PCT

WORLD INTELLECTUAL PROPERTY ORGANIZATION
International Bureau

INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : B60G 21/02, 21/04, 21/05, 21/06, 11/18 — A1

(11) International Publication Number: WO 97/06971

(43) International Publication Date: 27 February 1997 (27.02.97)

(21) International Application Number: PCT/AU96/00528

(22) International Filing Date: 21 August 1996 (21.08.96)

(30) Priority Data:

PN 4926	21 August 1995 (21.08.95)	AU
PO 0333	7 June 1996 (07.06.96)	AU

(71) Applicant *(for all designated States except US)*: KINETIC LIMITED [AU/AU]; 9 Clark Street, Dunsborough, W.A. 6281 (AU).

(72) Inventor; and
(75) Inventor/Applicant *(for US only)*: HEYRING, Christopher, Brian [AU/AU]; 11 Fern Road, Eagle Bay, W.A. 6281 (AU).

(74) Agent: WATERMARK PATENT & TRADEMARK ATTORNEYS; "Durack Centre", 4th floor, 263 Adelaide Terrace, East Perth, W.A. 6000 (AU).

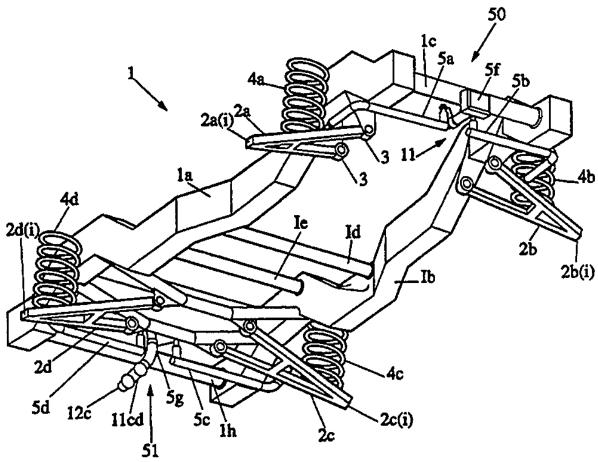
(81) Designated States: AL, AM, AT, AU, AZ, BB, BG, BR, BY, CA, CH, CN, CU, CZ, DE, DK, EE, ES, FI, GB, GE, HU, IL, IS, JP, KE, KG, KP, KR, KZ, LK, LR, LS, LT, LU, LV, MD, MG, MK, MN, MW, MX, NO, NZ, PL, PT, RO, RU, SD, SE, SG, SI, SK, TJ, TM, TR, TT, UA, UG, US, UZ, VN, ARIPO patent (KE, LS, MW, SD, SZ, UG), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, ML, MR, NE, SN, TD, TG).

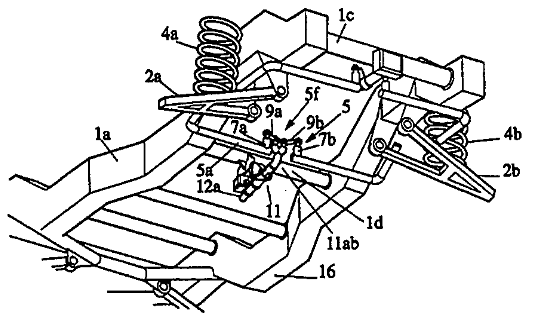
Published
With international search report.

(54) Title: IMPROVEMENTS TO ROLL STABILISATION MECHANISMS IN VEHICULAR SUSPENSION SYSTEMS

(57) Abstract

A vehicular suspension system for a vehicle respectively supported on at least one pair of transversely adjacent front support assemblies and at least one pair of transversely adjacent back support assemblies, the suspension system including a force transmitting means interconnecting the at least one pair of transversely adjacent front support assemblies and a force transmitting means interconnecting the at least one pair of transversely adjacent back support assemblies, the force transmitting means transferring forces between the interconnected support assemblies, wherein each force transmitting means includes connection means for progressively varying the magnitude and direction of the force transferred between the associated support assemblies by the force transmitting means as a function of the relative positions of and the load applied to at least two pairs of the interconnected support assemblies, the connection means being functionally linked such that the magnitude and the direction of the force transmitted between associated support assemblies by each of the force transmitting means is progressively varied to thereby maintain and return the attitude of the vehicle to a position that is at least substantially parallel to the average surface plane supporting the vehicle.





FOR THE PURPOSES OF INFORMATION ONLY

Codes used to identify States party to the PCT on the front pages of pamphlets publishing international applications under the PCT.

AM	Armenia	GB	United Kingdom	MW	Malawi
AT	Austria	GE	Georgia	MX	Mexico
AU	Australia	GN	Guinea	NE	Niger
BB	Barbados	GR	Greece	NL	Netherlands
BE	Belgium	HU	Hungary	NO	Norway
BF	Burkina Faso	IE	Ireland	NZ	New Zealand
BG	Bulgaria	IT	Italy	PL	Poland
BJ	Benin	JP	Japan	PT	Portugal
BR	Brazil	KE	Kenya	RO	Romania
BY	Belarus	KG	Kyrgystan	RU	Russian Federation
CA	Canada	KP	Democratic People's Republic of Korea	SD	Sudan
CF	Central African Republic			SE	Sweden
CG	Congo	KR	Republic of Korea	SG	Singapore
CH	Switzerland	KZ	Kazakhstan	SI	Slovenia
CI	Côte d'Ivoire	LI	Liechtenstein	SK	Slovakia
CM	Cameroon	LK	Sri Lanka	SN	Senegal
CN	China	LR	Liberia	SZ	Swaziland
CS	Czechoslovakia	LT	Lithuania	TD	Chad
CZ	Czech Republic	LU	Luxembourg	TG	Togo
DE	Germany	LV	Latvia	TJ	Tajikistan
DK	Denmark	MC	Monaco	TT	Trinidad and Tobago
EE	Estonia	MD	Republic of Moldova	UA	Ukraine
ES	Spain	MG	Madagascar	UG	Uganda
FI	Finland	ML	Mali	US	United States of America
FR	France	MN	Mongolia	UZ	Uzbekistan
GA	Gabon	MR	Mauritania	VN	Viet Nam

IMPROVEMENTS TO ROLL STABILISATION MECHANISMS IN VEHICULAR SUSPENSION SYSTEMS.

The present invention is generally directed to vehicular suspension systems, and in particular to suspension systems incorporating roll stabilisation mechanisms.

Modern vehicles are normally provided with one or more roll stabilisation or "torsion" bars which transversely link the wheels of one or more axles in order to afford some containment of roll motion to prevent uncomfortable and sometimes dangerous swaying motions when cornering. Roll stabiliser bars are typically manufactured as spring steel bars which permit torsional resilience so that as one wheel on an axle is caused to move in a generally upward or downward direction the other wheel on the same axle is induced to move in a similar direction at the same time. The extent to which the two wheels of a single axle move in a common direction is defined in part by the torsional rigidity of the roll stabiliser bar which transversely couples the two wheels in response to the lateral roll force exerted on the vehicle resulting from cornering.

Vehicles with high centres of gravity (such as trucks which are prone to excessive roll motions) and vehicles which are required to 'corner flat' without exhibiting excessive roll motions (such as sports cars) are normally provided with stiff roll stabiliser bars to prevent roll motion. An unbeneficial consequence of providing stiffer roll stabilisers is that the ride quality becomes harsher as both wheels of an axle become functionally linked (to an extent) and single wheel inputs are therefore not resolved by that single wheel alone which impacts on a bump or pothole.

Luxury passenger vehicles are, therefore, normally equipped with more compliant roll stabiliser bars so that single wheel inputs are absorbed by that single wheel's associated spring and damper unit which are relatively free to move in response to the single input without the additional resistance resulting from the stabiliser's torsional rigidity.

Regardless of the torsional rigidity of the torsion bar, the provisional of such bars do restrict the degree of movement of the wheels relative to each other. This can be a disadvantage in situations where a large degree of opposing vertical wheel motion is required, for example, when travelling over undulating surfaces. The limitations of the wheel movement due to the roll stabiliser bars interconnecting the wheels can lead to significant side to side jerking of the vehicle under such conditions. This movement limitation also restricts the amount of traction that the wheels will have when travelling over such surfaces.

It would be advantageous to have a vehicular suspension system which provides roll stability during cornering and also provides a comfortable ride when travelling in a near straight line or when transversing an undulating surface.

With this in mind, the present invention provides in one aspect a vehicular suspension system for a vehicle respectively supported on at least one pair of transversely adjacent front support assemblies and at least one pair of transversely adjacent back support assemblies, the suspension system including a force transmitting means interconnecting the at least one pair of transversely adjacent front support assemblies and a force transmitting means interconnecting the at least one pair of transversely adjacent back support assemblies, the force transmitting means transferring forces between the interconnected support assemblies, wherein each force transmitting means includes connection means for progressively varying the magnitude and direction of the force transferred between the associated support assemblies by the force transmitting means as a function of the relative positions of and the load applied to at least two pairs of the interconnected support assemblies, the connection means being functionally linked such that the magnitude and the direction of the force transmitted between associated support assemblies by each of the force transmitting means is progressively varied to thereby maintain and return the attitude of the vehicle to a position that is at least substantially parallel to the average surface plane supporting the vehicle.

Because of the functional linking of the connection means, this can provide "passive" control of the vehicle attitude. The suspension system can be self correcting without the need of any external control means. This avoids the need for components such as motion and displacement sensors, electronic control units for processing the sensor signals, and actuating components such as fluid pumps controlled by the electronic control units. Such arrangements are expensive and are relatively slow in their response to changes in surface conditions and vehicle motion.

The force transmitted by the force transmitting means may be a torsional force. To this end means that allow torsional forces to be transmitted may be used. Therefore, each force transmitting means may includes a pair of transverse torsion bars, each torsion bar being respectively connected to a support assembly, the torsion bars being interconnected by the connection means. The torsion bars may be rotatable about their elongate axes, the connection means preferably progressively controlling the axial rotation of the associated torsion bars relative to each other such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars.

According to a preferred embodiment of the present invention, each connection means may provide a mechanical interconnection of the associated pair of torsion bars. The connection means interconnecting a said pair of transversely adjacent front support assemblies and the connection means interconnecting a said pair of transversely adjacent back support assemblies may be functionally linked by a mechanical connection. This mechanical connection may be a longitudinal shaft interconnecting said connection means, each connection means preferably including a pair of linkage members respectively connected at one end thereof to one of the torsion bars, the other end of each pair of linkage members being interconnected to an end of the longitudinal shaft such that torsional forces can be transmitted between said connection means.

According to another preferred embodiment of the present invention, the connection means may alternatively provide a hydraulic connection of the torsion bars. The connection means may be a double acting ram, the ram having a cylinder and a piston assembly separating the cylinder into two fluid chambers therein. The cylinder may be connected to one of the torsion bars, and the piston assembly may be connected to the other torsion bar. A fluid communication may provided between the rams. To this end, the fluid communication may be provided by conduit means connecting the two fluid chambers of the double acting ram of the front torsion bar with the fluid chambers of the double acting ram of the back torsion bars whereby the transfer of fluid between the fluid chambers enables relative displacement between the piston assembly and the cylinder. The fluid chambers may be connected such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the piston assembly within each cylinder allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the piston assembly, while roll motions of the vehicle are reacted by an increase in the pressure differential across the piston assemblies generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle whilst minimising the changes in load on each support assembly Furthermore, fluid supply means for supplying fluid to the conduit means may also be provided so that fluid can be added to one conduit and fluid can be at least substantially simultaneously removed from the other conduit to thereby enable the roll angle of the vehicle to be controlled. This enable a degree of active control of the vehicle attitude if so required. Roll resilience means such as a hydropneumatic accumulator may also be provided in fluid communication with both of the conduit means, said roll resilience means including damping means for damping the rate of roll and isolating means for isolating the roll resilience means to thereby improve the roll control. It should however be noted that such a fluid supply means or roll resilience means is not essential to the operation of the vehicular suspension system of the present invention.

According to a further preferred embodiment of the present invention, the connection means may be a rotary actuation means including a housing supporting a rotor separating the housing into at least two fluid chambers, the housing being alternatively connected to one of the torsion bars, the rotor being connected to the other torsion bar. Conduit means may provide fluid communication between the two fluid chambers of the rotary actuation means of the front torsion bar with the fluid chambers of the rotary actuation means of the back torsion bars. The fluid chambers may be connected such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the rotor within each houising allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the rotor, while roll motions of the vehicle are reacted by an increase in the pressure differential across the rotor generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle whilst minimising the changes in load on each support assembly.

In all of the above described embodiments, the resilient support means may be provided between the support assemblies and the chassis of the vehicle for at least substantially supporting the weight of the vehicle.

Alternatively, a yoke means interconnecting each pair of torsion bars may be provided, and a resilient means connecting the yoke means to the chassis of the vehicle may be provided, the yoke means transferring the average load carried by the associated support assemblies to the resilient means such that the resilient means at least substantially supports at least a portion of the vehicle to thereby permit the vehicle to maintain an at least substantially uniform load on each support assembly regardless of any cross-axle articulation of the support assemblies. The yoke means may be provided by a lever arm respectively extending from each torsion bar, the lever arms being interconnected by a cross member arrangement. The resilient means can interconnect the cross member arrangement with the chassis of the vehicle, the resilient means preferably including a load support ram having an accumulator in fluid communication with the ram to provide said resilient support. Double acting rams may interconnect the yoke means with the chassis of the vehicle, and conduit means may connect corresponding chambers of the rams and valve means for controlling the fluid flow through each conduit means to thereby control the pitch motion of the vehicle. Accumulators in fluid communication with at least one of the conduits may also be provided. Fluid supply means for supplying and removing fluid from said conduit means, sensing means for sensing the attitude of the vehicle, and control means for controlling said fluid supply means may also be provided to thereby allow control of the attitude of the vehicle.

According to yet another embodiment of the present invention, the force transmitting means may include a single transverse torsion bar and the connection means may interconnect the torsion bar to at least one of the associated support assemblies. The connection means may provide a hydraulic connection of the torsion bar to the associated support assembly. Each said connection means may include a double acting ram located at one end of the torsion bar, the ram having a cylinder and a piston assembly separating the cylinder into two fluid chambers therein, the cylinder and the piston assembly being connected between one end of the torsion bar and the adjacent support assembly. The rams may be in fluid communication and the fluid communication may be provided by conduit means respectively connecting the two fluid chambers of the double acting ram of the front torsion bar with the fluid chambers of the double acting ram of the back torsion bars. The fluid chambers may be connected such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the piston assembly within each cylinder allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the piston assembly, while roll motions of the vehicle are reacted by an increase in the pressure differential across the piston assemblies generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle while minimising the changes in load on each support assembly.

The connection means may alternatively be a single acting ram located at each end of the torsion bars, each ram having a cylinder and a piston assembly supported therein to provide a fluid chamber within the cylinder, the cylinder and piston assembly being connected to one of the torsion bars and the adjacent support assembly. A fluid communication may be provided between the rams wherein said fluid communication is provided by conduit means respectively connecting the fluid chamber of each single acting ram of the front torsion bar with the fluid chamber of the longitudinally opposing single acting ram of the back torsion bar, the fluid chambers being connected such that the support assemblies are free to move when undergoing cross-axle articulation, while roll motions of the vehicle is reacted to by the torsion bars while minimising the changes in load on each support assembly.

Alternatively, the connection means may provide a mechanical coupling of the torsion bar.

According to another aspect of the present invention, there is provided a vehicular suspension system for a vehicle respectively supported on at least one pair of transversely adjacent support assemblies, the suspension system including a force transmitting means interconnecting the at least one pair of transversely adjacent support assemblies, the force transmitting means transferring forces between the interconnected support assemblies, wherein each force transmitting means includes connection means for progressively controlling the magnitude and varying the direction of the force transferred between the associated support assemblies by the force transmitting means as a function of the displacement of the associated support assemblies relative to each other and the load applied to the at least one pair of interconnected support assemblies.

The force transmitted by the force transmitting means may be a torsional force. Each force transmitting means may include a pair of transverse torsion bars, each torsion bar being respectively connected to a support assembly, the torsion bars being interconnected by the connecting means. The torsion bars may be rotatable about their elongate axes, the connection means progressively controlling the axial rotation of the associated torsion bars relative to each other such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars. The connection means may provide a mechanical interconnection of the pair of torsion bars although a hydraulic interconnection is also envisaged.

According to another preferred embodiment of the present invention, the connection means may include a pair of linkage members respectively connected at one end thereof to one of the torsion bars, the other end of the linkage members being connected to a stub shaft. Shaft control means may be provided for controlling axial and rotational movement of the stub shaft. The control means may include an axial braking assembly and a rotational braking assembly. The actuation of said braking assemblies may be controlled by electronic control means such that the stub shaft is allowed to move axially in the direction of its elongate axis and is at least substantially restrained from rotation about its elongate axis when the vehicle is primarily undergoing roll conditions, and such that the stub shaft is restrained from moving axially in the direction of its elongate axis when the associated support assemblies are displaced in opposing at least substantially vertical directions.

According to an alternative preferred embodiment of the present invention, the connection means may include a pinion gear assembly interconnecting the torsion bars for selectively allowing and restraining counter-rotation of the torsion bars relative to each other, the pinion gear assembly having a pinion gear respectively connected to an end of each of the torsion bars, with a rotatably supported control pinion bar meshing with both of the pinion gears. Rotation control means may be provided for controlling the rotation of the control pinion gear. The rotation control means may include a rotational braking assembly, the braking assembly being actuated by an electronic control means such that the rotation of each control pinion gear being controlled such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars. Resilient means may be provided at each of the support assemblies for at least substantially supporting the weight of the vehicle.

Resilient means may also be located between the pinion gear assembly and the chassis of the vehicle for at least substantially supporting the weight of the vehicle to thereby permit the vehicle to maintain an at least substantially uniform load on each support assembly. The resilient means may be a hydropneumatic strut having at least one associated accumulator providing damping for the vehicle.

It will be convenient to further describe the invention by reference to the accompanying drawing which illustrate possible embodiments of the invention. Other embodiments of the invention are possible and consequently the particularity of the accompanying drawings is not to be understood as superseding the generality of the preceding description of the invention.

Figure 1a. is an isometric view of the underside of a vehicle chassis showing a first embodiment of the suspension system according to the present invention;

Figure 1b. is an isometric view of a second embodiment of the suspension system according to the present invention;

Figure 1c. is a detail view of the suspension system of Figure 1b;

Figure 2a. is a third embodiment of the suspension system according to the present invention;

Figure 2b. is a detail view of the suspension system of Figure 2a;

Figure 3 is a plan view as seen from the underside of the vehicle chassis showing the general layout of a fourth embodiment of a suspension system according to the present invention;

Figure 4 is a schematic isometric view of a general layout showing a fifth embodiment of a suspension system according to the present invention;

Figures 5 and 6 are schematic diagrams respectively showing the fluid flow direction under cross-axle articulation and roll motions within the suspension system of Figure 4;

Figure 7 is a schematic isometric view of a general layout showing a sixth embodiment of a suspension system according to the present invention;

Figure 8 is a schematic isometric view of a general layout showing a seventh embodiment of a suspension system according to the present invention;

Figure 9 is a schematic diagram showing the fluid flow direction under articulation motion within the suspension system of Figure 8;

Figure 10 is a schematic isometric view of a general layout showing an eighth embodiment of a suspension system according to the present invention;

Figure 11 is a plan view of a ninth embodiment of a suspension system according to the present invention; and

Figure 12 is an isometric view of a tenth embodiment of a suspension system according to the present invention.

All the accompanying drawings and figures are similarly marked so that all identical components in all the drawings carry the same numbers and symbols for simplicity and only those parts that are relevant to the present invention are shown.

Referring initially to Figure 1, there is shown typical features of a vehicle chassis and a first embodiment according to the present invention. The front of the vehicle chassis 1 is shown facing the top right hand corner of the sheet. The chassis 1 includes two main longitudinal rails 1a, 1b and chassis cross members 1c, 1d, 1e, 1h interconnecting the rails 1a, 1b.

Lower wishbones 2a, 2b, 2c, 2d locate wheels (not shown) permitting them to move in a substantially vertical direction. The wishbones are in the shape of an "A" and are pivotally attached to the chassis 1 at the base 3 of each wishbone. The upper wishbone or "McPherson attachment" is omitted from the drawing for clarity reasons. Wishbone 2a therefore provides a movable location means for the front right wheel while wishbone 2c attaches the rear left wheel assembly to the chassis 1. Each wheel assembly is respectively attached to the outer ends 2a(i), 2b(i), 2c(i), 2d(i) of the wishbones.

It should be understood that other known linkage means may alternatively be used to locate the wheels relative to the chassis 1 such that they may move in a generally vertical direction. For example, the present invention is also applicable for vehicles fitted with multi-link wheel geometry such as trailing and leading arms, panhard rods and even leaf springs.

In Figure 1a, coil springs 4a, 4b, 4c, 4d are mounted on the upper surfaces of wishbones 2a, 2b, 2c, 2d respectively. It is to be understood that the upper ends of the coil springs are resolved against an attachment joined to the chassis 1 (although this is not represented) so that as the wishbones move up and down in a generally vertical direction about pivot points 3, the springs are compressed and allowed to extend between the wishbone and the chassis attachment as in known suspension systems. The coil springs support most of the weight of the vehicle.

The suspension system shown in Figure 1a is provided with torsion means 50, 51, interconnecting each pair of transversely adjacent wishbones 2a,2b and 2c,2d. Each torsion means includes two components 5.a,5b and 5c,5d which are similar to known transverse roll stabiliser bars. These are attached to the wishbones by known means such as ball joints, bushes or drop links so that as the wheel assemblies move in a generally vertical direction the main shaft of each component is urged to turn inside housings (not shown) which are attached to the chassis 1 as in known roll stabiliser bars. Roll stabiliser bars are normally manufactured from spring steel which provides some torsional resilience along its length.

The above described features of the suspension system do not significantly depart in design and function from known suspensions incorporating roll stabiliser bars. However, because the torsion means are each divided into two components 5a, 5b, and 5c, 5d, the 'roll stabiliser' functions of these components may be reversed. Accordingly it is more appropriate to refer to these components henceforth as 'transverse torsion bars' as at times they fulfil a function which is entirely contradictory to roll stabilisation. A pair of these transverse torsion bars are provided at each end of the chassis 1, and each pair of torsion bars are interconnected by means of a central connection means 11.

The connection means 11 provided in between the two transverse torsion bars at each end thereof, control the function of the transverse torsion bars such that at some times the two halves may be coupled as though the two halves were functionally one, and at other times the two halves are caused to contra-rotate about their elongate axes relative to each other. Therefore the functions of the transverse torsion bars can be reversible to either effectively increase roll motions on each axle independently or to prevent such divergent movements of both wheels on one axle simultaneously. Furthermore, according to the present invention, the connection means 11 for each pair of transverse torsion bars can be controlled simultaneously as a function of the vehicle motion and attitude. This will be made clear from the following description.

Alternative forms of connection means 11 are possible and various such means are illustrated. In Figures 1a, 1b, and 1c, for example, the components in the connection means 11 may be described as follows: The transverse torsion bars are rotatably fixed to the chassis and axle by any known means such as sleeves, and these details are omitted in most Figures. Lever ends are provided at the ends of the transverse torsion bars to enable torsional forces to be imparted to the transverse torsion bars. The lever ends may be fabricated in any known manner such as those commonly provided in torsion bar linkages. Figure 1a shows the connection means 11 connecting the front transverse torsion bars in a schematic fashion. An embodiment of the connection means 11 is shown in more detail in Figures 1b and 1c. It should be noted that while only the connection means 11 of the front transverse torsion bars is shown, a corresponding connection means arrangement can also be provided for the back transverse torsion bars. Referring now to Figures 1b and 1c, at the opposite ends of the transverse torsion bars lever ends 5a, 5b are provided lever arms 7a, 7b, attached to transverse torsion bars 5a, 5b respectively. The outer ends of the lever arms describe arcs in a plane generally parallel to the general longitudinal plane within which the wheels rotate. The pair of transverse torsion bars 5a, 5b is interconnected by a linkage assembly 5f. The ends of the levers arms 7a, 7b are provided with 'eye bolts' or ball joints or tie rod ends or bushings 8a, 8b which permit linkages or couplings to be flexibly joined.

The flexible joints attach each lever arm to a respective short linkage 9a, 9b. The opposite end of each linkage 9a, 9b is also provided with single or double flexible joints 10a, 10b which may be similar in construction to joints 8a, 8b.

Attached to these paired flexible joints at each end of the vehicle there is shown another lever mechanism 11a. This lever mechanism is attached by known means to a rotatable longitudinal stub shaft 12a which has a main rotational axis that is generally perpendicular to the axis of the transverse torsion bars 5a, 5b and which therefore generally follows the longitudinal axis of the vehicle.

Accordingly with reference to Fig 1b and 1c, if the right front wheel at supported by wishbone 2a moves in a downward direction then the transverse torsion bar 5a. (in Figs 1b and 1c) is caused to rotate about its elongate axis as a result. The associated lever arm 7a will therefore describe an arc towards the front of the vehicle, and this motion will pull on the linkage 9a. which in turn will pull the perpendicular lever 11a through an arc, transversely towards the front right wheel about its pivotal axis on stub shaft 12a.

The linkage 9b will in addition pull on pivot point 8b at the top of lever 7b. which will describe an arc towards the rear of the vehicle about the axis defined by the other half transverse torsion bar 5b thereby transmitting a torsional force into an upward direction onto the front left wheel assembly supported at wishbone 2b. Therefore, the vertical movement of one wheel can induce the opposite vertical movement of the other wheel on the same axle.

It should be understood that in order for one transverse torsion bar such as 5a to cause the other transverse torsion bar 5b to contra-rotate, the transverse torsion bars 5a, 5b and the longitudinal stub shaft 12a must be free to rotate about their own elongate axis although they should be restrained from moving axially, that is transversely in the case of torsion bars 5a, 5b and in the longitudinal plane of the vehicle in the case of stub shaft 12a. If the longitudinal stub shaft and transverse torsion bars are permitted to move axially this will result in the ultimate promotion of wheel movements in the same vertical direction simultaneously on both sides of the vehicle.

In order, therefore, to increase traction by forcing each wheel to maintain substantially consistent ground pressure at each tyre, (when travelling in an approximately straight line), it is necessary to cause the relative contra rotation of the two transverse torsion bars of each axle by permitting the longitudinal stub shaft 12a to rotate about its elongate axis whilst restraining it from moving axially.

Conversely, to improve roll stability (when this is desirable such as when cornering), it is necessary for both wheels on the same axle to attempt to move in the same vertical direction simultaneously and in this instance it is necessary to restrain the rotational movement of the longitudinal stub shaft 12a about its elongate axis whilst permitting this stub shaft to move in the axial direction instead. The transverse torsion bars still have to move rotationally but do not need to move axially in this instance.

Therefore, the wheels are free to move when udergoing cross-axle articulation while roll motion of the vehicle are reacted to by the transverse torsion bars. According to the present invention, the transverse torsion bars at both ends of the vehicle can also be simultaneously controlled in this manner under vehicle roll conditions.

However, under conditions where opposing vertical wheel movement is required, relative counter rotation between each pair of torsion bars is facilitated. Simultaneous control of the front and back transverse torsion bars is essential where substantial cross-axle articulation is required. In such situations, one pair of diagonally opposed wheels are required to move in the same general direction while the other pair of diagonally opposed wheels are required to move in the opposite direction or be held generally stationary relative to the former pair of wheels.

The linkage means 5f comprising the levers and links between the two transverse torsion bars 5a, 5b can therefore be controlled such that the suspension system can both resist roll motion or encourage opposite individual wheel movements, in particular, cross-axle articulation. Any differing mix of roll and load sharing can therefore be applied by restricting the movement of the longitudinal shaft 12a by either restraining the shaft axially, or rotationally, individually, collectively or not at all.

It should be noted that it is mechanically feasible, if not sometimes preferable, to terminate the transverse torsion bars in ways other than the described links, in order to provide a more positive and durable linkage and control mechanism which may work in similar ways or which may permit additional benefits. For example, the transverse torsion bars 5a, 5b, 5c, 5d, may optionally terminate in "T" junctions at their inboard ends instead of the "L" shaped levers as illustrated. The longitudinal shaft 12a may also be provided with matching double levers. In providing "T" shaped double levers either side of the shafts, these bars are caused to rotate on their axes without significant eccentric side loads. The cleats and attachment means which locate the shafts relative to the chassis may be minimised by the double lever terminations.

Additionally, roll control and wheel movement enhancement modes may also be selected in a different manner by functionally causing the double ended "T" junction on the end of the longitudinal shaft to be decoupled to function as required like the previously described "L" lever system as shown in Figure 1 and the different types of lever means may therefore be used as different kinds of control mechanisms.

There are various methods which can be used to select when and how to cause the central connection means 11 to enhance roll control or encourage vertical wheel travel motions. Now with reference to Figure 1c, it will be seen that shaft 12a has a grooved or splined portion 13 to slidably locate a brake unit 14 which for the purpose of demonstrating this part of the invention is drawn schematically as a segment of disc brake 14a and a brake calliper mechanism 14b.

The segment of disc brake 14a is provided with a splined inner surface which locates on the outer matching splined surface of shaft 12a. Brake segment 14a is caused to rotate in a limited arc along with the longitudinal shaft 12a. When the whole of shaft 12a. moves axially, however, the segment of disc brake 14a remains in its usual axial location relative to the chassis as it is located within the brake calliper mechanism 14b which is permanently fixed onto the vehicle body or chassis cross member by any convenient means such as by a cleat 15.

In order to promote roll resistance therefore, the brake calliper mechanism 14b contracts/presses onto, and restrains, brake segment 14a so that it cannot permit shaft 12a to turn, whilst still allowing this shaft to move axially thereby only permitting the torsion bar levers 7a,7b to move in unison in the same longitudinal plane, thereby causing both wheels to become predisposed towards moving in the same vertical direction at the same time, also in order to limit roll motion.

A second brake mechanism 16 is also located in such a way so as to prevent the axial movement of shaft 12a, whilst allowing the shaft to rotate about its own axis. The purpose of this then is to restrain the lever mechanism 11a extending from the longitudinal shaft 12a from moving freely in the longitudinal direction which encourages roll control whilst encouraging it to rotate about the axis of shaft 12a to promote the contra rotation of the two transverse torsion bars and to thereby promote the opposite vertical movements of the associated wheels.

The longitudinal brake mechanism 16 basically comprises a brake plate 16a, which moves in a longitudinal direction through the brake calliper mechanism 16b. The plate section is located on the shaft 12a by means of a tube 16c which is free to rotate about shaft 12a but which is located in between two stops or rings 16d which are permanently fixed to the shaft. All extraneous details such a the brake hoses and pads are omitted from the drawings.

It should be understood that the same functions may be achieved by other known and equivalent means including drum brakes, brake bands on the outer surface of a cylinder or simple pin and eye type locking mechanisms which locate the relative parts in one fixed position or help to centre the relative components to a preferred central position.

An additional example of location means which has additional advantages may be described in terms of a passive or a semi active system, in which the shaft's positions and movements are determined by two hydraulic cylinders mechanically connected between the vehicle chassis and longitudinal shaft in place of the brake mechanisms and hydraulically connected to a pressure source such as a pump or accumulator. The control system need not necessarily, therefore, rely on merely damping out forces by way of creating frictional losses as in a passive system, but rather by way of forcing positive roll correction and vertical wheel travel motions by actively rotating and moving the shafts appropriately (as described above) but in this instance via the hydraulic actuators.

In any event, whether the roll and wheel motions are passively or actively effected, the system (as described with reference to the Figures 1a to 1c) requires that roll forces and wheel travel motions are sensed or monitored so that the appropriate brake or actuator means is caused to work at the correct time. Under normal situations the wheel positions relative to the body are measured using any suitable known devices such as potentiometers attached between each wheel assembly and the body. Accelerometers, steering/throttle/brake pedal position sensors, G switches, and mercury switches are also now common and widely used in the automotive industry to detect cornering forces. The information from the wheel potentiometers and accelerometers may be collated along with information from the speedometer, (and any other input considered useful), to determine when each regulation or control means such as the brake units 14, 16 should be made to operate at any given time to provide the best mix of assisted wheel travel motion and roll control. An electronic control means (ECU) can typically be provided to receive the various sensor signals and to provide control signals to the connection means.

Figures 2a and 2b show an alternative central connection means 11 which is mechanically equivalent to the lever and link mechanisms as illustrated in Figure 1a, 1b, 1c. Figures 2a, 2b are also isometric views of the underside of part of the vehicle chassis as if looking up from the right hand side of the vehicle towards the front left hand wheel. Accordingly the transverse torsion bar 5b is located towards the lower left side of the sheet and the other transverse torsion bar 5a slopes upwards towards the top right of the sheet.

The levers and links shown in Figures 1a to 1c are replaced in Figure 2a and 2b by three meshed bevel gears 7bp, 7ap. and 11p. The lever 7a on the inner end of transverse bar 5a is therefore replaced by bevel 7ap while lever 7b is replaced by bevel 7bp joined onto the end of shaft 5b. Lever 11a is similarly replaced by bevel 11p. The two bevel gears 7ap and 7bp are caused to contra-rotate as they are both meshed with the common bevel gear 11p which has a rotational axis that is perpendicular to the axis of rotation of the two transverse torsion bars.

Although complete bevel gears are illustrated for clarity it is equally possible to install segments of bevel gears instead, as the axles in vehicles usually do not turn through arcs of greater than twelve degrees and consequently the bevel gears themselves do not need to rotate through greater angles. In order to provide a very positive meshing of the bevel gears they may be designed with helical gear teeth as in other known applications. Additionally extra bevel gears may be designed into the linkage unit 5f to provide a more positive meshing of the shaft and to resolve the loads with fewer eccentricities.

In Figure 2a and 2b, it will be seen that the coil springs (numbered 4 in Figures 1) have been optionally omitted and the main support for the vehicle is therefore provided by transverse torsion bars which are resolved into the chassis at cross member 1d by way of a resilient means in the following manner. The transverse torsion bars 5ab are meshed in the linkage assembly 5f (which in this drawing is represented as three bevel gears 7bp, 7ap, and 11p). These three bevel gears are located in a housing numbered 18. The bevel housing 18 is provided with any suitable cleat 18a which provides an anchorage point for a resilient means located between the cleat 18a and the chassis crossmember 1d.

A hydropneumatic strut 19 is shown in Figure 2b to represent the resilient means in combination with one or more gas springs or accumulators numbered 19a. Damping valves in the mouths of the accumulators 19a. (similar to those of known construction and not visible in this drawing) may be provided as a shock absorber for this central resilient means. The advantage of this central damping mechanism is that it provides damping for pitch direction inputs alone and can therefore be matched to this specific frequency requirement, without adversely effecting the roll damping at a different frequency. Roll damping may be provided at the wheels by way standard shock absorbers for example.

If the coil springs at the wheels are omitted, then the central resilient means 19 must provide the ultimate support for that end of the vehicle, and without this unit the entire assembly comprising of the twin transverse torsion bars 5a, 5b and central connection means 5f would rotate and would thereby permit the vehicle to drop onto the rubber jounce bumpers.

Any suitable combination of resilient means may be provided whereby outer coil springs could be relatively unsupportive in combination with a strong central resilient means such as the hydropneumatic strut 19, or conversely, the central hydropneumatic strut may provide little support while outer coil springs at the wheels may carry the majority of the vehicle's weight. Alternatively again, each central resilient means can carry the entire weight of the wheel that is born by the associated wheels.

The resilient and damper means such as the hydropneumatic strut 19 therefore provides an additional level of resilience in the pitch plane to significantly improve comfort in the longitudinal pitch direction whilst still maintaining firm roll resilience to maintain a high standard of handling so that the vehicle handles like a sports car in roll but is as comfortable as a limousine in pitch.

While a hydropneumatic strut 19. is represented in Fig. 2a, 2b it is equally convenient at times to replace this unit with any other type of known resilient means such as a rubber block, or coil spring, which may preferably also accommodate a damper (shock absorber).

One of the advantages of providing a hydropneumatic strut over say, a coil spring, is that the hydropneumatic strut may optionally be connected to a fluid pressure source (fluid pump) and reservoir (tank) so that additional hydraulic fluid may be introduced into the strut and accumulator to raise the vehicle up, or the fluid may be drained back to the tank to allow the vehicle to reduce its height. Such attitude and height adjustment is desirable to level the vehicle when weight is eccentrically applied onto one end of the machine or the vehicle may benefit from being lowered at both ends when being driven at high speed, for example. It should however be noted that it is possible to eliminate the resilient means between the torsion bars and the vehicle chassis, the transverse torsion bars being for example supported against the vehicle chassis by a solid bar. The resilience would then be provided solely by the resilience of the torsion bars.

Roll motion and individual dynamic wheel movements of low amplitude at high speed requiring a limited amount of wheel travel are thus resolved by relatively stiff transverse torsion bars (springs) and shock absorbers (dampers) which are omitted from drawings. A brake mechanism 14 prevents the central bevel gear turning freely and this prevents the contra rotation of transverse torsion bars to promote roll minimisation as required. Components 14a and 14b, in Figure 2b are similarly referenced to those referred to in the similar functional context with reference to the embodiment illustrated in Figure 1a to 1c.

When traversing uneven terrain the resilient mechanism 19 (Figure 2a and b) provides support for the contra rotational linkage 5f (bevel gear set) so that contra rotation of the two transverse torsion bars can be achieved more easily. This suspension system therefore exhibits multiple spring rates which differ in response appropriately as when the vehicle requires a stiff roll resistance response or when a softer pitch response for comfort when driving in a more straight line.

The above embodiment has an advantage in that it allows for at least substantially equal loadings to be applied to each wheel. In conventional suspension systems where a coil spring or other resilient means is provided at each wishbone, it is necessary for the suspension system to overcome the spring force of the resilient means before motion of the wheel can be effected. However, when the wishbones are free of any resilient means, this allows for free movement of the wishbone thereby allowing at least substantially equal loadings to be applied to each wheel by the suspension system.

Further developments of the present invention will now be described which enable the suspension system itself to intrinsically differentiate between circumstances when roll resistance should be enhanced or actively reversed, without the requirement for external sensors and an ECU or intelligence system. Moreover, the suspension system variations described below automatically and passively react to the various requirements of the system, to provide the required response without any outside influence, or intelligence, or requirement for energy.

Figure 3 shows a further embodiment of a suspension system according to the present invention. This embodiment is similar to the embodiment shown in Figures 1a to 1c in that the vehicle chassis 1 is supported on wishbones 2a, 2b, 2c, 2d, and coil springs 4a, 4b, 4c, 4d are also provided as in conventional suspension systems. Furthermore, the wishbones at each end of the vehicle chassis 1 are respectively interconnected by respective pairs of transverse torsion bars 5a, 5b, 5c, 5d. Each pair of torsion bars are also connected by connection means 11 in the form of linkage means 9a, 9b, 9c, 9d, 11a as in the earlier embodiments. The principal difference is that a longitudinal shaft 20 interconnects the connection means 11 of each pair of torsion bars.

The longitudinal shaft 20 is provided with a lever member 11a at each end thereof to thereby allow the shaft 20 be linked to the linkage means 9a, 9b, 9c, 9d in the same way as the longitudinal shaft 12a of Figure 1b and 1c. The longitudinal shaft 20 of Figure 3 further includes a spline joint 21 which allows a degree of movement in the longitudinal direction of the shaft 20.

The longitudinal functionally links the front and back transverse torsion bars in such a way so that the torsion bars of the suspension system can react in unison in dependence on the vehicle dynamics. The functional linking by the longitudinal shaft of the front and back transverse torsion bars allow the suspension system to maintain and return the attitude of the vehicle to a position that is at least substantially parallel to the average surface plane supporting the vehicle.

In particular, the wheels are free to move when undergoing cross-axle articulation when one pair of diagonally opposed wheels are displaced in the same general direction and in an opposite direction relative to the other pair of diagonally opposite wheels. In situations when the vehicle is primarily undergoing roll when the wheels on one side of the vehicle are both moving in the same general direction relative to the wheels on the other side of the vehicle, the torsion bars can act in the same manner as a conventional stabiliser to provide roll stiffness for the vehicle.

However, under cross-axle articulation situations which results in counter rotation of adjacent transverse torsion bars also results in rotation of the longitudinal shaft. This results in a transfer of force between the wheels to thereby facilitate the movement of the wheels. It should be noted that there is a progressive change in the degree of relative rotation between the torsion bars as the vehicle is subjected to varying combinations of roll and cross-axle articulation situations. The torsion bars will therefore only be allowed to counter rotate relative to each other when the support assemblies are undergoing cross-axle articulation if the vehicle is also undergoing roll to thereby maintain the roll attitude of the vehicle.

Figure 4 shows another embodiment of the suspension system. The vehicle suspension system is shown with the front of the vehicle being directed towards the bottom left corner of the drawing sheet. The vehicle is supported on wheels 20, 21, 22, 23. Coil springs 4a, 4b, 4c, 4d support the vehicle chassis (not shown) and provide for resilience in the ride of the vehicle.

Torsion means 50, 51 are provided at the front and rear of the vehicle. These torsion means at the front and rear axles of the vehicle are linked either mechanically or hydraulically. Since the hydraulic linkage system is easier to package and describe in a more generic form the ensuing description will refer to the hydraulic linkage system although other linkage systems may optionally be used to provide a method for the suspension system to distinguish between roll and **cross-axle** articulation motions of the vehicle in a similar manner. Further the suspension system is "passive" and not "active". In other words no external sensors are required to operate the system which reacts automatically to vehicle motion. It should be noted that the system shown in Figure 3 is also a "passive" system. Each arrangement includes a pair of transverse torsion bars 5a, 5b, 5c, 5d as in the arrangement shown in the previously described embodiment. The torsion bars are however interconnected by means of a hydraulic double acting ram assembly 62, 63. Each ram assembly has a cylinder 62a, 63a and a piston 62b, 63b supported therein to separate the cylinder into an inner chamber 62c, 63c and an outer chamber 62d, 63d. The lever arm 7b, 7c at one end of one of the torsion bar pairs 5b, 5c is coupled with and movable together with the cylinder 62a, 63a. The lever arm 7a, 7d of the other of the torsion bar pairs 5a, 5d is coupled with and movable together with the piston 62b, 63b. Conduits 64, 65 provide fluid communication between the two ram assemblies 62, 63. In the illustrated embodiment, each conduit 64, 65 connects the inner chamber 62c, 63c of one ram assembly with the outer chamber 62d, 63d of the other ram assembly. It is however to be appreciated that other conduit connection arrangements between the two ram assemblies are also possible depending on the design of the transverse torsion bars which may reverse the sense of rotation if, for example, one pair is located behind the axle while the other pair of transverse torsion bars is located in front of the other axle. Therefore, the outer chambers can be connected by one conduit whereas the inner chambers can be connected by another conduit.

A fluid pump and reservoir assembly 66 may optionally be provided to supply and remove fluid from the ram assemblies and associated conduits. This arrangement allows for additional control of the roll resilience and also provides some active roll control for the vehicle, and active assistance with cross-axle articulation to overcome the resistance of the springs (when used). In addition, the roll attitude of the vehicle can be controlled by varying the volume of fluid within the hydraulic circuit. To this end, the pump/reservoir assembly 66 is connected to the conduits 64, 65 connecting the two ram assemblies 62, 63 by two secondary conduits 67, 68, each respectively connected to one of the conduits 64, 65. Accumulators 69, 70 may also be provided on the secondary conduits to provide additional resilience in the suspension system. It should be noted that most of the resilience within the vehicle ride in terms of pitch and whole body motion is provided by the springs 4a, 4b, 4c, 4d and that roll resilience is only provided by the resilience in the optional accumulators 69, 70 or such resilience as there may be permitted in the transverse torsion bars 5a, 5b, 5c and 5d and linkages and bushes attached thereto.

Figures 5 and 6 show the fluid flow within the system and between the ram assemblies when there is cross-axle articulation motion of the wheels (Figure 5) and when there is roll motion of the wheels (Figure 6). The schematic diagrams show the suspension system in plan view with the front of the vehicle being directed to the top of the sheet. Thus as each wheel 20, 21, 22, 23 moves in a generally vertical direction, this would be a movement in the direction normal to the plane of the drawing sheet. Accordingly, upward wheel motion is therefore indicated by the symbol "-" whereas downward wheel motion is indicated by the symbol "+" .

Referring initially to Figure 5, when there is cross-axle articulation motion of the wheel, the diagonally adjacent wheels move together in the same direction which is opposite to the direction of movement of the other diagonally adjacent wheels of the vehicle. In this situation, because of the direction of movement of the wheels, and the way that the torsion bars are respectively connected to the cylinder and piston of the cylinder assembly, counter rotation of the torsion bars are possible.

For example, in the case of the front force transmitting arrangement 50, when the front left wheel 20 moves upwards and the front right wheel 21 moves downstairs, the counter rotation of the front transverse torsion bars 5a, 5b results in a decrease in volume of the outer chamber 62d of the front hydraulic ram assembly 62, and an increase in the volume of the inner chamber 62c thereof because of the relative motion of the piston 62b within the cylinder 62a. Fluid therefore flows from outer chamber 62d through conduit 64 to the inner chamber 63c of the rear hydraulic ram assembly 63 while fluid flows to the inner chamber 62c of the front hydraulic cylinder 62 through conduit 65 from the outer chamber 63d of the rear hydraulic ram assembly 63. This fluid flow is aided by the relative motion of the rear transverse torsion bars 5c, 5d. This fluid movement ensures that cross-axle articulation motion can be readily achieved by the suspension system.

By comparison, in Figure 6, the wheels on each side of the vehicle move in the same direction and opposite to the direction of movement of the wheels on the other side of the vehicle. The direction of movement of the wheels show that the vehicle is turning left resulting in roll motion of the vehicle. Because of the respective attempted rotation of each transverse torsion bar, and the interconnection between the chambers of each ram assembly, the fluid flow from each chamber is counter acted by the fluid flow from the opposing chamber such that there is little to no relative motion between the piston and cylinder of each ram assembly. Each pair of transverse torsion bars therefore acts in unison in a similar manner as a conventional roll stabiliser bar in roll but unlike conventional roll bars the system described permits simultaneous roll control stiffness and free wheel motions resulting from cross-axle articulation. The functional relationship of the linking between the two rams can also be viewed as a function of the pressure differential across the piston within each ram. When the vehicle is undergoing roll as shown in Figure 6, the pressure differential across the piston of each ram is relatively high, with the load and therefore the pressure carried by fluid chambers 63d and 62b being higher than for chambers 63b and 62d. However, under cross-axle articulation as shown in Figure 5, the pressure differential across the pistons will be relatively low. The pressure differential therefore progressively decreases as the vehicle moves from primarily roll situations to situation of cross-axle articulation.

It should be noted that in conventional roll stabiliser bar systems the torsion bars are "wound up" during cornering and if the road also happens to be undulating on a corner requiring a degree of cross-axle articulation this then requires that one end of the torsion bars is additionally wound up while the other end is temporarily relaxed and these alternating axle movements can cause rapid shifts of weight born by the wheels which in turn cause the lack of traction at the tyre contact patches. In the embodiments shown in Figs 3 and 4 the ground pressure at the wheels is therefore maintained more consistently thereby reducing the risk of skidding on the corners of poorly surfaced roads.

Additionally, in conventional roll stabilisation systems such as transverse torsion bars, when a single wheel impacts a bump or encounters a hollow, the torsion bar is rapidly wound up causing the impact to be resolved by the transversely adjacent wheel and spring assembly as well as by the support points on the chassis, and this results in single wheel input harshness being experienced by the vehicle occupants. It should be understood that single wheel input is, in effect an cross-axle articulation in that it requires two diagonally opposite wheels to move in one direction while the other diagonally opposite pair move in the other direction. Since conventional roll stabiliser bars of the front and back axles are independent they are intrinsically unable to differentiate and react differently to single wheel inputs and roll motions and all such inputs are therefore reacted to in similar manner causing harshness of ride quality and inferior road holding due to inappropriate weight shifting between wheels. In contrast the structural and functional relationship of the components in the system (shown in the accompanying Figures) requires that both axles interact such that roll and articulation are differentiated and reacted upon in different ways so that as minor articulations occur as when a single wheel input occurs this is not experienced as a roll motion (on one axle) requiring maximum stiffness of the roll stabiliser bar leading to undue harshness. In the invented system, therefore, a single wheel input may be reacted to as a minor high speed articulation motion which need not be resisted and which therefore does not need to resolved by the transversely adjacent wheel assembly causing unnecessary harshness.

The embodiment shown in Figure 7 is similar to the embodiment of Figure 4 except that the hydraulic double acting rams are replaced by rotary actuators or rams 62a, 62b. These rotary rams include a housing supporting a rotor rotatably supported therein which separates the housing into two fluid chambers. The housing and the rotor are respectively connected to one of the adjacent transverse torsion bars. Conduits 64, 65 interconnect the corresponding fluid chambers of each of the rotary rams 62a, 62b. This embodiment operates in the same way as the embodiment of Figure 4. In particular, the wheels are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars. The primary advantage is that the use of rotary rams eliminates the need for lever arms on the transverse torsion bars thereby reducing the amount of clearance required to accommodate the suspension system as well as leading to a neater overall arrangement.

Figures 8 and 9 illustrates a further embodiment of the vehicle suspension system wherein the vehicle is also supported by the suspension system according to the present invention. This embodiment is also similar to the embodiment shown in Figure 4 except that the coil springs are eliminated and replaced with a support arrangement 75, 76 interconnecting each pair of transverse torsion bars. This leads to the abovenoted advantage in that it is possible to provide at least substantially equal loading on each wheel. Each support arrangement may include linkage means 75b, 75c, 76b, 76c connecting each transverse torsion bar pair to a respective cross member 75d, 76d. The cross member 75d, 76d may be directly connected to the lever arms such as 7a, 7b to obviate the requirement of links 75c and 75b when the support means are located so as not to conflict with components 62 and 63. The cross members are each coupled to a load support assembly 75a, 76a. In this arrangement, the force to support the vehicle is resolved through the transverse torsion bars to the load support ram assemblies 75a, 76a which are secured by support means 82, 83 to the vehicle chassis and which permit some relative movement to accommodate to positional changes. The provision of the cross member arrangement "averages" the load carried by the associated load assemblies, this average load being resolved through the load support assembly to the vehicle chassis. The resilience of the suspension system is provided by means of accumulators 80, 81 in fluid communication with each load support ram assembly. While Fig 8 illustrates vehicle support means incorporating hydraulic rams and accumulators it should be understood that these resilient devices may be replaced by conventional steel or rubber or composite resilient mechanisms such as springs. It is noted that the conduit connections in this embodiment are similar to the connections shown in Figure 4. In particular, the inner chambers of each cylinder assembly 62, 63 are connected by one conduit to the outer chambers of the other cylinders.

According to the design of the torsion bars, the load support assemblies may be designed/engineered to support the vehicle by being in tension or compression. By way of example, the front load support assembly 75 and linkages 75b, 75c as seen in Figures 8 and 9 would normally be in tension when supporting the vehicle's static weight while the rear assembly 76 and linkages 76b, 76c would be in compression under the same conditions.

The embodiment shown in Figure 10 operates in a similar manner to the embodiment of Figure 8. The primary difference is that the load support ram assemblies 75a, 76a are replaced by coil springs 100,101.

Figure 11 shows yet another embodiment which is a further development of the embodiment of Figure 8. The weight of the vehicle primarily supported by the coil springs 4a,4b,4c,4d located at each wishbone 2a,2b,2c,2d. The load support rams 75a,76a are retained. Further conduits 103,104 are however provided to connect corresponding fluid chambers of those rams. A valve 105 controls the flowpath of fluid between the fluid chambers when the valve is in one position, the corresponding inner fluid chambers and the respective outer fluid chambers are in fluid communication. When the valve is in a second position, the connection is reversed such that each inner chamber is now in fluid communication with an outer fluid chamber. This arrangement allows control of the pitch of the vehicle by providing means to raise and lower each end of the vehicle.

Accumulators 106 may also be provided on each conduit. This provides an amount of damping to control the degree of pitch of the vehicle.

Figure 12 shows a further embodiment of the present invention which operates in a similar fashion to the embodiment of Figure 4. The main difference is that the transverse torsion bar pairs are substituted by a single front and rear torsion bar 90, 91. One end of each bar is connected to the wheel support assembly 98, 99 shown as a simple axle in the drawing by way of a drop link as commonly used to couple roll stabiliser bars to axle or wheel assemblies. The other end of the torsion bars is connected to the axle by way of a hydraulic double acting ram assembly 94, 95 which takes the place of the normal drop link component which links the roll stabiliser bar to the axle whilst accommodating changes in angle between the components. In this particular embodiment, the roll stabiliser bar is coupled to the hydraulic cylinder chamber housing while the piston rod 94b, 95b of each ram assembly is connected to the wheel support assembly 98, 99.

In the embodiment shown in Figure 12, it will be seen that the piston rods pass right through the rams so that the upper and lower piston faces have identical surface areas. In some situations it may be preferable to use a single piston rod extending out of one end of the ram end only or to use rod diameters of different outside diameters to overcome asymmetrics and/or provide specific roll split geometrises.

In this embodiment, when there is roll motion of the vehicle, fluid flow is prevented through the conduits 92, 93 connecting the ram assemblies 94, 95 in the same way as previously described with reference to Figure 4 and Figure 6. Movement of the end 90a, 91a of the torsion bar 90, 91 secured to the ram assembly relative to the wheel support is therefore prevented and the torsion bar therefore operates in a similar manner as a conventional stabiliser bar. During cross-axle articulation motion of the wheels, fluid movement through the conduits 92, 93 connecting the ram assemblies 94, 95 allow movement of the ram assembly end 90a, 91a of the torsion bars relative to the wheel support assemblies 98, 99 thereby facilitating the cross-axle articulation motion. The vehicle suspension system however otherwise operates in a similar way as the embodiment shown in Figures 4 and 8.

It should be noted that each of the double acting rams could be provided by two single acting rams respectively connecting the opposing ends of the roll stabiliser bar. By connecting the fluid chamber of each single acting ram with the corresponding ram of the opposite stabiliser bar, this arrangement can operate in the same general manner as the embodiment of Figure 12.

Although all of the above noted embodiments use torsion bar arrangements, it is also to be understood that the present invention can also encompass arrangements where transverse push/pull rods replace the torsion rods, with the connection means linking the rods applying tension or compression forces on each rod.

CLAIMS:

1. A vehicular suspension system for a vehicle respectively supported on at least one pair of transversely adjacent front support assemblies and at least one pair of transversely adjacent back support assemblies, the suspension system including a force transmitting means interconnecting the at least one pair of transversely adjacent front support assemblies and a force transmitting means interconnecting the at least one pair of transversely adjacent back support assemblies, the force transmitting means transferring forces between the interconnected support assemblies, wherein each force transmitting means includes connection means for progressively varying the magnitude and direction of the force transferred between the associated support assemblies by the force transmitting means as a function of the relative positions of and the load applied to at least two pairs of the interconnected support assemblies, the connection means being functionally linked such that the magnitude and the direction of the force transmitted between associated support assemblies by each of the force transmitting means is progressively varied to thereby maintain and return the attitude of the vehicle to a position that is at least substantially parallel to the average surface plane supporting the vehicle.

2. A vehicular suspension system according to claim 1 wherein the force transmitted by the force transmitting means is a torsional force.

3. A vehicular suspension system according to claim 2 wherein each force transmitting means includes a pair of transverse torsion bars, each torsion bar being respectively connected to a support assembly, the torsion bars being interconnected by the connection means.

4. A vehicular suspension system according to claim 3 wherein the torsion bars are rotatable about their elongate axes, the connection means progressively controlling the axial rotation of the associated torsion bars relative to each other such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars.

5. A vehicular suspension system according to claim 4 wherein each connection means provides a mechanical interconnection of the associated pair of torsion bars, and wherein the connection means interconnecting a said pair of transversely adjacent front support assemblies and the connection means interconnecting a said pair of transversely adjacent back support assemblies are functionally linked by a mechanical connection.

6. A vehicular suspension system according to claim 5 wherein the mechanical connection is a longitudinal shaft interconnecting said connection means, each connection means including a pair of linkage members respectively connected at one end thereof to one of the torsion bars, the other end of each pair of linkage members being interconnected to an end of the longitudinal shaft such that torsional forces can be transmitted between said connection means.

7. A vehicular suspension system according to claim 3 wherein the connection means provides a hydraulic connection of the torsion bars.

8. A vehicular suspension system according to claim 7 wherein the connection means is a double acting ram, the ram having a cylinder and a piston assembly separating the cylinder into two fluid chambers therein, the cylinder being connected to one of the torsion bars, the piston assembly being connected to the other torsion bar, and wherein a fluid communication is provided between the rams.

9. A vehicular suspension system according to claim 8 wherein the fluid communication is provided by conduit means connecting the two fluid chambers of the double acting ram of the front torsion bar with the fluid chambers of the double acting ram of the back torsion bars whereby the transfer of fluid between the fluid chambers enables relative displacement between the piston assembly and the cylinder.

10. A vehicular suspension system according to claim 9 wherein the fluid chambers are connected such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the piston assembly within each cylinder allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the piston assembly, while roll motions of the vehicle are reacted by an increase in the pressure differential across the piston assemblies generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle whilst minimising the changes in load on each support assembly.

11. A vehicular suspension system according to claim 10 further including fluid supply means for supplying fluid to the conduit means so that fluid can be added to one conduit and fluid can be at least substantially simultaneously removed from the other conduit to thereby enable the roll angle of the vehicle to be controlled.

12. A vehicular suspension system according to claim 11 further including roll resilience means such as a hydropneumatic accumulator in fluid communication with both of the conduit means, said roll resilience means including damping means for damping the rate of roll and isolating means for isolating the roll resilience means to thereby improve the roll control.

13. A vehicular suspension system according to claim 3 wherein the connection means is a rotary actuation means including a housing supporting a rotor separating the housing into at least two fluid chambers, the housing being alternatively connected to one of the torsion bars, the rotor being connected to the other torsion bar.

14. A vehicular suspension system according to claim 13 including conduit means providing fluid communication between the two fluid chambers of the rotary actuation means of the front torsion bar with the fluid chambers of the rotary actuation means of the back torsion bars.

15. A vehicular suspension system according to claim 14 wherein the fluid chambers are connected to thereby progressively vary the relative rotation of the rotor within the housing in each of the linked rotary actuation means such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the rotor within each houising allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the rotor, while roll motions of the vehicle are reacted by an increase in the pressure differential across the rotor generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle whilst minimising the changes in load on each support assembly.

16. A vehicular suspension system according to any one of the preceding claims wherein resilient support means are provided between the support assemblies and the chassis of the vehicle for at least substantially supporting the weight of the vehicle.

17. A vehicular suspension system according to any one of claims 3 to 15 including a yoke means interconnecting each pair of torsion bars, and a resilient means connecting the yoke means to the chassis of the vehicle, the yoke means transferring the average load carried by the associated support assemblies to the resilient means such that the resilient means at least substantially supports at least a portion of the vehicle to thereby permit the vehicle to maintain an at least substantially uniform load on each support assembly regardless of any cross-axle articulation of the support assemblies.

18. A vehicular suspension system according to claim 17 wherein the yoke means is provided by a lever arm respectively extending from each torsion bar, the lever arms being interconnected by a cross member arrangement.

19. A vehicular suspension system according to claim 18 wherein the resilient means interconnects the cross member arrangement with the chassis of the vehicle, the resilient means including a load support ram having an accumulator in fluid communication with the ram to provide said resilient support.

20. A vehicular suspension system according to claim 19 further including double acting rams interconnecting the yoke means with the chassis of the vehicle, conduit means connecting corresponding chambers of the rams and valve means for controlling the fluid flow through each conduit means to thereby control the pitch motion of the vehicle.

21. A vehicular suspension system according to claim 20 further including accumulators in fluid communication with at least one of the conduits.

22. A vehicular suspension system according to claim 20 or 21 further including fluid supply means for supplying and removing fluid from said conduit means, sensing means for sensing the attitude of the vehicle, and control means for controlling said fluid supply means to thereby allow control of the attitude of the vehicle.

23. A vehicular suspension system according to claim 2 wherein the force transmitting means includes a single transverse torsion bar and the connection means interconnects the torsion bar to at least one of the associated support assemblies.

24. A vehicular suspension system according to claim 23 wherein the connection means provides a hydraulic connection of the torsion bar to the associated support assembly.

25. A vehicular suspension system according to claim 24 wherein each said connection means includes a double acting ram located at one end of the torsion bar, the ram having a cylinder and a piston assembly separating the cylinder into two fluid chambers therein, the cylinder and the piston assembly being connected between one end of the torsion bar and the adjacent support assembly.

26. A vehicular suspension system according to claim 25 wherein the rams are in fluid communication and wherein said fluid communication is provided by conduit means respectively connecting the two fluid chambers of the double acting ram of the front torsion bar with the fluid chambers of the double acting ram of the back torsion bars.

27. A vehicular suspension system according to claim 26 wherein the fluid chambers are connected such that the support assemblies are free to move when undergoing cross-axle articulation with the movement of the piston assembly within each cylinder allowing the transfer of fluid between the connected fluid chambers with minimal change in the pressure differential across the piston assembly, while roll motions of the vehicle are reacted by an increase in the pressure differential across the piston assemblies generated by the increase in load on the support assemblies on one side of the vehicle and the similar reduction in the load on the support assemblies on the otherside of the vehicle to thereby control the roll attitude of the vehicle while minimising the changes in load on each support assembly.

28. A vehicular suspension system according to claim 27 wherein the connection means is a single acting ram located at each end of the torsion bars, each ram having a cylinder and a piston assembly supported therein to provide a fluid chamber within the cylinder, the cylinder and piston assembly being connected to one of the torsion bars and the adjacent support assembly.

29. A vehicular suspension system according to claim 28 wherein a fluid communication is provided between the rams wherein said fluid communication is provided by conduit means respectively connecting the fluid chamber of each single acting ram of the front torsion bar with the fluid chamber of the longitudinally opposing single acting ram of the back torsion bar, the fluid chambers being connected such that the support assemblies are free to move when undergoing cross-axle articulation, while roll motion of the vehicle is reacted to by the torsion bars while minimising the changes in load on each support assembly.

30. A vehicular suspension system according to claim 23 wherein the connection means provides a mechanical coupling of the torsion bar.

31. A vehicular suspension system for a vehicle respectively supported on at least one pair of transversely adjacent support assemblies, the suspension system including a force transmitting means interconnecting the at least one pair of transversely adjacent support assemblies, the force transmitting means transferring forces between the interconnected support assemblies, wherein each force transmitting means includes connection means for progressively controlling the magnitude and varying the direction of the force transferred between the associated support assemblies by the force transmitting means as a function of the displacement of the associated support assemblies relative to each other and the load applied to the at least one pair of interconnected support assemblies.

32. A vehicular suspension system according to claim 31 wherein the force transmitted by the force transmitting means is a torsional force.

33. A vehicular suspension system according to claim 32 wherein each force transmitting means includes a pair of transverse torsion bars, each torsion bar being respectively connected to a support assembly, the torsion bars being interconnected by the connection means.

34. A vehicular suspension system according to claim 33 wherein the torsion bars are rotatable about their elongate axes, the connection means progressively controlling the axial rotation of the associated torsion bars relative to each other such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars.

35. A vehicular suspension system according to claim 34 wherein the connection means provides a mechanical interconnection of the pair of torsion bars.

36. A vehicular suspension system according to claim 35 wherein the connection means includes a pair of linkage members respectively connected at one end thereof to one of the torsion bars, the other end of the linkage members being connected to a stub shaft.

37. A vehicular suspension system according to claim 36 further including shaft control means for controlling axial and rotational movement of the stub shaft.

38. A vehicular suspension system according to claim 37 wherein the control means includes an axial braking assembly and a rotational braking assembly.

39. A vehicular suspension system according to claim 38 wherein the actuation of said braking assemblies is controlled by electronic control means such that the stub shaft is allowed to move axially in the direction of its elongate axis and is at least substantially restrained from rotation about its elongate axis when the vehicle is primarily undergoing roll conditions, and such that the stub shaft is restrained from moving axially in the direction of its elongate axis when the associated support assemblies are displaced in opposing at least substantially vertical directions.

40. A vehicular suspension system according to claim 35 wherein the connection means includes a pinion gear assembly interconnecting the torsion bars for selectively allowing and restraining counter-rotation of the torsion bars relative to each other, the pinion gear assembly having a pinion gear respectively connected to an end of each of the torsion bars, with a rotatably supported control pinion bar meshing with both of the pinion gears.

41. A vehicular suspension system according to claim 40 further including rotation control means for controlling the rotation of the control pinion gear.

42. A vehicular suspension system according to claim 41 wherein the rotation control means includes a rotational braking assembly, the braking assembly being actuated by an electronic control means such that the rotation of each control pinion gear is controlled such that the support assemblies are free to move when undergoing cross-axle articulation while roll motion of the vehicle is reacted to by the torsion bars.

43. A vehicular suspension system according to any one of the preceding claims wherein resilient means are provided at each of the support assemblies for at least substantially supporting the weight of the vehicle.

44. A vehicular suspension system according to any one of claims 40 to 42 including resilient means located between the pinion gear assembly and the chassis of the vehicle for at least substantially supporting the weight of the vehicle to thereby permit the vehicle to maintain an at least substantially uniform load on each support assembly.

45. A vehicular suspension system according to claim 44 wherein the resilient means is a hydropneumatic strut having at least one associated accumulator providing damping for the vehicle.

Fig 1a.

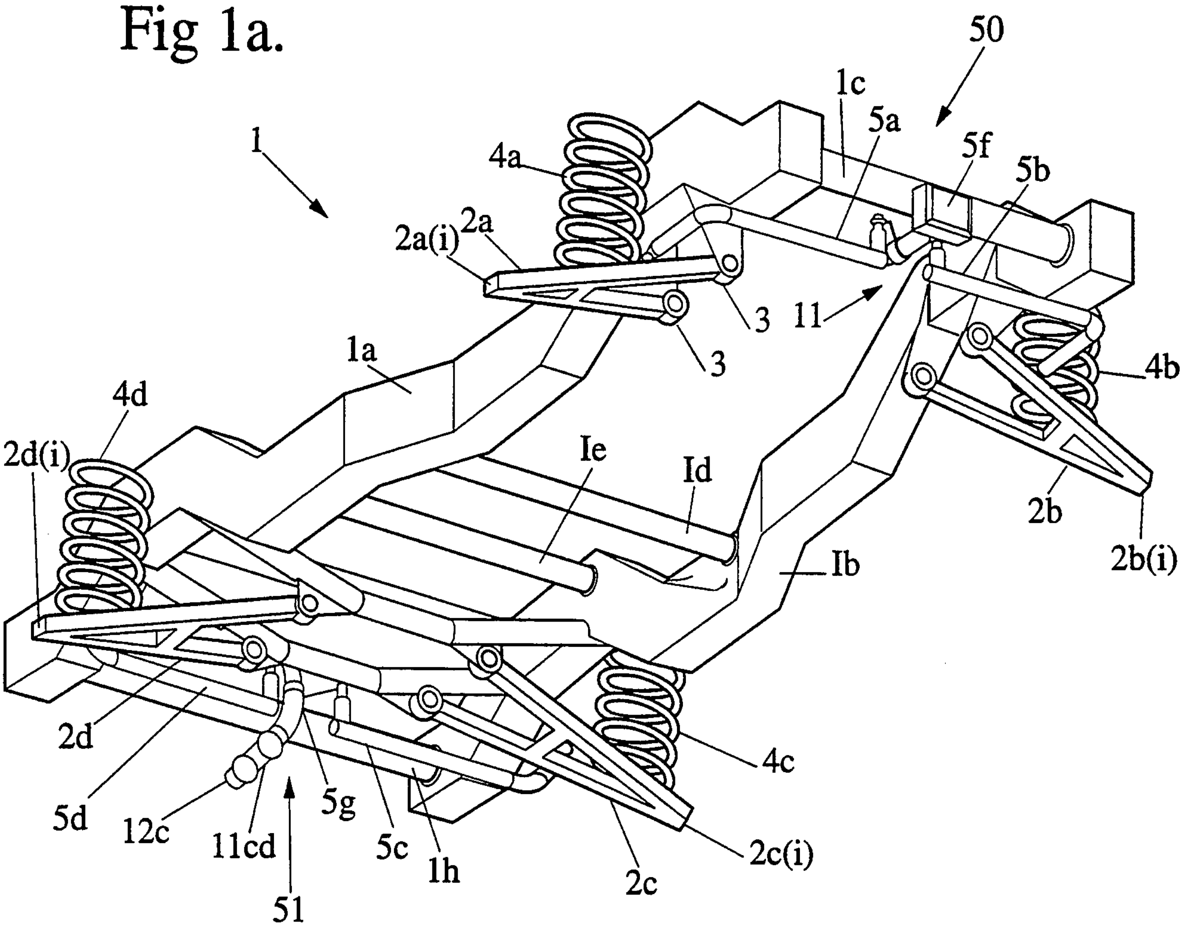

50
1c
5a
5f
5b
1
4a
2a(i)
2a
11
3
3
4b
1a
4d
Ie
Id
2d(i)
2b
Ib
2b(i)
4c
2d
5d
12c
11cd
5g
5c
1h
2c
2c(i)
51

Fig 1b.

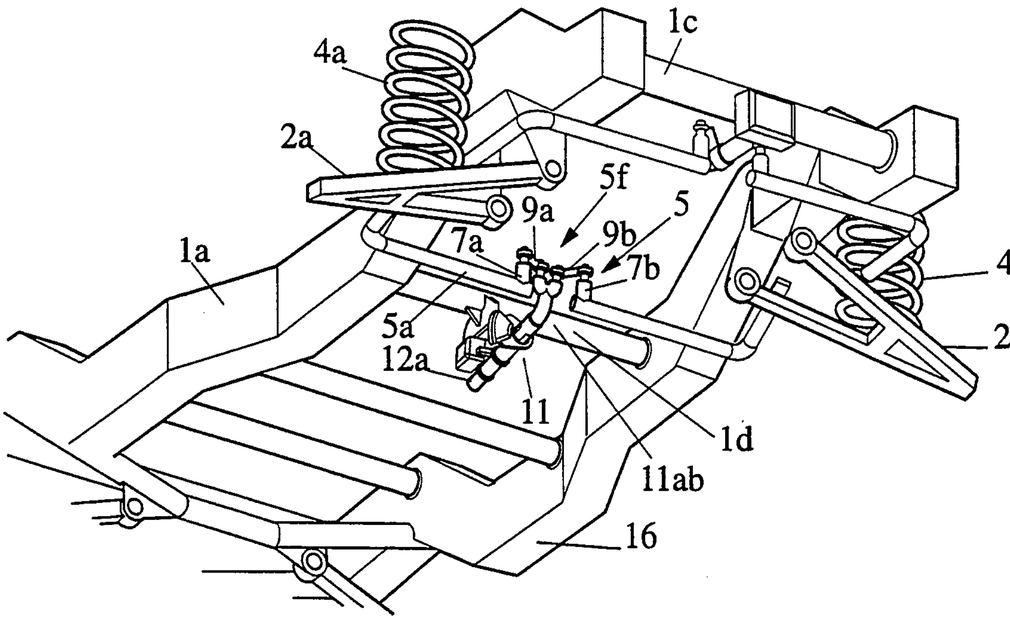

1c
4a
2a
5f
9a
9b
5
7a
7b
4b
1a
5a
12a
2b
11
1d
11ab
16

Fig 1c.

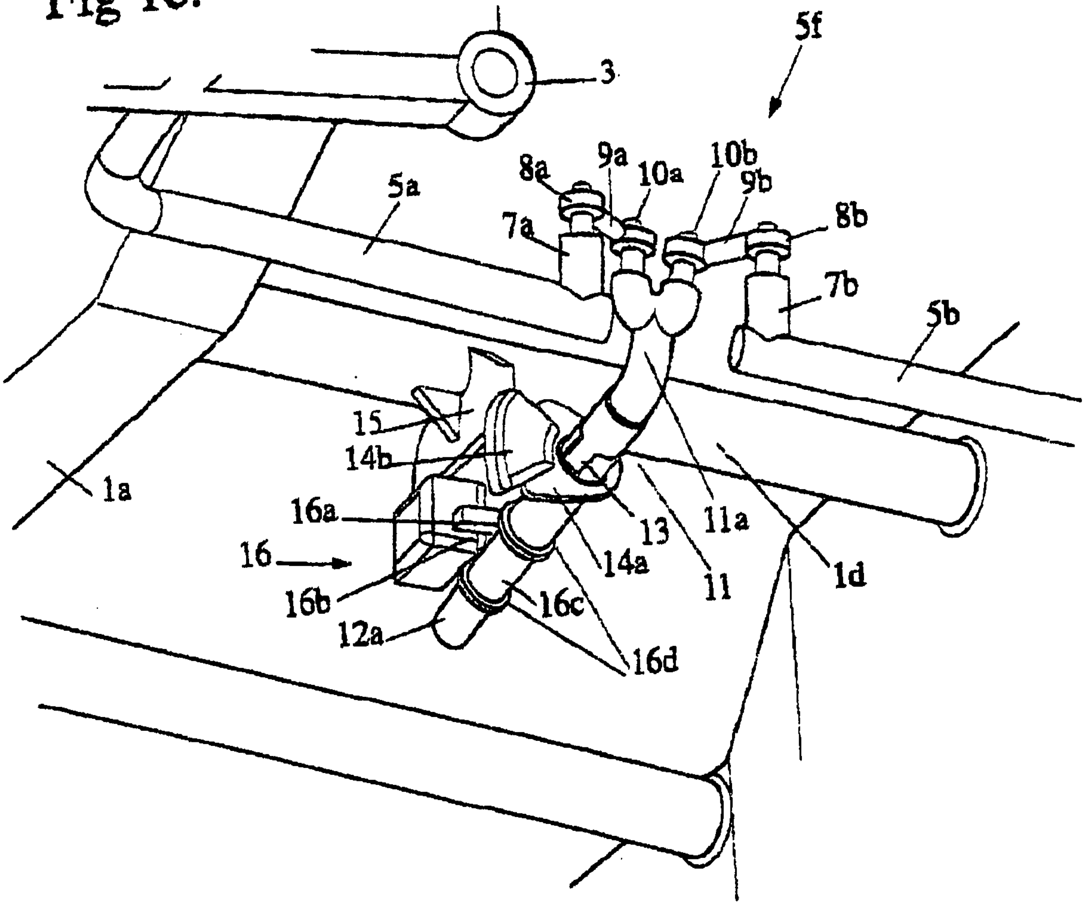

3
5f
8a
9a
10a
10b
9b
8b
5a
7a
7b
5b
15
14b
1a
16a
16
16b
12a
16c
16d
14a
13
11
11a
1d

Fig 2a.

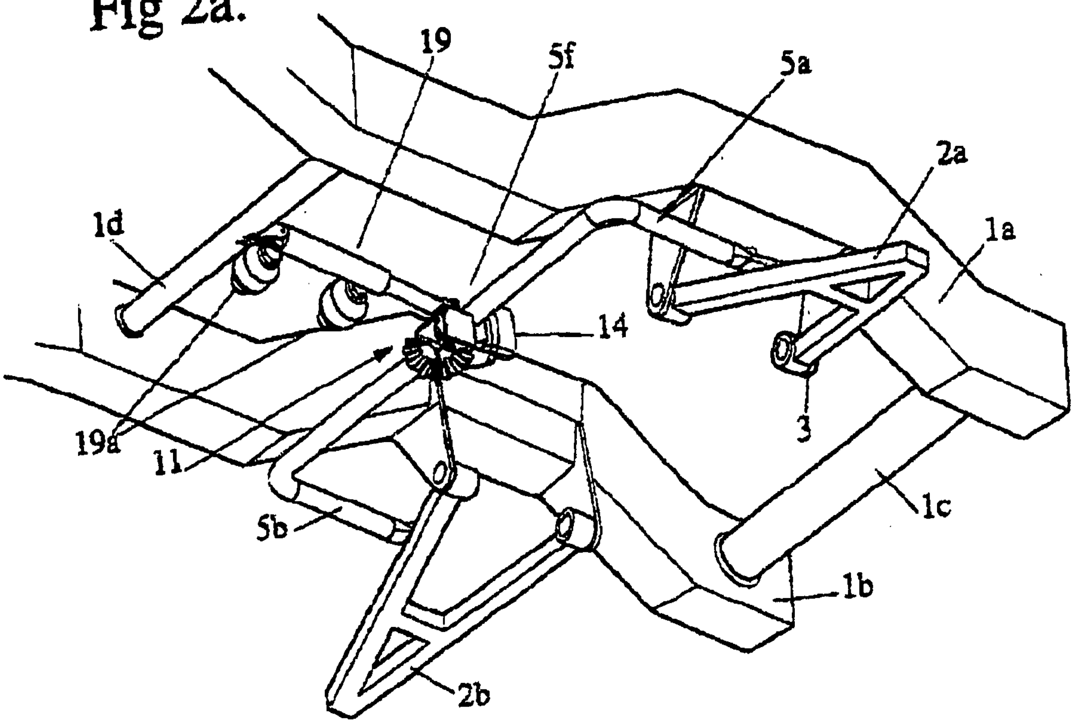

19
5f
5a
2a
1d
1a
14
3
19a
11
5b
1c
1b
2b

Fig 2b.

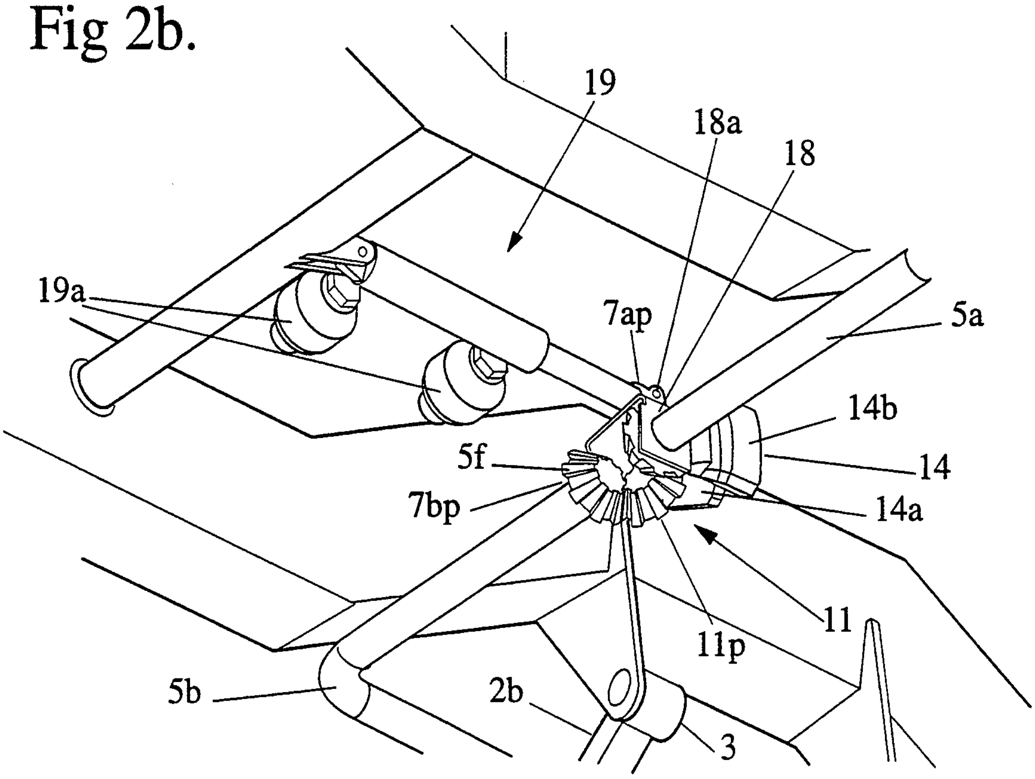
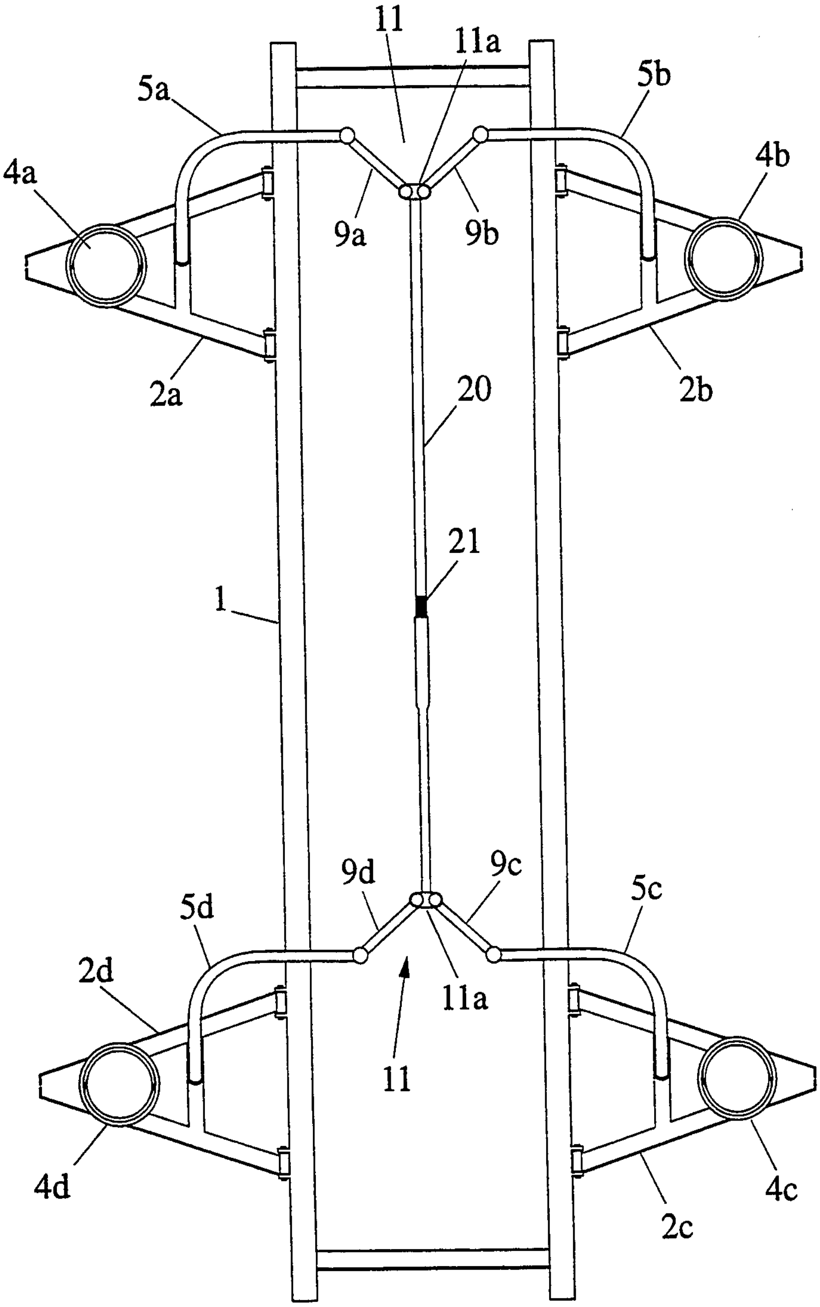


Fig 3.



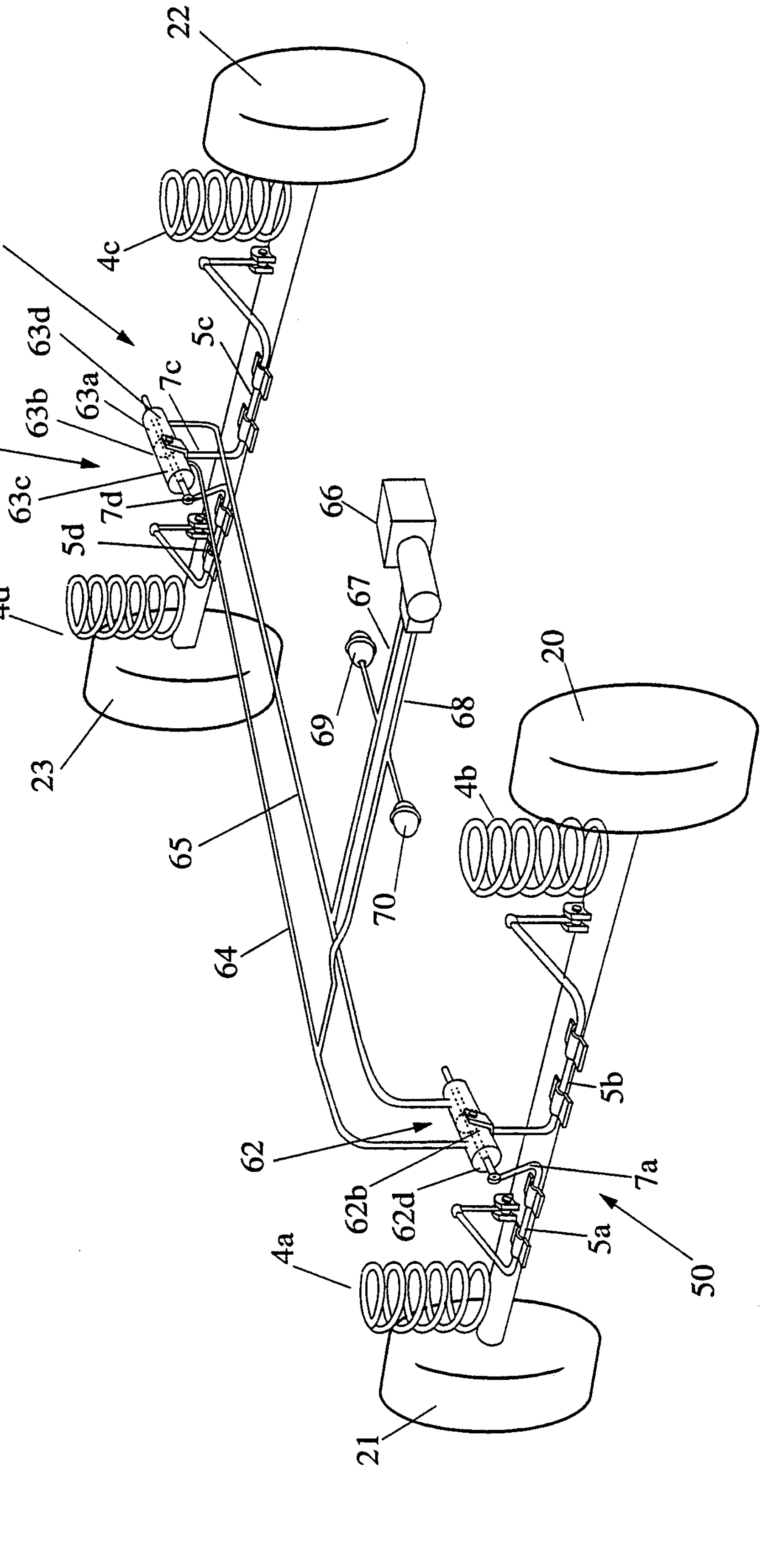

22
51
63d
63a
63b
63
63c
7c
5c
4c
7d
5d
4d
66
67
69
68
20
23
65
70
4b
64
62
62b
62d
5b
7a
5a
50
4a
21

Fig 4.

Fig 5.

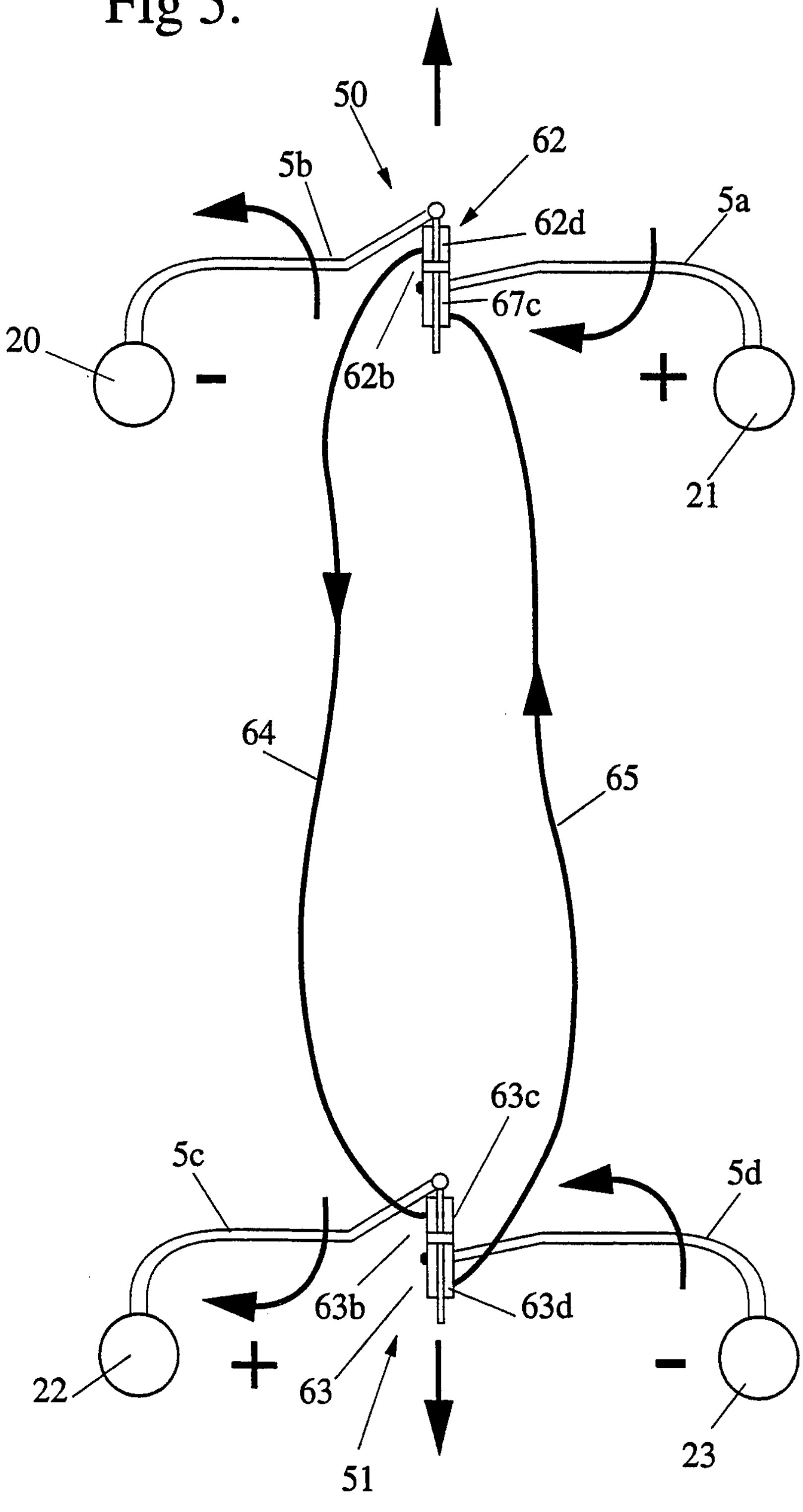
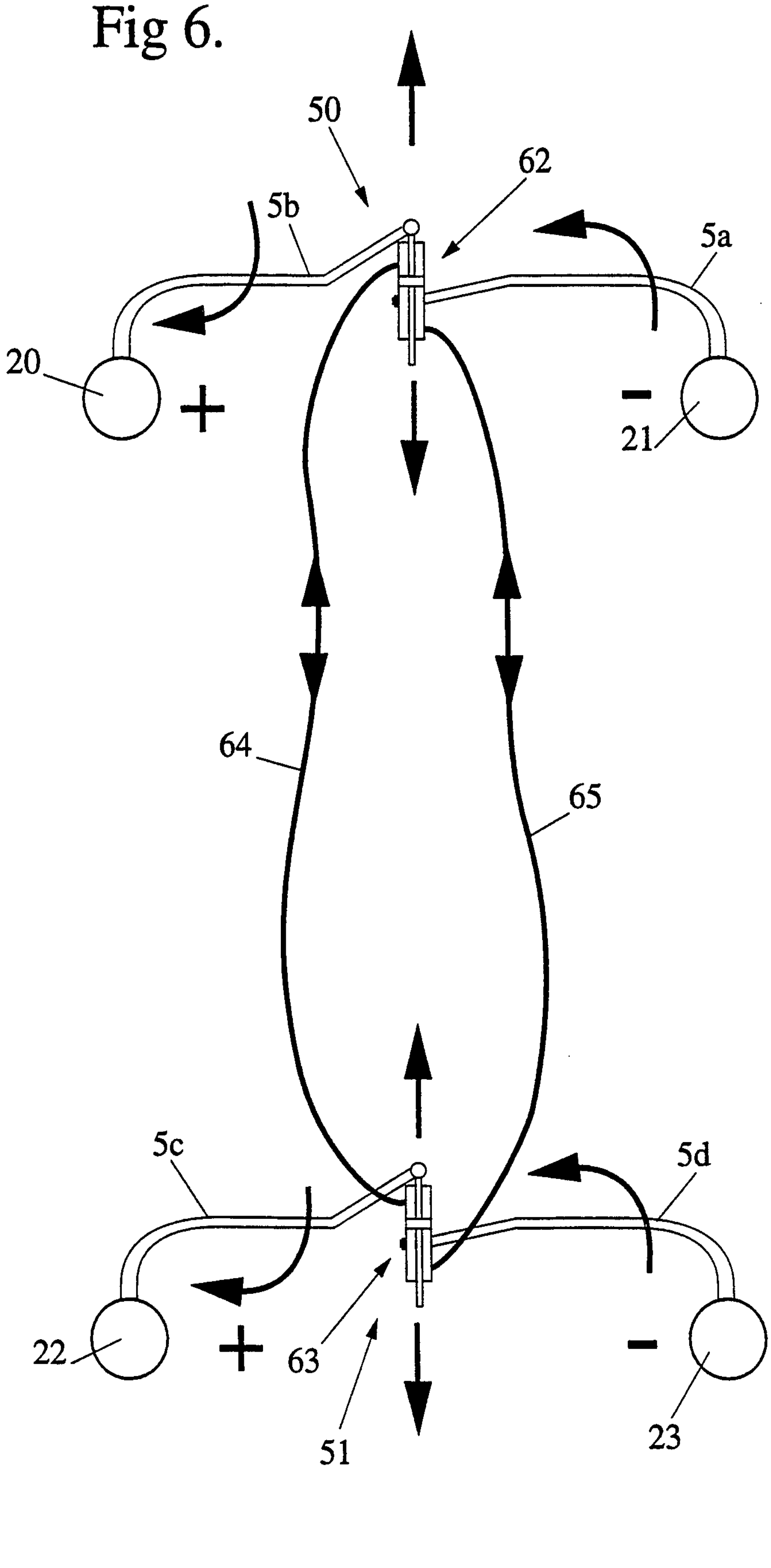


Fig 6.



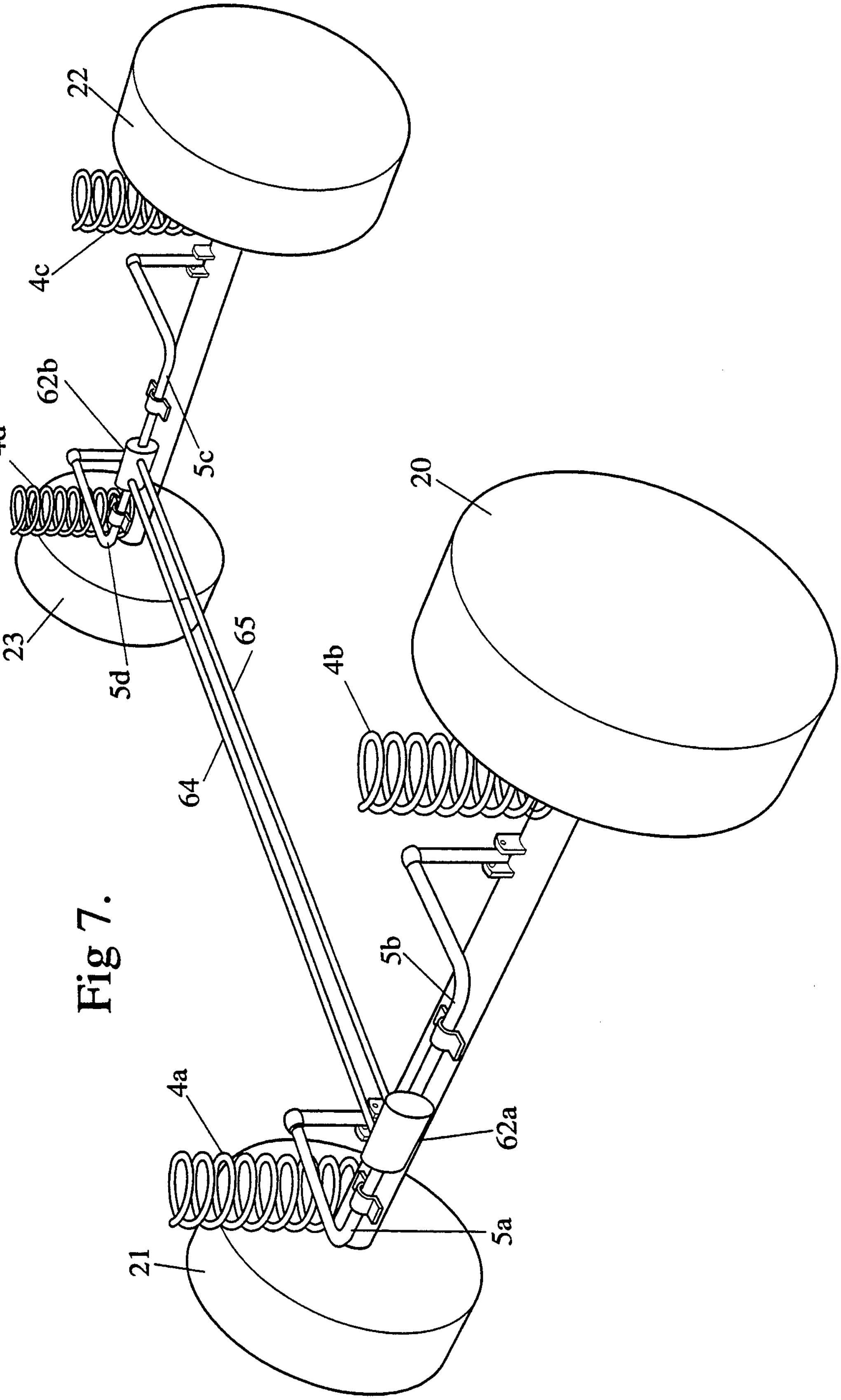


Fig 7.

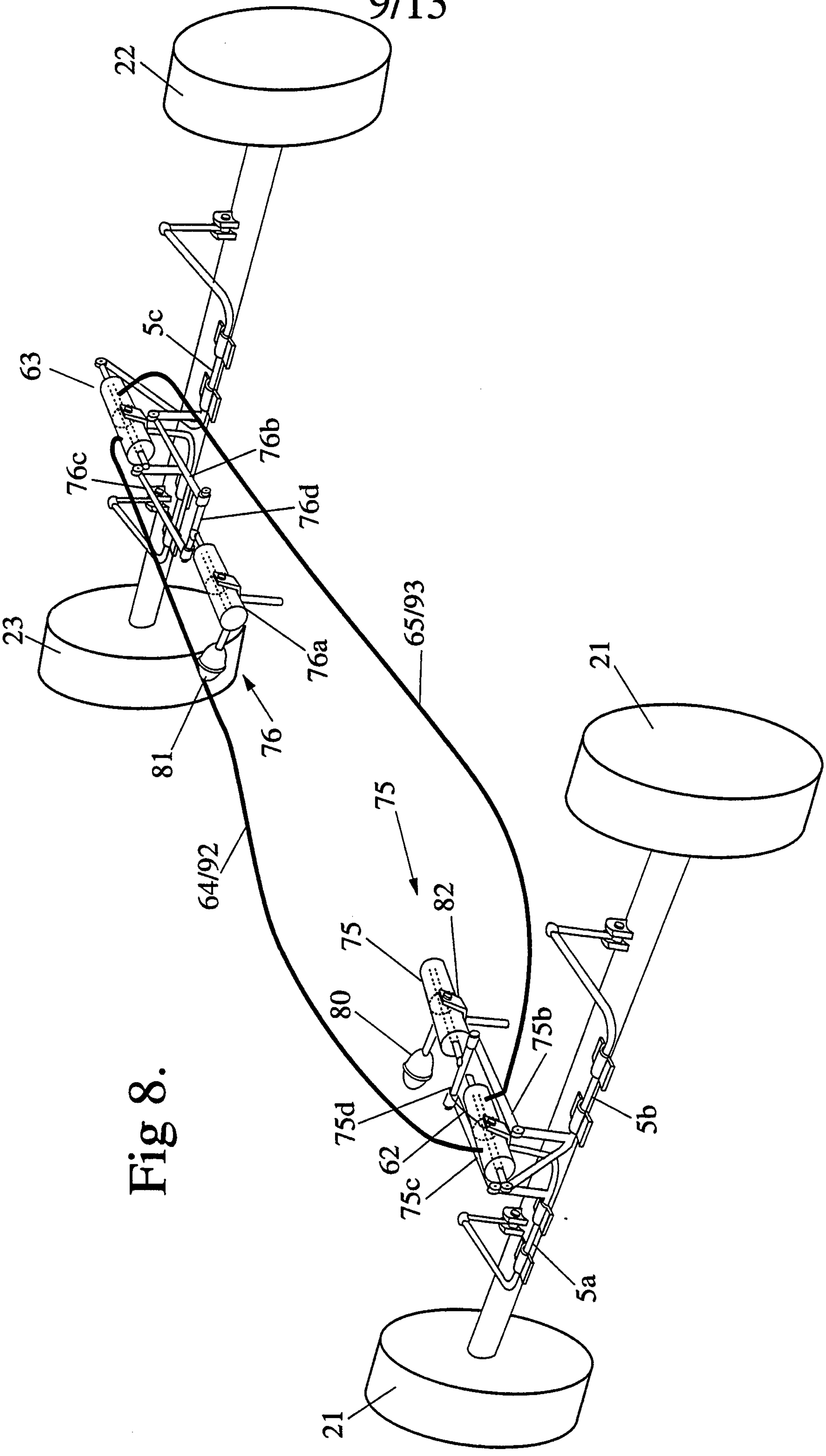
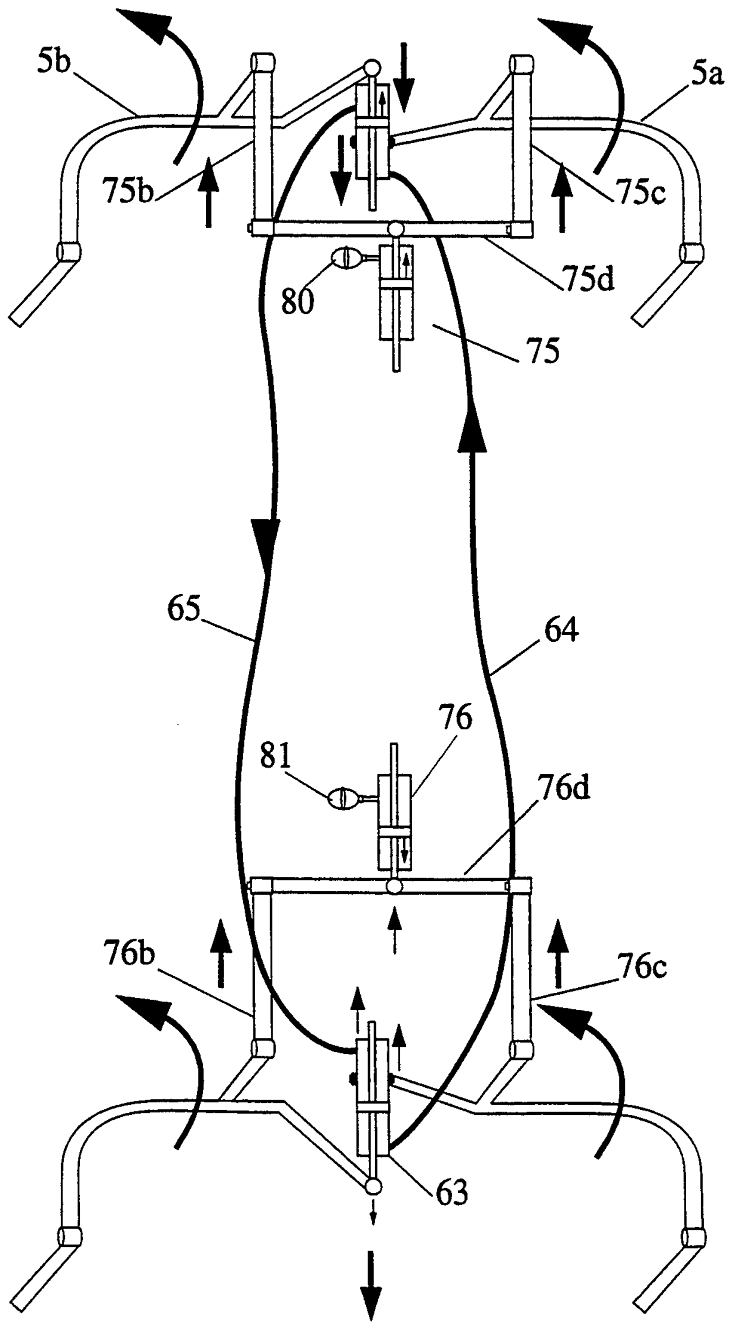

22
5c
63
76b
76c
76d
65/93
23
76a
21
76
81
75
64/92
82
75
80
75b
75d
62
75c
5b
5a
21

Fig 8.

Fig 9.



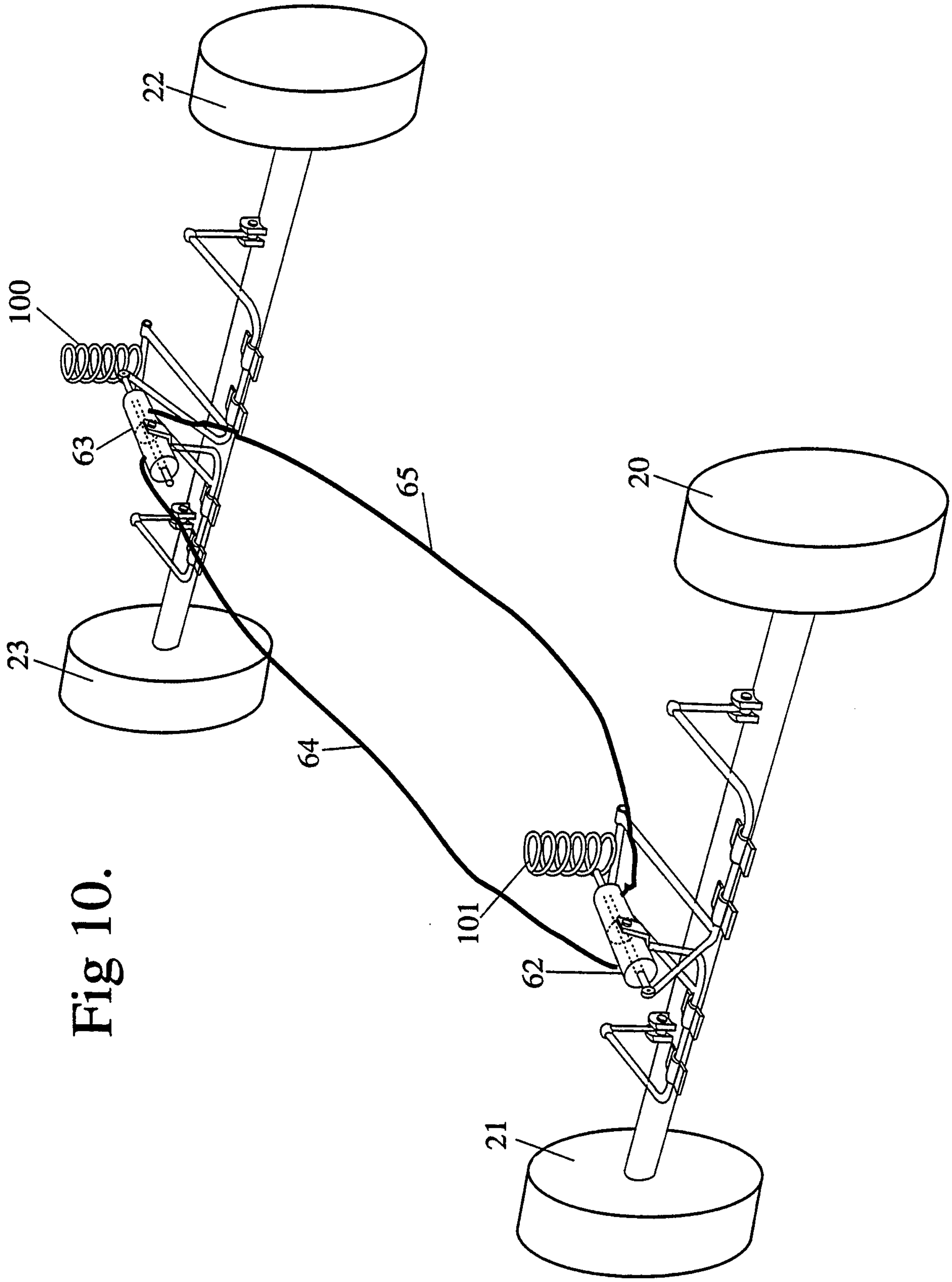
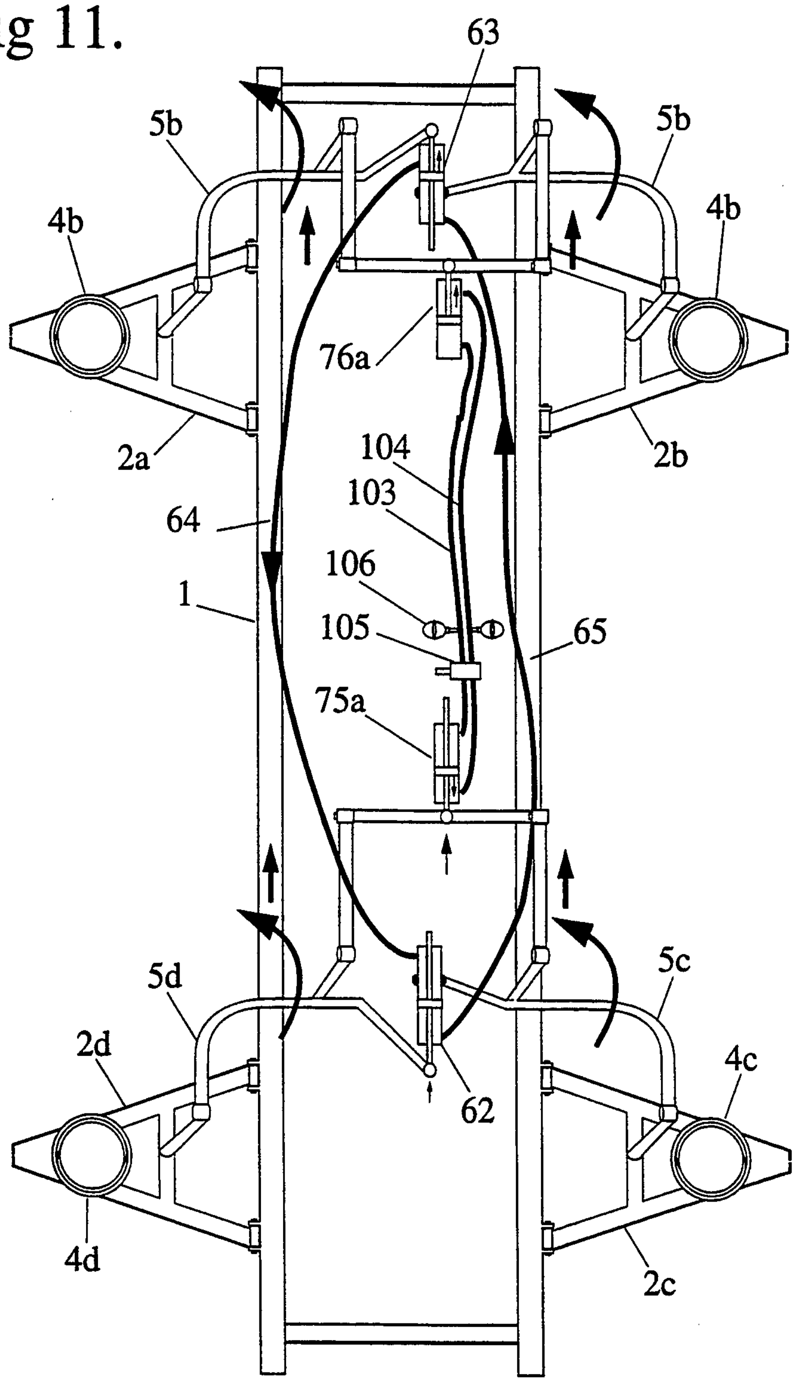


Fig 10.

Fig 11.


63
5b
5b
4b
4b
76a
2a
2b
104
103
64
106
1
105
65
75a
5d
5c
2d
4c
62
4d
2c

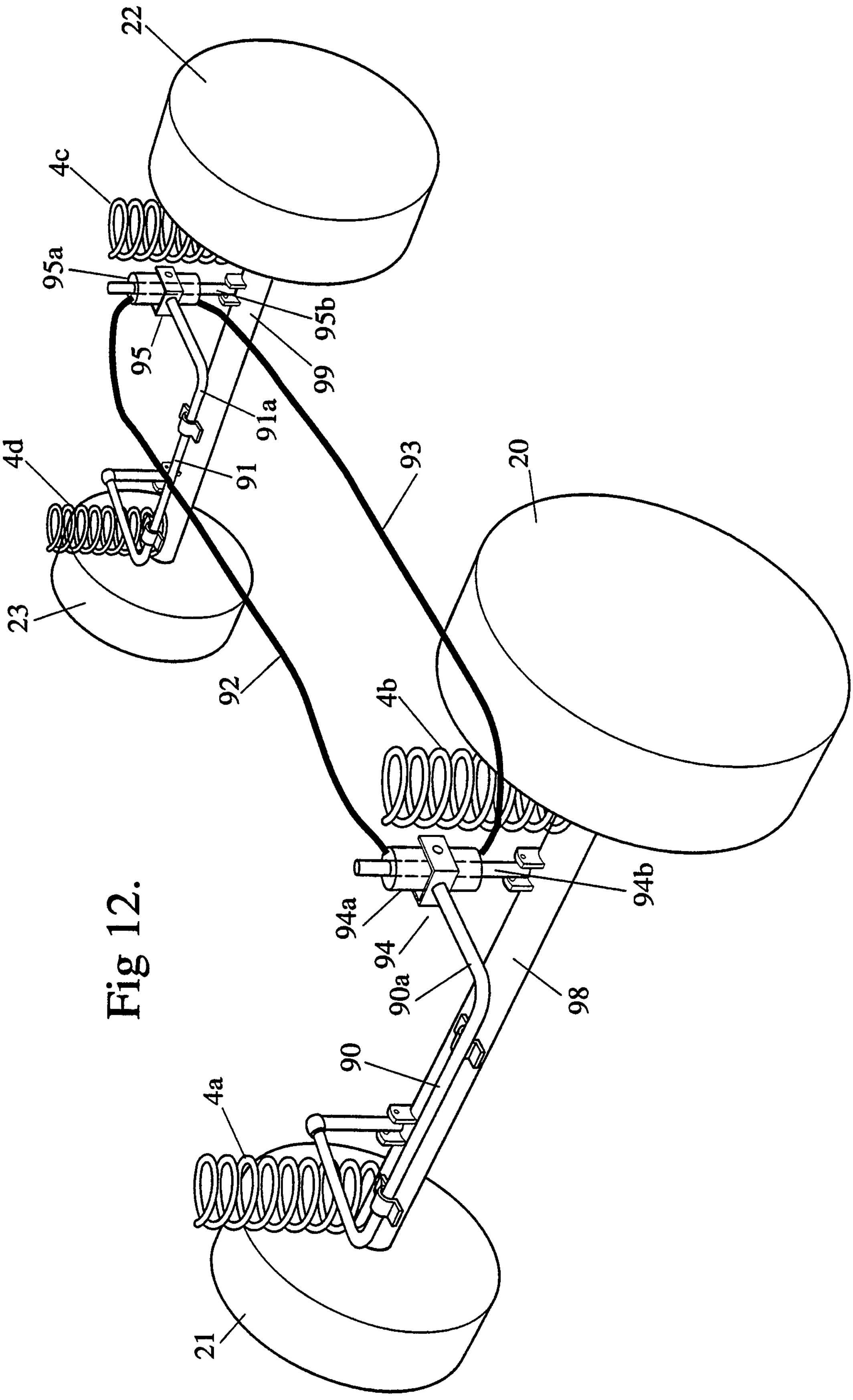

22
4c
95a
95b
95
99
91a
91
93
20
4d
23
92
4b
94b
94a
94
90a
98
90
4a
21

Fig 12.

INTERNATIONAL SEARCH REPORT

International Application No.
PCT/AU 96/00528

A. CLASSIFICATION OF SUBJECT MATTER

Int Cl[6]: B60G 21/02, 21/04, 21/05, 21/06, 11/18

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
B60G 21/02, 21/04, 21/05, 21/06, 11/18

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
AU : IPC as above

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
DERWENT

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
	WO 95/25020 A (KINETIC LIMITED) 21 September 1995	
X	Page 2 line 2 to page 6 lne 19, claims 1 & 5 and Figures 4, 6, & 9	1-15, 17-37, 40-42
Y	Page 1 lines 13-28	16, 38, 39, 43-45
	WO 95/23076 A (KINETIC LIMITED) 31 August 1995	
X	Page 4 line 19 to page 8 line 1, claims 1, 7 & 12 and Figures 10 & 11	1, 31
Y	Page 8 line 23 to page 15 line 5	7-12, 16-22, 24-29, 43-45
	WO 95/11814 A (KINETIC LIMITED) 04 May 1995	
X	Page 2 line 28 to page 7 line 29, claim 1 and Figure 1	1, 31
Y	Page 8 line 5 to 30	7-22, 24-29, 43-45

[X] Further documents are listed in the continuation of Box C   [X] See patent family annex

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier document but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
27 September 1996	15 October 1996 (15.11.96)

Name and mailing address of the ISA/AU
AUSTRALIAN INDUSTRIAL PROPERTY ORGANISATION
PO BOX 200
WODEN ACT 2606
AUSTRALIA Facsimile No.: (06) 285 3929

Authorized officer
LIONEL BOPAGE
Telephone No.: (06) 283 2153

C (Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	WO 93/01948 A (TOWER HILL HOLDINGS PTY. LTD) 04 February 1993 Page 2 line 29 to Page 5 line 22, Claim 1 and Figures 1 to 3	1, 31 7-15, 19-22, 24-29, 43-45
X Y	US 4014561 A (TOMIYA et al) 29 March 1977 Column 1 lines 29 to 59, Claim 1 and Figures 1 to 4	1-6, 23, 30-36 7-22, 24-29, 37-45
X Y	US 3197233 A (VAN WINSEN et al) 27 July 1965 Column 2 line 35 to Column 4 line 37, Claims 6 & 7 and Fig. 5	1-7, 13-15, 31-35 8-11, 16-21, 24-30, 36-45
X Y	FR 2109185 A (ÉTAT FRANCAIS) 26 May 1972 Page 1 lines 19 to 39, Claim 1 and Figures 1 and 4	1, 2, 31, 32 7-14, 19-22, 24-29
X Y	FR 1153372 A (DAIMLER BENZ AKTIENGESELLSCHAFT) 05 March 1958 Page 2 Column 1 line 55 to Column 2 line 51 and Figure 1	1-5, 7, 23, 31-36 8-22, 24-30, 37-44
X Y	EP 410675 A (KOPIECZEK) 30 January 1991 Column 2 line 20 Column 3 line 5, Claim 1 and Figures 1-3	1-5, 23, 31-36 6-22, 24-30, 41-45
X Y	EP 410676 A (KOPIECZEK) 30 January 1991 Column 2 line 31 to Column 3 line 36, Claim 1 and Figures 1-3	1-5, 23, 31-36 6-22, 24-30, 41-45
X Y	GB 926830 A (AKTIEBOLAGET VOLVO) 22 May 1963 Page 1 line 49 to Page 2 lien 45, Claim 1 and Figures 1 & 2	1-6, 17, 18, 31-36 7-16, 19-30, 37-45
X Y	GB 855257 A (DAIMLER BENZ AKTIENGESELLSCHAFT) 30 November 1960 Page 1 lne 9 to Page 2 line 66, Claim 1 and Figures 1 & 2	1, 31 2-30, 32-45
Y	858569 A (SOCIETE AUXILIAIRE DE L'ENTERPRISE) 11 January 1961 Page 1 line 10 to Page 2 line 87, Claim 5 and Figures 1 to 3	7-14, 40
X	Patent Abstracts of Japan, JP, 08011516 A (ISUZU MOTORS LTD) 16 January 1996 Whole abstract	1-5, 7, 31-35
X	Patent Abstracts of Japan, JP, 08002227 (NISSAN MOTOR CO LTD) 09 January 1996 Whole abstract	1-5, 7, 31-35

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No.
PCT/AU 96/00528

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member					
WO	95/25020	AU	19420/95				
WO	9523076	AU	79858/94	AU	17492/95	EP	725737
		WO	9511813				
WO	9511814	AU	80534/94	GB	2297299		
WO	9301948	AU	23664/92	CA	2112669	EP	599882
		US	5447332				
US	4014561	DE	2622736	FR	2315406	GB	1483172
		IT	1074909	SU	843717		
FR	2109185	DE	2149657	GB	1303239	US	3737173
EP	410676	AU	59717/90	BR	9003602	CA	2021969
		EP	410675	JP	3118206	US	5161818
EP	410675	AU	59717/90	BR	9003602	CA	2021969
		EP	410676	JP	3118206	US	5161818

END OF ANNEX