependency of the degree of axle damping on the temperature of the shock absorber.

Figure 23 of the drawings is a graph of dependency of the degree of axle damping on the tires of the vehicle.

Figure 24 of the drawings is a graph of dependency of the degree of axle damping on changes in the non-spring-suspended mass mU.

Figure 25 of the drawings is a graph of the direction of displacement of the parameters on the point of assessment.

DETAILED DESCRIPTION OF A PREFERRED EMBODIMENT

The method of examination is divided up into two principal steps, namely the recording of readings, in which readings of data are made and processed, and the evaluation, in which the processed data are further processed for making an assessment as regards the quality of the running gear. The examination means includes a vibratory platform, on which in addition to the platform's shear force sensors a pulse source for angularly equidistant scanning is provided.

During taking readings the static weight and the frequency curve of the one quarter vehicle over an exciting frequency of 35 Hz - 10 Hz range are recorded. The amplitude curve then corresponds to the shear force sensor signal on the vibratory platform in Newtons. The phase function corresponds to the phase shift between the exciting oscillation of the vibratory platform and the response oscillation of the one quarter vehicle in radians. From the frequency curve the characteristic frequencies or, respectively, frequency ranges are filtered out and further processed.

In the following evaluation the three characteristic frequency ranges are of special interest, namely the resonant frequency of the non-spring-suspended mass f_3 (here the amplitude curve reaches the absolute maximum), the frequency f_2 (here the phase function is equal to exactly 90° or, respectively, $\pi/2$ and the effective fraction of the amplitude curve is equal to zero at this frequency) and the higher frequencies up to the maximum frequency of 35 Hz. The measurement points are support points for the parameter estimation, with which the spring constant of the tire k_R is ascertained.

The frequency-equidistant scanning of the frequency range 35 Hz to 10 Hz is only a rough approximation. For high quality and reproducible measurements a special measurement algorithm is provided for each characteristic frequency and, respectively, each characteristic frequency range, the chart of such algorithm being illustrated in Figure 1.

The amplitude curve corresponds to the changing part of the overall signal. The unchanged part is taken into account by digital filtering. The static weight of the one quarter vehicle is included as m_s in the calculation of parameters.

At low environment temperature the viscosity of the damping liquid increases and consequently also the damping dissipation of power. The increases in power may here amount to several 100%. Consequently a warming up phase is provided for, in which in the environment of the upper resonant frequency f_3 the damping liquid is warmed up. The initially low force response to the vibratory platform increases on warming up and converges toward a maximum value, whose absolute value is irrelevant. After reaching a maximum value the measurement of the frequency curve as such is proceeded with.

In tests made so far at 0 °C the duration of warming up amounted partially to substantially less than 15 seconds. At even lower temperatures the warming up time may be assumed to be under 30 seconds.

Starting with the maximum frequency of 35 Hz the support points for the parameter estimation are measured. At the high frequencies so-called bounce effects are to be seen in the reading or data signal, which substantially falsify the signal in the tension and compression stage direction. These support points generally have the greatest weight for parameter estimation and must be correct. At each measurement point a quality test is hence performed. The quality test examines the departure of the actual signal for the required sinusoidal signal. If the departure is too large, the reading is rejected. The next target frequency is then 1 Hz below the actual frequency. If the quality of a signal is sufficient, scanning is further performed in 0.5 Hz steps.

For determination of the resonant frequency the following procedure is employed. If the amplitude curve in the compression and tension stage direction exceeds its maximum, the resonant point f_3 of the non-spring-suspended mass has been passed. This resonant range is then scanned again in 0.25 Hz steps. The resonant frequency is thus ascertained with a resolution of 0.25 Hz. The algorithm also provides that in the case of suspensions with a great change in track or shock absorbers with defective valves several maxima may be swept successively. In all problematical cases ultimately the actual resonant frequency is detected with the necessary resolution.

As an alternative to the above mentioned manner of proceeding the resonant frequency may be ascertained in such a manner that the resonant frequency is measured in a single frequency sweep with a resolution of 0.25 Hz. For this purpose it is possible, if the amplitude curve exceeds its point of inflection, for the assumed resonant range to be scanned in smaller frequency steps, for instance in 0.25 Hz steps in the to find the actual resonant frequency with the required resolution in one sweep.

In the case of extremely strong damping effects the phase function will not be able to reach the $\pi/2$ level or, in another extreme case with very weak damping, will run through the $\pi/2$ level very steeply. For both cases the recording algorithm must function. The $\pi/2$ frequency must always be below the resonant frequency f_3. If the two frequencies were of equal size then the vibrating system would be undamped. The phase function would rise with a falling exciting frequency. If the phase value of 1.2 radians is reached, the step pitch if reduced to 0.1 Hz. The resonant frequency f_3 will then be set from these support points.

During the taking of readings and preparing readings of a single excitation frequency, in addition to the evaluation of the spectral variations the measured force value is further processed. Such evaluation is based of the complex frequency curves of forced, damped, elastic oscillating systems. This implies strictly sinusoidal oscillations. The variations with time of the data signal is however in practice not sinusoidal. On the one hand the elongations in the compression and tension stage direction are quite different and on the other hand the duration in time of the compression stage stroke is slightly shorter than the duration of the tension stage stroke.

Figure 2 shows the taking of readings and the preparation of data therefrom in order to achieve optimum accuracy and to provide physically correct information for calculation.

In order to approach the next target frequency and to stabilize same the frequency converter is provided with the frequency value of the next scan point of the frequency spectrum. The taking of a reading is only enabled when the target point in time is reached and the system has become stabilized thereat. A frequency point is deemed to be stabilized when its frequency fluctuations no longer exceed a predetermined tolerance level. In this respect the different frequency ranges are differently evaluated. During run-up to the maximum frequency, attack or transient phenomena have the most marked effects and have the worst effects as regards the quality of the support points and consequently for the estimation of parameters. Here longer stabilizing times are to be preferred. In the case of low frequencies the vehicle may come in the range of disruption of road adherence. Here the road adherence of the suspension parts resists acceleration due to the vibratory platform and abruptly comes to an end as from a certain acceleration. The vehicle sags into its suspension system. The speed of rotation controller is not capable of dealing with this interfering factor when there are substantial disruption effects. Frequency points, which are not able to be stabilized, are omitted in measuring operations. The analog/digital conversion operations are in this case caused by an interrupt during measurement.

In order to be able to supply the physically correct information for the following calculations the overall signal is split up into two independent signal parts for the compression and tension stage and two different split algorithms are employed.

The basic notion of such splitting is the association of respectively half of the readings with the compression and, respectively, tension stage signal. In such splitting there is an exact maintenance of the exciting base frequency and the splitting is relatively simple and rapid to implement. In this type of split the compression stage signal is exaggerated and the tension stage signal is minimized to an even greater extent (see Figures 3A and 3B). The various different damping power dissipations in the stages are spread out even further. On the other hand the phase shifts produced are not plausible. At high power dissipation simulations on a simple one quarter vehicle model show a small phase shift and low power dissipations show a large phase shift. In the case of this type of stage split the phase shifts are almost identical.

The basic notion of the amplitude split is the association of the positive half wave as a compression stage signal and the association of the negative half wave as the tension stage signal. This is generally the same as the association in shock absorber tests in the non-installed state. A spread of the compression and tension stage amplitude does not take place here (see Figures 4A and 4B).

At the frequency of oscillation which becomes established of the two stages the compression stage is distinctly above the excitation frequency, whereas the tension stage is distinctly below the excitation frequency. This is however something which can be compensated for by phase shift, which is in agreement with the theoretical simulation. The narrow positive compression stage half wave must be shifted by $\delta\varphi$ to the right ("it must have a lead") in order to reach the basic frequency of the exciting signal. This is the same as a phase shift of $-\delta\varphi$. The wide negative compression tension stage half wave must be shifted by $\delta\varphi$ to the right ("it must have a lag") in order to reach the basic frequency of the exciting signal. This is the same as a phase shift of $+\delta\varphi$.

The digital filtering means that the signal of the exciting frequency is filtered out. The digital filtering must in this respect be adapted to the corresponding compression-tension stage split method, the classical form of Fourier transformation being able to be employed for frequency split, since the compression and tension portions are exactly equal to the exciting frequency taken as a basis. For amplitude split the higher basic frequency of the compression stage and the lower frequency of the tension stage must be taken in account for the calculation of the amplitude. For this purpose the Fourier transformation of the entire signal may be carried out and the above mentioned phase shifts are added.

The signal curves may be considerably distorted for the most various different reasons. Even the digital filtering for example by the Fourier transformation will then not provide any usable data, since the original signal differs very substantially from the sinusoidal form. These measurement points may then not be used for the further processing.

For assessment of quality for each measured value the relative departure from the ideal filtered value (sine-curve) is calculated. The transverse sum of all departures is then employed for an evaluation. Investigations have indicated that as from a critical value of 60% mean departure the interference, in contrast to 4% or 5% clearly increase on a scale which cannot be tolerated.

Alternatively for assessment of quality it is possible for the differential area of the filtered curve (sine-curve) and the actual value curve to be added on and related to the target area, only one division being necessary.

In the case of extremely poor running gear or, respectively, shock absorber performance it is possible in the frequency environment of the resonant point f_3 for the vehicle to lose contact with the vibratory platform and even to come clear of the test bed.

In the case of road adherence of under 10% the taking of measurements is discontinued for safety reasons. An evaluation of the readings made so far is then no longer necessary, since the running gear may then be classified ad hoc as being defective.

For each stored measurement point three complete cycles are measured. If at least two cycles fulfil the quality criterion, the qualitatively valuable measurement curve is employed to form the mean value for the amplitude and the phase. If only one or no cycle fulfills the quality criterion, then this frequency point is not included in the spectrum.

The so-called vibratory platform is mounted on the shear force sensors. If the system is excited, the oscillating mass of the vibratory platform also has an effect on the shear force sensors. This influence will increase in accordance with a square law with an increase in frequency:

$$F = m_{Platte} \cdot s_{Hub} \cdot \omega^2$$

In the course of the measurement of the frequency response of the vibratory platform however linear and constant fractions were included in the amplitude. In order to minimize this potential source of errors in a so-called 'dynamic' calibration run the frequency response of the vibratory platform is recorded and the amplitude spectrum is interpolated with a parabolic curve. The three coefficients of the interpolation polynomial must here be permanently stored. The phase shift is, but for the slight displacement effects of the low pass filter, constant and is employed as a zero transition reference of the vibratory platform in later determination of phase shift.

For compensation of the vibratory platform the filtered signal must now be corrected by the zero transition reference. Specifically, it is 'shifted to the left', that is to say the measured phase shift is added to the zero transition reference. Now the phase shift is correctly determined in relation to the zero transition reference of the vibratory platform.

The amplitude spectrum of the vibratory platform comprises only effective fractions and no so-called dummy or imaginary fractions. In order to eliminate the falsifying effect of the vibratory platform this real fraction must be subtracted from the real fraction of the measured amplitude spectrum.

From the resulting effective and imaginary fraction of the compensated signal the actual, so-called compensated amount and phase value of the oscillating one quarter vehicle is calculated.

The aim of the evaluation is a qualitative assessment of the measured running gear. More particularly, the quality of the installed shock absorber is to be assessed. In connection with such statement of quality a recommendation, on which the operator of the test bed may rely, is provided with respect to shock absorber replacement. The following three recommendations are provided:

The shock absorber is completely in order (no replacement required).

The reliability of the shock absorber is in the boarder area (replacement is recommended).

The shock absorber is certainly defective (replacement absolutely necessary).

The basis of valuation is the so-called degree of axle damping. This degree of axle damping is then related to the critical value degree of axle damping, which takes into account the particular running gear properties and test conditions. In particular, any attempts at manipulation are compensated for on the basis of this critical value degree of axle damping. The possibilities of manipulation are described in the following. The critical value degree of axle damping may be defined restrictively in a different fashion and is ultimately ascertained in a field test after consultation with the vehicle manufacturers. The characteristic curve of the critical value degree of axle damping is only defined by two specific parameters. They then apply for all types of vehicle and all test conditions.

The evaluation is based on a plurality of complex individual steps, which are represented in the flow chart of Figure 5. One feature of the evaluation method is a plausibility test, which comes before the calculation and evaluation of the degree of damping. This plausibility test has the purpose of eliminating any extreme situations of the running gear, which can possibly not be calculated. Such incalculable, extreme running gear is not necessarily due to bad or defective shock absorbers. Extremely good shock absorber performances, which may possibly be enhanced by a large fraction of external friction, offend against the one quarter vehicle model for parameter calculation.

The characteristic frequencies for the tension and compression stage are extracted separately for the measured amplitude and phase spectrum. If one stage is defective, then the shock absorber must be replaced.

The resonant frequency f_3 is the same as the frequency, at which the amplitude maximum is swept through. This extreme value must be local and global. If for example the first value at 35 Hz is a global maximum value wo, same is not accepted as being local and the series of measurements is rejected as being invalid.

The following problems impair the exact ascertainment of the $\pi/2$ frequency. In the course of recording the spectrums scanning takes place in frequency-equidistant steps. As a rule the frequency point is not exactly reached, at which a compensated phase shift of exactly $\pi/2$ becomes established. The disruption of road adherence, already mentioned several times, which occurs after passage through the upper resonant frequency, will particularly falsify the phase spectrum. Owing to disruption of road adherence the continuously extending phase function comprises erratic values, which may lie above the $\pi/2$ level, although the good damping action means that $\pi/2$ resonance will be reached much later or even not at all, as illustrated in Figure 6.

In the same manner certainly reaching the $\pi/2$ level may be prevented owing to an erratic measurement. By interpolation of the section of the phase function using a third order polynomial there is both an elimination of the erratic points and also an exact calculation of the π/2 frequency f_2. The support points for the interpolation start here at the resonant frequency and terminate at the local maximum of the phase function. The effect of a good shock absorber performance of the frequency curve is illustrated in Figure 7.

The frequencies f_2 and f_3 may be relied upon for a plausibility test. In this respect two extreme cases should be dealt with before the actual parameter calculation.

If the π/2 frequency is not present, this means that the π/2 frequency level was not reached. The damping of the running gear is exceptionally good. A calculation of parameters would not be possible and is unnecessary.

If the π/2 frequency f_2 is larger than or equal to the resonant frequency f_3 then in the oscillating system without damping the frequencies f_2 and f_3 are theoretically equal. If this behavior is found during the recording of the frequency curve, then the damping of the running gear is extremely poor. Calculation of parameters would not be possible and is unnecessary. The effect of poor shock absorber performance of the frequency curve is represented in Figure 8.

The two extreme cases could be provoked by manipulation of the tire pressure. Checking of air pressure is necessary before the plausibility test of the characteristic frequencies.

The estimation of parameters is a method, which estimates the parameters of a predetermined complex frequency curve on the basis of measured support points. The complex frequency curve is based on the simple one quarter vehicle model:

$$G=\frac{b_0+ib_1\omega-b_2\omega^2-ib_3\omega^3+b_4\omega^4}{1+ia_1\omega-a_2\omega^2-ia_3\omega^3+a_4\omega^4}$$

or, respectively, with the real parameters:

$$F_P=\frac{-m_S\omega^2-im_S\frac{d}{k}\omega^3+\frac{m_U\cdot m_F}{k}\omega^4}{1+i\frac{d}{k}\omega-(\frac{m_F}{k}+\frac{m_F}{k_R}+\frac{m_U}{k_R})\omega^2-im_S\frac{d}{k\cdot k_R}\omega^3+\frac{m_U\cdot m_F}{k\cdot k_R}\omega^4}$$

wherein:

m_F	spring-suspended mass [kg]
m_U	non-spring-suspended mass [kg]
m_s	static overall mass [kg]
k	spring constant of the suspension spring [N/m]
k_R	spring constant of the tire [N/m]
d	damping constant of the shock absorber [kg/s]
ω	circular frequency of the exciting oscillation [1/s]

The basis of the linear equation system is a loss function V, which calculates the departure of all support points for the complex frequency curve to be assessed:

$$V=\sum_{\nu=0}^{N}\psi_\nu[R_\nu(1-a_2\omega^2+a_4\omega^4)-I_\nu(a_1\omega-a_3\omega^3)-(b_0-b_2\omega^2+b_4\omega^4)]^2$$
$$+\psi_\nu[R_\nu(a_1\omega-a_3\omega^3)+I_\nu(1-a_2\omega^2+a_4\omega^4)-(b_1\omega-b_3\omega^3)]^2$$

wherein:

R_ν	is the measured real fraction of the support point ν
I_ν	is the measured imaginary fraction of the support point ν
Ψ_ν	is the weight fraction of the support point ν
N	is the number of support points - 1.

This loss function is differentiated after each coefficient to be estimated and set at zero. That is to say, every coefficient is to be so estimated that the resulting loss runs through a minimum.

For the nine coefficients to be estimated there are consequently nine linear equations, which are linked in a linear equation system. The coefficients a_1 - b_5 are determined by the solution of this linear equation system. The application of the method is extremely complex and is hence not particularly suitable for determination of parameters. The absolute values of the individual parameters of the frequency curve of the simple one quarter vehicle model differ to a great extent. This means that the accuracy of the individual parameters suffers. The parameters b_0 and b_1 are always zero in the case of the frequency curve of the simple one quarter vehicle model. The parameter estimation method however allocates values to these parameters and deals with the departures by means of the higher coefficients. Each support point is weighted with the function phi_v. That is to say, its influence on the loss function is set. In this respect the it is not the absolute amount but the ratio of the weight functions of the support points inter se which is relevant. By appropriate definition of the weight function the band width of the solutions may be practically varied at will. There is no unambiguous, optimum weight valid for all measurement curves.

The particular feature of the frequency curve for the simple one quarter vehicle model is the convergence of the amplitude spectrum toward the spring constant of the tire k_R at high frequencies. This asymptote may be calculated by a modified form of the parameter estimation method. Of the originally nine parameters to be estimated the parameters b_0 and b_1 are basically zero and the parameter b_2 is the static mass m_s, which is also known. The linear equation system is thus reduced to six linear equations. For estimation 20 different weight functions are swept. For each weight function the theoretical support points are calculated. The weight function with the minimum mean departure in the support points employed is then used.

For verification of the estimated spring constant values k_R of the tire the frequency curves of the tire are recorded under different amounts of bias. Three results may be derived from these results:

The spring constant is constant over the entire frequency range from 35 Hz to 1 Hz.

The value of the spring constant is in agreement with the estimated value with an accuracy of 2%.

The typical k_R characteristic curves of a tire are primarily dependent on the tire pressure, the dimensions of the tire and the bias, that is to say the weight, with which the tire is thrust against the ground. The specific manufacturer data such as a rubber blend, type of carcass or tread are only of secondary significance. Furthermore the wheel employed is without any significant influence on the spring constant k_R.

The spring constant k_R corresponds to the degree of coupling between the vibratory platform and the oscillating spring-shock absorber system of the non-spring-suspended mass m_U. If the degree of coupling too high, a large amount of oscillation energy will be supplied to the spring-shock absorber system. The excess energy is then interpreted as insufficient shock absorber performance. If the degree of coupling too low, too little oscillation energy will be supplied to the spring-shock absorber system. The deficit of energy is then attributed to the shock absorber performance.

As explained below, this erroneous assessment will be satisfactorily compensated for in the case of small air pressure departures as regards the assessment of the shock absorber, unlike other methods with the degree of axle damping.

The spring constant k_R of the tire is suitable for examination of the tire pressure. The basis of the plausibility test is the specification of dimensions of the tire types (for example 225/50 R15). The rated air pressures for a certain type of dimension vary on use of this tire on different types of vehicle only to an insignificant extent and are then coupled at least with the weight of the vehicle. Heavy motor vehicles imply in this case a higher rated tire pressure than lighter vehicles (Bridgestone™ "Ratgeber Reifen" 1992).

225/50 R16	Rated tire pressure VA (bar)	Rated tire pressure HA (bar)
Mercedes 500	3.0	3.3
Volvo	2.3	2.6
Porsche Carrera	2.0	2.5

For the tire type 225/50 R16 as indicated a minimum curve for the minimum value of 2.0 bar (for the front axle of the Porsche Carrera) and a maximum curve for the maximum value of 3.3 (for the rear axle of Mercedes 500) would be stored in a data base. Both of the curves represent the changes in the spring constant value against biasing weight (Figure 9). The two extreme working points are now connected by the rated k_R curve. All other vehicle types, which are fitted with these tires, then lie more or less on a rated k_R curve. For a fixed bias a change in the tire pressure has a linear relationship to a change in the spring constant k_R. Accordingly a tolerance band of, for example, 0.5 bar may be defined. If the estimated k_R value is outside the predetermined tolerance band, the measurement is not evaluated. An instruction to check tire pressures is issued as well.

Figure 10 shows the results of measurement with the tire type Uniroyal™ 155/70 R15 on as tire test bed. In the case of a test series with the Ford Escort the vehicle loading and the tire pressure were varied. The plausibility tests did not indicate any false assessment.

Attempts aimed at manipulating the final result by changing the load are rendered more difficult by this plausibility test. It is possible to see clearly whether an increased degree of coupling is due to an excessively high tire pressure or an extremely high load. A higher weight implies a higher bias and accordingly a higher degree of coupling with the tire pressure unchanged.

The actual curve of the amplitude and phase spectrum is substantially in accord with the theoretical target or rated curve of the simple one quarter vehicle model. It is only in the frequency range of disruption of road adherence that the simple one quarter vehicle model cannot be utilized. However since on reaching the disruption of road adherence the measurement is terminated in any case, this can be neglected. Any other departure of the spectrum is either due to defective running gear or to an unusual property of the running gear. The so-called signal form test is to classify irregularities in the amplitude spectrum.

In Figure 11 a satisfactory amplitude and phase spectrum is to be seen. At the resonant point the amplitude curve smoothly and unambiguously runs through its maximum. The phase function smoothly converges toward zero with increasing exciting frequency.

In Figure 12 the spectrums of a manipulated shock absorber are represented. The resonance maximum of the amplitude spectrum is not unambiguous. The phase spectrum is practically free of interference in this range. This leads to a spread of the wrongly determined characteristic frequencies f_2 and f_3. The resulting degree of axle damping is then clearly above or outside the calculable value range. This case cannot be employed for calculation.

Owing to a camber angle some running gear sweeps through a plurality of resonant points , in which the vehicle is subject to powerful horizontal oscillations. Such a spectrum is illustrated in Figure 13. In this case such so-called camber flutter could also be interpreted as due to a defective shock absorber. The difference between this and the amplitude curve of a mechanically defective shock absorber is however the fact that between the characteristic frequencies f_2 and f_3 no local minimum is swept through. Such series of measurements are made available for further parameter calculation.

The parameter calculation is based on the complex frequency curve of the simple degree one quarter vehicle model:

$$F_P = \frac{-m_S\omega^2 - im_S\frac{d}{k}\omega^3 + \frac{m_U \cdot m_F}{k}\omega^4}{1 + i\frac{d}{k}\omega - (\frac{m_F}{k} + \frac{m_F}{k_R} + \frac{m_U}{k_R})\omega^2 - im_S\frac{d}{k \cdot k_R}\omega^3 + \frac{m_U \cdot m_F}{k \cdot k_R}\omega^4}$$

wherein:

m_F	spring-suspended mass [kg]
m_U	non-spring-suspended mass [kg]
m_s	static overall mass [kg]
k	spring constant of the suspension spring [N/m]
k_R	spring constant of the tire [N/m]
d	damping constant of the shock absorber [kg/s]
ω	circular frequency of the exciting oscillation [1/s]

All the following calculation steps were performed using computer algebra:

- calculate the amplitude curve A from the complex frequency curve F_p.
- For calculation of the characteristic equation for the above resonant point f_3 differentiate the amplitude curve A and make A' equal to zero.
- Solve this equation in terms of the damping constant d and the following applies:
  $d_3 = g(\omega_3, m_S, m_U, k_R, k)$.
- For calculation of the characteristic equation for the π/2 frequency f_2 calculate the real fraction of A and make A_{re} equal to zero.
- Solve this equation in terms of the damping constant d and the following applies:
  $d_2 = h(\omega_2, m_S, m_U, k_R, k)$.

In both equations g and h the exciting frequencies omega and the static weight m_s are measured. The spring constant of the tire $_{kR}$ is determined in the parameter estimation method and is accordingly known as well.

Two equations are now available for d_2 and d_3 with two unknowns m_U and k_R. The same may however not be linked directly, since the damping constant is frequency-dependent. The linking together occurs in the amplitude curve A. Generate the amplitude curves A_2 (replace d by d_2) and A_3 (replace d by d_3). Since the amplitudes at both frequency points are also measured, there are still two equations with two unknowns m_I and k, which may be now linked. It is now possible to calculate all parameters.

The calculation of the parameters is now able to be resolved to several zero position problems as far as the fourth order. If the parameters are not able to be calculated (for example no real zero point or division by zero), a series of measurements is to be considered, which corresponds to an extreme situation. In such cases evaluation on the basis of the degree of axle damping is omitted. From an extreme situation with a valid signal curve and with characteristic frequencies present it is possible to conclude that there is either very good or very poor running gear. For the classification in this special case recourse is had to the ground standing value. If this value is relatively large the running gear is in order. If the value is on the other hand low, the running gear must be classified as poor and replacement of the shock absorber is recommended. If the ground standing value is not able to be classified, an error message is issued.

Using the measurements described so far basically all measurement series may be calculated, whose signal curves and characteristic frequencies were satisfactory. If the above mentioned error should be due to an unusual measuring situation, then the original measurement values must be communicated to the software partner for detailed investigations. The respective saving of data is automatically performed when there is an error.

The results of the parameter calculation may not be employed directly for an assessment of the measured running gear. The reasons for this are the selection of the correct calculation model, since the simple one quarter vehicle model does not differentiate between the external friction of the suspension parts and the internal friction of the damping liquid and the lack of comparability of the parameters in the installed condition with measured parameters in the dismounted condition. The dynamics of both examination methods are extremely different. For instance, when recording bell curves measurement is normally performed with large strokes at relatively low exciting frequencies, whereas the vibratory platform only implies extremely small strokes at high frequencies at the shock absorber. If running gear is to be assessed using the calculated parameters, then the comparison values for this running gear must be held in a data base. In view of the large number of types of running gear and installed shock absorbers the production of a user friendly data base is not possible for use in the workshop.

The aim is consequently the assessment of the complete running gear situation, which is not tied to the type of vehicle and takes into account attempts at manipulation or takes same into account in assessment. The result of the assessment is to be a recommendation to replace the installed shock absorber.

This aim is to be reached using the so-called degree of axle damping ($Achs_{DG}$):

$$Achs_{DG} = \frac{d}{2 * \sqrt{(k + k_R) * m_U}}$$

The parameters employed here are the result of the so-called model iteration.

For familiar reasons the results of the parameter calculation may not be utilized directly for the determination of the degree of axle damping. This will be clear, when the solution quantity is represented in an m_U-k field. Theoretically all solution curves for the different characteristic frequencies should intersect at a single (m_U,k) point, as illustrated in Figure 14.

In fact there is no unambiguous solution point (m_U,k). In practice a solution field is defined by the three solution curves, as shown in Figure 15. As will be seen, the solution straight lines f_2 and f_3 are spread apart and the solution straight line f_1 is substantially above the target solution point (m_I,k). With the aid of computer algebra it is possible to show that principally the influence of external friction is responsible for these shifts. The solution range for the non-spring-suspended mass can however be limited to a range of under 10 kg.

In the case of vehicles, which are measured with shock absorbers of varying quality, there is a shift of the entire solution field in the m_U dimension. In the case of running gear whose external friction is to be assumed to be large this effect is particularly clear.

Non-spring-suspended mass M_U (kg) with different shock absorber performances

	good	medium	poor
BMW 740i	32.0	39.5	46.0
Ford Escort	29.0	30.5	29.5

For the calculation of the correct solution point a so-called model iteration is swept through. Starting with the original measured data the measured data are interactively modified in order to eliminate the effects of the external friction. The algorithm terminates, when all three solution straight lines intersect precisely at one point and more particularly the f_1 iteration straight line converges to the f_1 target solution straight line.

Model iteration is difficult to handle in this form. The resulting parameters are to a large extent in agreement with manufacturers' data. A problematical parameter is the spring constant k, which in the some vehicle types is subject to a relative variation of 50%. Model iteration may require computing times of up to one minute. In order to be able to reliably use the terminating model iteration, the measurement of the resonant frequency f_1 of the spring-suspended mass is necessary). The amount of time then required for measurement would then be doubled.

The resonant maximum of the resonant frequency of the spring-suspended mass is not distinct in the case of some vehicle types so that some search algorithms must be utilized, which find the resonant frequency f_1 on the basis of empirically determined facts.

The work involved in determining the spring constant f_1 bears no relation to its influence on the final result. Neither the damping constant d nor the degree of axle damping are significantly influenced by the varying spring constant k or, respectively, the absence thereof. As an example some data for the BMW 3.28 i are relevant:

BMW 3.28i front axle right

Parameter	Target value measurement	measurement series 1	measurement series 2
Non-spring-suspended mass m_U (kg)	ca. 40	40.7	41.5
Suspension spring k (N/mm)	19.2	18.8	24.6
Damping coefficient d(kg/s)	?	1202	1216
Degree of axle damping without k (%)	?	13.2	13.3
Degree of axle damping with k (%)	?	13.0	12.9

Owing to the small influence on the degree of axle damping and the substantial disadvantages involved in the determination of the spring constant k the latter is omitted so that the degree of axle damping is equal to:

$$Achs_{DG} = \frac{d}{2 * \sqrt{k_R * m_U}}$$

The model iteration may be simplified to a great extent and consequently performed more rapidly. The simplification is due to the fact that the solution curve for the characteristic resonant frequency f_1 is replaced by a constant straight line at k = 100000 N/m (Figure 16).

The non-spring-suspended mass applying here is calculated using the customary iteration algorithm. This iteration algorithm leads to the solution points (m_U,10000) with the least measured value departure and the best cover of the damping constants of the remaining solution curves f_3.

The following statements may be made on the basis of the degree of axle damping for an assessment of the running gear. There is a high reproducibility in the determination of the parameters d, k_R and m_u. There is no limitation t particular types of vehicles. The larger k_R ,the higher the fraction of the coupled oscillation energy in the spring-shock absorber system and the higher must the shock absorber performance be. The larger m_U, the higher must the shock absorber performance be.

For purposes of comparison vehicles with manipulated shock absorbers were measured. The degrees of axle damping of the defective shock absorbers were then employed as basic data for the determination of a critical or border value line. The critical or border value line must take into account the characteristic properties of the running gear, which are not inherent in the degree of axle damping.

Test measurements have indicated that the ratio between spring-suspended to non-spring-suspended mass m_F/m_U represents this decisive property. Vehicles with a large mass ratio m_F/m_U made do with substantially smaller degrees of axle damping than vehicles with relatively small mass ratios, independently of the overall weight of the vehicle.

This fact was included in a critical or border value line for the mass ratio in the assessment. The critical value line was in this case to possess a hyperbolic form with on the one hand the minimum possible mass ratio as the pole point and on the other hand the minimum degree of axle damping as the asymptote, as shown in Figure 17.

On the basis of the basic data non-linear regression was employed to explicitly represent the linear boundary value line. The regression model utilized as a basis had to be a member of the so-called growth models. These models are characterized by a strictly monotonous, asymptotic form, which is in agreement with the previously made observations. The model with the best agreement is the so-called saturation model. This model renders possible the description of the critical value line by the two parameters a and b ($Achs_{DGGrenz}$=critical value for degree of axle damping):

$$Achs_{DGgrenz} = \frac{a * {}^{m_F}\!/_{m_U}}{b + {}^{m_F}\!/_{m_U}}$$

wherein:

a is the asymptote, that is to say the minimum degree of axle damping.

b is the pole point, that is to say the minimum mass which is physically possible.

In Figures 18 and 19 the degrees of axle damping of different measured vehicle types are represented for the compression the tension stage. All types of vehicle were measured in this case with their rated tire pressure and unloaded weight and, respectively, with a driver.

The basic data of the critical or border value line are the 50% shock absorber of the BMW 3.28, the 60% shock absorber of the Ford Escort, the minimum mass ratio of 5 and the minimum degree of axle damping of 7%. The running gear and axle designs in an as-new condition were all found to be poor in the EUSAMA test, as may be seen from the following table:

Running gear	EUSAMA rating (%)
BMW 3.28i HA	22
BMW 5.28T HA	25
Audi A & VA	37
Audi A6 HA	32
Passat HA	39
Golf 3 HA	35

The degrees of axle damping are on the contrary distinctly in the range applying for satisfactory running gear. Problematical cases are only the rear axle of the BMW 5.23T and the front axle of the Audi A6. In these two cases the running gear the degree of axle damping is less than for example the 50% damping of the BMW 3.28i. However both mass ratios are substantially above that of the BMW 3.28i and hence substantially above the critical value line.

In order to have further measurement points below the critical value line the rear axle of the BMW 5.28 was measured without the shock absorber. The vehicle could however only be measured with a tire pressure decrease of -0.5 bar and -1.0 bar, since at normal pressure the tire rose clear of the vibratory platform.

The problem in the calculation of the parameters, in which the non-spring-suspended mass m_U fluctuates in accordance with the quality of the installed shock absorbers, is however not dealt with by using model iteration. However this is actually an advantage in the case of the use of the mass value m_U within the degree of axle damping, as may be seen from the three measurement points for the BMW 740i. Using an interface it was possible to select between the shock absorber settings of hard, medium and comfortable:

The poor shock absorber performance of the comfortable setting implies a small mass ratio and displaces the degree of axle damping to be nearer to the critical line.

The good damping performance with the hard setting implies a large mass ratio and displaces the degree of axle damping further away from the critical line. This effect occur to a greater degree in the case of vehicles with a high degree of external friction.

The rating in the running gear in separate stages and made to apply for different vehicle type on the basis of the degree of axle damping and including the mass ratio is a substantial advance over the EUSAMA Method.

For maximum objectivity in rating of the test method it is necessary additionally to compare the reactions to attempts at manipulation with the new test method on the one hand with the corresponding EUSAMA ratings on the other hand. The following parameters were manipulated for this purpose:

Tire pressure - in set 0.5 bar steps from 1.0 bar under the rated tire pressure to 1.0 above the rated tire pressure;

Loading - all vehicle were empty, some of them additionally with a driver and one vehicle in its predetermined design position;

Temperature - at the start of the development phase (in winter 1996/97) one vehicle at 0°C and at 15°C;

Tires - one vehicle with three different tire designs. Some vehicles with a special purpose wheel clip to vary the non-spring-suspended mass.

The illustrations mentioned only constitute the tension stage results. The corresponding values for the compression stage hardly differ from those of the tension stage. The critical value lines for the tension stage are extracted from the measurement points, which were recorded at rated tire pressure and without a load (at the most one driver, see Figure 19).

The values in brackets correspond to the ground standing values at these measurement points. Same are necessary in order to compare the reactions of the EUSAMA method with those of the new method.

The tire pressure is one of the principal problem in the measurement of running gear. The results of the assessment may be manipulated to suit the particular intention. The preliminary check of the tire pressure is to detect departures from the rated pressure value and is the primary advantage of the new method of testing. With the present stock of experience it is possible to assume that departures in tire pressure of 0.5 bar may be detected without difficulty. When further experience has been gained it may be possible to detect a departure of 0.25 bar. Even smaller intervals in the departure are theoretically not possible, since the sum of the departures in the estimation of the parameters, the preset rated value and the temperature drift constitute a natural lower limit.

Figure 20 shows the measurement points for the rated tire pressure and the measurement points for a pressure departure of 0.5 bar. Major variations in tire pressure are not represented, since same are certainly filtered out by the preliminary check. The direction of the arrow indicates the increase in tire pressure by 0.5 bar.

The following details may be derived from the above graph. With an increase in tire pressure, the degree of damping and the mass ratio decrease. Good running gear is more influenced by the tire pressure than poor running gear. A change in the rating does not occur. The influence of the tire pressure on the EUSAMA values is on the contrary particularly great in the case of poor running gear so that even a recommendation to replace may be manipulated. The EUSAMA values were to some extent tripled by the change in tire pressure. The variation in the rating points ($Achs_{DG}, m_F/m_U$) is certainly below 50%.

The influence of the tire pressure is not eliminated in the result, but is made lower by a factor of 4 in comparison with the EUSAMA method on average. Substantial departures from the rated tire pressure can be detected. A manipulation of the recommendation to replace on the basis of the tire pressure is not possible. In fact poor running gear varies only around the error of propagation as based on errors of measurement when the tire pressure is changed.

The BMW 3.28i was measured in its design position, with a load of ca. 230 kg and with a driver. The difference between the empty vehicle and the vehicle with a driver was so slight that it was not taken into account in the figure.

The following details may be gathered from Figure 21. The degrees of axle damping were reduced by 2% owing to the load. This is in contrast to the EUSAMA data, which increase slightly with a load. The calculated value for the non-spring-suspended mass m_U is practically the same with and without a load. The weight of the load means that the mass ratio is increased and the assessment point is moved into the favorable field.

Assessment reacts to load of a vehicle with the better classification of the running gear properties. The assessment of this running gear situation is basically correct, but there is no possibility of distinguishing the added load part of the mass from the mass of the unloaded vehicle. The reaction is more significant than with the EUSAMA method.

The manipulation of the recommendation to replace on the basis of the loading is therefore possible in the case of running gear in the environment of the critical value line.

The influence of the temperature of the surroundings was investigated by making measurements at approximately 0°C and then at the temperature of the surroundings of approximately 15°C. From Figure 22 it will be seen that the degrees of axle damping increases on cooling and that the mass ratio is not changed by cooling down. The EUSAMA values remain practically unchanged. Manipulation of the recommendation to replace on the basis of the environmental temperature of the surroundings can consequently not be excluded.

With a change in tires the tire spring constant for the vehicle and the non-spring-suspended mass will be changed. In this connection two test series were run:

- The BMW 3.28i was measured with the complete sets of tires in the design position.
- Several vehicles were measured with a special purpose wheel clip (5.6 kg) and, respectively, a screwed-on steel plate (14.3 kg) on the wheel.

The following details may be gathered from Figure 23. The rating points ($Achs_{DG}, m_F/m_U$) are not changed in their classification. The slightly improved degree of axle damping which heavy tires is compensated for by the decrease in the mass ratio.

This effect is particularly clear in the case of small low mass ratios. The EUSAMA values decrease with an increase in the tire spring constant k_R. A change in tires mainly causes changes in the spring constant k_R and thus in the degree of coupling. The influence on the non-spring-suspended mass is within the range of accuracy of measurement owing to the small differences in measured values. The influence of the spring constant k_R is taken into account in the degree of axle damping, whereas in the case of such an increase in coupling the EUSAMA method provide poorer road adherence data. A manipulation of the recommendation is not possible.

The following may be seen from Figure 24. The degrees of axle damping are, like the road adherence values, slightly reduced by the additional weight. The additional weight is fully taken up in the non-spring-suspended mass and in part leads to a drastic reduction in the mass ratio.

Bad running gear can be imitated by manipulation of the non-spring-suspended mass. The EUSAMA values tend to react to manipulation in the same manner but not to such a significant extent. The result of this is that highly defective running gear may still be classified as good.

By way of conclusion it is possible to say the following on the possibilities of manipulation. The manipulation of the tire pressure would be detected by the preliminary examination in view of k_R and m_s. As compared with the EUSAMA method there is the advantage that statement as regards the tire pressure is possible and the sensitivity of the examination method for tire pressure is less. Manipulation by way of the temperature of the shock absorber is excluded by the warming up phases. To be possible manipulation on the basis of the load would involve massive external manipulation and may therefore also be excluded. A further point is that the load affects the mass ratio, see the above mentioned graph. Manipulation by way of changing the tires is not possible, since there is a check in this respect. As compared with the EUSAMA method there is the advantage that the influence of the changed spring constant is removed by calculation.

Manipulation of the mass m_U could make it seem that the running gear is poorer than is the actually case, something which would have to involve massive manipulation from the outside and can consequently be excluded. As compared with the EUSAMA method there is the advantage that the load is included in the mass ratio and is therefore covered by the test method. Figure 25 shows by way of conclusion the directions of displacement of the effective parameters in the field of degree of axle damping to the degree of axle damping mF/m_U. Manipulations are consequently either excluded or they have a substantial effect of the results of measurement.

It is to be understood that the above description is intended to be illustrative and not restrictive. Many embodiments will be apparent to those skilled in the art upon reviewing the above description. Those skilled in the art will recognize as an equivalent or alternative method of tire testing and combining a tire testing machine with a wheel balancing machine. The scope of the invention should, therefore, be determined not as reference to the above description, but should instead be determined with reference to the appended claims along with the full scope of equivalence to which such claims are entitled.

THE EMBODIMENTS OF THE INVENTION IN WHICH AN EXCLUSIVE PROPERTY OR PRIVILEGE IS CLAIMED ARE DEFINED AS FOLLOWS:

1. A method for the examination of shock absorbers installed on a vehicle using a vibratory platform, on which one wheel of the vehicle is stood and which is able to be reciprocated in a vertical direction at a suitable amplitude and with a variable frequency, an oscillating action is exerted of the wheel, the damping of the shock absorber being derived from the force response of the running gear to the oscillations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the movement of the vibratory platform, the data signal which represents the force response of the shock absorber being subjected to the evaluation, characterized in that, as a basis for assessment of the quality of the installed shock absorber, the degree of axle damping as related to the quotient: suspended mass divided by non-suspended mass, is calculated, the measured values of the degree of axle damping being related to a characteristic curve of a critical degree of axle damping, which delimits the range of non-acceptable degrees of axle damping.

2. The method as set forth in claim 1, characterized in that prior to the actual evaluation the road adherence of the vehicle being tested is examined during the measurement and in that, if the road adherence is below a predetermined value, evaluation is terminated and the shock absorber is classified as being sub-standard.

3. The method as set forth in claim 1, characterized in that the data signal is split by tension-compression stage splitting into a compression stage signal, which is characteristic for the

compression stage of the shock absorber and a tension stage signal, which is characteristic for the tension stage of the shock absorber, and in that the compression stage signal and the tension stage signal are evaluated separately as regards the degree of axle damping.

4. The method as set forth in any one of the preceding claims, characterized in that for extraction of the characteristic frequency the measured amplitude and phase spectrums are processed by interpolating the curve section of the phase function by a polynomial.

5. The method as set forth in claim 4, characterized in that the support points for the interpolation start at the resonant frequency and terminate at the local maximum of the phase function.

6. The method as set forth in claim 4, characterized in that the frequencies f_2 and f_3 are subjected to a plausibility test, the shock absorber being deemed to be good, if the π/2 frequency f_2 does not exist, and is deemed to be poor, if the π/2 frequency f_2 is larger than or equal to the resonant frequency f_3.

7. The method as set forth in any one of claims 1 through 4, characterized in that a parameter calculation is performed on the basis of the complex frequency curve of the simple one quarter vehicle model:

$$F_P = \frac{-m_S\omega^2 - im_S\frac{d}{k}\omega^3 + \frac{m_U\cdot m_F}{k}\omega^4}{1 + i\frac{d}{k}\omega - (\frac{m_F}{k} + \frac{m_F}{k_R} + \frac{m_U}{k_R})\omega^2 - im_S\frac{d}{k\cdot k_R}\omega^3 + \frac{m_U\cdot m_F}{k\cdot k_R}\omega^4}$$

wherein:

m_F suspended mass [kg]

m_U	non-suspended mass [kg]
m_S	static overall mass [kg]
k	spring constant of the suspension spring [N/m]
k_R	spring constant of the tire [N/m]
d	damping constant of the shock absorber [kg/s]
ω	circular frequency of the exciting oscillation [1/s]

and in that from such parameters the degree of axle damping ($Achs_{DG}$):

$$Achs_{DG} = \frac{d}{2 * \sqrt{(k + k_R) * m_U}}$$

is found, the parameters utilized here being a result of model iteration.

8. The method as set forth in claim 7, characterized in that for parameter estimation of the value k_R a loss function:

$$V = \sum_{\nu=0}^{N} \psi_\nu [R_\nu (1 - a_2\omega^2 + a_4\omega^4) - I_\nu (a_1\omega - a_3\omega^3) - (b_0 - b_2\omega^2 + b_4\omega^4)]^2 \\ + \psi_\nu [R_\nu (a_1\omega - a_3\omega^3) + I_\nu (1 - a_2\omega^2 + a_4\omega^4) - (b_1\omega - b_3\omega^3)]^2$$

is employed, which calculates the departure of all support points for the complex frequency curve to be estimated, wherein:

R_ν	is the measured real fraction of the support point ν
I_ν	is the measured imaginary fraction of the support point ν
Ψ_ν	is the weight fraction of the support point ν
N	is the number of support points - 1.

and the loss function is differentiated after every coefficient to be estimated and set at zero, the asymptote of the converging form of the amplitude spectrum as plotted against the spring constant k_R of the tire at high frequencies being able to be calculated using a modified form of the parameter estimation method, in which of the original nine parameters to be estimated the parameters b_0 and b_1 are basically zero and the parameter b_2, is the also known static mass m_s so that the linear equation system is thus reduced to six linear equations.

9. The method as set forth in claim 8, characterized in that for such estimation a plurality, as for example 20, different weight functions are swept, in that for each weight function the theoretical support points are calculated and in that then the weight function is employed with the smallest mean departure in the support points used.

10. The method as set forth in claim 8 or in claim 9, characterized in that after such estimation of the parameter k_R this value is compared in the k_R/mS field with a target k_R curve and the measurement is not evaluated, if the estimated k_R value is outside a predetermined tolerance band.

11. The method as set forth in claim 7, characterized in that the signal form of the frequency curve is examined and in that measurements, which are characteristic for a disruption of road adherence, and measurements, which have a frequency curve without any unambiguous resonance maximum of the amplitude spectrum are rejected.

12. The method as set forth in claim 7, characterized in that the frequency curve, which does not have any local maximum between the characteristic frequencies f_2 and f_3, is made available for further calculation of parameters.

13. The method as set forth in claim 7, characterized in that for parameter estimation the following calculation steps are performed using computer algebra:

- Calculate the amplitude curve A from the complex frequency curve F_p.
- For calculation of the characteristic equation for the upper resonant point f_3 differentiate the amplitude curve A and make A' equal to zero.
- Solve this equation in terms of the damping constant d and the following applies:
  $d_3 = g(\omega_{3}, m_S, m_U, k_R, k)$.
- For calculation of the characteristic equation for the $\pi/2$ frequency f_2 calculate the real fraction of A and make A_{re} equal to zero.
- Solve this equation in terms of the damping constant d and the following applies:
  $d_2 = h(\omega_{2}, m_S, m_U, k_R, k)$;

in both equations g and h the exciting frequencies Ω and the static weight m_s are measured, the spring constant of the tire k_R is known from the parameter estimation method so that two equations remain for d_2 and d_3 with two in unknowns m_U and k_R;

- generate the amplitude functions S2, d being replaced by d_2, and A_3, d being replaced by d_3, two equations with two unknowns m_U and k being obtained, which are linked for calculation of the parameters.

14. The method as set forth in claim 7, characterized in that owing to small influence of the spring constant k on the degree of axle damping the spring constant k is omitted so that the degree of axle damping is equal to:

$$Achs_{DG} = \frac{d}{2 * \sqrt{k_R * m_U}}$$

the model iteration being simplified to a great extent and consequently performed more rapidly since the solution curve for the characteristic resonant frequency f_1 is replaced by a constant straight line at k = 100000 N/m, the non-suspended mass applying here being calculated using the customary iteration algorithm, this iteration algorithm leading to the solution points (m_U,10000) with the least measured value departure and the best cover of the damping constants of the remaining solution curves f_3.

15. The method as set forth in claim 7, characterized in that if the parameters are not able to be calculated, road adherence is checked and if a high degree of road adherence is found, the shock absorber is deemed to be good, whereas in the other case evaluation is terminated and the shock absorber is deemed to be sub-standard.

16. The method as set forth in claim 7, characterized in that for the determination of a critical value line measurements are made on vehicles with manipulated shock absorbers and the degrees of axle damping of the defective shock absorbers are employed as basic data.

17. The method as set forth in claim 16, characterized in that the critical value line possesses a hyperbolic form with on the one hand the minimum practical value as a pole point and on the other hand the minimum degree of axle damping as an asymptote.

18. The method as set forth in claim 16 or claim 17, characterized in that the critical value line is represented by a non-linear regression across the basic data employed, a growth model being employed as the regression model, the growth model having a strictly monotonous, asymptotic form, which is in agreement with the previously made measurements

19. The method as set forth in claim 18, characterized in that as a model the so-called saturation model is utilized, the critical value line being able to be described by the two parameters a and b:

$$Achs_{DGgrenz} = \frac{a * {}^{m_F}/_{m_U}}{b + {}^{m_F}/_{m_U}}$$

wherein:

a is the asymptote, that is to say the minimum degree of axle damping; and,

b is the pole point, i.e. the minimum mass ratio which is physically possible.

20. The method as set forth in claim 14, characterized in that in accordance with the calculation of the degree of axle damping and its incorporation in the diagram of the degree of axle damping to the quotient of the suspended mass to the

non-suspended mass a recommendation is made as regards the necessity of replacement of the shock absorber.

21. The method as set forth in claim 3, characterized in that the data signal, which represents the change with time of the force response at a target frequency, is split by tension-compression stage splitting into a compression stage signal, which is characteristic for the compression stage of the shock absorber and a tension stage signal, which is characteristic for the tension stage of the shock absorber, and in that the compression stage signal and the tension stage signal are subjected to further processing.

22. The method as set forth in claim 21, characterized in that the tension-compression stage split is performed on the basis of a frequency split, the basic frequency being retained and the measured data during the time $\pi/2$ being associated with the positive half wave of the compression stage and the measured data of the negative half wave during the time $\pi/2$ being associated with the tension stage.

23. The method as set forth in claim 21, characterized in that the tension-compression stage split is performed on the basis of the amplitude split, the higher basic frequency of the compression stage and the lower frequency of the tension stage being taken into account for the calculation of the amplitude by displacement of the narrower, positive compression stage half wave by $\delta\varphi$ to the right, and displacement of the wider, negative tension stage half wave by $\delta\varphi$ to the left.

24. The method as set forth in claim 22, characterized in that the signals obtained by the frequency split are subjected to digital filtering, preferably a Fourier transformation and more especially a fast Fourier transformation.

25. The method as set forth in claim 23, characterized in that the signals obtained by the frequency split are subjected to digital filtering, preferably a Fourier transformation and more particularly a fast Fourier transformation and the phase displacements $\delta\varphi$ are added.

26. The method as set forth in claim 24 or claim 25, characterized in that in the fast Fourier transformation only the basic oscillation is filtered out.

27. The method as set forth in any one of the preceding claims, characterized in that after such digital filtering a quality assessment of the processed signal is carried out as to the sine-conformity, wherein for each measured value the relative departure from the sine-signal, which represents the ideal filtered value, is calculated, the transverse sum of all departures is found and those measurement points are rejected, whose mean departure from the ideally filtered value exceeds a predetermined value, such predetermined value preferably being 5%.

28. The method as set forth in any one of the preceding claims, characterized in that the road adherence, i.e. the constant of the wheel with the vibratory platform is examined and in that in the case of a road adherence below a predetermined value, preferably 10%, measurement is terminated.

29. The method as set forth in any one of the preceding claims, characterized in that three complete cycles are measured, n measured values, preferably 256 measured values, being produced per revolution and in that several revolutions, preferably three revolutions are measured and the results employed in forming an average.

30. The method as set forth in any one of the preceding claims, characterized in that the influence of the vibratory platform on the result of measurement is compensated for by recording a frequency curve for the vibratory platform in a dynamic calibration run and the amplitude spectrum is interpolated by a parabolic curve.

31. The method as set forth in any one of the preceding claims, characterized in that the taking of measurements is only started when, after approaching a frequency point, the frequency fluctuations no longer exceed a predetermined tolerance level.

32. The method as set forth in any one of the preceding claims, characterized in that frequency points, at which the frequency fluctuations are not able to be stabilized, are rejected.

33. The method as set forth in any one of the preceding claims, characterized in that prior to a measuring run per se a warm-up phase for the shock absorber takes place in order to warm the shock absorber up to a predetermined temperature for measurement.

34. The method as set forth in claim 33, characterized in that during such warm-up phase the shock absorber is moved in the environment of the upper resonant frequency in order to warm up the damping liquid thereof.

35. The method as set forth in any one of the preceding claims, characterized in that on recording the support points for estimation of parameters a quality test is performed, the departure of the actual signal from the required sinusoidal signal being ascertained and , should the departure be excessively large, the measured value is rejected and the next target frequency will then be taken to be a predetermined amount, for example 1 Hz, below the actual frequency, whereas if the quality of the signal is satisfactory, further scanning is performed in smaller steps, for example of 0.5 Hz.

36. The method as set forth in any one of the preceding claims, characterized in that when the amplitude curve in the compression stage and the tension stage direction exceeds its maximum, the resonant range then recorded is scanned again in small frequency steps, for example of 0.25 Hz, in order to detect the actual resonant frequency with the required resolution.

37. The method as set forth in any one of the preceding claims, characterized in that when the amplitude curve exceeds its point of inflection, the resonant range then assumed is scanned in small steps, for example of 0.25 Hz, in order to detect the actual resonant frequency with the required resolution.

38. The method as set forth in any one of the preceding claims, characterized in that for recording the π/2 frequency f_2 the pitch of scanning is reduced to a small value, for example of 0.25 Hz, when a predetermined phase value, for example of 1.2 rad, is reached and in that the step pitch is reduced to a smaller value, for example 0.1 Hz, when the phase value reaches a larger value, for example of 1.4 rad.

39. The method as set forth in any one of the preceding claims, characterized in that during recording the frequency curve the following steps are performed in the order given:

- Measurement of the static weight;
- Running up to starting frequency of for example 10 Hz;
- Warming up phase of shock absorber;
- Run up of exciting vibration of the vibratory platform to the maximum frequency of for example 35 Hz;
- Recording of support points for the estimation of parameters;
- Recording of resonant frequency f_3;
- recording π/2 frequency f_2 and
- Transmission of data to evaluation system.

40. The method as set forth in any one of the preceding claims, characterized in that for recording the frequency points the following steps are performed in the order given;

- Run up to target frequency;
- Check to see if frequency becomes stable;
- Recording readings;
- Separation of tension from compression stage signal;
- Fourier transformation of the separated signals;

- Quality assessment of the signals;
- Check of road adherence;
- Check to see whether measurements have been recorded for a given number of revolutions;
- Check to see whether measurements conform to the quality standard for at least a fraction of the revolutions;
- Extraction of mean value for the measurements classified as being good;
- Compensation of the effect of the vibratory platform.

41. The method as set forth in any one of the preceding claims, characterized in that:

- Check to see whether the road adherence is within predetermined tolerance limits;
- Extraction of the characteristic frequencies;
- Parameter estimation as regards the calculation of k_R, the spring constant of the tire;
- Check of air pressure in tire;
- Check of signal form of the amplitude spectrum;
- Check on whether the $\pi/2$ phase state is reached;

  Check on whether the equation $f_2 = f_3$ is fulfilled;
- Parameter calculation
- Check to see whether parameters can be calculated;
- Calculation of critical degree of axle damping;
- Assessment of degree of axle damping and issue of recommendation concerning the need for replacement of shock absorber.

42. An apparatus for the performance of the method for the examination of shock absorbers installed as set forth in any one

of the preceding claims on a vehicle using a vibratory platform, on which one wheel of the vehicle is stood and which is able to be reciprocated in a vertical direction at a suitable amplitude and with a variable frequency, an oscillating action is exerted of the wheel, the damping of the shock absorber being derived from the force response of the running gear to the oscillations of the vibratory platform and the relative phase function between the force exerted by the vibratory platform and the movement of the vibratory platform, characterized by shear force sensors on the vibratory platform and a pulse source for angularly equidistant scanning and a means by which the data is split into a compression stage signal and a tension stage signal, and a means in order to supply the compression stage signal and the tension stage signal separately to the means for further processing of the signals.

43. The apparatus as set forth in claim 43, characterized in that the evaluating means assesses the quality of the installed shock absorber by reference to the quotient: suspended mass divided by non-suspended mass, measured values of the degree of axle damping being related to a characteristic curve of a critical degree value of axle damping, which relationship delimits the range of non-acceptable degrees of axle damping.

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FIG 1

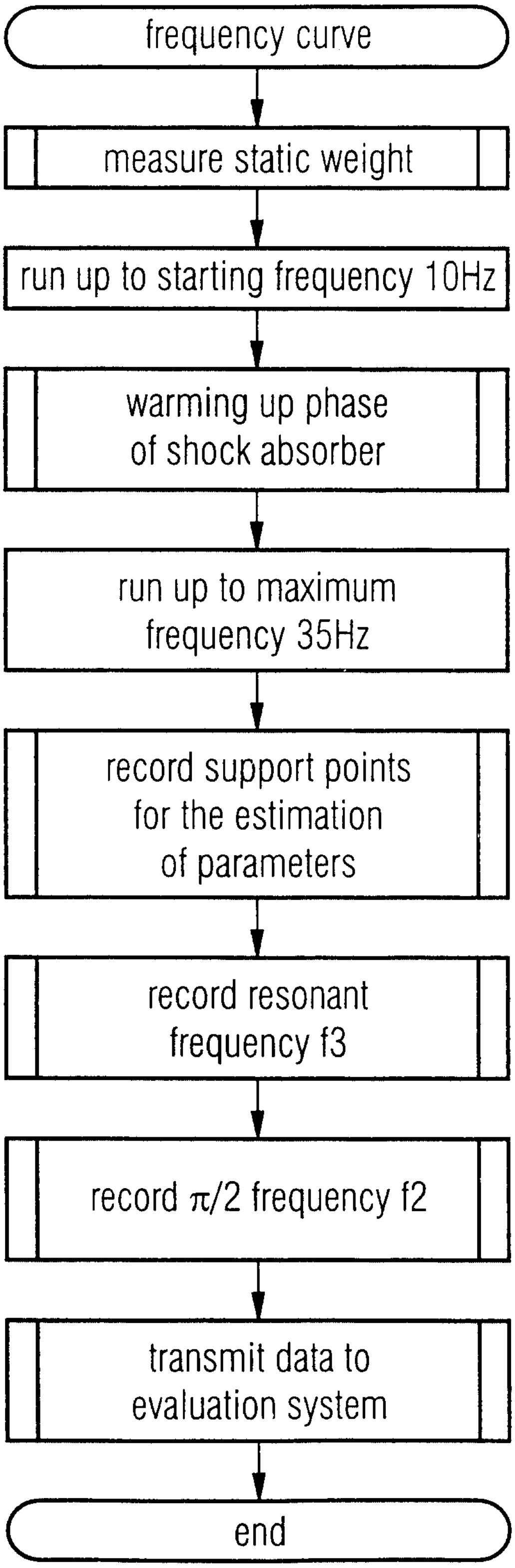

frequency curve
measure static weight
run up to starting frequency 10Hz
warming up phase of shock absorber
run up to maximum frequency 35Hz
record support points for the estimation of parameters
record resonant frequency f3
record π/2 frequency f2
transmit data to evaluation system
end

FIG 2

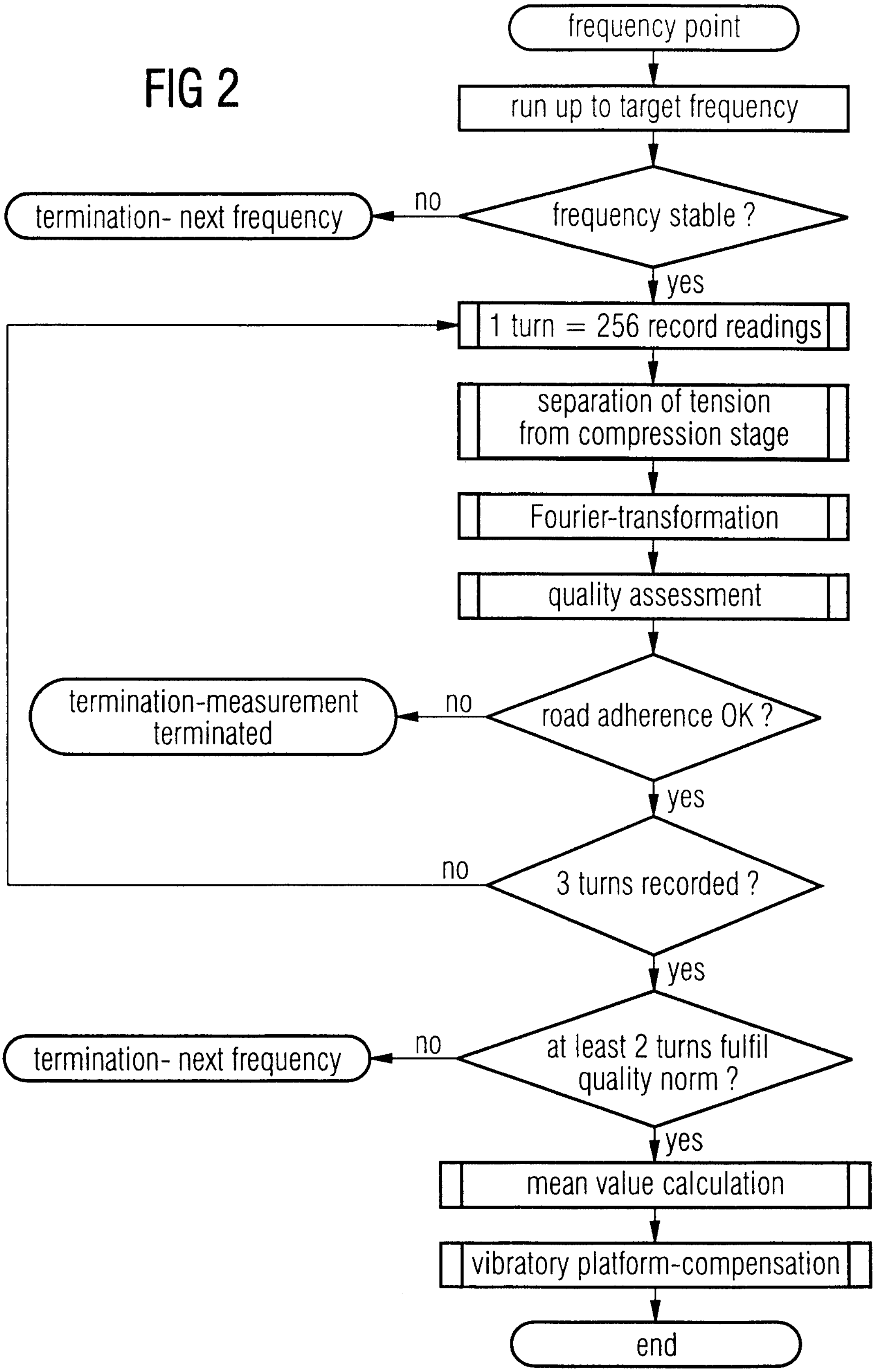

frequency point
run up to target frequency
frequency stable ?
no
termination- next frequency
yes
1 turn = 256 record readings
separation of tension from compression stage
Fourier-transformation
quality assessment
road adherence OK ?
no
termination-measurement terminated
yes
3 turns recorded ?
no
yes
at least 2 turns fulfil quality norm ?
no
termination- next frequency
yes
mean value calculation
vibratory platform-compensation
end

FIG 3A

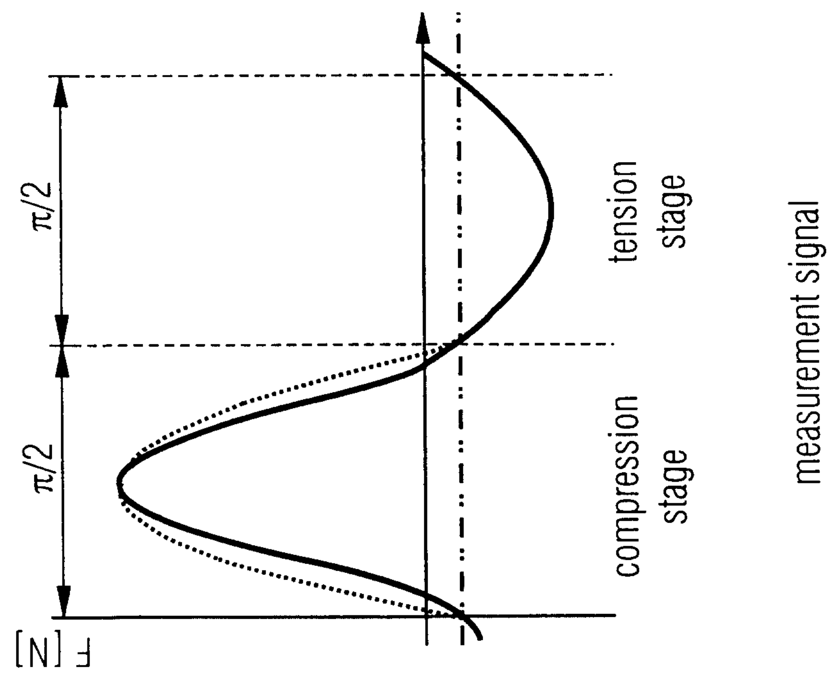

F [N]
π/2
π/2
compression stage
tension stage
measurement signal

FIG 3B

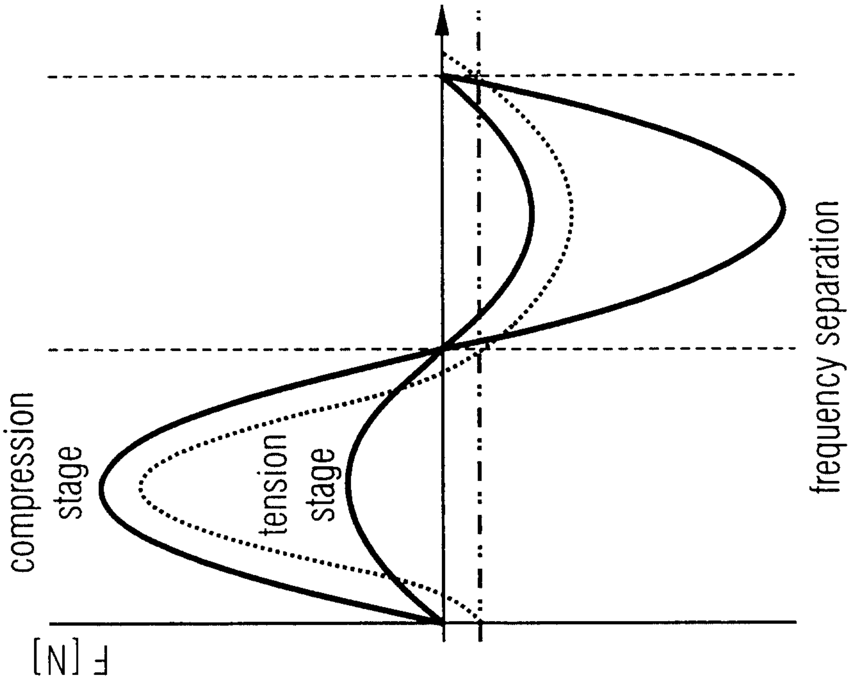

F [N]
compression stage
tension stage
frequency separation

FIG 4A

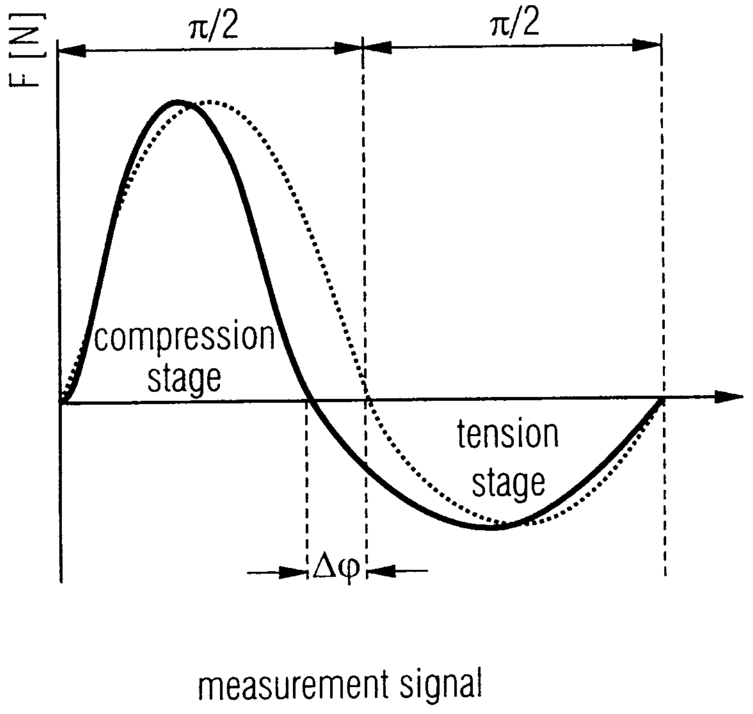

F [N]
π/2
π/2
compression
stage
tension
stage
Δφ
measurement signal

FIG 4B

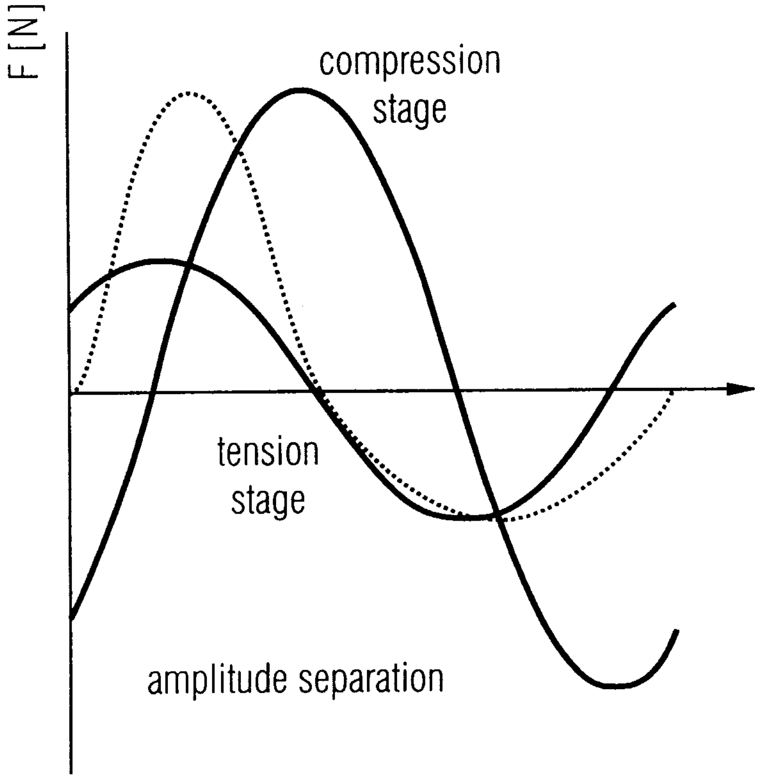

F [N]
compression
stage
tension
stage
amplitude separation

FIG 5A

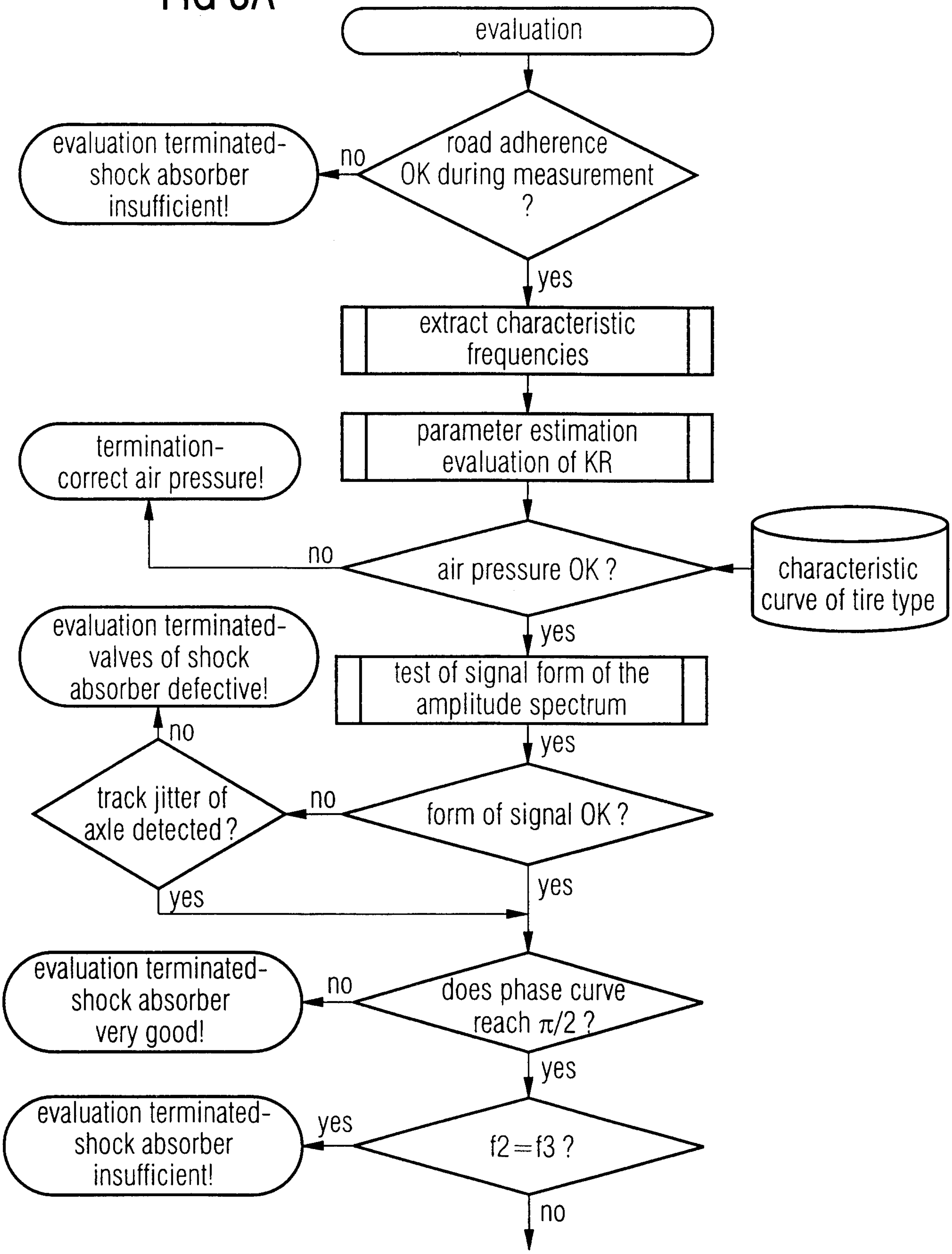

evaluation
road adherence OK during measurement ?
no
evaluation terminated- shock absorber insufficient!
yes
extract characteristic frequencies
parameter estimation evaluation of KR
air pressure OK ?
no
termination- correct air pressure!
characteristic curve of tire type
yes
test of signal form of the amplitude spectrum
yes
form of signal OK ?
no
track jitter of axle detected ?
no
evaluation terminated- valves of shock absorber defective!
yes
yes
does phase curve reach π/2 ?
no
evaluation terminated- shock absorber very good!
yes
f2=f3 ?
yes
evaluation terminated- shock absorber insufficient!
no

FIG 5B

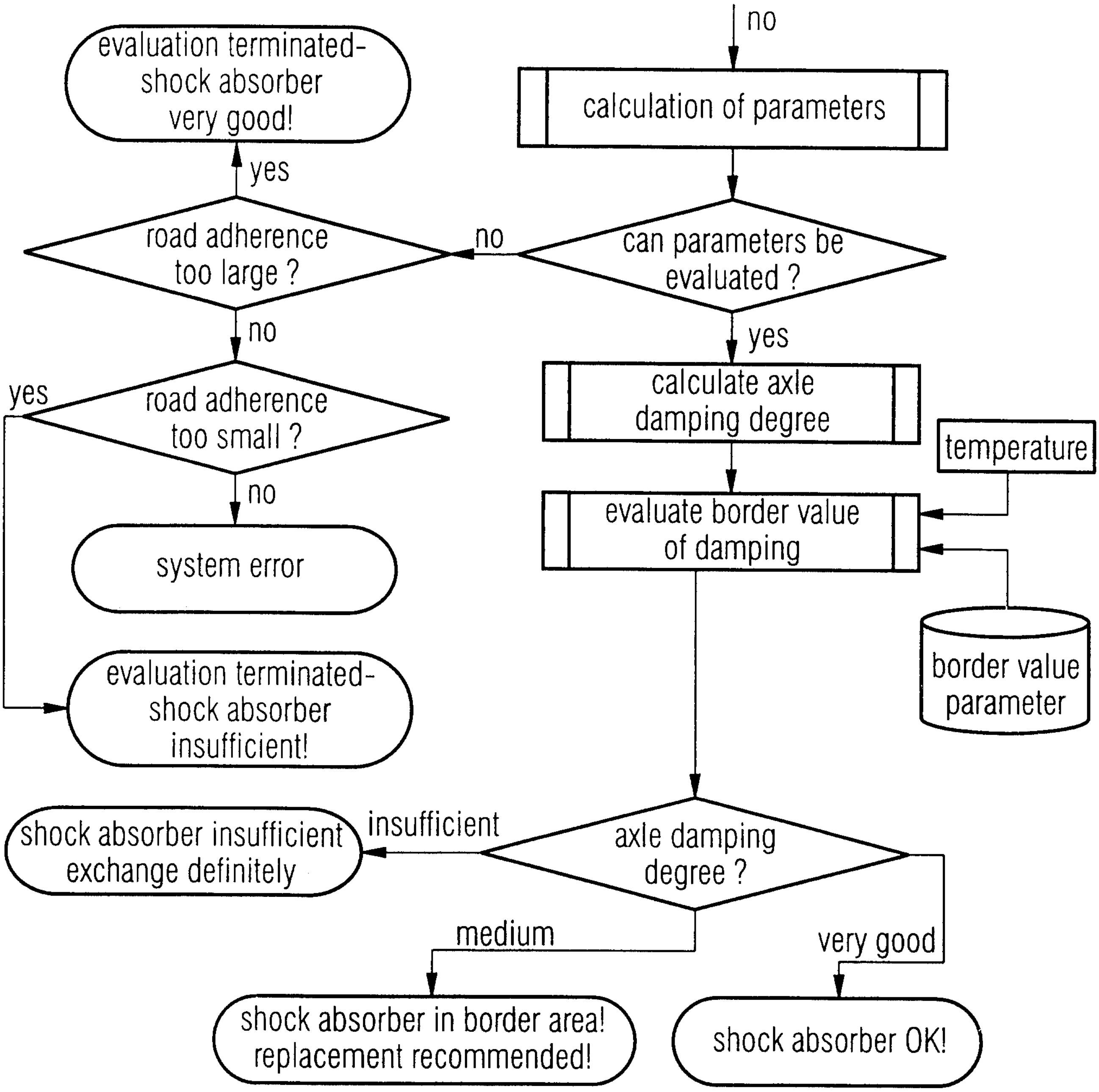

no
calculation of parameters
can parameters be evaluated ?
no
yes
road adherence too large ?
yes
evaluation terminated-shock absorber very good!
no
road adherence too small ?
yes
no
system error
evaluation terminated-shock absorber insufficient!
calculate axle damping degree
temperature
evaluate border value of damping
border value parameter
axle damping degree ?
insufficient
shock absorber insufficient exchange definitely
medium
shock absorber in border area! replacement recommended!
very good
shock absorber OK!

FIG 6

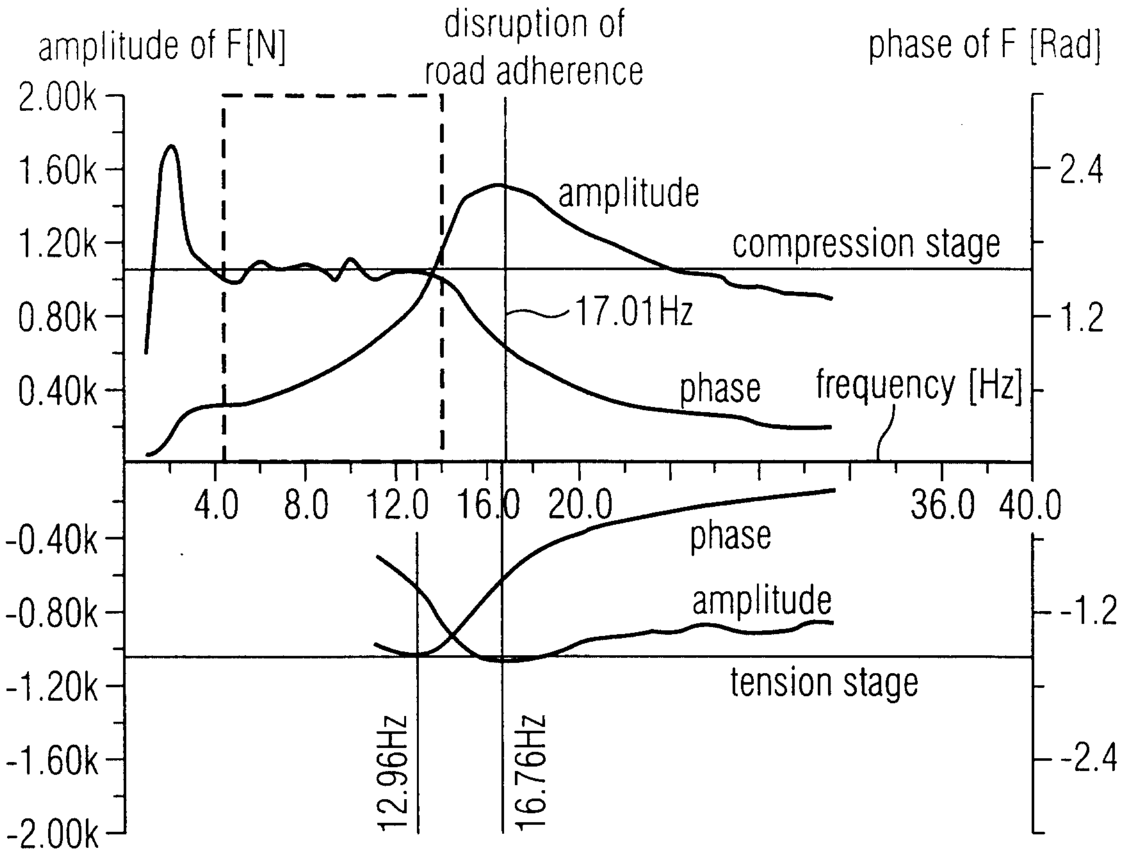

amplitude of F[N]
disruption of road adherence
phase of F [Rad]
2.00k
1.60k
1.20k
0.80k
0.40k
-0.40k
-0.80k
-1.20k
-1.60k
-2.00k
2.4
1.2
-1.2
-2.4
amplitude
compression stage
17.01Hz
phase
frequency [Hz]
4.0
8.0
12.0
16.0
20.0
36.0
40.0
phase
amplitude
tension stage
12.96Hz
16.76Hz

FIG7

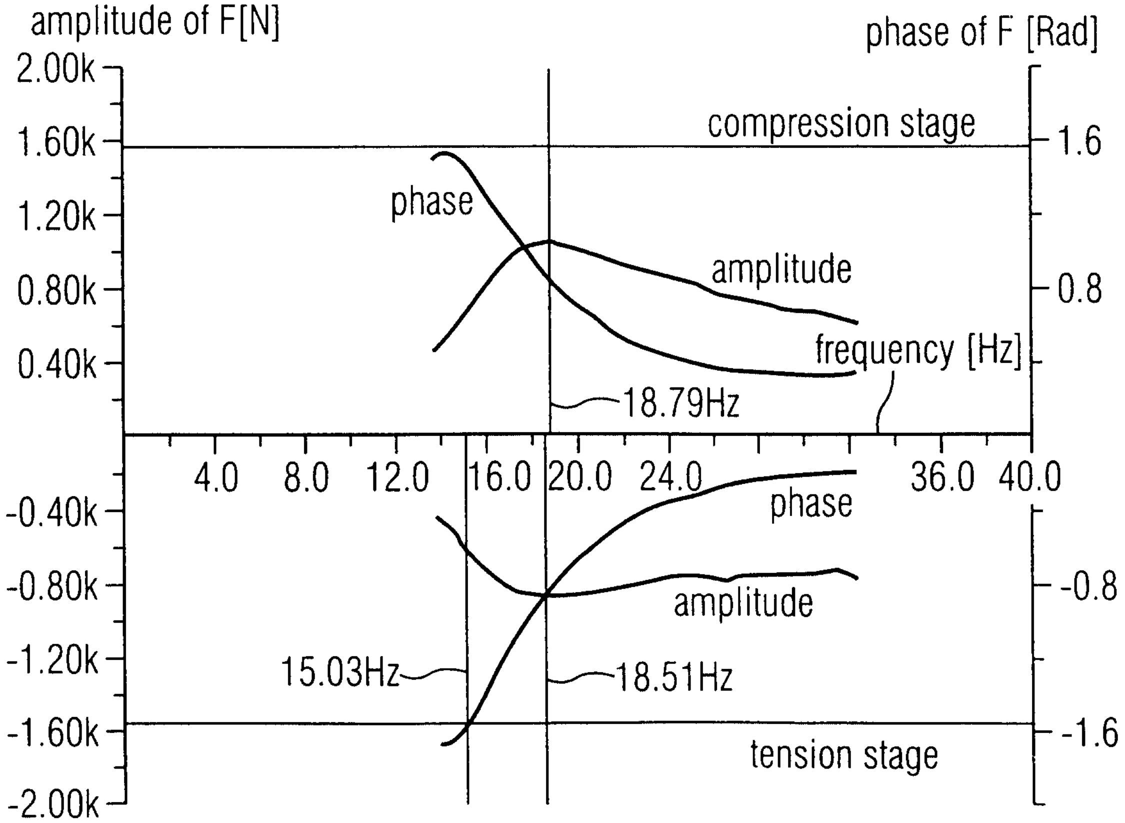

amplitude of F[N]
phase of F [Rad]
2.00k
1.60k
1.20k
0.80k
0.40k
-0.40k
-0.80k
-1.20k
-1.60k
-2.00k
1.6
0.8
-0.8
-1.6
compression stage
phase
amplitude
frequency [Hz]
18.79Hz
4.0
8.0
12.0
16.0
20.0
24.0
36.0
40.0
phase
amplitude
15.03Hz
18.51Hz
tension stage

FIG 8

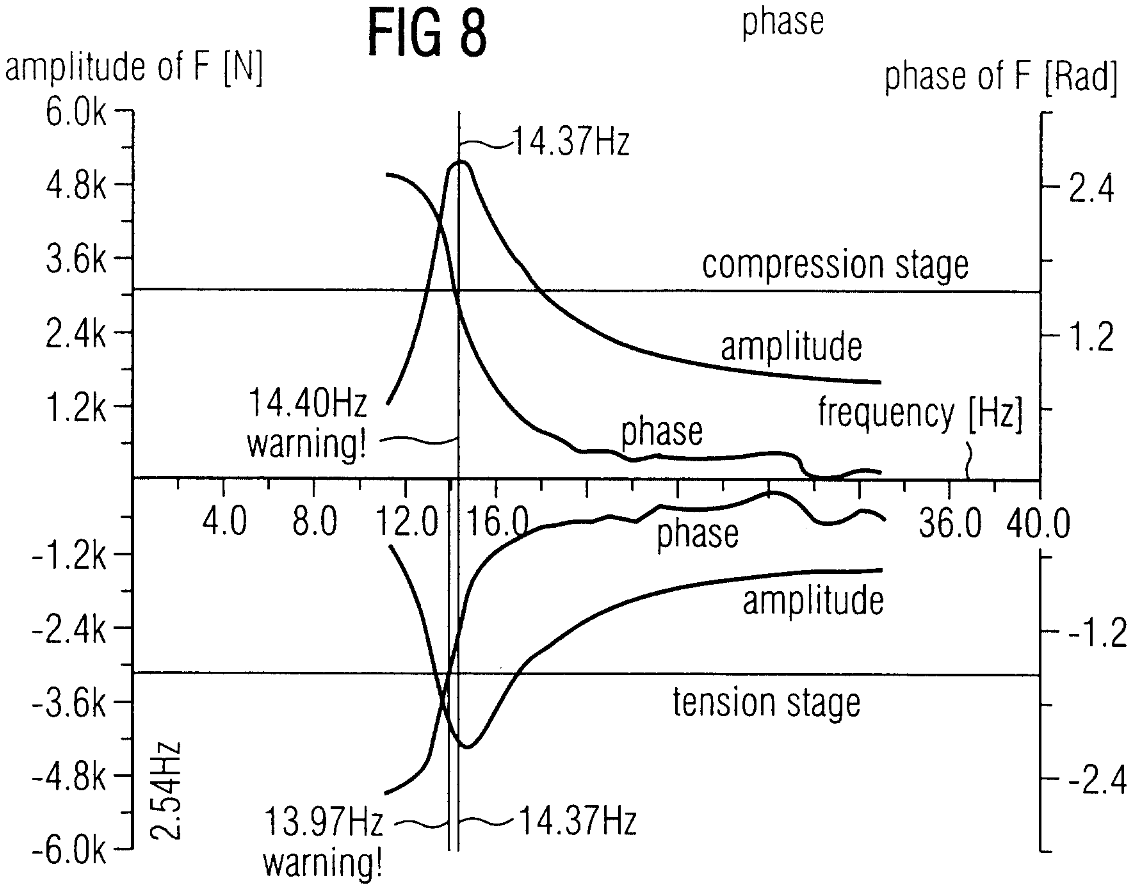

phase
amplitude of F [N]
phase of F [Rad]
6.0k
4.8k
3.6k
2.4k
1.2k
-1.2k
-2.4k
-3.6k
-4.8k
-6.0k
2.4
1.2
-1.2
-2.4
14.37Hz
compression stage
amplitude
14.40Hz
warning!
phase
frequency [Hz]
4.0
8.0
12.0
16.0
36.0
40.0
phase
amplitude
tension stage
2.54Hz
13.97Hz
warning!
14.37Hz

FIG9

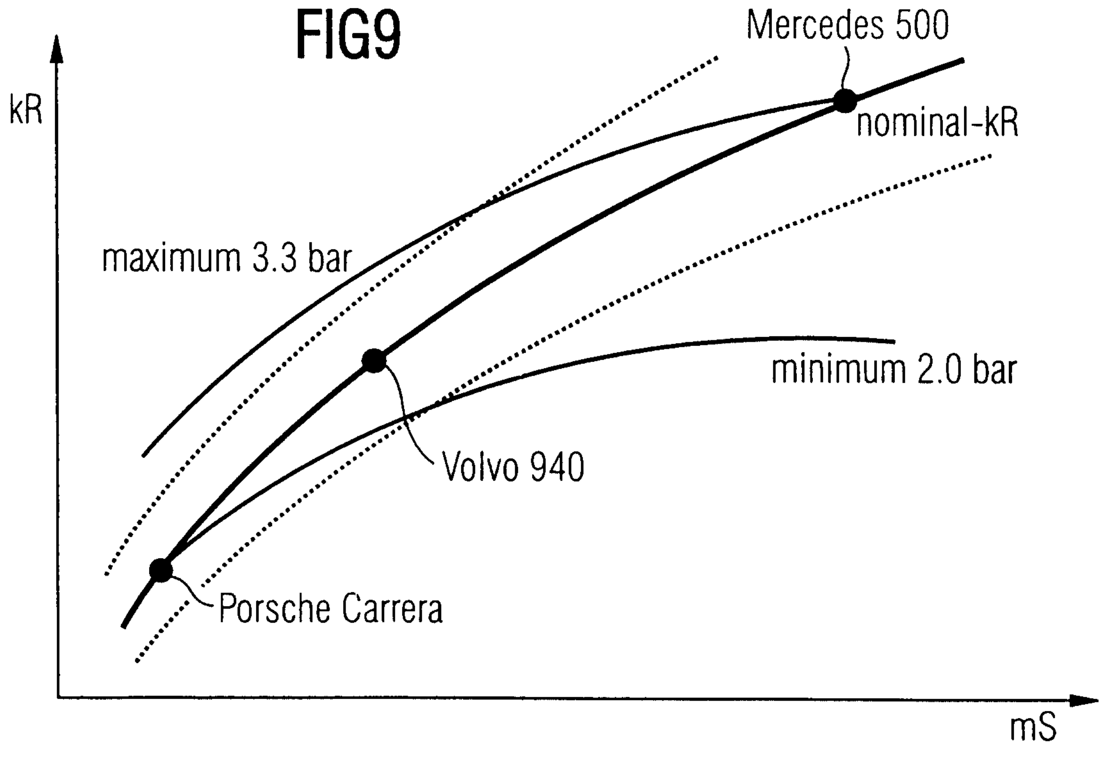

Mercedes 500
kR
nominal-kR
maximum 3.3 bar
minimum 2.0 bar
Volvo 940
Porsche Carrera
mS

FIG10

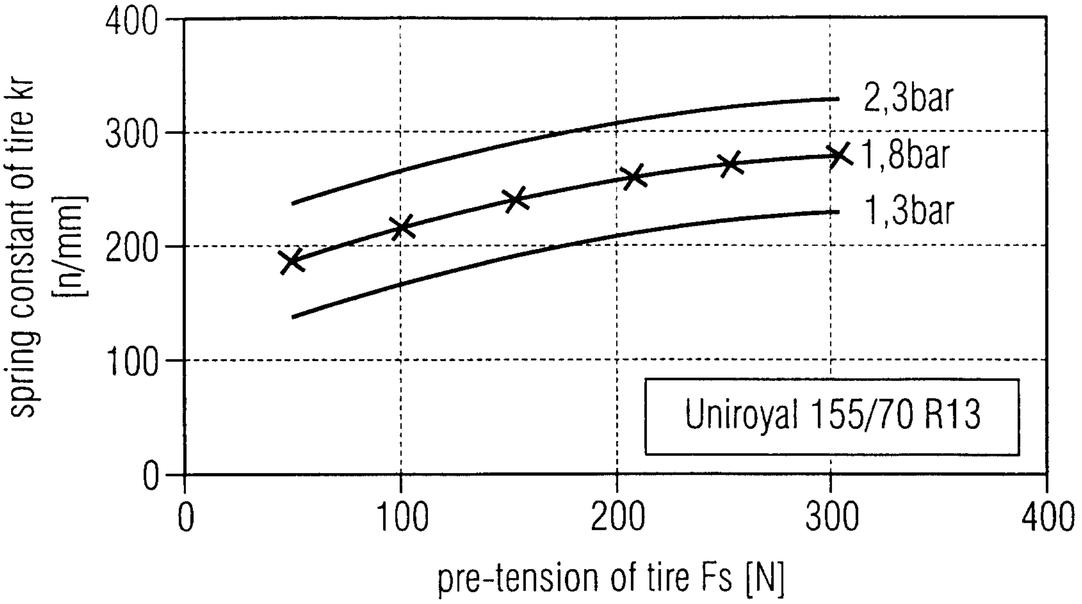

400
300
200
100
0
spring constant of tire kr [n/mm]
2,3bar
1,8bar
1,3bar
Uniroyal 155/70 R13
0
100
200
300
400
pre-tension of tire Fs [N]

FIG11

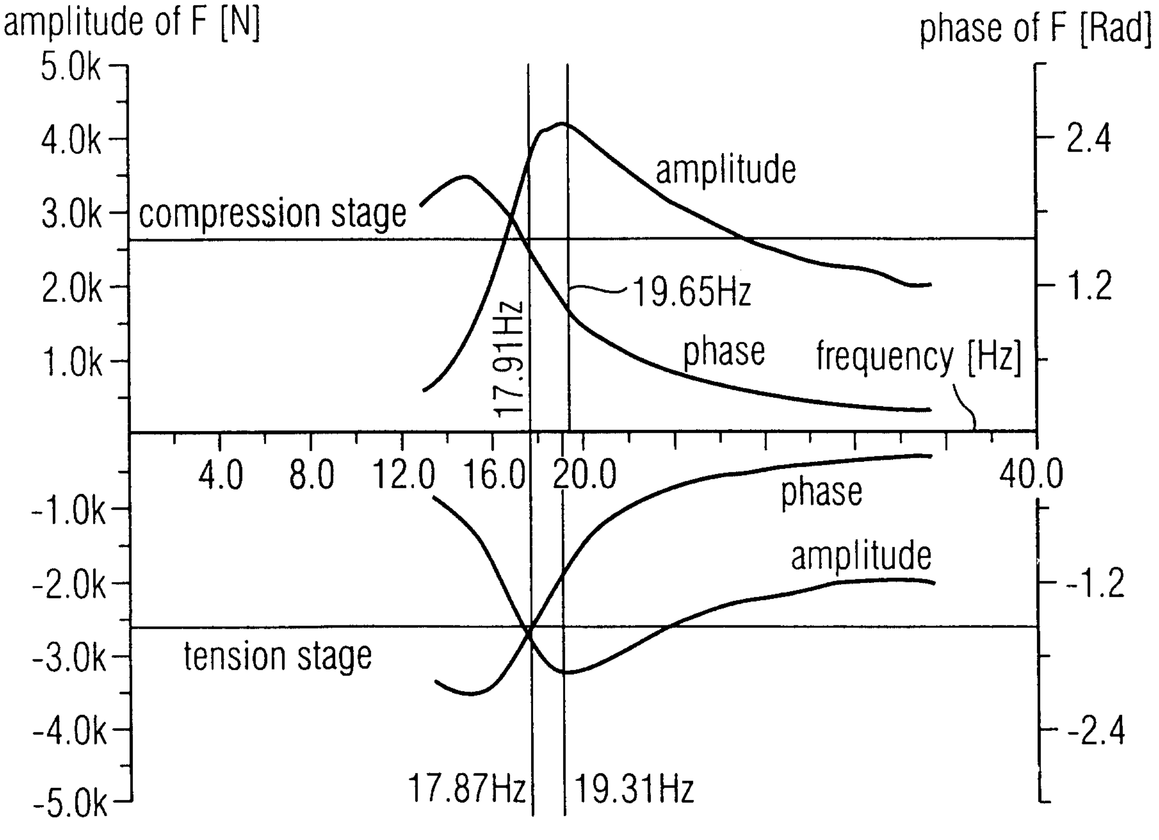

amplitude of F [N]
phase of F [Rad]
5.0k
4.0k
3.0k
2.0k
1.0k
-1.0k
-2.0k
-3.0k
-4.0k
-5.0k
2.4
1.2
-1.2
-2.4
compression stage
amplitude
19.65Hz
17.91Hz
phase
frequency [Hz]
4.0
8.0
12.0
16.0
20.0
40.0
phase
amplitude
tension stage
17.87Hz
19.31Hz

FIG12

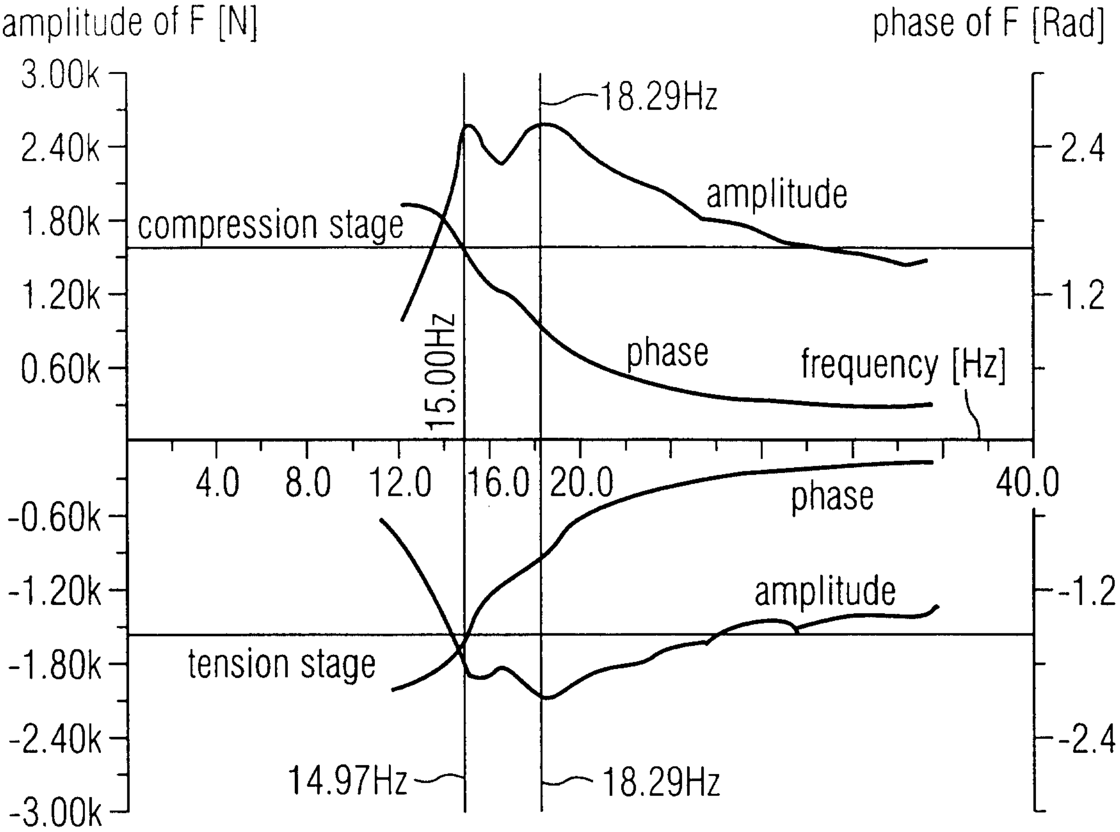

amplitude of F [N]
phase of F [Rad]
3.00k
2.40k
1.80k
1.20k
0.60k
-0.60k
-1.20k
-1.80k
-2.40k
-3.00k
2.4
1.2
-1.2
-2.4
18.29Hz
amplitude
compression stage
15.00Hz
phase
frequency [Hz]
4.0
8.0
12.0
16.0
20.0
40.0
phase
amplitude
tension stage
14.97Hz
18.29Hz

FIG13

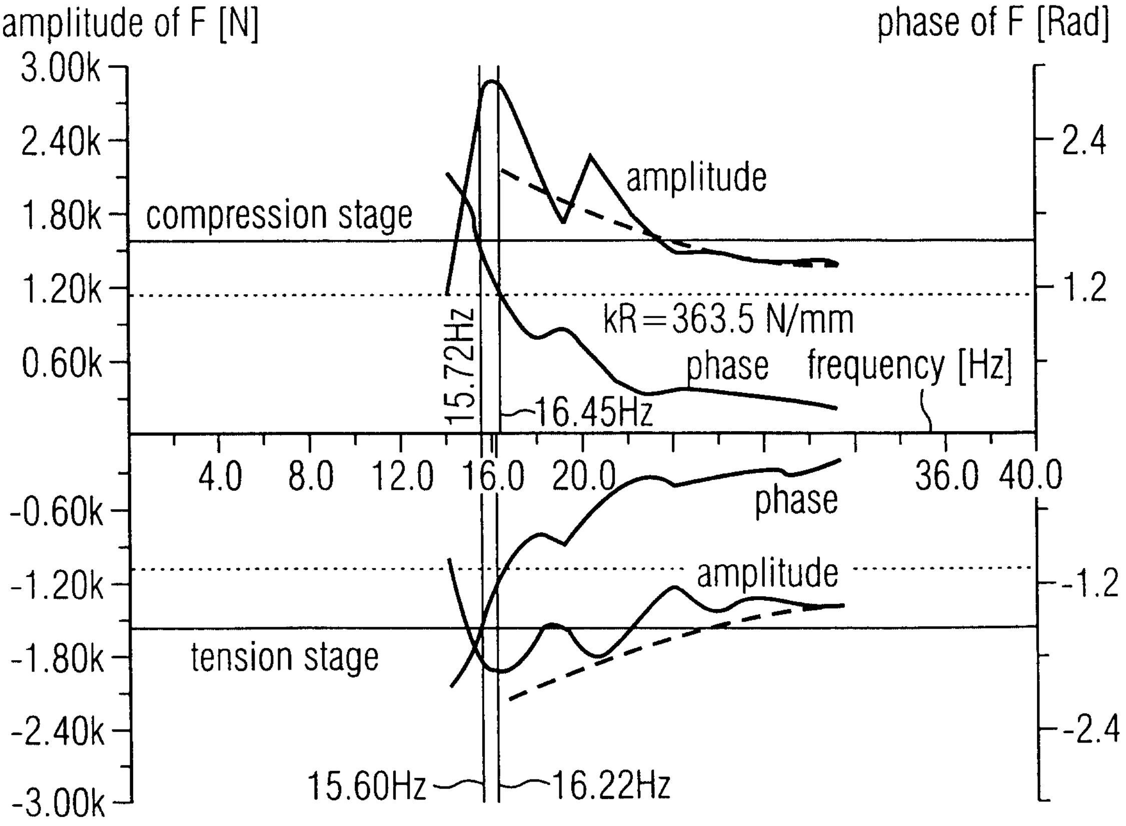

amplitude of F [N]
phase of F [Rad]
3.00k
2.40k
1.80k
1.20k
0.60k
-0.60k
-1.20k
-1.80k
-2.40k
-3.00k
2.4
1.2
-1.2
-2.4
amplitude
compression stage
kR=363.5 N/mm
15.72Hz
phase
frequency [Hz]
16.45Hz
4.0
8.0
12.0
16.0
20.0
36.0
40.0
phase
amplitude
tension stage
15.60Hz
16.22Hz

FIG 14

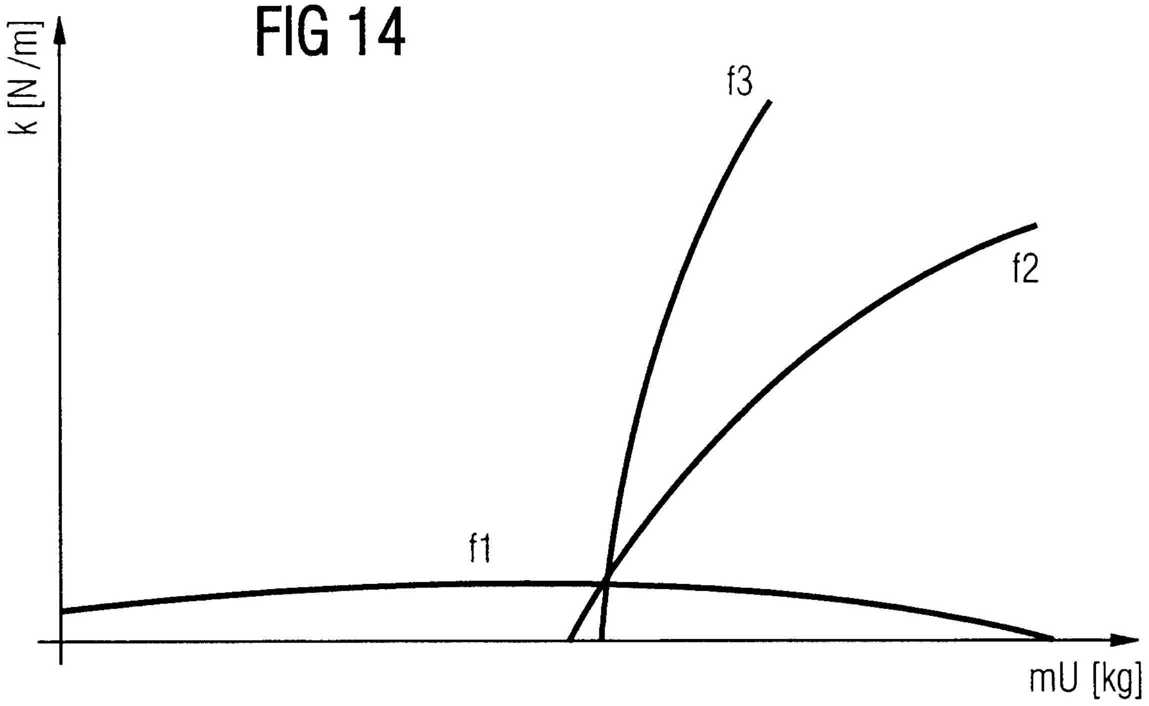


FIG 15

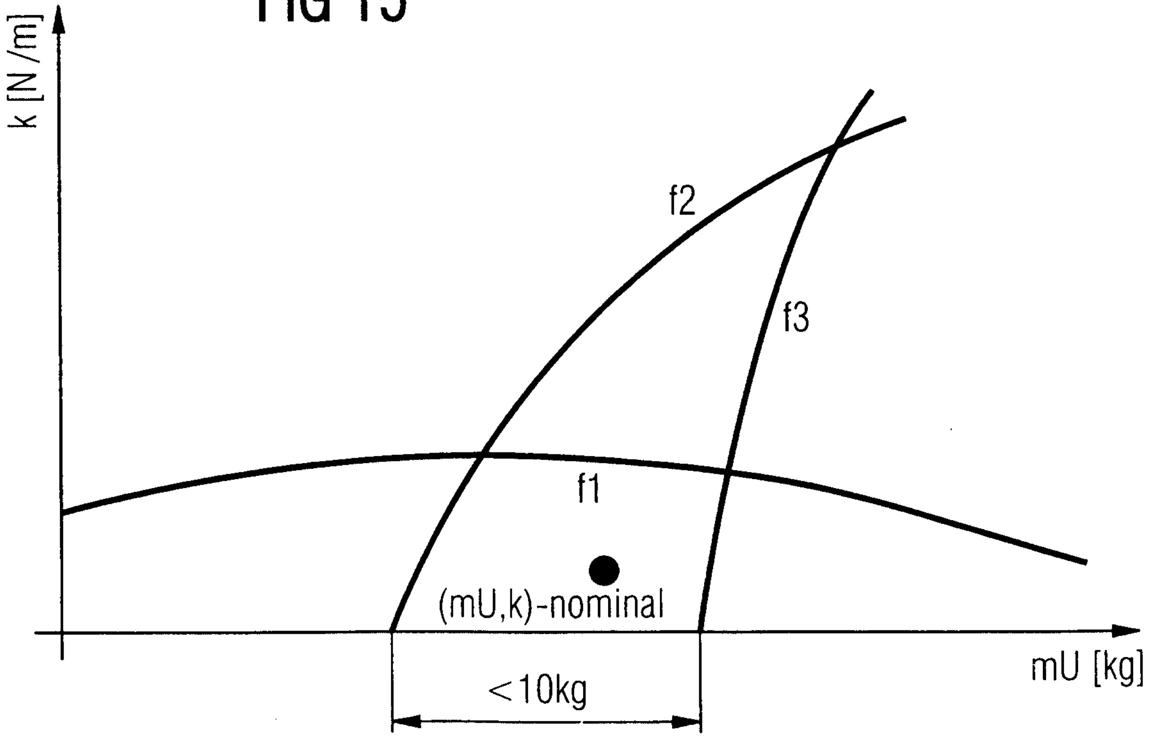


FIG16

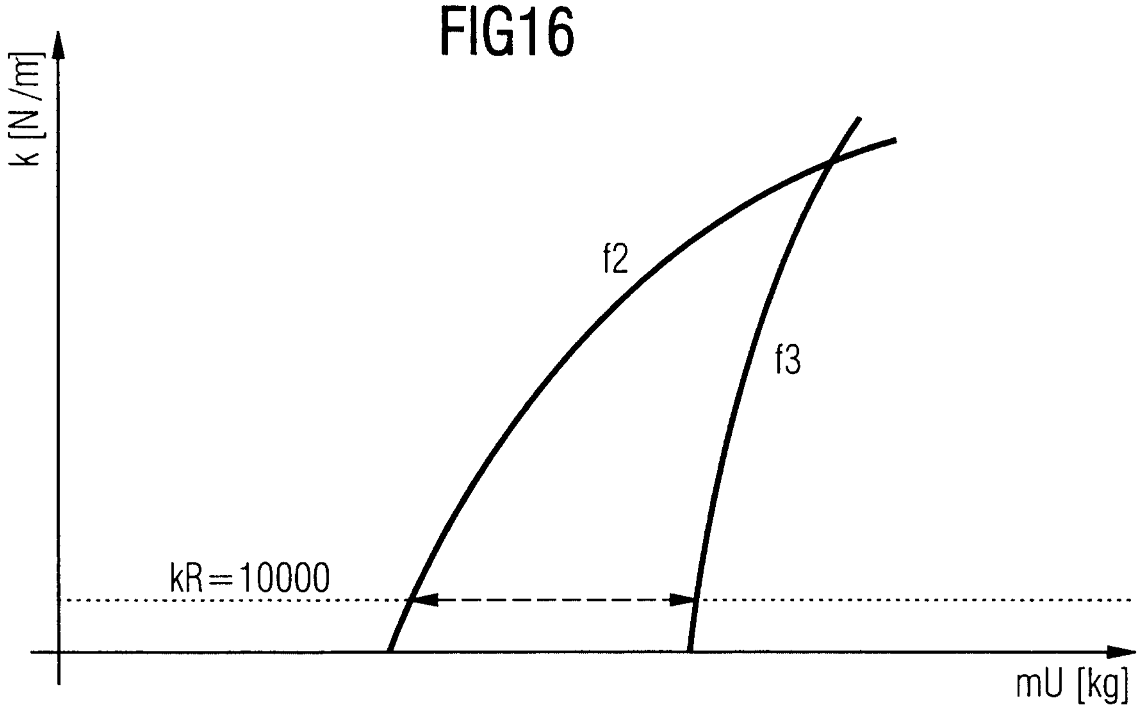

k [N/m]
f2
f3
kR=10000
mU [kg]

FIG17

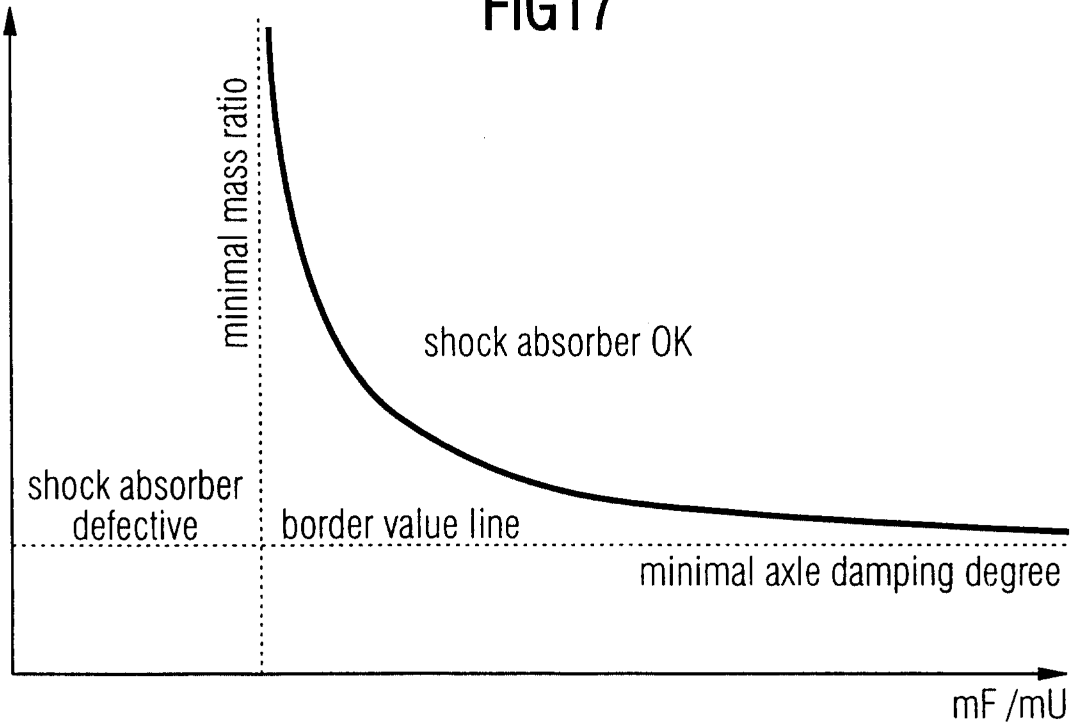

minimal mass ratio
shock absorber OK
shock absorber defective
border value line
minimal axle damping degree
mF /mU

FIG 18

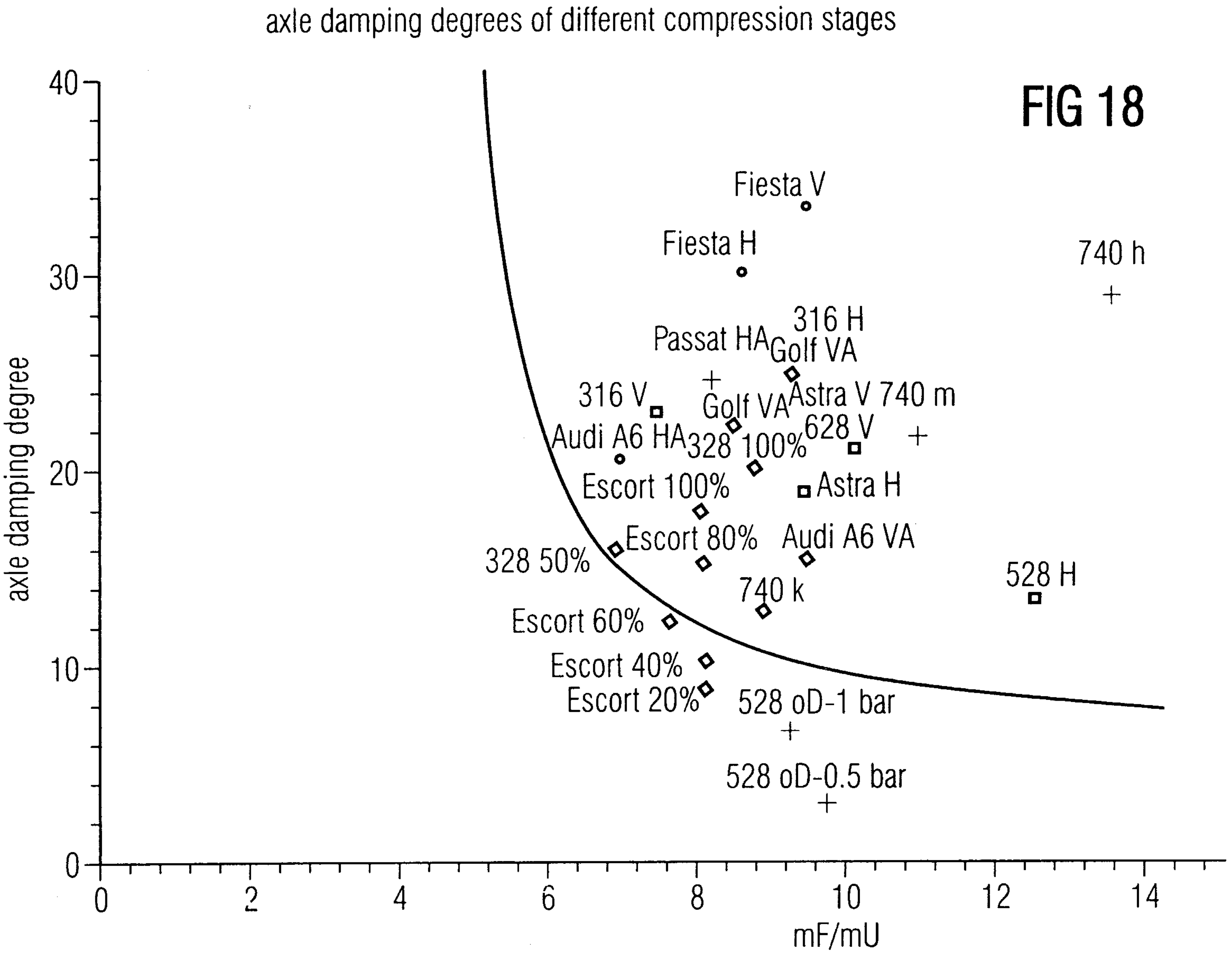

axle damping degrees of different compression stages
axle damping degree
mF/mU
0
10
20
30
40
0
2
4
6
8
10
12
14
Fiesta V
Fiesta H
740 h
316 H
Passat HA
Golf VA
316 V
Golf VA
Astra V
740 m
Audi A6 HA
328 100%
628 V
Escort 100%
Astra H
Escort 80%
Audi A6 VA
328 50%
740 k
528 H
Escort 60%
Escort 40%
Escort 20%
528 oD-1 bar
528 oD-0.5 bar

FIG 19

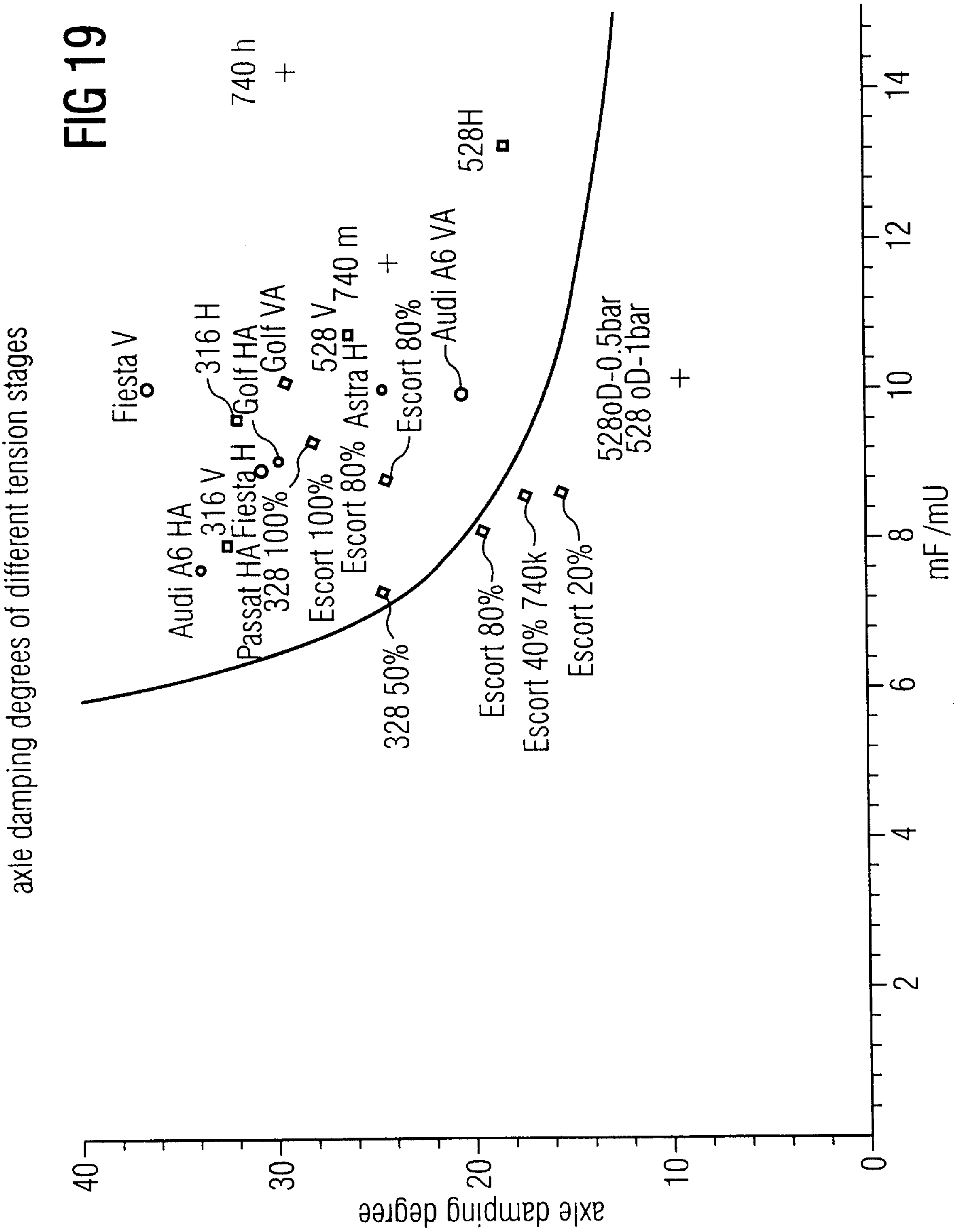

axle damping degrees of different tension stages
axle damping degree
mF /mU
0
10
20
30
40
2
4
6
8
10
12
14
Fiesta V
316 H
Golf HA
Golf VA
528 V
740 m
740 h
528H
Audi A6 HA
316 V
Fiesta H
Passat HA
328 100%
Escort 100%
Escort 80%
Astra H
Escort 80%
Audi A6 VA
328 50%
Escort 80%
Escort 40% 740k
Escort 20%
528oD-0.5bar
528 oD-1bar

FIG 20

pressure dependance of axle damping degree

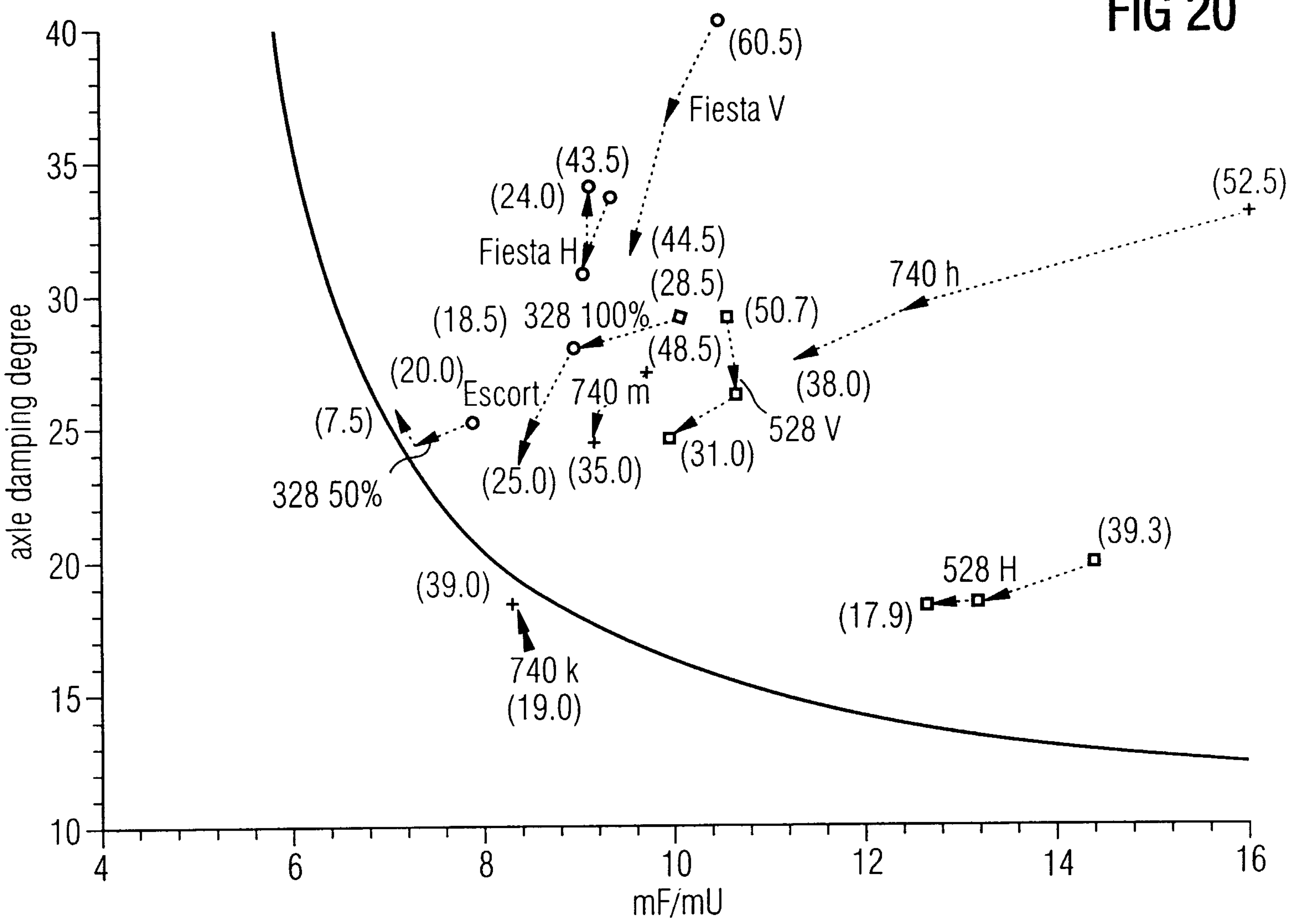


FIG 21

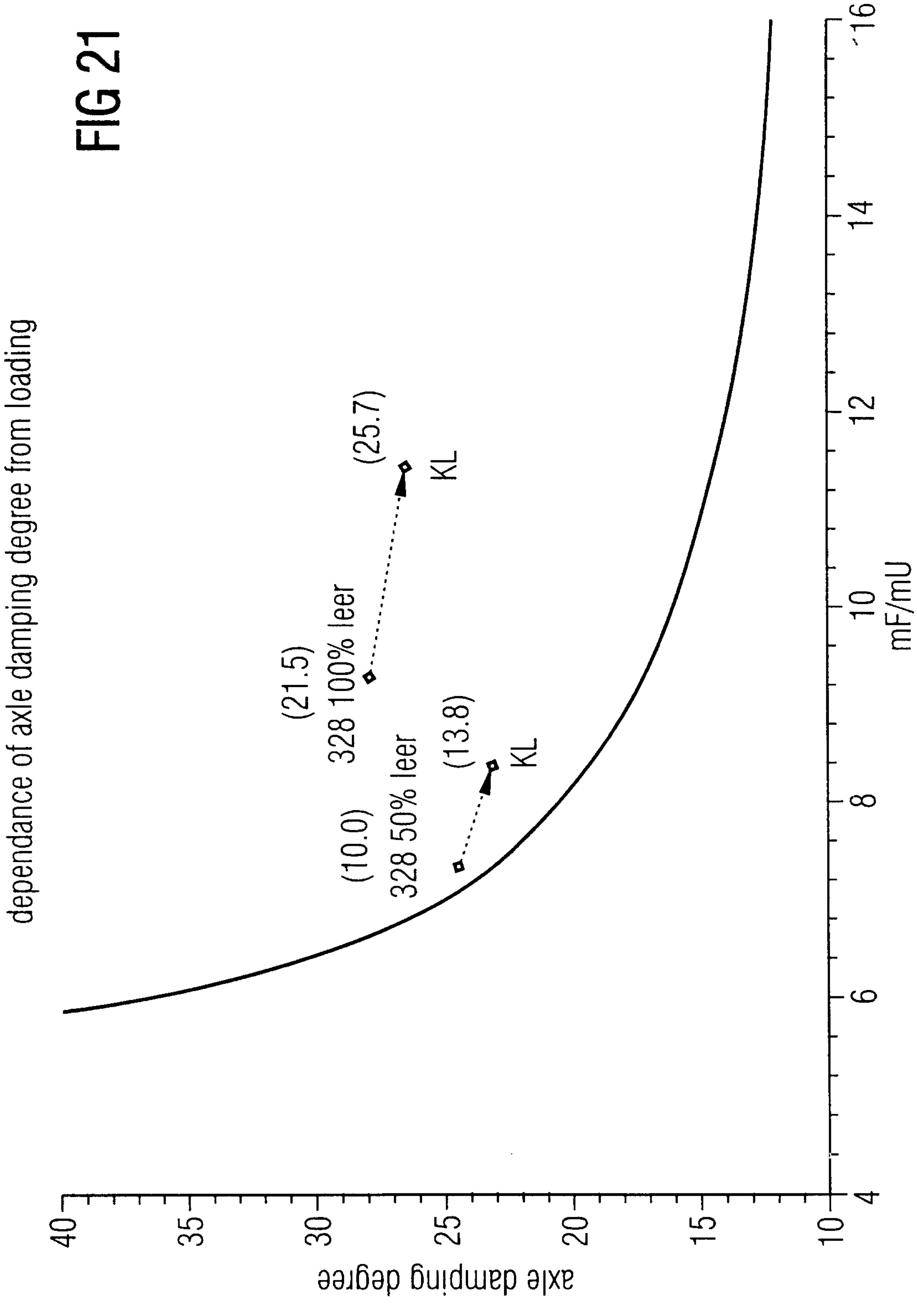

dependance of axle damping degree from loading

FIG 22

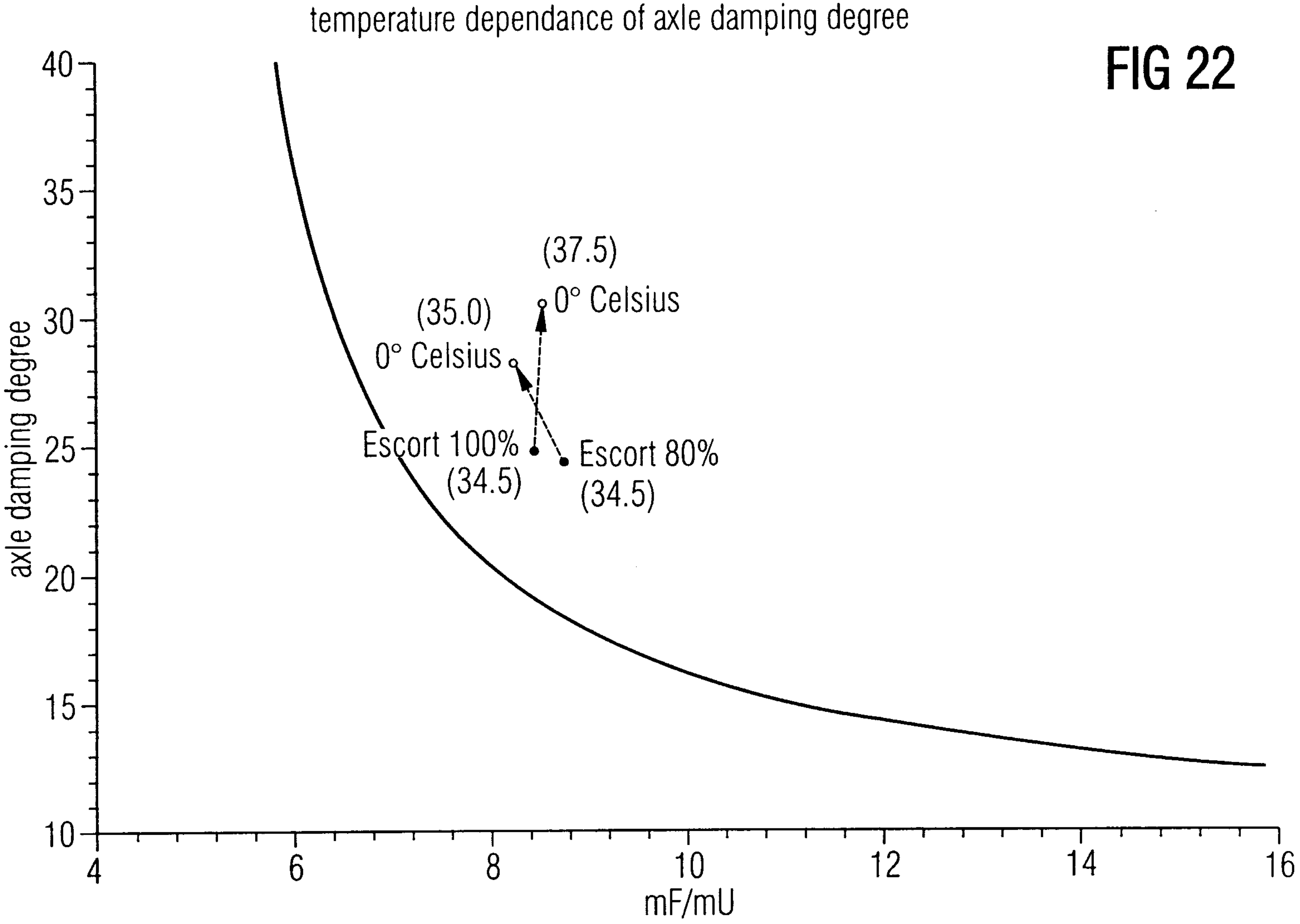

temperature dependance of axle damping degree
axle damping degree
40
35
30
25
20
15
10
4
6
8
10
12
14
16
mF/mU
(37.5)
0° Celsius
(35.0)
0° Celsius
Escort 100%
(34.5)
Escort 80%
(34.5)

FIG 23

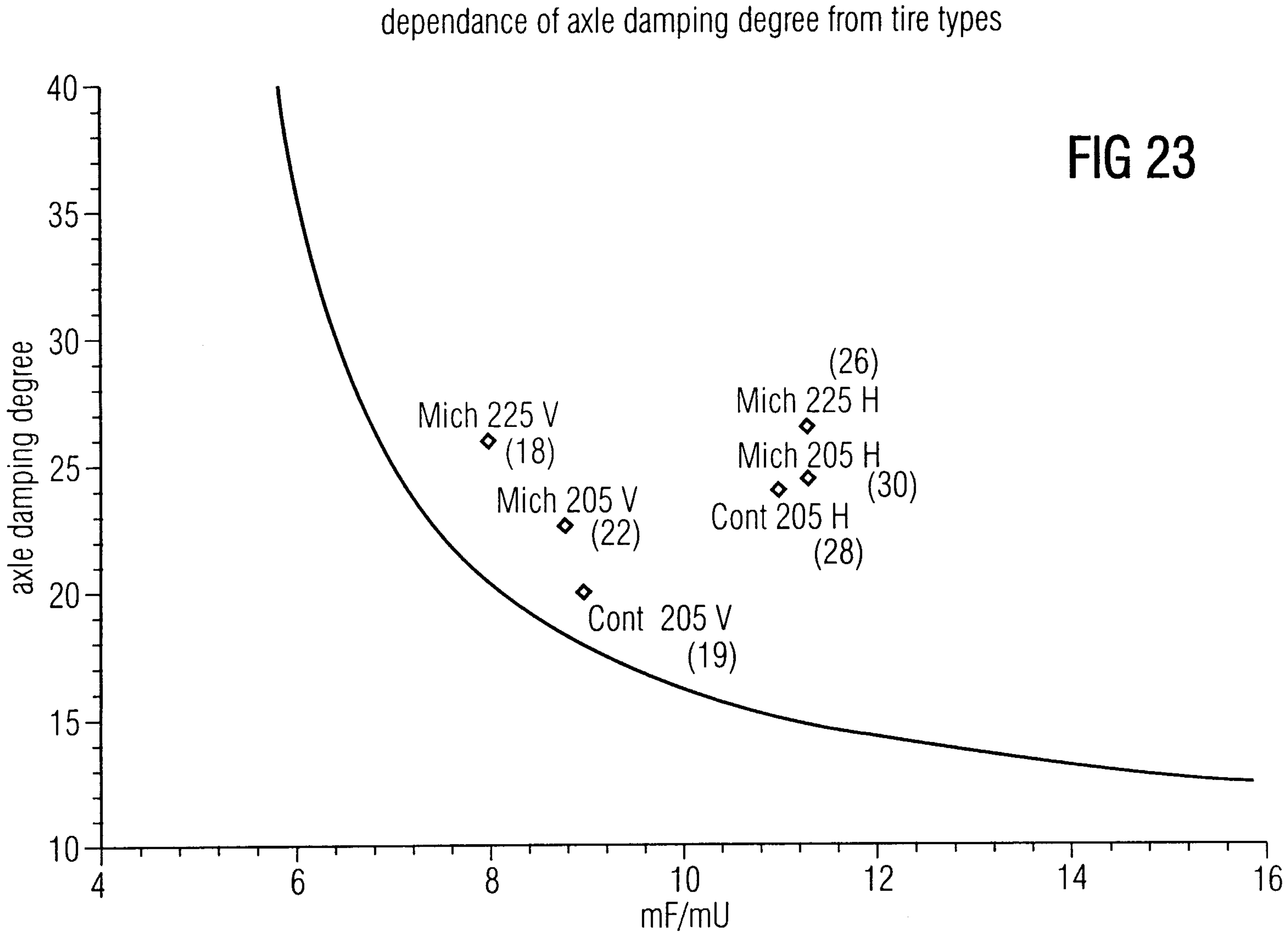

dependance of axle damping degree from tire types
axle damping degree
40
35
30
25
20
15
10
4
6
8
10
12
14
16
mF/mU
Mich 225 V
(18)
Mich 205 V
(22)
Cont 205 V
(19)
(26)
Mich 225 H
Mich 205 H
(30)
Cont 205 H
(28)

FIG 24

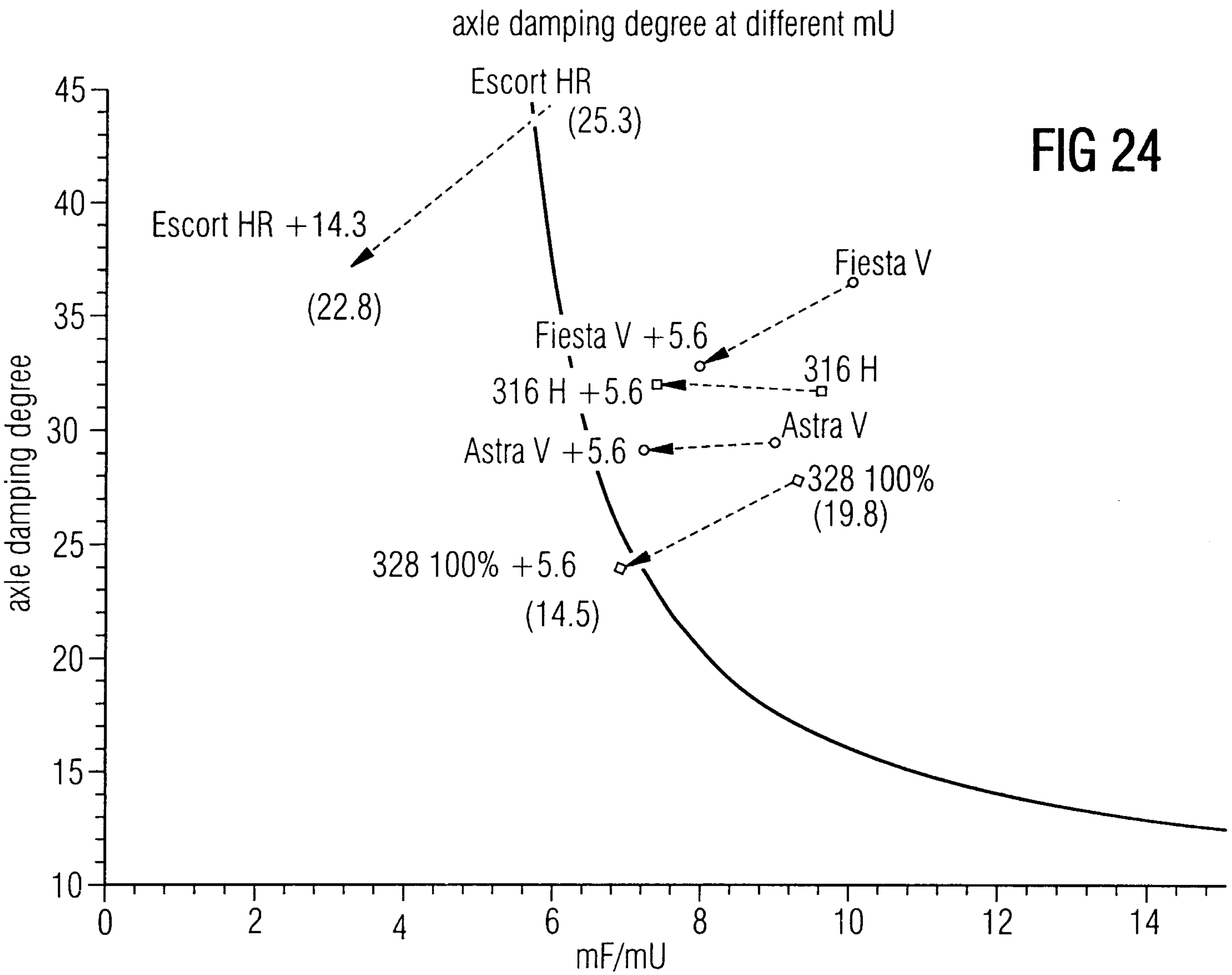
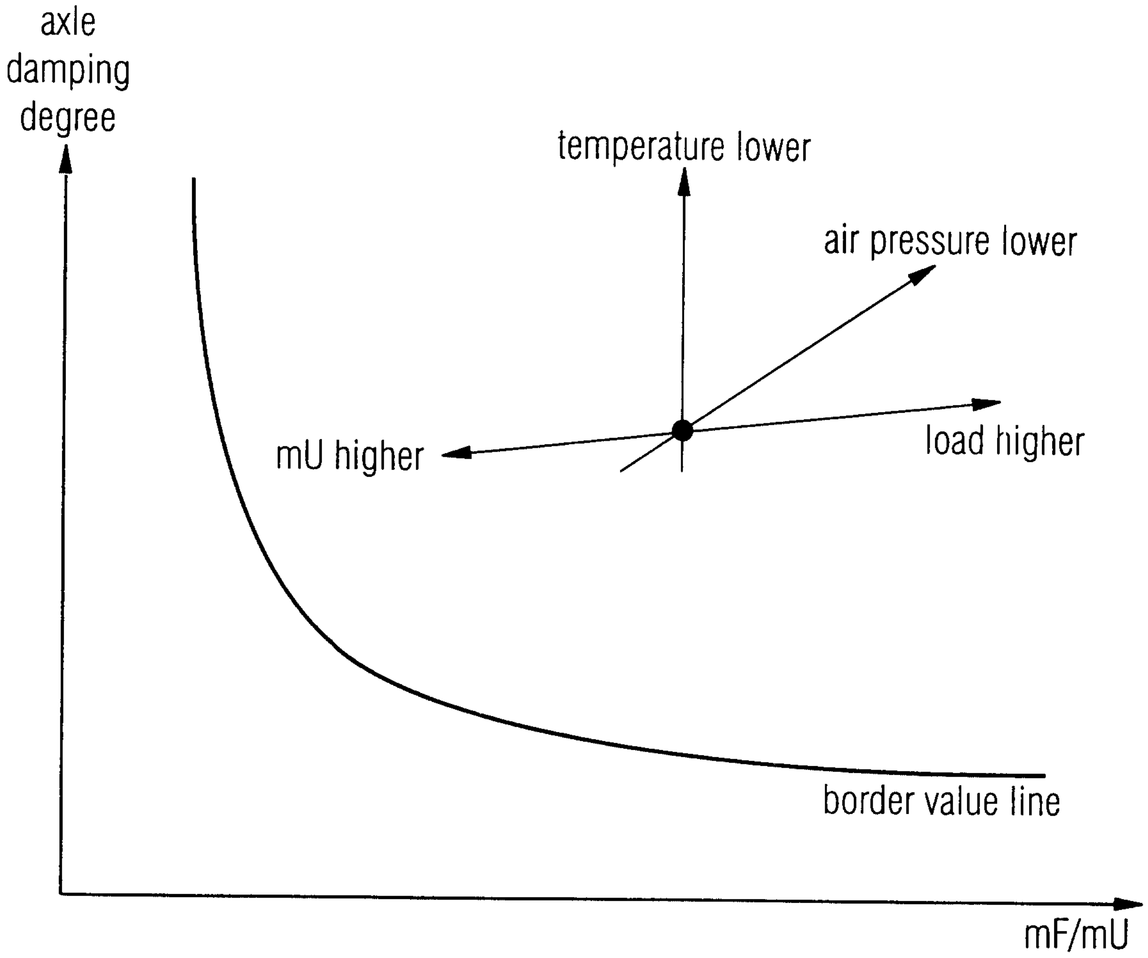
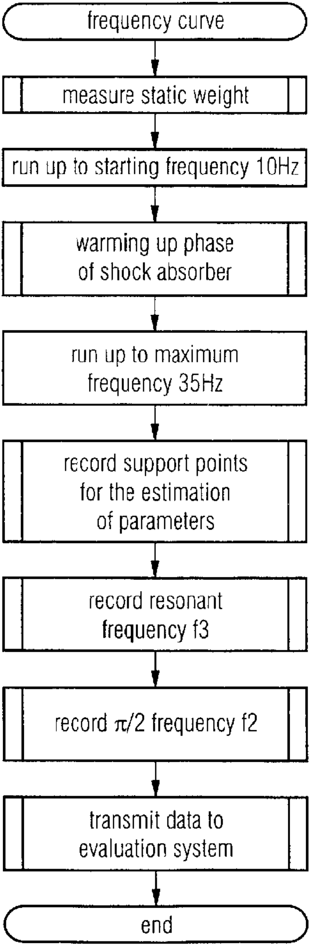

axle damping degree at different mU
axle damping degree
mF/mU
45
40
35
30
25
20
15
10
0
2
4
6
8
10
12
14
Escort HR
(25.3)
Escort HR +14.3
(22.8)
Fiesta V
Fiesta V +5.6
316 H
316 H +5.6
Astra V
Astra V +5.6
328 100%
(19.8)
328 100% +5.6
(14.5)

FIG25


axle
damping
degree
temperature lower
air pressure lower
mU higher
load higher
border value line
mF/mU


frequency curve
measure static weight
run up to starting frequency 10Hz
warming up phase of shock absorber
run up to maximum frequency 35Hz
record support points for the estimation of parameters
record resonant frequency f3
record π/2 frequency f2
transmit data to evaluation system
end