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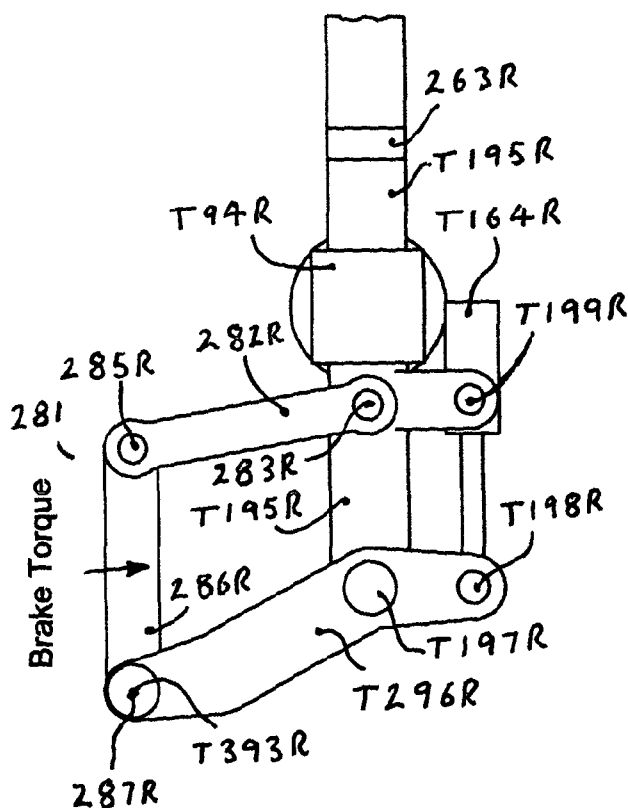


Figure 61

(57) Abstract: The invention is a new suspension system for the paired front wheels of motorcycle type vehicles employing a transversely pivoted swing arm for each wheel of the said pair where the invention improves road holding by comparison to the prior art for an equivalent vehicle mass. The invention can also provide a much better suspension deflection than the prior art can deliver without compromising the associated motorcycle type vehicle's angle of lean.

FRONT SUSPENSION SYSTEM

In this document the following terms have their associated meanings

Vehicle 1	As described in EP01998472.3 EP10003570.8 EP03253106.3
Swing arms	Paired transversely pivoted swing arms
Balance	When cornering, vehicle1 is balanced by gravity countering the centrifugal force and any other destabilising forces such as crosswinds via the stabilising actions of the rider in the same manner as a two wheeled motorcycle
Swing Arm Advantages A to F	
A	Controls destabilising events
B	Maximises stability
(i)	Negates the need for a separate chassis
(ii)	Negates the need a separate chassis with a headstock
C	Improves agility without compromising straight line stability
D	Minimises mass
E	Has an anti catapult effect
F	Allows for suspension biasing
Friction	Generated by the exponentially increasing or decreasing slip angle
Discernable roll resistance	The rider experiencing more leaning resistance than is normal for a two wheeled motorcycle
Prototype	Full size working prototype of vehicle1
Lowside	Gravity destabilises the balance to cause a motorcycle to fall down
Highside	Sudden increase in the resultant force causes a motorcycle to be initially flicked towards the upright position before falling down on its side.
Limiting width	The vehicle width that determines how easy it is for a motorcycle to filter through congested traffic
Suspension biasing	The progressive biasing of the suspension towards the wheels on one side of vehicle1 as it leans

Section G to J

G	Constant steering trail and rake
H	Steering
I	Suspension distribution
J	Damping/anti tilt brake
Steering trail	Distance from where the steering axis intersects the ground to the contact patch centre of the associated tyre with the ground.
Third transverse axis	Between each strut and associated swing arm
Fourth transverse axis	Between each strut and associated upper arm
Fifth transverse axis	For the rear ends of each upper swing arm
Straight line distance	The distance between the first and fifth transverse axes
Parallelogram or Quadrilateral	Formed between, first transverse axis, swing arm, third transverse axis, strut or upright, fourth transverse axis, upper swing arm, fifth transverse axis, straight line distance between first and fifth transverse axes.
First Problem	The greater the unsprung mass the poorer the road holding or traction
Traction	Roadholding, i.e. the time the tyre is in contact with the ground.
Sundry parts	Small unsprung components
Momentum	Force exerted by the movement of unsprung components
Second Problem	Suspension deflecting into a bump
Vehicle2	Vehicles other than vehicle1 that use paired front transversely pivoted swing arms
Prior art	Vehicle1 and vehicle2 unless individually referred to

First variation of the invention

- 1.1** Each front swing arm is supported by a resilience means
- 2.1** Each front swing arm does not move as one with the associated wheel's suspension movement
- 3.1** Each swing arm moves as one with the associated wheel's leaning movement

Second variation of the invention

- 1.2a** Each swing arm is supported by a primary resilience means.
- 1.2b** Each swing arm supports a secondary resilience means.
- 2.2** Each swing arm moves as one with the associated wheels secondary suspension movement.
- 3.2** Each swing arm moves as one with the associated wheel's leaning movement.

Embodiment 1/1 First embodiment of the first variation of the invention Figures 32-40 which use straight guides.

Embodiment 2/1 Second embodiment of the first variation of the invention figures 41-44, as per the first embodiment but using hydraulic displacement devices.

Embodiment 3/1 Third embodiment of the first variation of the invention figure 45 where the piston rod is also the king pin

Embodiment 1/2 First embodiment of the second variation of the invention is where embodiments 1/1 2/1 3/1 provide a primary suspension movement and each swing arm support a secondary suspension means.

Embodiment 2/2 Second embodiment of the second variation of the invention is where the primary suspension movement is provided by a lever arm but can also be applied to the first variation of the invention.

Lower bump frequency Means shallower incline from the start of the bump to the top of the bump and shallower decline from the top of the bump to the end of the bump of a larger bump by comparison to a smaller bump.

In the Figures the following numbers refer to their associated components and in the Figures L = left side R = right side T = trailing

1	left front wheel
2	right front wheel
201	right tyre
202	right rim
203	right rim mounted brake disc
3	ground
4	left exponentially increasing slip angle
104	left greater exponentially increasing slip angle
5	right exponentially increasing slip angle
105	right greater exponential increasing slip angle
6	left exponentially decreasing slip angle
106	left greater exponential decreasing slip angle
7	right exponentially decreasing slip angle
107	right greater exponential decreasing slip angle
8	left back wheel
9	right back wheel
10	left front swing arm
11	right front swing arm
12	left back swing arm
13	right back swing arm
14	first transverse axis
15	second transverse axis
16	represents vehicle 1's main body
17	left steering axis
18	centre of wheel 1
19	wheel 1's contact centre with ground 3
20	steering axis 17 intersection with ground 3
21	left third transverse axis
22 122	left strut or upright
23	left fourth transverse axis
24	left upper swing arm
25	fifth transverse axis
26	straight line distance
27	right third transverse axis
28	right strut or upright
29	right fourth transverse axis

30	right upper swing arm
31	right steering axis
32	left rear swing arm joint
33	right rear swing arm joint
34	left front swing arm joint
35	right front swing arm joint
36	left strut and upper arm joint
37	right strut and upper arm joint
38	left upper arm rear joint
39	right upper arm rear joint
40	left stub axle assembly
41	left king pin
42	right stub axle assembly
43	right king pin
44	left steering rod
45	left steering rod joint
46	right steering rod
47	right steering rod joint
48 480	left lug
49 490	left connecting link
50 500	left lug joint or rotational connection
51 151	balance beam
52	left balance beam joint
53 530	right lug
54 540	right connecting link
55 550	right lug joint or rotational connection
56	right balance beam joint
57	carrier – linear
58	balance beam axis
59	guide rods for linear carrier
60	resilience means
61	left suspension damper
62	right suspension damper
63 163 263	upper frame
64 164	resilient/damping means
65	bracket
66	location fasteners
67 167	guide
68 168	housing
69 169	lower frame

70	ball joint
71 171 371	hydraulic displacement device or resilience/damping means
72 172 272 372	cylinder
73 173	piston
74 174 274 374	piston rod
75 175	cavity
76 176	opening
77	hydraulic line
78	damper device
79	resilience means

80	bridging portion
81 181 281	torque arm assembly
82 182 282	upper link
83 183 283	rotational connection
84	bracket
85 185 285	rotational connection
86 186 286	lower link
87 187 287	rotational connection
88 188	bracket
89	bracket

90	bracket
91	drop link
92	boss
93 193 293 393	stub axle
94	knuckle joint
95 195	king pin
96 196 296	lever arm
97 197	leading arm rotational axis or rotational connection
98 198	rotational connection
99 199	rotational connection

100	brake plate
601	damping/resilience means
602	damping/resilience means

Background Information

This invention relates to the various embodiments of the motorcycle type vehicle described in EP01998472.3, EP10003570.8, and EP03253106.3, hereinafter referred to as vehicle1, which use paired transversely pivoted swing arms, hereinafter also referred to as swing arms. When cornering, vehicle1 is balanced by gravity countering the centrifugal force and any other destabilising forces such as cross winds via the stabilising actions of the rider in the same manner as a two wheeled motorcycle, hereinafter referred to as balance. Vehicle1 uses paired transversely pivoted swing arms which advantageously:

- A. Controls destabilising events.
- B. Maximises stability.
- C. Improves agility without compromising straight line stability.
- D. Minimises mass.
- E. Has an anti catapult effect.
- F. Allows for 'suspension biasing'.

The above advantages are explained below:

- A. Controls destabilising events.

Brief explanation of A

As vehicle1 leans away from the upright position its track exponentially increases and as vehicle1 leans toward the upright position its track exponentially decreases. This exponentially increasing and exponentially decreasing track generates an exponentially increasing and exponentially decreasing slip angle, respectively. Should vehicle1 experience lowside and highside types of destabilising events the exponentially increasing and exponentially decreasing slip angle generates sufficient friction between the associated tyres and ground to absorb much of the energy of the said destabilising events to slow the said destabilising events down which allows the rider to maintain control.

Full explanation of A

Vehicle1's paired transversely pivoted swing arms provide an increasingly wide track between their associated paired wheels as vehicle1 increasingly leans away from the upright position. Thus, by way of a non limiting example, a 460mm track as per Figure 1 when vehicle1 is upright or vertical increases to, for instance, 531mm when vehicle1 as per Figure 2 leans by 30 degrees in either direction. Figure 1 depicts the end view of the said paired wheels 1 and 2 in contact with the ground 3 generating a 460mm track when vehicle1 is at 0 degrees angle of lean, i.e. vertical or upright. Figure 2 shows Figure 1 but with vehicle1 leaning at 30 degrees from the vertical in both directions with wheels 1 and 2 generating a 531mm track, which is 71mm more than in Figure 1.

The track increase is also exponential. Based on the Figure 1 example of a 460mm track, a 1 degree increase in lean from 0 degrees to 1 degree causes the track to widen by 0.07mm. Whereas the same 1 degree increase in lean from, for instance, 22 degrees to 23 degrees causes the track to widen by 3.6mm. The graph in Figure 3 shows the exponential differences from 0 degrees to 45 degrees in 5 degree increments for the Figure 1 example of a 460mm track. Track dimensions that are bigger than the 460mm Figure 1 example generate bigger exponential differences and track dimensions that are smaller than the 460mm Figure 1 example generate smaller exponential differences. But whatever vehicle1's upright track dimension is, the exponential difference is a feature of vehicle1's paired transversely pivoted swing arms.

Thereby, as vehicle1 leans away from the upright position the track exponentially increases, and as vehicle1 leans towards the upright position the track exponentially decreases.

When vehicle1 is travelling, the exponentially increasing track moves the associated tyres through an exponentially increasing slip angle which generates an exponentially increasing amount of friction between the said tyres and the surface they are in contact with, hereinafter referred to as friction.

Figure 4 shows the plan view of Figure 1 combined with Figure 3's exponentially increasing track dimensions to show; the resulting exponentially increasing slip angles 4 and 5 that are generated between the 460mm track and 650.54mm track as vehicle1 leans away from the upright position to a 45 degree incline respectively.

When vehicle1 is travelling the exponentially decreasing track moves the associated tyres through an exponentially decreasing slip angle which generates an exponentially decreasing amount of friction, hereinafter referred to as friction.

Figure 5 shows the 650.54mm exponentially increased track of Figure 4 combined with Figure 3's exponentially decreasing track dimensions to show the resulting exponentially decreasing slip angles 6 and 7 generated between the 650.54mm track and 460mm track as vehicle1 leans from a 45 degree incline to the upright position respectively.

The faster the rate of lean away from or towards the upright position the greater is the exponential slip angle and resulting friction for a given travelling speed of vehicle1.

Figure 6 shows Figure 4 but with a faster rate of lean which produces a greater exponentially increasing slip angle 104 and 105 and resulting friction.

Figure 7 shows Figure 5 but with a faster rate of lean which produces a greater exponentially decreasing slip angle 106 and 107 and resulting friction.

The travelling speed of vehicle1 also determines the resulting friction. For instance Figures 6 and 7 can also be interpreted as having the same rate of lean as Figures 4 and 5 respectively but with vehicle1 travelling slower than in Figures 4 and 5.

It can be appreciated that if the rate of lean was great enough the resulting friction would resist the rate of lean by providing vehicle1 with discernable roll resistance. Roll is a term commonly used to describe a motorcycle's leaning action. Discernable roll resistance is a term herein used to convey that the rider is experiencing more leaning resistance than is normal for a two wheeled motorcycle.

Experiments, over a wide range of speeds with a full size prototype of vehicle1, show that in normal and even aggressive leaning movements the rider does not experience any increase in the roll resistance by comparison to an equivalently dimensioned two wheel motorcycle. This is because slip angles are a normal function of motorcycle tyres anyway when cornering, and the rate of lean that a rider can generate is too slow for vehicle1 to produce a sufficiently steep slip angle and resulting friction to create discernable roll resistance. This conclusion is also supported by calculations. Furthermore, the prototype's tyre wear favourably compares to that of a two wheeled motor scooter with the same tyres. This would not be the case if the prototype's tyres were generating a greater slip angle than a two wheeled motor scooter with the same tyres. Just like a two wheeled motorcycle at slow speeds, vehicle1's angle of lean is minimal and therefore avoids the greater slip angles associated with the greater angles of lean of faster speeds.

However, experiments on very slippery surfaces over a range of speeds with the same and unaltered prototype show that in destabilising events such as lowsides and highsides the rider does experience a clear increase in the roll resistance compared to a similarly destabilised two wheeled motorcycle. This is because the rate of lean in such destabilising events is much faster than a rider can generate. Consequently the resulting friction is greater too which absorbs much of the energy from a lowside and highside which causes discernable roll resistance by resisting the rate of lean. The increase in the roll resistance during lowsides and highsides is extremely beneficial because it proportionally slows down these destabilising events to provide the rider with a lot more time to regain control in what would otherwise be uncontrollable destabilising events for a two wheeled motorcycle.

A lowside destabilising event is when gravity destabilises the balance and causes a two wheeled motorcycle to fall down on its side. A highside destabilising event is when a sudden increase in the resultant force causes a two wheeled motorcycle to be initially flicked towards the upright position before falling down on its side. A highside always starts from a leaning position.

In a lowside the rate of fall is initially at its least and then exponentially increases under the accelerating influence of gravity. In relation to vehicle1 the exponentially increasing rate of fall is conveniently resisted by the exponentially increasing amount of friction. Conversely, the resultant force that causes a highside is initially at its greatest and then its influence dissipates. In relation to vehicle1 the initially high resultant force of a highside is conveniently resisted by the initially high amount of friction derived from vehicle1's leaning position and associated slip angles. The friction then exponentially decreases with the dissipating resultant force as vehicle1 moves towards the upright position. Therefore in both lowsides and highsides the changing magnitude of the destabilising force is tracked by a proportional magnitude of friction.

B. Maximises stability

Brief explanation of B

Because vehicle1's swing arms are transversely pivoted they are ideally positioned to connect with vehicle1's main structure at a very low point without requiring a two wheeled motorcycle style headstock. The headstock is the part of a conventional motorcycle's frame where the telescopic front forks are rotationally connected, to provide steering. The low centre of mass that can be achieved from the absence of a headstock and exponentially widening track can keep vehicle1 from falling over even beyond a 45 degree angle of lean and even with the addition of a rider without becoming unduly heavy or wide.

Full explanation of B

Vehicle1's transversely pivoted swing arms can also promote a very low centre of mass by being connected to vehicle1's main structure, or sprung mass, at a very low point. This negates the need for (i) a separate chassis because the swing arms and associated components can be connected directly to, for instance, the engine/transmission unit or (ii) a separate chassis with a high level headstock. The absence of (i) or (ii) reduces mass and lowers vehicle1's centre of mass. The combination of vehicle1's low centre of mass and exponentially increasing track provides a great amount of stability. The actual amount of stability is determined by the centre of mass height and track dimension. Using the 460mm upright track from Figure 1 as a non limiting example; calculations show that a stationary 190kg vehicle1 would not topple over when leaning up to and beyond 45 degrees with a 90kg rider sitting in the conventional two wheeled motorcycle riding position and vehicle1 having 130mm of ground clearance. The advantage here is that if control could not be regained in a destabilising event vehicle1 would then not topple over onto its rider as it came to rest, on level ground, despite being fully leant over and with only gravity acting through the combined centre of mass of vehicle1 and its rider. Furthermore, in such instances all of the tyres would remain in contact with the ground which means they can still impart steering, braking and acceleration actions to vehicle1 before vehicle1 came to rest which would not be possible if vehicle1 was on its side. The same would apply on non level ground until gravity acting through the combined centre of mass of vehicle1 and its rider acted outside of vehicle1's footprint formed by its wheels.

The Figure 8 calculation determines the overall centre of mass height com between vehicle1 and its rider to be 459.2mm from the ground when; vehicle1's 190kg mass has its centre (M1) at 240mm from the ground and the rider's 90kg mass has its centre (M2) at 925mm from the ground.

Figure 9 shows the Figure 8 com result applied to Figure 1 wherein the arrow represents gravity acting through com to a midway position P between the respective contact patches r and q of wheels 1 and 2 with the ground 3.

Figure 10 shows Figure 9 but with vehicle1 leaning by 45 degrees and generating the Figure 3 track dimension for 45 degrees wherein; the line com-P is parallel to and midway between the 45 degree inclined wheels 1 and 2. The distance 459.2mm of com-P is the same as in Figure 9 because as vehicle1 leans, wheel 1 moves upward by the same amount as wheel 2 moves downward, or visa-versa if vehicle1 was leaning

in the other direction. Thereby, the distance of com from ground 3 in a parallel plane to wheels 1 and 2 is unaltered from Figure 9. The arrow in Figure 10 represents gravity acting through com to form a right angle with ground 3 at position S. A right angled triangle can be formed between com-S and P-S by which P-S can be calculated, as shown in Figure 11.

In Figure 11 the base P-S of right angle triangle P-S-com equals 324.7mm which is less than half of the 650.54mm Figure 10 track r-q ($650.54\text{mm} / 2 = 325.27\text{mm} - 324.7\text{mm} = 0.57\text{mm}$). Stability is therefore generated because gravity acts between the two wheels 1 and 2 to prevent vehicle1 from falling over.

In this calculation the threshold of stability appears only to be 0.57mm ($325.27\text{mm} - 324.7\text{mm} = 0.57\text{mm}$) but as vehicle1 leans each tyre's contact patch moves away from the centre line of the associated tyre. Therefore, using a 120mm wide tyre with a round profile as a non limiting example, the actual threshold becomes 43mm as the tyre adds 42.43mm to the said 0.57mm threshold ($42.43\text{mm} + 0.57\text{mm} = 43\text{mm}$) at a 45 degree angle of lean before any additional stability from the squashing of the tyres is considered. The moving of the tyre's contact patch away from the tyre's centre line provides a useful margin of stability especially if the ground is not completely flat. Lowering of the centre of mass by the suspension's compression due to the addition of centrifugal force has not been included because the aim here is to determine the extent of stability when vehicle1 has come to a rest at a 45 degree angle of lean which would not include centrifugal force.

The very low centre of mass result com in Figure 8 can be best achieved by vehicle1's suspension and steering being connected to an engine configuration where the cylinder/s are horizontal and in close horizontal proximity to the other engine and transmission components such as disclosed in EP10250390.1.

If vehicle1 had a static track instead of the exponentially widening track, then when upright vehicle1 would need a much wider track to generate the same amount of stability. This would compromise vehicle1's ability to filter through traffic which greatly depends on vehicle1's upright width remaining narrow enough to provide easy filtering, with the exception of the handlebars.

For instance, for a maximum angle of lean of 30 degrees a static track would have to measure 531mm when vehicle1 was upright to generate the same amount of stability as the exponentially widening track measuring 460mm when vehicle1 was upright. Thereby a static track would add 71mm to vehicle1's width.

For a maximum angle of lean of 45 degrees a static track would have to measure 650mm (650.54mm) when vehicle1 was upright to generate the same amount of stability as the exponentially widening track measuring 460mm when vehicle1 was upright. Thereby a static track would add 190mm to vehicle1's width.

The addition of, for instance, 120mm wide tyres to vehicle1 with the said 460mm upright track would result in vehicle1 being 580mm wide when upright.

The addition of the 120mm wide tyres to vehicle1 with the said 531mm upright track results in vehicle1 being 651mm wide when upright. This is 71mm more than vehicle1 with the said 580mm width.

The addition of the 120mm wide tyres to vehicle1 with the said 650mm upright track results in vehicle1 being 770mm wide when upright. This is 190mm more than vehicle1 with the said 580mm width.

On a conventional two wheeled motorcycle without a fairing and side boxes, the width that determines how easy it is for a motorcycle to filter through congested traffic, hereinafter referred to as the limiting width, is usually taken from the outside of one of the rider's knees to the outside of the rider's other knee. Limiting widths below 610mm makes a little difference in the associated motorcycle's ability to filter through congested traffic because the average rider's width is still the same. However, when the 610mm threshold, or thereabouts, has been breached the ability of the associated motorcycle to filter through traffic is drastically reduced.

The 610mm maximum limiting width conclusion is based on a study of a major European capital's motorcycle couriers, and their companies, who business it is to continually filter through congested traffic to quickly deliver important packages.

The above 651mm and 770mm limiting widths of vehicle1 with a static track would both exceed the said 610mm limiting width conclusion from the motorcycle courier study.

Based on the Figure 8 results shown in the Figure 10 example of stability, the compromises that are required to achieve the same stability from a 460mm static track are either, increasing vehicle1's weight from 190kg to an extreme 640kg, or reducing vehicle1's centre of mass to 43mm from the ground, or biasing the suspension, see section F below, to an extreme amount so that the two wheels lying on the outside of the bend carry 86.4% of vehicle1's weight. All of which are impractical.

Therefore the stability that can be generated by vehicle1's exponentially widening track cannot be matched by a static track without significant negative compromises elsewhere in vehicle1's design, even with combinations of the said compromises.

Vehicle1's exponentially increasing and decreasing track maximises stability whilst leaning, which is when stability is most needed, and minimises vehicle1's width when upright which is when stability is less needed but where the narrower width is more beneficial for traffic filtering when leaning is negligible.

The above relationship between vehicle1's width and its ability to filter through congested areas excludes the width of the handlebars. Handlebars can be manoeuvred around obstacles like car mirrors etc more easily than vehicle1's main structure, and are therefore not considered to be a limiting factor in filtering through congested areas. Nevertheless, vehicle1's commuting ability would be further enhanced by using the narrowest practical handlebars.

C. Improves agility without compromising straight line stability

Brief explanation of C

Vehicle1's transversely pivoted swing arms shortens the wheelbase when cornering to enhance cornering agility, and lengthens the wheelbase when upright to enhance straight line stability.

Full explanation of C

It is well known that for a given wheelbase an associated two wheeled motorcycle's cornering agility will be increased if the said wheelbase was shortened and the straight line stability will be increased if the said wheelbase was lengthened.

In two wheeled motorcycle design the wheelbase measurement is largely a compromise between the need for straight line stability and the need for cornering agility for any particular purpose such as touring, sport, trials etc. This is because the wheelbase measurement on two wheeled motorcycles remains virtually unchanged between straight line travelling and cornering.

Another advantage of vehicle1's paired transversely pivoted swing arms is that vehicle1's average wheelbase exponentially shortens as vehicle1 progressively leans away from the upright position, and exponentially lengthens as vehicle1 progressively leans towards the upright position. The reference to average wheelbase means the wheelbase as measured between the mid front track and mid rear track.

Consequently, vehicle1's cornering agility is improved for a given amount of wheelbase generated straight line stability by comparison to a two wheeled motorcycle. The shortening and lengthening of vehicle1's wheelbase is maximised when the front and rear ends of vehicle1 both employ paired transversely pivoted swing arms as shown in Figures 13 and 15.

Based on Figure 2, Figure 12 shows that the up and down movement of vehicle1's paired wheels 1 and 2 generated by a 30 degree angle of lean is 133mm when vehicle1's upright track, as in Figure 1, measures 460mm.

The simplified diagram in Figure 13 depicts vehicle1 as seen from the direction of Figure 12's arrow. In Figure 13 the upwards and downwards path taken by each front wheel 1, 2, and back wheel 8, 9 is determined by their 310mm long transversely pivoted swing arms 10, 11, 12, 13, respectively. Whereby, the front pair of the transversely pivoted swing arms rotate about a first transverse axis 14 and the rear pair of the transversely pivoted swing arms rotate about a second transverse axis 15.

The first and second transverse axes 14 and 15 respectively are separated from each other by a non limiting measurement example of 850mm of vehicle1's main body which in Figure 13 is represented by shortened box 16.

In Figure 13 the wheelbase measures 1410mm. The 1410mm wheelbase has been calculated from the right angle triangle derived from the 310mm swing arm length (hypotenuse), the 133mm upright derived from Figure 12, whereby the base measurement must equal 280mm as shown in Figure 14, therefore $280\text{mm} \times 2 = 560\text{mm} + 850\text{mm} = 1410\text{mm}$ as shown in Figure 13.

It can be appreciated that when vehicle1 as depicted in Figure 13 moves back to the upright position its transversely pivoted swing arms will be horizontal and generate a longer wheelbase. Figure 15 shows Figure 13 but where vehicle1 is upright wherein the wheelbase measures 1470mm ($310\text{mm} \times 2 = 620\text{mm} + 850\text{mm} = 1470\text{mm}$) which is 60mm longer than in Figure 13.

The measurements in Figures 12, 13, 14, 15 are for the purpose of demonstration only and are non limiting examples. Different dimensions and angles of lean may apply without diverting from the theme. For instance, a greater angle of lean, shorter swing arm lengths, and wider upright track, are all variations that would increase the difference between the upright wheelbase measurement and cornering wheelbase measurement. Conversely, a smaller angle of lean, longer swing arms, and narrower upright track are all variations that would reduce the difference between the upright wheelbase measurement and cornering wheelbase measurement.

D Minimises mass.

Brief explanation of D

Vehicle1's transversely pivoted swing arms can be positioned sufficiently low and are ideally aligned to be connected directly to the engine crankcase. This enables vehicle1 not to need a separate chassis which reduces mass. The reduction of mass has many advantages.

Full explanation of D

As mentioned in B above, vehicle1's transversely pivoted swing arms do not require a separate chassis or frame. Therefore a considerable saving in mass can be made by comparison to vehicle1 having a separate chassis or frame. The mass reduction improves acceleration, fuel consumption, tyre wear, braking, handling, manoeuvrability, and reduces production costs. The mass reduction also permits vehicle1 to gain the extra mass of a fourth wheel while comfortably remaining within overall acceptable mass limits. For instance, calculations show that a 400cc version of vehicle1 with four wheels need not exceed 190kg. Vehicle1's fourth wheel doubles the lowside and highside control of advantage A above, doubles the stability of advantage B above, increases the gyroscopic wheel stability by a third, and improves braking and cornering, by comparison to a three wheel version of vehicle1.

The absence of a separate chassis allows for a lower centre of mass design which, better distributes the braking force between the front and rear wheels to further improve braking, centralises vehicle1's mass between its front and back wheels which reduces understeer or/and oversteer, and thereby improves handling. Vehicle1's transversely pivoted swing arms are ideally aligned to be directly connected to the, for instance, engine/transmission unit which is why a separate chassis is not required and hence the great saving in weight. For the purpose of just reducing mass the transversely pivoted swing arms of vehicle1 can be directly connected to any known engine/transmission configuration with substantially upright cylinder/s. Where the low mass has to be combined with a very low centre of mass the transversely pivoted swing arms of vehicle1 can be directly connected to the engine/transmission unit where at least one cylinder is horizontal and in close horizontal proximity to the other engine and transmission components as per EP10250390.1.

E. Has an anti catapult effect

When a motorcycle type vehicle with front forks mounted in a conventional headstock or similar crashes into an obstacle the tendency is for the rear of the vehicle to lift upwards and catapult the rider off. In such scenarios the resistance of the front forks causes the vehicle's high centre of mass, by comparison to vehicle1, to rotate around the axle of the front wheel which causes the rear of the vehicle to lift and in the process catapult the rider forwards and upwards.

Should such an accident occur on vehicle1 the first transverse axis 14 of the front swing arms 10 and 11 will tend to ride up and over the front wheels 1 and 2. This is because first transverse axis 14 would not offer any resistance, unlike conventional headstock mounted forks or similar, combined with vehicle1's low centre of mass. Consequently the front of vehicle1 will lift instead of the rear resulting in the rider not being catapulted off, which also makes vehicle1's use of an airbag more effective than a conventional motorcycle.

F. Suspension biasing**Brief explanation of F**

Vehicle1's suspension linkages can be so arranged to allow vehicle1 to lean when cornering at a reduced angle from the vertical than a two wheeled motorcycle for an equivalent, resultant force, trail, wheelbase, tyre section, centre of mass height, and environmental conditions. The reduced leaning angle has many advantages.

Full explanation of F

Vehicle1's arrangement of its interconnecting suspension linkages and rotational axis for paired wheels can be designed to allow vehicle1 to lean when cornering at a reduced angle from the vertical than a two wheeled motorcycle when the other factors that influence the angle of lean, as detailed above in the brief explanation, are the same. This is achieved by the said arrangement progressively shortening the leverage from the resilience means (e.g. suspension spring) to the associated wheel of each pair laying on the outside of a bend as vehicle1 progressively leans. Consequently the suspension acting on the wheels laying on the outside of a bend becomes progressively stiffer, as vehicle1 progressively leans, than the suspension acting on the wheels laying on the inside of a bend. The progressive biasing of the suspension as vehicle1 leans is hereinafter referred to as suspension biasing.

Therefore, to maintain balance the resultant force derived from gravity and the centrifugal force is positioned closer to the wheel of each pair lying on the outside of a bend than otherwise would be the case.

This means vehicle1 does not have to lean so far to negotiate a corner for a given resultant force than otherwise would be the case. This has the following benefits; improved agility, promotes rider confidence, allows for flatter section tyres which improves grip and reduces tyre wear, less need for angular ground clearance which allows for some of vehicle1's components to be positioned lower resulting in a lower centre of mass with the previously mentioned benefits thereof.

A further advantage is as follows: As is well known different types of tyres interact with the ground with sufficient difference that their individual characteristics are discernable to the rider. Should the characteristics of a desired tyre type actually cause discernable roll resistance during aggressive riding involving quick and extreme angles of lean, derived from the associated friction generated by the associated steeper slip angles, then the suspension biasing could be designed to avoid the offending steep slip angles and yet still allow vehicle1 to corner at a speed normally associated with the said offending slip angle.

A side effect of suspension biasing is that the overall friction is reduced during cornering. This is because with suspension biasing the vehicle's weight is increasingly carried by the two wheels lying on the outside of a bend as vehicle1 progressively leans. In an extreme design of suspension biasing all of vehicle1's weight could be carried by just the two wheels lying on the outside of a bend during cornering. Therefore, the suspension biasing design has to ensure that the distribution of vehicle1's weight between paired wheels stays within desirable parameters. The suspension biasing may need any of the, rake, trail, Ackermann angle, steering offset, camber, settings to be altered from their normal non suspension biasing settings. Different amounts of suspension biasing could be applied a different ends of vehicle1.

Experiments with the said prototype of vehicle1 with its anti tilt suspension brake activated at full lean whilst cornering a little faster than normal for the angle of lean emulated suspension biasing, whereby no undue cornering effects were experienced.

Conversely, the suspension could be biased towards each wheel of a pair lying on the inside of a bend. This can be achieved by designing vehicle1's interconnecting suspension linkages and rotational axis for paired wheels to progressively shorten the leverage from the resilience means (e.g. suspension spring) to the associated wheel of each pair lying on the inside of a bend as vehicle1 progressively leans. Consequently the suspension acting on the wheels lying on the inside of a bend becomes progressively stiffer, as vehicle1 progressively leans, than the suspension acting on the wheels lying on the outside of a bend. This arrangement would require vehicle1 to lean at a greater angle than the resultant force that it is producing and may aid, for instance, an off-the-highway learner to become familiar with leaning vehicle1 at very slow speeds in the interest of safety.

Unlike advantages A, B, C, D, E, the advantage F is not unique to transversely pivoted swing arms as the same could apply to longitudinally pivoted swing arms although not to the same extent; the static track of longitudinally pivoted swing arms being a handicap to the scope of suspension biasing.

Note: Another advantage of paired transversely pivoted swing arms is the pitch, not track, between the associated paired wheels remains constant irrespective of the independent up and down movement of the associate wheels which makes the drive train to typically the rear wheels easy. The said constant pitch also ensures that suspension deflections do not cause the unsprung mass to generate a destabilising transverse force across vehicle1.

Additional components required to deliver the advantages A, B, C, D, E, F

As can be seen from A, B, C, D, E, F, above the use of transversely pivoted swing arms provides many unique benefits to vehicle 1. To practically install transversely pivoted swing arms for paired wheels requires additional components to make provision for:

- G. Constant steering trail and rake**
- H. Steering**
- I. Suspension distribution**
- J. Damping/anti tilt brake**

As follows:

G. Constant steering trail and rake

As can be seen from Figure 12 the leaning movement of vehicle 1 causes one wheel of a pair to move upwards and causes the other wheel of the same pair to move downwards relative to vehicle 1's main structure. The actual amount of the said upward and said downward movement is mainly determined by vehicle 1's upright track measurement and angle of lean. Thus the 133mm upward movement and 133mm downward movement in Figure 12 is derived from vehicle 1 leaning by 30 degrees and having a 460mm upright track, because Figure 12 is based on the 30 degree angle of lean of Figure 2 which in turn is based on the 460mm upright track of Figure 1.

The suspension then adds further upward and downward movement. For instance, a total suspension movement of 125mm of which 75% is in compression and 25% is in extension would add a further 94mm ($125\text{mm} \times 75\% = 94\text{mm}$) to the 133mm upward movement and add a further 31mm ($125\text{mm} \times 25\% = 31\text{mm}$) to the 133mm downward movement. Thereby the total upward movement would equal 227mm ($133\text{mm} + 94\text{mm} = 227\text{mm}$) and the total downward movement would equal 164mm ($133\text{mm} + 31\text{mm} = 164\text{mm}$).

The said upward and downward movement would disastrously alter vehicle 1's positive steering trail for paired steered wheels if the angle between the steering axis and associated swing arm was fixed. Figure 16 shows, wheel 1, swing arm 10, and the first transverse axis 14, from Figure 15 combined with a steering axis 17 and ground (e.g. road surface) 3. The steering axis 17 is generating a typically average and positive steering trail value of +75mm. For calculation purposes the radius of wheel 1 is 235mm. The angle of the steering axis from the vertical is +17.7 degrees which is derived from Figure 16's right angled triangle formed between, the centre 18 of wheel 1, the centre 19 of wheel 1's contact patch with the ground 3, and the intersection 20 of the steering axis 17 with the ground 3.

Figure 17 shows Figure 16 but with wheel 1 raised upwards by the said 227mm which causes swing arm 10 to rotate upwards by 47.1 degrees about the first transverse axis 14. The said 47.1 degrees has been derived from Figure 17's right angled triangle which is formed from the 310mm length of swing arm 10 and the said 227mm upwards movement. If the steering axis 17 was in a fixed angular relationship with the swing arm 10 then the said 47.1 degree angular movement of swing arm 10 from

Figure 16 to Figure 17 would increase the steering axis 17 angle to +64.8 degrees ($17.7 \text{ degrees} + 47.1 \text{ degrees} = 64.8 \text{ degrees}$) which increases the positive steering trail to +499.4mm ($\tan 64.8 \times 235\text{mm} = 499.4\text{mm}$).

Figure 18 shows Figure 16 but with wheel 1 lowered by the said 164mm which causes swing arm 10 to rotate downward by 31.9 degrees about the first transverse axis 14. The said 31.9 degrees has been derived from Figure 18's right angled triangle which is formed from the 310mm length of swing arm 10 and the said 164mm downward movement. If the steering axis 17 was in a fixed angular relationship with the swing arm 10 then the said 31.9 degree angular movement of swing arm 10 from Figure 16 to Figure 18 would reduce the steering axis 17 angle to -14.2 degrees ($17.7 \text{ degrees} - 31.9 \text{ degrees} = -14.2 \text{ degrees}$) which reduces the +75mm Figure 16 positive steering trail value to a -59.5mm negative steering trail value ($\tan 14.2 \times 235\text{mm} = 59.5\text{mm}$).

In the above calculations the trail of wheel 1 changes between +499.4mm to -59.5mm whilst vehicle1 leans by 30 degrees in one direction to 30 degrees in the other direction. Wheel 1 is one of a pair of wheels, the other wheel being wheel 2 as per Figure 13. In the above calculation as wheel 1 is generating a +499.4mm trail, wheel 2 would be generating a -59.5mm trail, and visa-versa.

The above non limiting calculations demonstrate that if the steering axes were in a fixed angular relationship with their associated swing arms it would drastically alter the trail of the associated wheels.

The steering trail, sometimes known as the castor effect, is the distance between where the projected steering axis would intersect the ground and the contact patch centre of the associated tyre with the ground. As vehicle1 travels, the said contact patch will always try to trail behind the said steering axis intersection away from the direction of travel like a castor. The steering trail is the major factor that causes a steered wheel to self centre or self align. Therefore, vehicle1 is designed to travel with a positive steering trail, examples of which are shown in Figures 16, 17, 19, 20.

The alternative of a negative steering trail as shown in Figure 18 would cause the associated steered wheel to try and spin around its own steering axis to form a positive steering trail. Therefore, negative steering trail would be very dangerous and has to be avoided.

In relation to Figure 17, if the positive steering trail is too great it can also cause problems by generating too much self aligning force which can continually over shoot the in-line to the direction of travel position with increasing force and generate a dangerous and increasingly violent left and right steering oscillation.

It may be argued that the varying trail between vehicle1's paired wheels does not matter because as vehicle1 leans the detrimental aspects associated with the increasing positive trail of one steered wheel of a pair is remedied by the opposite detrimental aspects associated initially with the decreasing positive trail and then with the increasing negative trail of the other steered wheel of the same pair, and visa versa. The effect from the trail of one wheel of a pair is transferred to the other wheel of the same pair by the interconnecting steering linkages. However should one wheel

of a pair lose or reduce traction with the ground then any detrimental trail aspects from the other wheel generating more traction would dominate.

Therefore, vehicle1 needs a means to keep the trail of both steered wheels within workable positive parameters despite the up and down movement of the associated paired wheels. To achieve this, the angular relationship between vehicle1's paired steering axes and their associated swing arms is not fixed.

Instead, each steering axis can rotate about a third transverse axis located towards or at the front of the associated swing arm in the form of a predominately upright strut.

The upper end of each strut can rotate about a fourth transverse axis located towards or at the front end of its own dedicated upper swing arm.

The rear ends of the two upper swing arms rotate about a common fifth transverse axis laying above the swing arm's common first transverse axis. Thereby, the first transverse axis, swing arms, third transverse axes, struts, fourth transverse axes, upper swing arm, fifth transverse axis, and straight line distance between the first and fifth transverse axes, form a quadrilateral or parallelogram for their associated wheel by which the associated steering trail can remain within workable positive parameters.

Figure 19 shows Figure 17 but with the addition of the above mentioned parallelogram mechanism comprised from first transverse axis 14, swing arm 10, third transverse axis 21, strut 22, fourth transverse axis 23, upper swing arm 24, fifth transverse axis 25, straight line distance 26. The first transverse axis 14 and fifth transverse axis 25 are joined to vehicle1's sprung mass which is represented by distance 26. Figure 20 shows Figure 19 but with swing arm 10 moved to the Figure 18 position.

In Figures 19 and 20 the third transverse axis 21 is shown in-line with the wheel's centre 18 but this does not have to be the case, the Figures are only to convey the theme rather than to focus on a specified embodiment of the theme.

Figures 19 and 20 show that the addition of the said parallelogram mechanism maintains the +75mm Figure 16 positive steering trail despite swing arm 10 moving either upward by 227mm, as per Figure 17, or downward by 164mm, as per Figure 18. The parallelogram mechanism also ensures the trail remains constant throughout the said upward and said downward movement of the swing arm 10 and beyond these extremities if required. The important aspect is to keep the steering trail within workable parameters. Therefore a strict parallelogram is not absolutely necessary as some very small variance in the trail would be tolerable.

Figure 21 shows an oblique and raised view of the Figure 19 and Figure 20 parallelogram mechanism for wheel 1 but where vehicle1 is in the upright position as in Figure 16. In Figure 21 wheel 1 is paired with Figure 1's wheel 2. Wherein wheel 2 has its own parallelogram mechanism comprised of, first transverse axis 14, swing arm 11, third transverse axis 27, strut 28, fourth transverse axis 29, upper swing arm 30, fifth transverse axis 25, straight line distance 26, which is a mirrored version of wheel 1's parallelogram mechanism. In Figure 21 the wheels 1 and 2 are positioned

away from their respective parallelograms so the two parallelograms can be easier seen, and wheel 2's steering axis 31 has been added.

The above described parallelogram mechanism also maintains a constant rake. The rake is the angle of the steering axis from the vertical when vehicle 1 is upright. The rake greatly determines the trail and determines the angle the associated tyre interfaces with the ground when steered. The rake also has other benefits.

A vertical rake would ensure that the steered wheel remains upright when steered, excluding the influence of vehicle 1's angle of lean. If the rake was inclined from the vertical in a direction parallel to vehicle 1's longitudinal vertical plane, the associated wheel would also lean sideways, excluding vehicle 1's angle of lean, as it is steered. The greater the said incline the greater the wheel would lean sideways in relation to the amount it was steered. If the said rake's incline was horizontal then any steering inputs would only lean the wheel sideways without steering the wheel at all. A said incline to one side of the vertical would cause the associated wheel to lean when steered in an opposite direction to an opposite said incline from the vertical.

If vehicle 1 had a fixed angular relationship between each of its steering axes and associated swing arm it would cause the two steering axes to be raked differently from each other when vehicle 1 was leaning. Whereby, any steering actions would cause one wheel of a pair to steer less and lean more than the other wheel of the same pair. Furthermore the said steer less and lean more scenario would vary as vehicle 1 leaned by varying angles and steered by varying angles. These said variable steer less and lean more differences between paired steered wheels are not likely to coincide with a workable steering geometry. Hence the need to maintain a near constant rake irrespective of vehicle 1's angle of lean to either side.

The trail and rake can be kept within workable parameters by the above described parallelogram type steering system.

Components required for constant steering trail and rake.

In Figure 21 the components that are required to keep the steering trail and rake within workable parameters are, joints 32 and 33, swing arms 10 and 11, joints 34 and 35, struts 22 and 28, joints 36 and 37, upper swing arms 24 and 30, joints 38 and 39.

H. Steering

Figure 21 shows vehicle 1's steering axis 17 being in-line with strut 22 and steering axis 31 being in-line with strut 28. Figure 22 shows Figure 21 but with the addition of stub axle assembly 40 rotationally connected to strut 22 and thereby steering axis 17 too via kingpin 41, and stub axle assembly 42 rotationally connected to strut 28 and thereby steering axis 31 too via kingpin 43. Steering movements are delivered to stub axle assembly 40 by steering rod 44 via joint 45. Steering movements are delivered to stub axle assembly 42 by steering rod 46 via joint 47. The length of the steering rods 44 and 46 have to be compatible with their associated said parallelogram to ensure the upward and downward movement of wheels 1 and 2 does not independently change the steering angle selected by the rider. The free ends of steering rods 44 and 46 would be interconnected with a handlebar steering mechanism but said mechanism is not relevant to the main purpose of this application and is therefore not shown.

Components required for steering

In Figure 22 the components that are required to provide steering are, stub axle assemblies 40 and 42, steering rods 44 and 46, joints 45 and 47, wheels 1 and 2. Figure 23 shows Figure 22 but with wheels 1 and 2 joined to their associated stub axle assemblies 40 and 42.

I. Suspension distribution

Two wheeled motorcycle type vehicles that employ a swing arm for the rear wheel normally have a suspension spring acting between the swing arm structure and the sprung mass, typically chassis. Likewise, two wheeled motorcycle type vehicles that employ a swing arm for the front wheel normally have a suspension spring acting between the swing arm structure and the sprung mass, typically chassis.

If vehicle1 used the above normal suspension spring arrangement for each swing arm of a pair the result would be a resistance to leaning that would progressively increase as vehicle1 leaned, e.g. when cornering.

This is because as the swing arm of each pair closest to the inside of a corner rises it would compress its associated suspension spring which then resists the lean. Such a situation would make it very difficult for the rider to complete a cornering manoeuvre, i.e. a case of extreme discernable roll resistance. Furthermore, the cornering balance of vehicle1 is mainly maintained by gravity countering the centrifugal force. Because gravity is constant its ability to counter varying centrifugal forces depends on varying the gravitational leverage between the combined centre of mass of vehicle1 and its rider and the distance between the contact patches of paired tyres where the resultant force intersects the ground to maintain balance. This is achieved by varying vehicle1's angle of lean just like a two wheeled motorcycle.

If vehicle1 employed the above normal suspension arrangement then additional leverage would be required to counter the force of the said compressed suspension spring in addition to the centrifugal force to maintain cornering balance. Thereby vehicle1 would have to lean by a greater angle by comparison to employing a suspension arrangement that did not compress the suspension on one side when vehicle1 leaned.

To ensure the suspension springs do not resist vehicle1 from leaning, vehicle1's suspension system is not arranged in the above normal manner. Instead vehicle1's suspension system distributes the force from its resilience means to a position midway, or close to midway if suspension biasing is used, between the associated paired wheels throughout vehicle1's range of lean in both directions. This is achieved via a balance beam and associated components.

Figure 24 shows Figure 21 but without showing wheels 1 and 2 or axes 14, 21, 23, 25, 27, 29 to help clarity by de-cluttering but with the following additions, lug 48 rigidly connected to swing arm 10, connecting link 49 jointed between lug 48 via joint 50 and one end of balance beam 51 via joint 52, lug 53 rigidly connected to swing arm 11, connecting link 54 jointed between lug 53 via joint 55 and the other end of balance beam 51 via joint 56, the balance beam 51 centrally pivoted to carrier 57 about axis

58, carrier 57 linearly guided by guide rods 59. Acting on carrier 57 is resilience means 60 represented by the arrow. The other end of resilience means 60 connects to vehicle1's main structure which is the sprung mass.

Thereby the force from resilience means 60, represented in Figure 24 by the arrow, is equally distributed between each wheel 1 and 2 irrespective of vehicle1's angle of lean or road surface within the parameters of suspension biasing if applicable and within the suspension's range of movement.

Components required for the suspension distribution

In Figure 24 the components that are required to equally distribute the force of resilience means 60 between wheels 1 and 2 (not shown) are, lugs 48 and 53, joints 50 and 55, connecting links 49 and 54, joints 52 and 56, balance beam 51, rotational axis 58, carrier 57, guide rods 59, resilience means 60.

J. Damping/anti tilt brake.

Figure 25 shows Figure 24 but with the addition of suspension dampers 61 and 62 positioned at opposite ends of balance beam 51 from each other. The suspension dampers 61 and 62 can also be locked to prevent tilting of the vehicle as per EP03253106.3 and EP07824787.1

Components required for damping/anti tilt brake

The components relevant to the main purpose of this application that are required to provide the suspension damping and anti tilt brake, are the suspension dampers 61 and 62.

Total components required for front transversely pivoted swing arms

Figure 26 combines Figure 23, 24, 25 to show the total amount of components that are required to provide the advantages A, B, C, D, E, F, are as follows:

Joints 32 and 33, swing arms 10 and 11, joints 34 and 35, struts 22 and 28, joints 36 and 37, upper swing arms 24 and 30, joints 38 and 39, stub axle assemblies 40 and 42, kingpins 41 and 43, steering rods 44 and 46, joints 45 and 47, wheels 1 and 2, lugs 48 and 53, joints 50 and 55, connecting links 49 and 54, joints 52 and 56, balance beam 51, axis 58, carrier 57, guide rods 59, resilience means 60, dampers 61 and 62.

First Problem with EP01998472.3, EP10003570.8, and EP03253106.3

Figure 26 typifies the amount of components employed by the various embodiments of vehicle1's front suspension system for paired transversely pivoted swing arms as described in EP01998472.3, EP10003570.8, and EP03253106.3.

A common feature of these embodiments as typified in Figure 26 is that the resilience means 60, normally a suspension spring, has to control the momentum of many unsprung components which, proportional to their combined mass, limits road holding which hereinafter will be referred to as traction. The unsprung mass is the combined mass of the components that are not supported by the resilience means.

Figure 26, which typifies EP01998472.3, EP10003570.8, and EP03253106.3, has the following unsprung components: Wheels 1 and 2, stub axle assemblies 40 and 42,

kingpins 41 and 43, joints 45 and 47, steering rods 44 and 46, struts 22 and 28, joints 34 and 35, joints 36 and 37, swing arms 10 and 11, joints 32 and 33, upper swing arms 24 and 30, joints 38 and 39, lugs 48 and 53, joints 50 and 55, connecting links 49 and 54, joints 52 and 56, balance beam 51, axis mechanism 58, carrier 57, suspension dampers 61 and 62 (unsprung proportion thereof). Plus all the other associated unsprung components such as, brake parts, wiring, cables, mudguards, bearings, seals, washers, bolts, nuts, circlips, studs, split pins, fluids, etc. hereinafter referred to as sundry parts.

All of the above unsprung components interconnect without being interrupted by resilience means 60.

Therefore, as wheel 1 moves up and down in response to bumps and dips in the ground the associated unsprung components 1, 40, 41, 45, 44, 22, 34, 36, 10, 32, 24, 38, 48, 50, 49, 52, 51, 58, 57, sundry parts, move as one with wheel 1.

Similarly, as wheel 2 moves up and down in response to bumps and dips in the ground the associated unsprung components 2, 42, 43, 47, 46, 28, 35, 37, 11, 33, 30, 39, 53, 55, 54, 56, 51, 58, 57, sundry parts, move as one with wheel 2.

As both wheels 1 and 2 simultaneously move up and down in response to bumps and dips in the ground the associated unsprung components 1, 2, 40, 42, 41, 43, 45, 47, 44, 46, 22, 28, 34, 35, 36, 37, 10, 11, 32, 33, 24, 30, 38, 39, 48, 53, 50, 55, 49, 54, 52, 56, 51, 57, 58, sundry parts, move as one with the two wheels 1 and 2.

Components 1, 2, 40, 42, 41, 43, 45, 47, 44, 46, 22, 28, 34, 35, 36, 37, 10, 11, 32, 33, 24, 30, 38, 39, 48, 53, 50, 55, 49, 54, 52, 56, 51, 58, sundry parts, also move as one with the two wheels 1 and 2 in relation to vehicle1's sprung mass when, for instance, vehicle1 leans.

The unsprung portion of resilience means 60 and suspension dampers 61 and 62 would also contribute toward the unsprung mass where applicable.

Each of the above combinations of unsprung components has an unsprung mass. As the unsprung components move, their unsprung mass generates force which hereinafter will be referred to as momentum. The greater the unsprung mass the greater the momentum for a given speed of movement.

When wheel 1 or/and 2 has risen to the highest point of a bump on the ground the associated momentum will not immediately dissipate. Thereby, the associated wheel 1 or/and 2 if uninhibited would form a curve from the influence of, the momentum, gravity, and vehicle1's speed.

However, such curves seldom correlate with the usually more acute irregularities of the ground's surface. In such circumstances, the said curve would cause the associate tyre to partially or completely part company with the ground to proportionally cause the same tyre's traction to be partially or completely lost respectively.

Without traction it would be impossible for the associated tyre to impart any of the rider's desired steering, braking, acceleration, and stability actions to vehicle1. The

extent by which the rider can impart their desired steering, braking, acceleration, and stability actions to vehicle1 is proportionally dependant on the amount of traction vehicle1's tyres can generate with the ground.

To ensure that each tyre remains in traction with the ground, the momentum of the associated unsprung mass has to be reversed after the apex of a bump.

A resisting force is supplied by the resilience means which pushes against both the unsprung mass and the sprung mass. The sprung mass being the combined mass of the components that are supported by the resilience means, including people and any other carried load.

The effectiveness of the resilience means in resisting the momentum of the unsprung mass greatly depends on the ratio between the unsprung mass and the sprung mass. For instance, if the unsprung mass was greater than the sprung mass the resilience means would move the sprung mass more, when the associated wheel/s encountered a bump on the ground, than moving the unsprung mass after the associated wheel/s encountered a bump on the ground. This would not be very useful in resisting the momentum to maintain traction.

To ensure that the resilience means moves the unsprung mass more than the sprung mass the unsprung mass must be less than the sprung mass. The less the unsprung mass is in relation to the sprung mass the more the resilience means is able to resist the momentum and maintain traction

The ratio between the unsprung mass and sprung mass could be increased by adding to the sprung mass. But this would increase manufacturing costs, tyre wear, fuel consumption, and reduce acceleration, braking, handling, and manoeuvrability. It is much better to increase the ratio between the unsprung mass and sprung mass by reducing the unsprung mass. This can be achieved by reducing the mass of the unsprung components or/and changing some of the unsprung mass into sprung mass.

Second Problem with EP01998472.3, EP10003570.8, and EP03253106.3

To help bias the resultant force towards the outside wheels whilst cornering the swing arms could slope downwards from the first transverse axis 14 to wheel centre 18 and likewise at the rear of vehicle1 as shown in the Figure 27 variation of Figure 15. Thereby the front and back wheels that lay towards the inside of the corner whilst cornering generate a longer wheelbase than those lying toward the outside of the corner as shown in the Figure 28 variation of Figure 13 where vehicle1 is cornering left. It can be seen that the inside wheels 1 and 8 are producing longer levers L2 than the L1 levers associated with the outside wheels 2 and 9. Consequently the force from resilience means 60 to the said inside wheels travels through the longer leverage L2 than to the said outside wheels via leverage L1. This means that the resultant force would act closer to the said outside wheels whilst cornering than otherwise would be the case which can be a desirable characteristic as described in F.

The said swing arm's downward slope also shortens the swing arm in any of the EP01998472.3 and EP03253106.3 upwardly looped swing arm variations of vehicle1, which makes the swing arm stiffer or/and saves weight and manufacturing costs.

The said swing arm's downward slope more equally distributes the suspension's compression and extension movements to each side of the horizontal plane when the suspension's compression potential exceeds the suspension's extension potential, as in Figures 19 and 20.

As can be seen there are advantages to swing arms 10 and 11 having a downwards slope away from the first transverse axis 14 and likewise at the rear of vehicle 1 as in Figure 27. But when encountering a bump the said downward slope causes wheels 1 and 2 to deflect into the bump which is not as smooth as deflecting away from the bump and also unduly compresses the tyre which hinders the maintenance of constant traction.

Based on Figure 27, the upper diagram in Figure 29 shows the said downwards slope of swing arm 10 with wheel 1 on ground 3. The lower diagram in Figure 29 shows the said upper diagram but where the bump causes wheel 1 to move upwards as represented by vector X and to move forwards as represented by vector W. The resultant of vectors X and W lies between them as represented by vector V.

However, the bump produces a rearward force on wheel 1 which is opposite to vector W in Figure 29. This rearward force is represented in Figure 30 by vector Y. When vector Y is combined with vector X from Figure 29 the resultant lays in between as represented by vector Z. Also included in Figure 30 is vector V from Figure 29. It can be seen that vector V moves forward from the vertical and vector Z moves backwards from the vertical. Consequently, vehicle 1 suspension deflections akin to vector V will not yield the smoothest possible ride or provide the most consistent traction.

Other prior art

Although specific reference in the Background Information has been made to vehicle 1 as disclosed in EP01998472.3, EP10003570.8, and EP03253106.3 the Background Information also relates to the vehicles (vehicles 2) disclosed in the later applications WO/03/018390, US2004/0160030, EP1884457, WO2010015986, US2010/0147615, which also have paired front transversely pivoted swing arms and a similar amount of associated unsprung components as EP01998472.3, EP10003570.8, and EP03253106.3. Hereinafter vehicle 1, and vehicles 2 will collectively be referred to as the prior art unless referred to individually.

This concludes the Background Information.

The invention

The invention is a new suspension system for the paired front wheels of motorcycle type vehicles employing transversely pivoted swing arms for said paired wheels such as those described in but not limited to the prior art. The invention can provide all of the above advantages A, B, C, D, E, F, of paired transversely pivoted swing arms and can also make provision for G, H, I, J, above.

However, the invention differs from the prior art in that it provides a much greater potential for enhancing traction than the prior art can deliver for an equivalent vehicle mass and thereby redresses the First Problem. The invention can also provide a much

better suspension deflection than the prior art can deliver without compromising the associated vehicle's angle of lean which redresses the Second Problem.

In relation to overcoming the First Problem, the enhanced traction of the invention is achieved by increasing the ratio between the sprung mass and unsprung mass by comparison to the prior art by converting some of the unsprung components associated with the prior art into sprung components. In the invention the resilience means acts on the associated wheels with fewer components interconnected between the resilience means and the associated wheels than in the prior art.

In the prior art each front transversely pivoted swing arm of a pair:

- 1 Supports the resilience means.
- 2 Moves as one with the associated wheel's suspension movement.
- 3 Moves as one with the associated wheel's leaning movement.

In a first variation of the invention each front transversely pivoted swing arm of a pair:

- 1.1 Is supported by the associated resilience means (unlike prior art).
- 2.1 Does not move as one with the associated wheels suspension movement (unlike prior art).
- 3.1 Moves as one with the associated wheel's leaning movement (similar to prior art).

In a second variation of the invention each front transversely pivoted swing arm of a pair:

- 1.2a Is supported by a primary resilience means (unlike prior art).
- 1.2b Supports a secondary resilience means (unlike prior art).
- 2.2 Moves as one with the associated wheel's secondary suspension movement (unlike prior art).
- 3.2 Moves as one with the associated wheel's leaning movement (similar to prior art).

Thereby a resilience means in both the first and second variations of the invention interrupts the prior art's non compressible (or static length) linkages that interconnect each transversely pivoted front swing arm of a pair to its associated wheel.

Consequently, the resilience means in both variations of the invention acts closer to the associated wheels than in the prior art. This reduces the unsprung components, which permits an increase in the ratio between the unsprung mass and sprung mass, which ensures the associated tyres will maintain traction with the ground for more of the time, which improves, stability, braking, and steering, and in the case of front wheel drive or four wheel drive also improves acceleration, beyond the limitations of the prior art for an equivalent vehicle mass.

In relation to overcoming the Second Problem, each of the embodiments of the invention can allow the front wheels of the associated vehicle to deflect in closer proximity to the resulting force generated by the associated front wheel/s encountering a bump on the ground as per vector Z in Figure 30 than in the prior art.

The Background Information of the invention has been described with reference to Figures 1 to 30 where:

- Figure 1 Represents vehicle1's paired wheels with a 460mm track example.
- Figure 2 Represents Figure 1 but leaning to a 30 degree example to demonstrate the increased track.
- Figure 3 Represents the exponentially increasing and decreasing track dimensions as vehicle1 leans based on Figure 1's track dimension.
- Figure 4 Represents a plan view of Figure 1 combined with the exponentially increasing track dimensions of Figure 3 to show the resulting slip angle.
- Figure 5 Represents the increased track of Figure 4 with Figure 3's exponentially decreasing track dimensions to show the resulting slip angle.
- Figure 6 Represents Figure 4 with a faster rate of lean or/and slower vehicle speed.
- Figure 7 Represents Figure 5 with a faster rate of lean or/and slower vehicle speed.
- Figure 8 Represents a visual calculation to determine a combined centre of mass height generated between vehicle1 and its rider.
- Figure 9 Represents Figure 1 combined with the Figure 8 result.
- Figure 10 Represents Figure 9 but with vehicle1 leaning by 45 degrees.
- Figure 11 Represents a visual stability calculation derived from figure 10.
- Figure 12 Represents how far the paired wheels in Figure 2 move up and down.
- Figure 13 Represents vehicle1 viewed from the direction of Figure 12's arrow.
- Figure 14 Represents a visual calculation to prove vehicle1's shortening wheelbase.
- Figure 15 Represents Figure 13 but when vehicle1 is upright.
- Figure 16 Represents a trail for vehicle1.
- Figure 17 Represents the effect to Figure 16's trail as the swing arm rises if the steering axis and swing arm were in a fixed angular relationship.
- Figure 18 Represents the effect to Figure 16's trail as the swing arm lowers if the steering axis and swing arm were in a fixed angular relationship.

- Figure 19 Represents Figure 17 but with the inclusion of a parallelogram and without the steering axis and swing arm being in a fixed angular relationship.
- Figure 20 Represents Figure 19 but in the Figure 18 position.
- Figure 21 Represents an oblique view of Figure 19 and 20 but in the Figure 16 position and for paired wheels which are shown away from their locations.
- Figure 22 Represents Figure 21 with the addition of relevant steering components.
- Figure 23 Represents Figure 22 with the wheels in place.
- Figure 24 Represents Figure 21 with the addition of suspension components and without the wheels.
- Figure 25 Represents Figure 24 with the addition of the suspension dampers.
- Figure 26 Represents the combination of Figures, 23, 24, 25.
- Figure 27 Represents Figure 15 but with downward sloping swing arms.
- Figure 28 Represents Figure 27 combined with Figure 13.
- Figure 29 Represents a part of Figure 27 encountering a bump.
- Figure 30 Is a vector diagram incorporating Figure 29 vector conclusions.

First embodiment of the first variation of the invention (embodiment1/1)

- Figure 31 Figure 31 represents the outward facing side of wheel 2 comprising tyre 201, rim 202, rim mounted brake disc 203, and identifies cross section AA.
- Figure 32 Represents the cross section AA of Figure 31 to reveal the top view of embodiment1/1 comprising stub axle assembly 42, upper frame 63R, upright 28, resilience/damping means 64R comprising compression coil spring and suspension damper - typically a shock absorber, bracket 65R, locating fasteners 66R, kingpin 43, guides 67R, transversely pivoted swing arm 11 (truncated in Figure 32). Note: R = right side
- Figure 33 Represents Figure 32 from the direction of arrow B but without wheel 2 to depict the outward facing side of embodiment1/1 and reveal housing 68R and lower frame 69R.

- Figure 34 Represents Figure 33 from the direction of arrow C to depict the rearward facing side view of embodiment1/1 and reveal joint 35 and angle1 of the resilience/damping means.
- Figure 35 Represents Figure 34 but with the resilience/damping means 64R more compressed.
- Figure 36 Represents Figure 34 from the direction of arrow D to depict the inward facing side view of embodiment1/1.
- Figure 37 Represents an isometric view of Figure 33 combined with the prior art components as typified in Figures 21 and 22. Also included are embodiment1/1's counterpart left side components identified by L. Although the locating fasteners 66L are obscured their left side position can easily be extrapolated by one skilled in the art.
- Figure 38 Represents Figure 37 with the addition of the of, lug 48, lug 53, joint 50, joint 55, connecting link 49, connecting link 54, joint 52, joint 56, balance beam 51, axis 58, from Figure 24. To provide visual access for these additions the, rear portion of the upper swing arm 30, joint 39, and the rear portion of steering rod 46, have been omitted. Although joint 50 is obscured its position can easily be extrapolated by one skilled in the art.
- Figure 39 Represents angle2.
- Figure 40 Represents angle3.

Second embodiment of the first variation of the invention (embodiment2/1)

- Figure 41 Represents a common hydraulic fluid displacement device.
- Figure 42 Shows section BB of Figure 41.
- Figure 43 Represents Figure 42 with the addition of a damping means.
- Figure 44 Shows the remotely positioned resilience means from the displacement devices.

Third embodiment of the first variation of the invention (embodiment3/1)

- Figure 45 Represents the use of the king pin as the piston rod.

The following Figures show further developments of the invention

- Figure 46 Shows a one guide rod version of Figure 33

- Figure 47 Shows view C1 of Figure 46
- Figure 48 Shows Figure 46 but with suspension more compressed
- Figure 49 Shows an embodiment where the guide rod is also the piston rod and shows an assembly to transfer steering torque
- Figure 50 Shows view E of Figure 49
- Figure 51 Shows Figure 49 but with the suspension more compressed
- Figure 52 Shows Figure 49 where the steering torque assembly is also an anti brake dive mechanism
- Figure 53 Shows a leading arm arrangement
- Figure 54 Shows view C2 of Figure 53
- Figure 55 Shows Figure 53 but with the suspension more compressed
- Figure 56 Shows Figure 53 but where the brake torque is isolated from the leading arm
- Figure 57 Shows a trailing arm arrangement
- Figure 58 Shows view E1 of Figure 57
- Figure 59 Shows Figure 57 but with the suspension more compressed
- Figure 60 Shows Figure 57 but where the brake torque is isolated from the trailing arm
- Figure 61 Shows a trailing arm arrangement where the brake torque is used to generate an amount of anti brake dive.

The following Figures show how the first and second variations of the invention can be applied to a vehicle having paired front wheels combined with paired rear wheels but the vehicle only using one balance beam.

- Figure 62 Shows how one balance beam can be used for paired front and back wheels for the first variation of the invention
- Figure 63 Shows Figure 62 but relating to the second variation of the invention
- Figure 64 Shows a development of Figure 62 where the connecting links do not cross
- Figure 65 Shows Figure 64 but relating to the second variation of the invention
- Figure 66 Shows how an offset balance beam can be applied to Figures 64 and 65

A first embodiment of the first variation of the invention (embodiment1/1) will now be described with reference to Figures 31 to 40 and in relation to the prior art as typified by Figures 26:

Whereby on the associated vehicle's right side: wheel 2 is rigidly connected to stub axle assembly 42, stub axle assembly 42 is rotationally connected to housing 68R to provide wheel 2 with its rotational movement, housing 68R can linearly slide along guides 67R, guides 67R are statically fixed to upper frame 63R and lower frame 69R, the fixed positional relationship of the two guides 67R with each other is determined by upper frame 63R and by lower frame 69R, bracket 65R is connected to housing 68R, the lower end of resilience/damping means 64R is rotationally connected to bracket 65R via locating fastener 66R, and the upper end of resilience/damping means 64R is rotationally connected to upper frame 63R via the locating fastener 66R, upper frame 63R is connected to upright 28, rotational joint 37 (which in Figure 37 can be one of the joint types described in GB2435021) interconnects the top of upright 28 with the front of upper swing arm 30, the rear of upper swing arm 30 is connected to joint 39, the upper end of kingpin 43 is rotationally connected to upper frame 63R and the lower end of kingpin 43 is rotationally connected to lower frame 69R, the position of kingpin 43 in relation to the guides 67R is determined by upper frame 63R and by lower frame 69R, kingpin 43 is connected to joint 35 which isolates steering axis 31 from being in a fixed angular relationship with swing arm 11 as previously discussed in G, swing arm 11 interconnects between joint 35 and joint 33, steering rod 46 is connected to upright 28 via joint 47, lug 53 is in a fixed angular relationship with swing arm 11, connecting link 54 connects to lug 53 via joint 55, the free end of connecting link 54 connects to the right side of balance beam 51 via joint 56, balance beam axis 58 is in a fixed relationship to the associated vehicle's main structure (unlike Figures 24, 25, and 26), each component has a counterpart left side component as seen in Figures 26, 37, 38, identified by L in Figure 37. When the associated vehicle is vertical and perpendicular to the ground the above right side components are the mirror image of their counterpart left side components as if a mirror was positioned vertically along the vehicle's longitudinal central plane.

Suspension actions occur when deflections of tyre 201 in response to the ground surface's dips and bumps cause wheel 2, stub axle assembly 42, housing 68R, and bracket 65R, to move upwards and downwards as one unit in relation to upper frame 63R and lower frame 69R via guides 67R. The said upward and downward movements, which is sometimes referred to as bump and rebound, are controlled by resilience/damping means 64R.

Therefore embodiment1/1 only has four completely unsprung components associated with wheel 2 whose numbers are, 2, 42, 68R, 65R.

By comparison the prior art, as typified by Figure 26, has nineteen completely unsprung components associated with wheel 2, whose numbers are, 2, 42, 43, 47, 46, 28, 35, 37, 11, 33, 30, 39, 53, 55, 54, 56, 51, 58, 57.

Because components 2 and 42 are common to both the prior art as typified by Figure 26 and embodiment1/1 they can be removed from the comparison of unsprung components. Thus the comparison of unsprung components is between the two components 68R and 65R of embodiment/1 and the seventeen components 43, 47, 46,

28, 35, 37, 11, 33, 30, 39, 53, 55, 54, 56, 51, 58, 57, of the prior art as typified by Figure 26. Although the actual amount of unsprung components in some instances of prior art may be slightly less than those in Figure 26 the amount of unsprung components in the prior art are still considerably more than in any of the embodiments of the invention.

In terms of enhancing traction the ratio between the unsprung mass and sprung mass is the important aspect and not the amount of unsprung components. For instance, a vehicle with a 1:10 ratio between its unsprung mass and sprung mass and having a total of four unsprung components would have worse traction than a vehicle with a 1:15 ratio between its unsprung mass and its sprung mass but having a total of say nineteen unsprung components (assuming the tyres, tyre pressure, suspension, and any other relevant parameters were optimised for both 1:10 and 1:15 ratio scenarios).

However by comparison to the prior art as typified by Figure 26 the reduced unsprung components of embodiment1/1 would increase the unsprung/sprung mass ratio and thereby improve traction as follows. Calculations show that if housing 68R and bracket 65R in embodiment1/1 were made from solid steel they could weigh about 1.28Kg and only 0.65kg if they had sections removed for lightening with the remaining mass suitably webbed and radiused for strength. It is not possible that the seventeen unsprung components, or there about, of the prior art could weigh less. If specialised lightweight materials were used to reduce the unsprung mass of the prior art then the same specialised lightweight materials could also reduce the unsprung mass of embodiment1/1 to maintain the unsprung mass to sprung mass ratio prior to the use of specialised lightweight materials.

Similarly embodiment1/1 also only has four completely unsprung components associated with wheel 1 whose numbers are, 1, 40, 68L, 65L, and the prior art, as typified by Figure 26, has nineteen completely unsprung components associated with wheel 1, whose numbers are, 1, 40, 41, 45, 44, 22, 34, 36, 10, 32, 24, 38, 48, 50, 49, 52, 51, 58, 57. Consequently embodiment1/1 also improves the unsprung mass to sprung mass ratio on the left side of the associated vehicle by the same extent as on the right side.

The above comparisons only include the mass of completely unsprung components. The partly unsprung resilience/damping means were omitted to help the reader quickly appreciate the unsprung mass and sprung mass ratio difference between embodiment1/1 and prior art. The addition of the partly unsprung resilience/damping means would add very close to the same amount of unsprung mass to both embodiment1/1 and prior art. Therefore the omission of the unsprung portion of the resilience/damping means does not undermine the advantage of embodiment1/1 over the prior art in terms of increasing the ratio between the unsprung mass and sprung mass to improve traction. The above comparisons also omit the sundry parts for the same reason of clarity as for the omission of the partly unsprung components. However, it will be appreciated that there will be fewer sundry parts associated with embodiment1/1 than associated with the prior art.

Embodiment1/1 does not employ the unsprung rotational axis 58, carrier 57, guides 59, resilience means 60, dampers 61 and 62, of Figure 26. Instead, balance beam 51 in embodiment1/1 rotates about axis 58 that is fixed to a sprung part of the associated

vehicle as shown in Figure 38. In embodiment1/1 resilience means 60 and dampers 61 and 62 associated with vehicle1 as per Figure 26 have been replaced by resilience/damping means 64R/L as shown in Figures 37 and 38.

Non limiting examples of the scope of embodiment1/1

As can be seen in Figure 32, embodiment1/1 is positioned within the internal space of rim 202. Consequently embodiment1/1's size is limited by the internal space of rim 202.

Where the ability of resilience/damping means 64R to deliver the desired suspension characteristics depends on its spring having a length, with or without pre-load, that cannot fit within the internal space of rim 202 if resilience/damping means 64R was mounted parallel to guides 67R, then resilience/damping means 64R can instead be angled to the guides 67R (angle/1) as shown in Figure 34. Angle/1 also compensates for the greater increase in resistance of a shorter spring by comparison to a longer spring for the same rate of compression.

For instance, a spring with a free length of 100mm and a 1kg/mm spring rate would generate a 40kg resistance if compressed to a length of 60mm [$100\text{mm} - 60\text{mm} = 40\text{mm} \times 1\text{kg} = 40\text{kg}$]. To replicate the 40kg resistance at a compressed length of 60mm with a spring that had a free length of 80mm would require a spring rate of 2kg/mm [$80\text{mm} - 60\text{mm} = 20\text{mm} \times 2\text{kg} = 40\text{kg}$].

However, a further and equal compression of the two springs would yield different resistances. For instance, a further 30mm compression of the 100mm spring and of the 80mm spring would yield 70kg [$40\text{mm} + 30\text{mm} = 70\text{mm} \times 1\text{kg} = 70\text{kg}$] and 100kg [$20\text{mm} + 30\text{mm} = 50\text{mm} \times 2\text{kg} = 100\text{kg}$] resistances respectively.

But when the above 30mm compression is applied to an angled resilience/damping means 64R such as shown in Figure 34 the result can be considerably different. For instance if resilience means 64R in Figure 34, had a distance of 150mm between the centres of the two location fasteners 66R, was angled by 45 degrees to guides 67R, had the above 80mm free length spring compressed to 60mm, was generating 40kg of resistance, then a 30mm upward movement of housing 68R would increase the spring's resistance to 79kg instead of the 100kg in the above example, [$1\text{kg}(40 + (2(150 - (\text{sqr}((\text{sqr}((150\text{sq})/2) - 30)\text{sq} + \text{sqr}((150\text{sq})/2)\text{sq})))) = 79\text{kg}$] where sqr means square root, and sq means squared. Thereby angle/1 broadens the scope for the associated spring's design in two ways. Firstly, by providing a longer distance for the spring within the limitations of rim 202 and secondly, by progressively reducing the effective spring rate as the suspension compresses. Consequently, the greater the angle of angle/1, within workable parameters, the broader is the scope for the spring's design.

Resilience/damping means 64R could alternatively be angled (angle/2) in relation to guides 67R as in Figure 39. In Figure 39 the lower end of resilience/damping means 64R is located to housing 68R via a spacer or similar (unseen) that is long enough for resilience/damping means 64R to clear kingpin 43. The upper end of resilience/damping means 64R is located to the right side of upper frame 63R.

To further increase the length and therefore the design scope of resilience/damping means 64R within the confines of rim 202 resilience/damping means 64R can be positioned to combine angle/1 and angle/2 to form a compound angle/3. Figure 40 shows angle/3 where resilience/damping means 64R locates to a bracket 65R (or similar) extending from housing 68R as in Figure 34 but to the other side of kingpin 43 with the upper end of resilience/damping means 64R located to upper frame 63R in a similar position to that shown in Figure 34. In Figure 40 the location fasteners of Figure 34 have been replaced with ball joints 70R.

There will also be other design considerations associated with embodiment1/1 such as but not limited to: The amount of steering lock, steering momentum, lowering the centre of mass, further reduction of the unsprung mass, material costs, tooling costs, side force on guides 67R, camber, offset of wheel 2's rotational axis from steering axis 31, offset of steering axis 31 from the centre of tyre 201's contact patch with ground 3, rake, trail, position and type of the braking mechanism, drive shaft if there is a drive to wheel 2, weather protection of parts, electric or hydraulic drive of wheel 2, and combinations thereof. To accommodate these and other variables the position of resilience/damping means 64R can depart from those already discussed to further broaden the design scope without departing from the theme of embodiment1/1 as per the following non limiting examples and combinations thereof.

In Figure 36 resilience/damping means 64R is shown far to the right of kingpin 43 to provide a large steering lock. Where a particular application of embodiment1/1, for instance when applied to a sports variation of vehicle1, does not need as great a steering lock as Figure 36 would provide then resilience/damping means 64R can be positioned closer to kingpin 43. This would better centralise resilience/damping means 64R with the suspension deflections caused by wheel 2 which reduces the side loading on guide rods 67R and also reduces the steering momentum. Resilience/damping means 64R can also be positioned to the left side of kingpin 43 when viewed from the perspective of Figure 36.

In Figure 39 resilience/damping means 64R is positioned towards the inside of vehicle/1 from kingpin 43. Alternatively, resilience/damping means 64R could be positioned towards the outside of vehicle/1 from kingpin 43 to provide a greater steering lock before the minimal desired clearance between resilience/damping means 64R and lower swing arm 11 is reached.

To maximise angle/2, and thereby maximise the design scope for resilience/damping means 64R, Figure 39 shows the lower end of resilience/damping means 64R connected to housing 68R to the left of kingpin 43 and the upper end of resilience/damping means 64R connected to the far right of upper frame 63R. However, where angle/2 does not need to be so steep for resilience/damping means 64R to deliver the desired suspension characteristics the lower end of resilience means 64R can instead be mounted to housing 68R to the right of kingpin 43 or/and the upper end of resilience/damping means 64R can be mounted closer to kingpin 43 rather than as far away as possible.

In relation to Figure 39, if the connection between kingpin 43 and lower swing arm 11 was sufficiently far from upper frame 63R then, the upper end of resilience/damping

means 64R could be mounted to upper frame 63R but instead to the left of kingpin 43 with the lower end of resilience/damping means 64R mounted to housing 68R via bracket 65R or similar to the right of kingpin 43 with the resilience/damping means 64R crossing over the said connection between kingpin 43 and lower swing arm 11 instead of underneath.

With reference to Figure 34, angle/1 could be reversed in relation to guides 67R so that the lower end of resilience/damping means 64R connects instead directly to housing 68R via a location fastener 66R with the upper end of resilience/damping means 64R still connecting to the upper frame 63R via a location fastener 66R but instead closer to upright 28 and not as far to the right as possible as shown in Figure 34. This would eliminate the need for bracket 65R which would further reduce the unsprung mass and associated material/tooling costs as well as reducing the steering momentum and increasing the room for other nearby components. This might reduce the steering lock in relation to Figure 34 but for a sport's variation of vehicle/1, for instance, it may not be relevant and may not reduce the steering lock at all if the connection between kingpin 43 and lower swing arm 11 was lower in relation to the upper frame 63R as previously discussed.

Alternatively if the lower end of resilience/damping means 64R was connected to housing 68R as per said reversed angle/1 but with the upper end of resilience/damping means 64R connecting to upper frame 63R so that a parallel relationship between resilience/damping means 64R and guides 67R existed this would also eliminate the need for bracket 65R with the aforementioned associated benefits. The said parallel relationship could also exist with the inclusion of bracket 65R if the upper end of resilience/damping means 64R was moved closer to upright 28 when viewed from the perspective of Figure 34.

In any of the above examples of angle/1 angle/2 angle/3 or combinations thereof resilience/damping means 64R can be the other way up than shown in the Figures, and either location fasteners 66R or/and ball joints 70R can be used.

A stub axle for wheel 2 can be integral with or connected to housing 68R with at least one associated wheel bearing housed in stub axle assembly 42. A stub axle for wheel 2 can be integral or connected to the stub axle assembly 42 with at least one associated wheel bearing housed in housing 68R. A stub axle for wheel 2 could interconnect between at least one associated wheel bearing housed in housing 68R and at least one associated wheel bearing housed in stub axle assembly 42. The stub axle, or the rotational axis of wheel 2 if different from the stub axle's longitudinal centre line, does not have to be perpendicular to the length of the guide rods 67R or kingpin 43.

Rim 202 can either bolt to the flange of stub axle assembly 42 or screw onto stub axle assembly 42, the latter with or without a knock-off type hub or similar. The flange of stub axle assembly 42 can be any feasible diameter with the corresponding widening of the central hole of rim 202. An extreme example of the central hole of rim 202 being widened would be akin to that used on Lambretta scooters and Volkswagen cars and vans during the 1950/60s. Rim 202 could also be an integral unit with stub axle assembly 42 with or without the associated said integral or connected stub axle.

In Figure 31 brake disc 203 is mounted via its outer circumference to rim 202. This arrangement provides space around wheel 2's rotational axis for embodiment 1/1 and kingpin 43 etc as well as maximising the leverage of the associated brake calliper to help arrest the associated vehicle. Brake disc 203 can either be solidly connected to rim 202 or be connected to rim 202 but allowed to float sideways to the rotation of wheel 2. The associated brake calliper is not shown but would be either solidly or floatingly connected, such as provided by the invention described in GB2436672, to an unsprung component such as housing 68R. Brake torque can be either resisted by upright 28 or by the calliper being rotationally and concentrically connected around wheel 2's rotational axis via a plate or similar that has an arm directly or indirectly connected, the latter via a joint, to housing 68R or indirectly connected via a joint to a sprung component such as upper frame 63R. The brake disc can instead be of a conventional type, and thereby located by its inner circumference or inner flange, either connected to or integral with stub axle assembly 42 or associated sub axle. Alternatively a drum brake can be used. Where wheel 2 has an electric drive then braking can be applied electrically by reclaiming the kinetic energy or/and reverse drive. Where wheel 2 has a hydraulic drive then braking can be applied by restricting the flow of hydraulic fluid. Outside of legal requirements, a hydraulic drive would not need a separate brake as a complete shut off of the hydraulic flow would prevent the wheel from turning. Outside of legal requirements and depending on the strength of the permanent magnets, if used, an electric drive may or may not need a separate brake.

In the Figures, housing 68R slides along guides 67R which are circular in cross section. Alternatively guides 67R could have any other feasible cross sectional shape that is uniform along their length. In the Figures, there are two guides 67R to prevent housing 68R from twisting around on the circular cross section of a single guide 67R. However, a single suitably designed non circular in cross section guide 67R could be used instead. The guides 67R can also be, more than two, of different cross sectional shapes and areas from each other, positioned at different distances from the central plane of wheel 2 to each other.

In the Figures lower frame 69R provides positional consistency between the lower ends of guides 67R and kingpin 43. The rotational joint between kingpin 43 and upper frame 63R could be designed to negate the need for the rotational joint between lower frame 69R and the lower end of kingpin 43. This would then allow kingpin 43 to be much shorter than implied in the Figures which can save weight and reduces manufacturing costs. In this instance lower frame 69R would provide positional consistency just between guides 67R. However, if each guide 67R had sufficient rigidity and was adequately secured to upper frame 63R the need for lower frame 69R could be negated which saves weight and reduces manufacturing costs and reduces steering momentum. In such instance the lower ends of guides 67R may benefit from a stop to prevent housing 68L from sliding off. The said stops could be integral (or only removable by the manufacturer or similar) with their guide 67R or detachable with a screw, circlip, split pin, cotter pin, etc fixing.

Kingpin 43 does not have to be parallel to guides 67R because an angular relationship between kingpin 43 and guides 67R is also envisaged. Therefore in relation to Figures 34, 35, 40, the top end of kingpin 43 can be further away from guides 67R than the lower end of kingpin 43 or visa-versa, or/and in relation to Figures 33, 36, 39,

the top end of kingpin 43 can be further away from guides 67R than the lower end of kingpin 43 or visa-versa.

Rotational axis 35 between lower swing arm 11 and kingpin 43 does not have to be perpendicular. Instead the angular relationship can be greater than ninety degrees or less than ninety degrees. With reference to Figure 39, the steering axis defined by kingpin 43 does not have to be parallel to guides 67R or/and perpendicular to the ground.

More than one resilience means 64R may be used for wheel 2 whereby the associated suspension spring's wire cross sectional area can be smaller than the wire cross sectional area of a single resilience means 64R. For instance, where two springs are used instead of one spring the wire cross sectional area of the two springs can be approximately half that of the one spring for the same overall spring rate when other associated parameters are the same. The smaller the wire's cross sectional area the less is the room taken up by the associated spring's coils. This then allows each spring's movement to be greater than for a single spring for a given free spring length which can allow the angles of angle/1 angle/2 angle/3 to be reduced or maybe negated or/and to provide more suspension movement. A similar advantage can be achieved by two or more springs being inside of each other within one resilience/damping means 64R.

Multiple resilience/damping means 64R can each be smaller in diameter than a single resilience/damping means 64R which may allow them to be positioned in places too small for a single larger resilience/damping means 64R. Unlike a single resilience/damping means 64R multiple resilience/damping means 64R do not have to be centrally aligned with either wheel 2's rotational axis or/and guides 67R to minimise the twisting force of housing 68R acting on guides 67R caused by suspension deflections. Multiple resilience/damping means 64R can employ different spring wire cross sectional areas from each other. Where multiple resilience/damping means 64R are used one or more may just contain the suspension spring/s and one or more may just contain the damping mechanism.

If resilience/damping means 64R was instead just a suspension damper then a compression or bump suspension coil spring could encompass one or each guide 67R between housing 68R and upper frame 63R with or without an extension or rebound suspension coil spring encompassing one or each guide 67R between housing 68R and lower frame 69R, or said stop.

The above variables and combinations thereof provide a lot of scope for a designer to pursue many specific design goals without departing from the theme of embodiment1/1.

All the above variations of embodiment1/1 are equally applicable to the counterpart components associated with wheel 1 on the left side of the associated vehicle. In embodiment1/1 the resilience/damping means associated with wheel 1 and wheel 2 would turn with the steering and contribute to the steering momentum as well as contributing to the unsprung mass.

In embodiment1/1, when for instance the left wheel encounters a bump the associated suspension spring acts in series, via the rotating balance beam 51, with the right wheel's suspension spring. This leads to the unsprung mass associated with the left wheel being resisted by half the spring rate, i.e. unsprung mass = 1 and spring rate = 0.5. But when both wheels simultaneously encounter the same bump the associated suspension springs act in parallel with each other via the non rotating balance beam 51. This leads to the unsprung mass associated with both wheels being resisted by double the spring rate, i.e. unsprung mass = 2 and spring rate = 2. A comparison of the two scenarios shows that when the unsprung mass doubles it causes the spring rate to quadruple. This inconsistency, previously covered by EP01998472.3, hampers designing the best spring rate for both scenarios. If the designer optimises one scenario it will inevitably compromise the other scenario, and visa-versa. Nevertheless, a workable spring rate compromise can be achieved for both scenarios as shown by virtually every car suspension design which also suffer from the same inconsistent unsprung mass to spring rate ratio between the two said scenarios.

In embodiment1/1 the role of balance beam 51 is to control the up and down relationship between lower swing arms 10 and 11 as the associated vehicle leans, and also to balance the force from the suspension deflections from one side of the associated vehicle to the other side.

An anti tilt brake can act between balance beam 51 and a sprung part of the associated vehicle in the form of a hydraulic cylinder where an automatically or/and manually activated valve intermittently interrupts the hydraulic flow in the said hydraulic cylinder to further slow down destabilising events like lowsides and highsides in addition to that described in A or to stop the said hydraulic flow for parking. Alternatively the anti tilt brake can comprise a disc and brake calliper or brake drum and brake shoes acting between balance beam 51 and a sprung part of the associated vehicle.

A second embodiment of the first variation of the invention (embodiment2/1) will now be described with reference to Figures 41, 42, 43, 44 and in relation to the prior art as typified by Figures 26:

In embodiment2/1 the damping/resilience means 64R/L are replaced in any of their previously described positions and by any of their previously described multiples with hydraulic displacement devices (displacement device). For instance, and by way of a non limiting example, Figure 41 shows a well known hydraulic displacement device 71L. Figure 42 shows Figure 41's cross section BB of cylinder 72L to reveal piston 73L connected to piston rod 74L. Within cylinder 72L and above piston 73L is cavity 75L which contains hydraulic fluid. The volume of cavity 75L varies according to the up and down movement of piston 73L in cylinder 72L in response to suspension deflections moving piston rod 74L up and down. Opening 76L in cylinder 72L allows hydraulic fluid to be displaced to and from cavity 75L via hydraulic line 77L in response to the associated suspension deflections. Hydraulic line 77L, shown in Figure 41 in truncated form, interconnects the displacement devices 71L/R with their associated resilience means. Figure 43 shows Figure 42 but with the addition of damping device 78L fixed in cavity 75L to control the bump and rebound, initiated by the associated suspension deflections activating the resilience means, via through-holes in damping device 78 L that may or may not have valves. Typically a piston

seal or/and close manufacturing tolerances ensures that the hydraulic fluid does not bypass piston 73L, and typically a linear bearing will interface between piston rod 74L and cylinder 72L. As with resilience/damping means 64R/L each displacement device 71L/R is held in position by location fasteners 66R/L or/and ball joints 70R/L.

In embodiment2/1 resilience means 79R/L are remotely positioned from their associated displacement devices 71L/R as shown in Figure 44. To minimise the unsprung mass resilience means 79R/L could be connected to any sprung part of the associated vehicle. To minimise the steering momentum resilience means 79R/L can also be remotely positioned from any steering components. In Figure 44 resilience means 79R/L are by the well known method of compressed gas contained in a spherical chamber wherein the compressed gas is separated from the hydraulic fluid by a membrane, bladder, sealed piston, etc. The combined force from the compressed gas in resilience means 79R/L via hydraulic lines 77R/L is equally divided between their associated displacement devices 71R/L by balance beam 51, so that suspension deflections on say the right side of the associated vehicle are deterred from causing the associated vehicle to tilt to the left by the equal pressure in displacement device 71L on the left side of the same vehicle, and visa versa.

Embodiment2/1 employs the same balance beam 51 arrangement as embodiment1/1 and can incorporate the same anti tilt brake as embodiment1/1.

In embodiment2/1, when for instance wheel 1 encounters a bump the associated unsprung mass 1M compresses resilience means 79L by a distance 1D, i.e. unsprung mass = 1 and compression = 1. When both wheels 1 and 2 simultaneously encounter the same bump the associated unsprung mass doubles 2M and the distance resilience means 79L is compressed = 1D and the distance resilience means 79R is compressed = 1D making the total compression 2D, i.e. unsprung mass = 2 and compression = 2. It can be seen that as the unsprung mass increases the ratio between the unsprung mass and the compression of the resilience means does not change. This allows the designer to better optimise the suspension for one wheel and two wheel scenarios more than embodiment1/1.

Non limiting examples of the scope of embodiment2/1

In Figure 43 the damping means 78L is positioned within cylinder 72L. In reference to Figure 44 the damping means could alternatively be positioned along hydraulic lines 77R/L or within resilience means 79R/L or combinations thereof. In Figure 44 single hydraulic lines 77R/L are shown but alternatively two or more hydraulic lines can be used. Where one or more hydraulic line is used for bump and one or more hydraulic line is used for rebound the bore diameters of the hydraulic lines for the bump and rebound can be different from each other to provide different flow rates which may negate the need for a separate suspension damper. In Figure 44 spherical resilience means 79R/L are shown but any known shape of a similar functioning resilience means can be used and not exclusive to just one resilience means per displacement device. Alternatively one compressed gas resilience means per set of paired wheels can be used where the compressed gas fills a cavity in a cylinder between two, pistons, bladders, or diaphragms, etc. Whereby, the hydraulic fluid from the displacement device/s associated with one wheel of a pair acts on one of the said pistons etc and the hydraulic fluid from the displacement device/s associated with the other wheel of the same pair acts on the other said piston etc. Where the resilience

means is via compressed gas a separate accumulator may not be required to compensate for a change in the volume of the hydraulic fluid due to a change in the said fluid's temperature. In Figure 44 a closed loop hydraulic system is shown but an open loop system can be used with a recirculation pump, oil tank, and associated valves. A suspension brake could be added via a valve interrupting the hydraulic fluid flow along either or both the hydraulic lines 77R/L. Where a compressed gas resilience means is not used then resilience means 79R/L can be replaced with hydraulic cylinders akin to 72L/R to form a master/slave cylinder arrangement to activate a mechanical resilience means such as a coil or leaf spring etc.

Embodiment2/1 can make a similar improvement in traction as embodiment1/1 by comparison to the prior art because embodiment2/1 has a similar number of unsprung components as embodiment1/1.

A third embodiment of the first variation of the invention (embodiment3/1) will now be described with reference to Figure 45. In Figure 45 stub axle assembly 40 is connected to bridging portion 80L in the same way as to housing 68L. Lower frame 169L is rigidly connected to, or is integral with, the lower end of bridging portion 80L and upper frame 163L is rigidly connected to, or is integral with, the upper end of bridging portion 80L. Upright 122 is rigidly connected to, or is integral with, upper frame 163L. Upright 122 connects to upper swing arm 24 in the same way as upright 22. In Figure 45 the king pin for steering is provided by piston rod 174L which rotates between upper frame 163L and lower frame 169L via suitable bearings. Fitted to piston rod 174L is piston 173L which can slide up and down the inside of cylinder 172L. Cylinder 172L is connected to swing arm 10 via rotational joint 34. Above piston 173L is cavity 175L which contains hydraulic fluid. The hydraulic fluid can enter or exit cavity 175L via opening 176L to and from resilience means 79L in the same way as the hydraulic fluid in displacement device 71L via opening 76L in Figure 44. Embodiment3/1 is a more compact design than the previous embodiments but by comparison the following components, lower frame 169L, upper frame 163L, upright 122, upper swing arm 24, steering rod 44, joint 45, become unsprung and thereby contribute to the unsprung mass. Nevertheless embodiment3/1 has far less unsprung components than the prior art which improves traction. Furthermore, the unsprung components in embodiment3/1 do not include the comparatively heavy swing arms 10 and 11. Consequently the overall unsprung mass of embodiment3/1 would not have to be much more than in embodiment1/1 and embodiment2/1.

Non limiting examples of the scope of embodiment3/1

Embodiment3/1 includes any practical variations associated with embodiment1/1 and embodiment2/1. For instance by way of a non limiting example, coil springs could be encompass piston rod 174L between displacement device 171L and upper frame 163L and lower frame 169L. Each component identified by L for the left side of the associated vehicle has a counterpart component for the right side.

In the previous embodiments the suspension movement occurs without axis 58 of balance beam 51 needing to move in relation to the sprung mass. In other words axis 58 is fixed to the sprung mass. This means that swing arms 10 and 11 are sprung components, i.e. supported by the resilience means. As already mentioned the space within the associated wheel rim is limited. Previously described angle1 angle2 and

angle3 are ways where the use of the limited space within the associated wheel rim can be maximised.

Another way of using the said limited space is to limit the movement of the suspension within the said limited space with the remaining suspension movement being provided elsewhere.

A first embodiment of the second variation of the invention (embodiment1/2) delivers some of the required suspension movement by any of the means described in the previous embodiments, hereinafter referred to as the primary suspension movement, combined with the rest of the required suspension movement being delivered by the method shown in Figure 26 where a resilience means 60 acts on carrier 57 which linearly slides on guides 59, hereinafter referred to the secondary suspension movement. Unlike the previous embodiments axis 58 in embodiment1/2 is not fixed to the sprung mass and swing arms 10 and 11 are sprung in relation to the primary suspension movement but unsprung in relation to the secondary suspension movement.

Thereby, 40% for a non limiting example, of the required suspension movement can be delivered by the primary suspension movement and 60% of the required suspension movement can be delivered by the secondary suspension movement. The advantage of embodiment1/2 is that less room is required within the confines of the associated wheel rims than required by the previous embodiments. Based on the above 40% example, only 40% of the total room needed for the suspension movement is required by the primary suspension movement of embodiment1/2 by comparison to the previous embodiments.

When the secondary suspension movement is activated the unsprung mass would be similar to that in the prior art. Therefore at first glance there does not appear to be an advantage to embodiment2/1. However, such a situation would arise when the associated wheel has encountered a large bump in the ground. Very often larger bumps have a lower frequency, which hereinafter means a shallower incline from the start of the bump to the top of the bump and a shallower decline from the top of the bump to the bottom of the bump, by comparison to smaller bumps.

The lower the frequency the more time the resilience means has to reverse and accelerate downwards the unsprung mass after the top of the bump has been passed to maintain traction.

By comparison the higher frequency more commonly associated with smaller bumps requires the resilience means to reverse and accelerate downwards the unsprung mass quicker to maintain traction than with lower frequency bumps. The less the unsprung mass is the quicker it is to accelerate. Therefore the reduced unsprung mass associated with the primary suspension of embodiment1/2 by comparison to the prior art helps to maintain traction better than the prior art.

Non limiting examples of the scope of embodiment1/2

Instead of axis 58 being mounted to carrier 57 which has a linear movement via guides 59, axis 58 can rotate about an axis that is parallel to first axis 14 or/and second axis 15 in a similar way as disclosed in EP0606191.

Side loadings on guide rods cause friction which has to be overcome before any suspension movement can occur. Thereby such side loading has a similar undesirable effect as increasing the unsprung weight.

In a second embodiment of the second variation of the invention (embodiment 2/2) the primary suspension movement is provided by a lever arm that causes the associated wheel to move through a curved path according to the length of the said lever arm when the primary suspension operates. This differs from the previous embodiments where the suspension movement causes the associated wheel to move in a linear path. The radially moving end of the lever arm would hold the stub axle assembly which may or may not include part of the steering mechanism such as a king pin. An advantage of a lever arm is the almost complete absence of friction from side loadings which allows the suspension to work just as well under braking as when not braking.

Figures 46 to 61 show further developments of the invention. Figure 47 is view C1 of Figure 46. In reference to Figure 46 and Figure 47, a single guide 167R having a circular cross sectional area is possible when combined with torque arm assembly 81R to prevent housing 168R from rotating about steering axis 31 independently from the steering means controlled by the rider and without limiting the linear suspension movement of housing 168R on guide 167R. Wherein, torque arm assembly 81R comprises at least two links.

In Figure 46 torque arm assembly 81R comprises upper link 82R which is rotationally connected at 83R to a sprung part of the vehicle which as a non limiting example in Figure 46 is bracket 84R extended from and fixed to upper frame 163R. The other end of upper link 82R is rotationally connected at 85R to an end of lower link 86R. The other end of lower link 86R is rotationally connected at 87R to a bracket 88R that is extended from and fixed to housing 168R. Wherein, at least two of the three rotational connections 83R, 85R, 87R, would each have a single axis of rotation. Torque arm assembly 81R transfers steering torque to housing 168R.

To accommodate steering movements, guide 167R is rotationally connected to lower frame 169R and fixed to upper frame 163R which is linked to the steering means controlled by the rider. Alternatively guide 167R could be fixed to lower frame 169R and rotationally connected to upper frame 163R. Alternatively guide 167R could be rotationally connected to the lower and upper frames 169R and 163R respectively.

Stub axle assembly 42 is rotationally connected to housing 168R. Resilience/damping means 64R is at one end rotationally connected to bracket 89R extended from and fixed to housing 168R. The other end of resilience/damping means 64R is rotationally connected to bracket 90R extended from and fixed to upper frame 163R.

In Figure 47 lower frame 169R is fixed to drop link 91R which is rotationally connected at 35 to swing arm 11. Thereby steering axis 31 is not confined to a fixed angular relationship with swing arm 11 as previously discussed under G. Thereby third rotational axis 27 of rotational connection 35 forms a corner of the parallelogram or quadrilateral that was also previously discussed under G.

Figure 48 shows Figure 46 but with the suspension compressed more which reduces the angle formed between rotational connections 83R, 85R, 87R, of torque arm assembly 81R by comparison to Figure 46.

Alternatively, steering movements can be imparted by a suitably designed resilience/damping means without the need for a torque arm assembly like 81R. For instance and with reference to Figure 46 as a non limiting example, steering torque imparted by the vehicle's operator to upper frame 163R could be transferred to housing 168R by just the interconnecting resilience/damping means 64R to negate the need for torque arm assembly 81R. The said suitably designed resilient/damping means can be of a through-piston rod design as per residual/damping means 371R in Figure 56 to cope with the steering torque. Alternatively the steering torque can be shared between resilience/damping means 64R and torque arm assembly 81R.

Brake torque can be transferred from the associated brake (not shown) to housing 168R by an interconnecting bracket (not shown).

In Figures 49 to 56 stub axle assembly 42 is not shown to reveal other components that would otherwise be hidden.

In Figure 49 cylinder 272R of the resilience/damping means has a piston rod 274R which also provides linear guidance for the suspension movement. Piston rod 274R is fixed to boss 92R. Stub axle 93R is fixed to boss 92R. Bracket 188R is fixed to boss 92R.

Cylinder 272R is fixed to upper frame 263R which is linked to the steering means controlled by the rider. To allow steering movements, cylinder 272R rotates in knuckle joint 94R about steering axis 31. Thereby cylinder 272R also acts as a kingpin.

Figure 50 is view E of Figure 49. In Figure 50 knuckle joint 94R rotates about axis 27 of rotational connection 35. Thereby steering axis 31 is not confined to a fixed angular relationship with swing arm 11 as previously discussed under G. Thereby the third rotational axis 27 of rotational connection 35 forms a corner of the parallelogram or quadrilateral that was also previously discussed under G.

Where piston rod 274R has a circular cross sectional area torque arm assembly 81R from Figures 46, 47, 48, can be used to prevent piston rod 274R from rotating within cylinder 272R to alter the steering independently from the steering means controlled by the rider and without limiting the linear suspension movement of piston rod 274R in cylinder 272R.

Torque arm assembly 81R is not required if piston rod 274R is non circular in cross section and is instead confined solely to linear movement by a matching female part that is fixed to cylinder 272R.

Brake torque can be transferred from the associated brake (not shown) to boss 92R by an interconnecting bracket (not shown).

The rotational axes of rotational connections 83R, 85R, 87R, are perpendicular to the suspension movement. When the said axes are also parallel to the rotational axis of their associated wheel the associated torque arm assembly can be used to limit brake dive.

When braking, a resultant force proportional to the rate of deceleration combined with gravity acts closer to the front of the vehicle than just gravity alone when not decelerating. Consequently the front suspension is compressed more when braking, commonly known as brake dive, than when not braking. Brake dive can be reduced or eliminated as the manufacturer desires by an anti dive mechanism. There are several known anti dive mechanisms which are often built into the resilience/damping means. Any of the known anti dive mechanisms can be incorporated into the various embodiments of the invention.

In relation to Figures 46 to 51 brake dive would reduce the angle between rotational joints 83R, 85R, 87R, of torque arm assembly 81R.

Anti dive can be achieved by harnessing the braking torque. For instance, and with reference to Figures 46 to 51, by attaching the brake (not shown) to lower link 86R of torque arm assembly 81R where rotational connection 87R of lower link 86R has the same rotational axis as the associated brake drum or disc.

Such an arrangement is shown in the Figure 52 variation of Figure 49 where torque arm assembly 181R comprises an upper link 182R which has one end rotationally connected at 183R to a bracket fixed to cylinder 272R and the other end rotationally connected at 185R to one end of lower link 186R. Wherein, the other end of lower link 186R is rotationally connected at 187R about the same longitudinal axis as stub axle 93R.

The Figure 52 arrangement would transfer the braking torque, indicated by the arrow, to a sprung part of the vehicle via torque arm assembly 181R. Consequently the braking torque would try to increase the angle between rotational connections 183R, 185R, 187R, which in turn would extend the front suspension by extending piston rod 274R from cylinder 272R.

Thereby although brake dive tries to reduce the angle between rotational joints 183R, 185R, 187R, of torque arm assembly 181R the associated brake torque, shown by the arrow, tries to increase the angle between rotational joints 183R, 185R, 187R, of torque arm assembly 181R. Thus anti dive has been achieved by harnessing the braking torque

The desired amount or range of anti dive can be achieved by calculating the length and positions of the lower link 186R and upper link 182R in relation to each other and the resulting force generated by the deceleration caused by braking.

Torque arm assembly 81R in Figures 46 to 51 can be combined with at least one torsion spring where one leg of the torsion spring would bare on the upper link 82R and the other leg of the torsion spring would bare on the lower link 86R. The said torsion spring could provide all of the required suspension resilience means, whereby piston rod 274R becomes a guide, or just some of the required suspension resilience.

Alternatively, using Figure 49 as a non limiting example, a torsion spring could act between the upper link 82R and cylinder 272R or/and between the lower link 86R and bracket 188R, or at all of the rotational joints 83R, 85R, 87R.

In a further development at least one of the rotational connections 83R, 85R, 87R, of torque arm assembly 81R could include a friction suspension damper. Wherein, the associated friction in suspension compression can differ from the associated friction in suspension extension. The said difference could be provided by each interfacing friction surface comprised from directional friction differences. Wherein the said directional friction difference is provided by the associated friction surface comprised from, for instance, abrasions that are not concentric to themselves but are uniformly aligned with each other. When such interfacing friction surfaces have the said uniform alignment but in opposite directions to each other it would generate more friction in one direction than the other direction. The said interfacing friction surfaces could be mounted to discs or quadrants. Alternatively two opposed directional clutches/cams/ratchets can be used where each one of the pair can provide a different resistance to movement from the other. Alternatively at least one of the rotational connections 83R, 85R, 87R, could include a known rotary suspension damper such as those that function using a piston/s or a vane. The above further developments can all be applied to torque arm assembly 181R in Figure 52.

Torque arm assembly 181R from Figure 52 can also be used with any of the embodiments of the invention shown in Figures 32 to 48 to transfer steering torque, and to provide anti dive if the brake was attached to lower arm 186.

In the Figures any of the shafts that interface with a linear bearing or linear seal can be protected from the ingress of contaminants by a gaiter or bellows that are suitably flexible and suitably resilient to said containments.

Figure 53 shows king pin 95R rotationally connected to knuckle 94R to provide steering movement. At one end king pin 95R is fixed to upper frame 263R which is connected to the steering means operated by the rider. The other end of king pin 95R is rotationally connected at 97R to one end of leading arm 96R. The other end of leading arm 96R is rotationally connected at 98R to one end of resilience/damping means 64R. The other end of resilience/damping means 64R is rotationally connected at 99R to king pin 95R via a bracket fixed to king pin 95R. Stub axle 193R is connected to leading arm 96R.

Figure 54 shows Figure 53 from the direction of C2. Figure 55 shows Figure 53 but with the suspension more compressed to show the movement of leading arm 96R about rotational connection 97R.

Referring to Figure 53, anti dive can be achieved by mounting the brake (not shown) directly to leading arm 96R. Thereby the clockwise braking torque applied to leading arm 96R about rotational connection 97R would try to lengthen resilience/damping means 64R but the associated brake dive would try to shorten resilient/damping means 64R by applying an anti clockwise torque to leading arm 96R about rotational connection 97R. The clockwise brake torque counters the anti clockwise brake dive torque. The amount or range of anti dive depends on the position of rotational

connection 97R from stub axle 193R in relation to the resulting force generated by braking deceleration which can be calculated by the manufacturer.

In the non limiting example of Figure 53 king pin 95R is dog legged but alternatively king pin 95R can be straight. Stub axle 193R can be either lower than rotational axis 97R as depicted in Figure 53 or share the same horizontal plane as rotational axis 97R, or be above rotational axis 97R.

In Figure 56 the brake torque has been isolated from leading arm 196R by the brake (not shown) being mounted to brake plate 100R which is connected to piston rod 374R of resilience/damping means 371R. In the same way as resilience/damping means 64R is rotationally connected at 98R to leading arm 96R in Figure 53 the resilience/damping means 371R is rotationally connected to leading arm 196R where said rotational connection is aligned with the longitudinal axis of stub axle 293R. If the braking torque applied to piston rod 374R is detrimental to the associated piston then piston rod 374R can be a through-rod as shown in Figure 56 and be supported at both ends of cylinder 372R by linear bearings against side thrust generated by the brake torque.

Figures 57 to 60 show the trailing arm alternative to the previously described leading arm arrangement in Figures 53 to 56. The components in Figures 57 to 60 are the same as those in Figures 53 to 56 and share the same component numbers. However, the components in Figures 57 to 60 that are specific to the trailing arm are prefixed with T to identify trailing arm. Figure 58 shows Figure 57 from the direction of E1. Figure 59 shows Figure 57 but with the suspension more compressed to show the movement of trailing arm T96R about rotational connection T97R.

With reference to Figure 57, if the brake (not shown) was connected to trailing arm T96R then the braking torque, shown by the arrow, would turn trailing arm T96R clockwise about rotational connection T97R. The brake dive would also turn trailing arm T96R clockwise about rotational joint T97R. Thereby the brake torque increases the brake dive. However, the trailing arm scenario deflects the associated wheel away from bumps, as per vector Z in Figure 30, when stub axle T193R is lower than the trailing arm's rotational axis T97R, which overcomes the previously described Second Problem.

In Figure 60 the brake (not shown) is mounted to brake plate 100R as in Figure 56 to isolate the brake torque from trailing arm T196R. In Figure 60, resilient/damping means 371R is mounted in the same way to trailing arm T196R as resilience/damping means 371R is to leading arm 196R in Figure 56.

In Figure 61 the brake (not shown) is connected to lower link 286R which has one end rotationally connected at 287R, to either stub axle T393R or trailing arm T296R, about the longitudinal axis of stub axle T393R. The other end of lower link 286R is rotationally connected at 285R to one end of upper link 282R. The other end of upper link 282R is rotationally connected at 283R to king pin T195R. Stub axle T393R can be fixed to or rotationally connected to trailing arm T296R. In the latter case lower link 286R can be fixed to stub axle T393R. Trailing arm T296R is rotationally connected at T197R to king pin T195R. One end of resilient/damping means T164R is rotationally connected at T198R to trailing arm T296R on the opposite side of rotational connection T197R from stub axle T393R. The other end of

resilience/damping means T164R is rotationally connected at T199R to kingpin T195R via a bracket fixed to king pin T195R. Thereby as trailing arm T296R rotates clockwise about rotational connection T197R caused for instance by a bump in the road stub axle T393R moves upwards whilst rotational connection T198R moves downwards. Consequently, resilience/damping means T164R would operate in tension which is the reverse of, for instance, resilience/damping means 64R in Figure 57 which would operate in compression. However, if T198R was connected to the same side of T197R as 287R then T164R would act in compression.

When the braking torque, as indicated by the arrow, is applied to lower link 286R it tries to increase the angle between the lower link 286R and upper link 282R which applies an anti clockwise torque on trailing arm T296R. The anticlockwise torque counters the clockwise torque on trailing arm T296R caused by brake dive. Thereby anti dive is achieved for trailing arm T296R. As previously discussed the amount of anti dive can be pre-determined to work within desired parameters by calculating the lengths and positions of trailing arm T296R, upper link 282R, lower link 286R, relative to each other in relation to the resultant force generated by braking deceleration.

Where the brake is mounted to lower link 286 then lower link 286 can also provide the associated brake with side float by rotational joint 283 and 287 both being spherical bearings, Thus negating the need to engineer side float into a single piston calliper which would permit the installation of a smaller calliper than otherwise would be the case. Thereby the small size advantage of a single piston calliper is further enhanced by the absence of a side float mechanism built into the single piston calliper. The same would apply to torque arm assembly 181 but only when torque arm assembly 181 is not used to transfer steering torque. Side float can also be provided by the brake plate 100 by being rotationally connected to piston rod 374 or by piston rod 374 being rotationally connected to lever arm 196 via a spherical bearing and where piston rod 374 can also rotate in cylinder 372.

Other prior art

The vehicle described in EP1918187 has a front suspension and steering system akin to vehicle 1. In one embodiment of EP1918187 (Figure 7a therein) the front suspension is integrally positioned with the two components 38 that are equivalent to vehicle 1's struts or upright which herein are numbered 22 and 28. This means that the three previously described characteristics (1.1) (2.1) (3.1) of the first variation of the invention are equally true for EP1918187. In this regard the said one embodiment of EP1918187 achieves a similar unsprung component reduction as the invention by comparison to vehicle 1.

However, EP1918187 does not give any reason for the said one embodiment's suspension position. Furthermore, the subject commonly known as the unsprung mass is not referred to at all in any form despite the applicant, Piaggio, being an established vehicle manufacturer and thereby surely knowing about the subject of unsprung mass. Evidently the minimising of unsprung mass, i.e. the previously described First Problem, was not a goal of EP1918187.

Although the said one embodiment in EP1918187 and the invention both reduce the unsprung components by comparison to vehicle1, it is only the invention that is aimed at overcoming the First Problem of minimising the unsprung mass.

Unlike the said one embodiment in EP1918187, the invention minimises the unsprung mass by also minimising the size of the unsprung components. This, results in an intersection between the unsprung and sprung mass being as close as possible to the associated wheel's rotational axis in the first and second variations of the invention.

In Figure 33 the said intersection is between components 68R and 67R. In Figure 45 the said intersection is between components 174L and 172L. In Figure 47 the said intersection is between components 167R and 168R. In Figures 49 and 52 the said intersection is between components 274R and 272R. In Figures 53/56/57/60/61 the said intersection is between components 96R/196R/T96R/T196R/T296R and components 95R/T95R/T195R respectively.

Consequently all of the embodiments of the invention have an intersection between the unsprung and sprung mass laying within the circular plane dimensionally defined by the diameter of the associated tyre. For instance where a ten, twelve, thirteen, fourteen, fifteen, sixteen, eighteen, twenty inch etc diameter tyre is used then the circular plane would respectively be ten, twelve, thirteen, fourteen, fifteen, sixteen, eighteen, or twenty inches in diameter and so on. These sizes relate to the inside diameter of the tyre's beading. The same would also apply to tyres with metric sizes. Wherein, the circular plane's parallel centre line is positionally defined as being one and the same as or in-line with the associated tyre's rotational axis.

By comparison, EP1918187 with reference to Figure 7a therein has the intersection 76 between the unsprung components 38 and the sprung components (the other numbered components) laying outside of the said circular plane.

The significance of the above comparison is that the greater the distance between said intersection and associated stub axle the greater the unsprung mass must be when other factors are the same. Consequently, the intersection of the invention laying within the said circular plane must result in less unsprung mass, and thereby better road holding, than in EP1918187 or any other similar embodiment where the said intersection lays outside of the circular plane.

Furthermore, to position the invention within the limited space of the associated wheel rim and yet still provide sufficient suspension travel was a practical challenge identified in this description which the various embodiments of the invention overcome.

By comparison, EP1918187 does not identify the provision of sufficient suspension travel within any limited space as a practical challenge to overcome.

None of the previously disclosed secondary suspension aspects (1.2a) (1.2b) (2.2) (3.2) of the invention's second variation are true for EP1918187. Figure 7a of EP1918187 and every embodiment of the invention can overcome the Second Problem of suspension deflection,

To summarise, EP1918187:

- * Does not overcome the previously discussed First Problem of minimising unsprung mass.
- * Does overcome the previously discussed Second Problem of suspension deflection.

By comparison, the invention

- * Overcomes the First Problem of minimising unsprung mass.
- * Overcomes the Second Problem of suspension deflection.

Any of the discussed embodiments of the invention relating to Figures 47 to 61 inclusive can be used with either the first or second variations of the invention.

The anti dive mechanism provided by torque arm assemblies 181 and 281 can be applied to the front suspension of a car which would make the ride more comfortable by reducing or by the total absence of brake dive in all braking situations. Without an anti dive mechanism a driver has to gradually ease off braking as the car slows until the very last motion is arrested by just the vehicle's rolling resistance in order to arrest the vehicle without brake dive. Not only is this practice difficult to master, usually only by best chauffeurs, it is not possible on a downwards slope or when having to brake rapidly or with automatic transmission.

So far the invention has been described in relation to front paired wheels but the invention in any of the embodiments herein described can be used at the rear of the associated vehicle.

Normally the invention would be interconnected with a single balance beam or hydragas equivalent for the front paired wheels, and where paired back wheels are also used the associated vehicle would employ another balance beam or another hydragas equivalent for the rear paired wheels, thus requiring two balance beams or two hydragas gas equivalents or one of each. Thereby the suspension deflections of one wheel of a pair are countered by the other wheel of the same pair via their common balance beam as per vehicle 1 or via their hydragas equivalent. However, the balance beam that interconnects with the invention can also be simultaneously used for paired rear wheels when the interconnections from the single balance beam to one pair of wheels crosses over. Thus requiring only one balance beam despite the associated vehicle having paired front wheels' and paired back wheels'. In this scenario one pair of wheels, for instance at the front, are connected to the balance beam as herein described but with the right rear wheel connected to the left end of the balance beam and with the left rear wheel connected to the right end of the balance beam. So that the right front wheel and left rear wheel are connected to the right end of the balance beam and the left front wheel and rear right wheel are connected to the left end of the balance beam. In this way the suspension deflections of any one wheel of the four are countered by the responding suspension deflections of the other three wheels to ensure suspension deflections on just one side of the associated vehicle do not tilt the associated vehicle sideways. The single balance beam can rotate about a static axis. But to provide the previously described secondary suspension movement

the single balance beam axis can rotate about a linear moving balance beam axis as per vehicle 1 or rotate about a radially moving balance beam axis as per EP0606191. Whereby, the resilience means of the secondary suspension movement pushes the balance beam axis, or carrier thereof, towards a stop or where separate resilience means of the secondary suspension movement push the balance beam axis, or carrier thereof, in opposite directions to each other so that the movement of the balance beam axis is determined by the opposing resilience means of the secondary suspension. Secondary suspension movement can also be achieved with a single static balance beam axis where the connection from each primary suspension movement includes means for the secondary suspension movement between the associated swing arms and balance beam's ends and where the rear suspension either employs the same primary and secondary suspension movement arrangement or where all of the rear suspension movement is provided by an arrangement akin to the associated secondary suspension movement. The single balance beam cross over arrangement can take many forms and can be used without the invention being included.

Figure 62 shows the front right swing arm 11 and left rear swing arm 12 linked to the right end 56 of balance beam 51 and the front left swing arm 10 and rear right swing arm 13 linked to the left end 52 of balance beam 51. Thereby the suspension deflections of any one wheel of the four wheels are countered by the responding suspension deflections of the other three wheels to ensure that suspension deflections on just one side of the associated vehicle do not tilt the vehicle sideways. The single balance beam rotates about axis 58 which is statically fixed in relation the vehicle's main frame and where connecting links 49, 54, 490, 540, are all under compression. To prevent connecting links 490 and 540 from colliding with each other one or both connecting links may employ a portion that is offset from one end or from both ends to bridge the other connecting link. Alternatively, one connecting link 490 or 540 can provide clearance for the other by employing opposing bridging portions as shown by, for instance, connecting link 540 in Figure 62. Wherein connecting link 490 and 540 incorporate damping/resilience means 601 and 602 respectively.

Figure 62 shows the first variation of the invention where at the vehicle's front end the intersection between the unsprung mass and sprung mass is located within the circular plane dimensionally defined by the associated tyre's diameter and where said circular plane is positionally defined by its parallel centre line being aligned with the rotational axis of the associated wheel, and axis 58 of balance beam 51 is static in relation the associated vehicle's main frame.

Figure 63 shows Figure 62 but with the addition of a secondary suspension as per the second variation of the invention. Alternatively, the secondary suspension can instead be accommodated by axis 58 of balance beam 51 moving linearly on a carrier as per Figure 24 or radially on a carrier as previously discussed where the secondary suspension can act on one side of the said carriers pushing them towards a stop or on opposite sides of the said carriers to resist their linear or radial movement in both directions.

Figure 64 shows a variation of Figure 62 where lug 480 is in a fixed angular relationship with rear swing arm 12 but projected downwards from transverse axis 15. Similarly, lug 530 is in a fixed angular relationship with rear swing arm 13 but is projected downwards from transverse axis 15. This means that the left connecting

link 490 can be rotationally connected between lug 480 and the left end of balance beam 51 via rotational connections 500 and 52 respectively and the right connecting link 540 can be rotationally connected between lug 530 and the right end of balance beam 51 via rotational connections 550 and 56 respectively. Thereby connecting links 540 and 490 do not cross over each other. With this arrangement connecting links 540 and 490 are under tension. Where connecting links 540 and 490 incorporate resilience means 602 and 601 respectively 602 and 601 will work under tension rather than in compression. In Figure 64 transverse axes 14 and 15 are on the same horizontal plane as in Figure 62. Therefore, axis 58 of balance beam 51 can be perpendicular to connecting links 49, 54, 490, 540, if balance beam 51 was angled from the vertical with connecting link 49 being parallel to connecting link 490 and with connecting link 54 being parallel to connecting link 540.

Figure 65 shows Figure 64 but with the addition of the secondary suspension from Figure 63. Alternatively, the secondary suspension can instead be accommodated by axis 58 of balance beam 51 moving linearly on a carrier as per Figure 24 or radially on a carrier as previously discussed where the secondary suspension acts on one side or on opposite sides of the said carriers to resist their linear or radial movement.

Suspension biasing can be achieved for the embodiments shown in Figures 64 and 65 by axis 58 of balance beam 51 being offset from the ends 52 and 56 of balance beam 51 as previously discussed under F and as shown by balance beam 151 in Figure 66. Although this could be applied to the embodiments shown in Figures 62 and 63 it would result in the suspension at one end of the vehicle being biased in an opposite direction to the other end of the vehicle. However suspension biasing for the embodiments shown in Figures 62 and 63 can be achieved in the same direction at both ends of the vehicle by making appropriate non perpendicular alignments between, each swing arm and associated lug, each lug and associated connecting link, each connecting link and associated end of the balance, or combinations thereof.

In any of the single balance beam scenarios the arrangement of the rear connecting links 540 and 490 can swap positions with the arrangement of the front connecting links 54 and 49.

In any of the embodiments the rear suspension can employ the suspension mechanism for either the first or second variations of the invention.

The single balance beam arrangement shown in Figures 62, 63, 64, 65, 66, can be used separately from any of the embodiments of the invention shown in Figures 32 to 61.

The invention can be further described by the following numbered clauses 1 to 15

1. A suspension system for the paired front wheels of motorcycle type vehicles where to cause the paired wheel's track to increase whilst cornering and to ensure the trail and rake of each wheel of the pair always remain within workable parameters each wheel of the pair has its own movable quadrilateral or parallelogram mechanism comprised from (i) a transversely pivoted swing arm (ii) a transversely pivoted upper swing arm (iii) a steerable strut that is jointed to the free ends of the swing arm and upper swing arm (iv) a main

frame or prime move to which the said transverse pivots are connected; where to ensure suspension deflections do not unduly tilt the associated vehicle sideways there are also means to disperse the suspension force between paired wheels, characterised by the suspension being able to move independently of the swing arms.

2. A suspension system according to clause 1 where each swing arm is supported by a resilience means.
3. A suspension system according to clause 1 where each swing arm is supported by a primary resilience means and supports a secondary resilience means.
4. A suspension system according to any of the preceding clauses where the damping means is integral with its resilience means.
5. A suspension system according to any of the clauses 1 to 3 where the resilience means is remotely positioned from one or more associated hydraulic fluid displacement device or/and damping means.
6. A suspension system according to any of the preceding clauses where a piston rod in a hydraulic displacement device or/and damping means also acts as a kingpin for steering.
7. A suspension system according to any of the preceding clauses that causes the associated wheel to move in a linear path.
8. A suspension system according to any of the clauses 1 to 6 that causes the associated wheel to move in a radial path.
9. A suspension system according to any of the preceding clauses where the resilience means is by one or more, coil spring, leaf spring, gas spring, electromagnetic spring, rubber or similar type material spring, pneumatic spring, or combination thereof.
10. A suspension system according to any of the preceding clauses that incorporates one or more accumulators to accommodate changes in the hydraulic fluid's volume.
11. A suspension system according to any of the preceding clauses that incorporates an anti tilt brake to either slow down lowsides or/and highside, or for parking that can be activated either manually or/and automatically.
12. A suspension system according to any of the preceding clauses where the means to disperse the suspension force between paired wheels is by a balance beam or similar mechanism.
13. A suspension system according to any of the preceding clauses where the associated vehicle has paired rear wheels.

14. A suspension system according to clause 13 that uses the invention for the paired rear wheels.
15. A suspension system according to clause 13 and 14 where one balance beam is used.

The invention can be further described by the following numbered clauses 16 to 32

- 16 A suspension system for a motorcycle type vehicle having two front wheels, two stub axles one for each front wheel, two swing arms one for each stub axle, each swing arm confined for radial movement about a transverse axis of the vehicle, means to constrain each swing arm relative to each other for equal and opposite movements as the vehicle lean, each stub axle supporting a suspension means, the suspension means supporting each swing arm, the suspension movement of the suspension means being independent of the swing arms, characterised by the suspension movement being within the internal space of the associated front wheel's rim.
- 17 A suspension system according to clause 16 above where the suspension movement causes the associated wheel to move through a linear path.
- 18 A suspension system according to clause 16 where the suspension movement is provided by two lever arms one for each wheel that causes each wheel to move through a radial path according to the length of the associated lever arm.
- 19 A suspension system according to clause 17 where the suspension means operates parallel to the suspension movement.
- 20 A suspension system according to clause 17 where the suspension means operates in a non parallel alignment with the suspension movement
- 21 A suspension system according to any of the preceding clauses 16 to 20 where each suspension means is provided by a hydraulic cylinder that displaces hydraulic fluid to and from a remotely positioned resilient means via one or more hydraulic lines.
- 22 A suspension system according to clauses 21 where the remote resilience means is a hydragas unit.
- 23 A suspension system according to any of the preceding clauses 16 to 22 where a piston rod in the suspension means also acts as a kingpin for the steering.
- 24 A suspension system according to any of the preceding clauses 16 to 23 where on encountering a bump the suspension movement is angled rearwards from vertical.
- 25 A suspension system according to any of the preceding clauses 16 to 24 where the brake torque is transferred to a sprung component.

- 26 A suspension system according to any of the preceding clauses 16 to 25 where the suspension means is a primary suspension means and each swing arm supports a secondary suspension means.
- 27 A suspension system according to any of the preceding clauses 16 to 26 where a balance beam provides the means to constrain the swing arms for equal and opposite movement.
- 28 A suspension system according to any of the clauses 16 to 26 where an accumulator provides the means to constrain the swing arms for equal and opposite movement.
- 29 A motorcycle type vehicle according to any of the preceding clauses 16 to 28 having at least one rear wheel.
- 30 A motorcycle type vehicle according to clause 29 using any of the disclosures in clauses 16 to 28 for the vehicle's one or more rear wheels.
- 31 A motorcycle type vehicle according to any of the clauses 27 to 30 having, two rear wheels, a rear swing arm for each rear wheel, each rear swing arm confined for radial movement about a transverse axis of the vehicle, where each rear swing arm is constrained for equal and opposite movements relative to each other as the vehicle leans by being interconnected by the same balance beam or hydragas equivalent that interconnects each front swing arm.

Claims

- 1 A suspension system for a motorcycle type vehicle having two front wheels, each front wheel supporting a suspension means, the suspension means supporting a swing arm one for each front wheel, each swing arm confined for radial movements about a transverse axis of the vehicle, means to constrain the two swing arms for equal and opposite movement relative to each other as the vehicle leans, the suspension means providing suspension movement for each front wheel independently from the associated swing arm, characterised by the suspension means having an intersection between the unsprung mass and sprung mass within the circular plane of the associated front tyre.
- 2 A suspension system according to claim 1 where the suspension movement lays within the internal space of the associated wheel rim.
- 3 A suspension system according to claim 1 and claim 2 where suspension movement causes the associated wheel to move through a linear path and the suspension means operates parallel to the suspension movement.
- 4 A suspension system according to claim 1 and claim 2 where suspension movement causes the associated wheel to move through a linear path and the suspension means operates non parallel to the suspension movement.
- 5 A suspension system according to any of the preceding claims where a piston rod of the suspension means also acts as a kingpin for the steering.
- 6 A suspension system according to claim 1 and claim 2 where the suspension movement is provided by two lever arms one for each front wheel that cause each front wheel to move through a radial path according to the length of the associated lever arm.
- 7 A suspension system according to any of the preceding claims where a cylinder of the suspension means also acts as a king pin for the steering.
- 8 A suspension system according to any of the preceding claims where each suspension means incorporates a cylinder that displaces fluid to and from a remotely positioned resilient means via one or more hydraulic lines.
- 9 A suspension system according to any of the preceding claims where on encountering a bump the resulting upward suspension movement is also simultaneously rearwards.
- 10 A suspension system according to any of the preceding claims where a torque arm assembly transfers steering torque between sprung and unsprung components.
- 11 A suspension system according to any of the preceding claims where brake torque is transferred from unsprung components to sprung components via a torque arm assembly or via the suspension means.

- 12 A suspension system according to any of the preceding claims where brake torque is used to generate any amount of anti brake dive.
- 13 A suspension system according to any of the preceding claims where the said suspension means is a primary suspension means and each swing arm supports a secondary suspension means.
- 14 A motorcycle type vehicle according to any of the preceding claims having at least one rear wheel.
- 15 A motorcycle type vehicle according to claim 14 using any of the disclosures in claims 1 to 13 for the vehicle's one or more rear wheels.
- 16 A motorcycle type vehicle according to any of the preceding claims having, two rear wheels, a rear swing arm for each rear wheel, each rear swing arm confined for radial movements about a transverse axis of the vehicle, where each rear swing arm is constrained for equal and opposite movements relative to each other as the vehicle leans by the same means that constrains the two front swing arms for equal and opposite movements relative to each other as the vehicle leans.

Figure 1

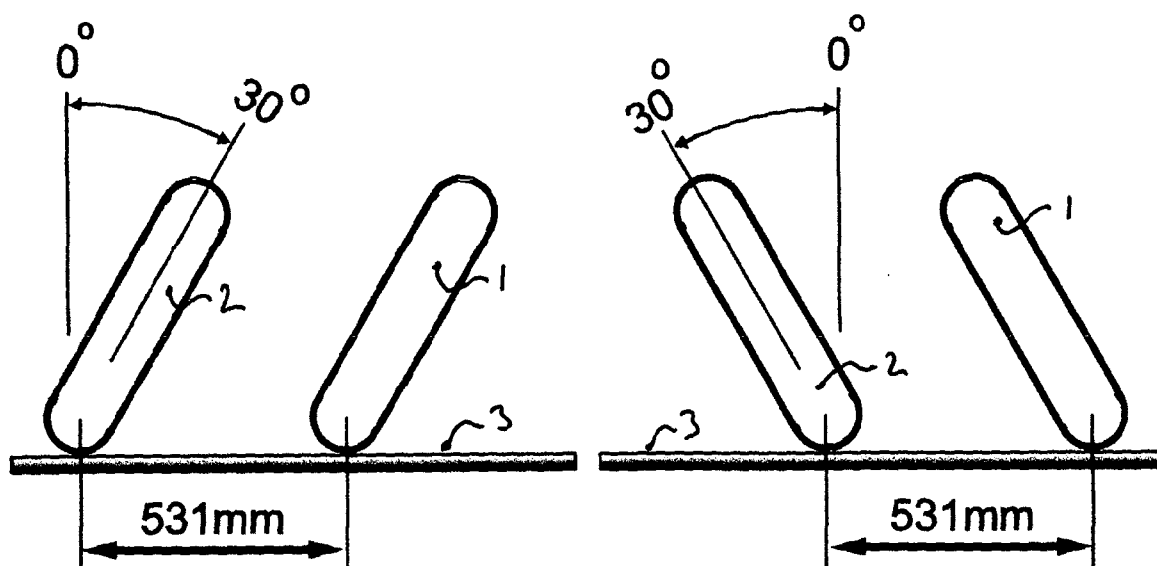
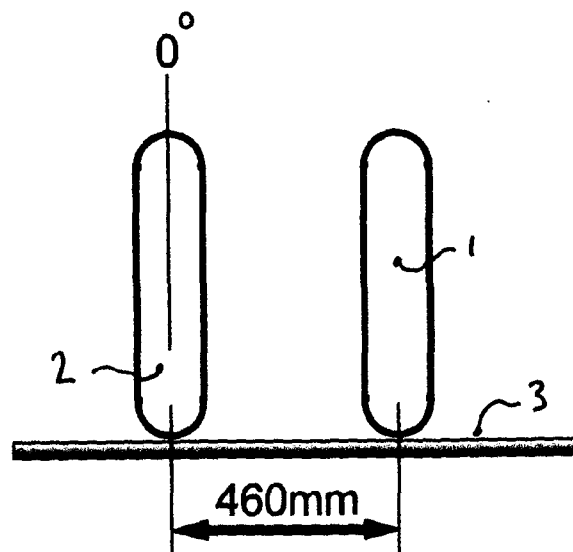


Figure 2

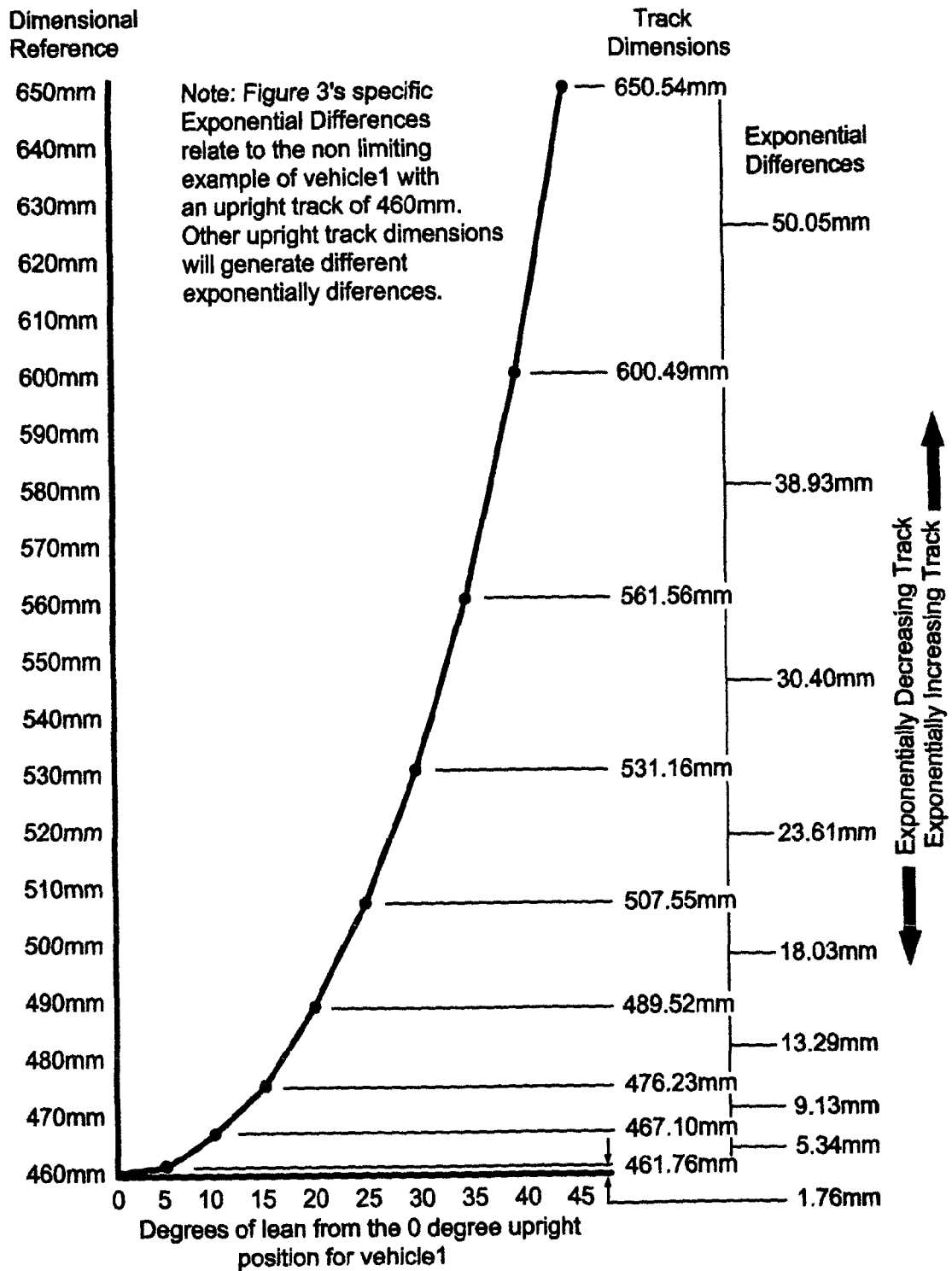


Figure 3

3/36

Figure 4

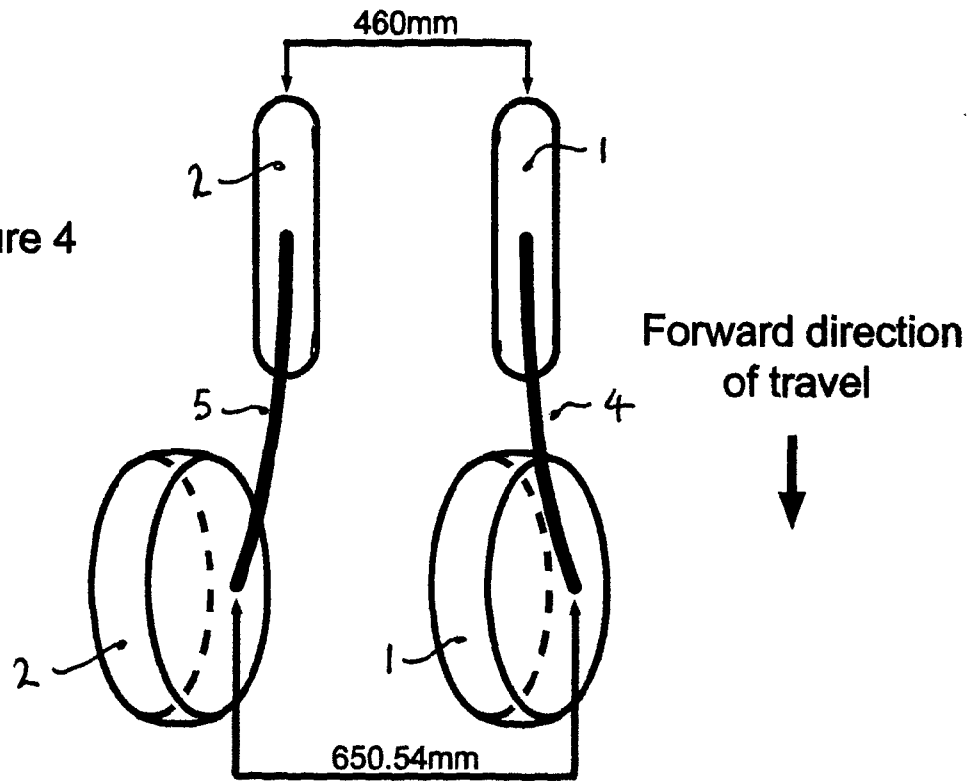


Figure 5

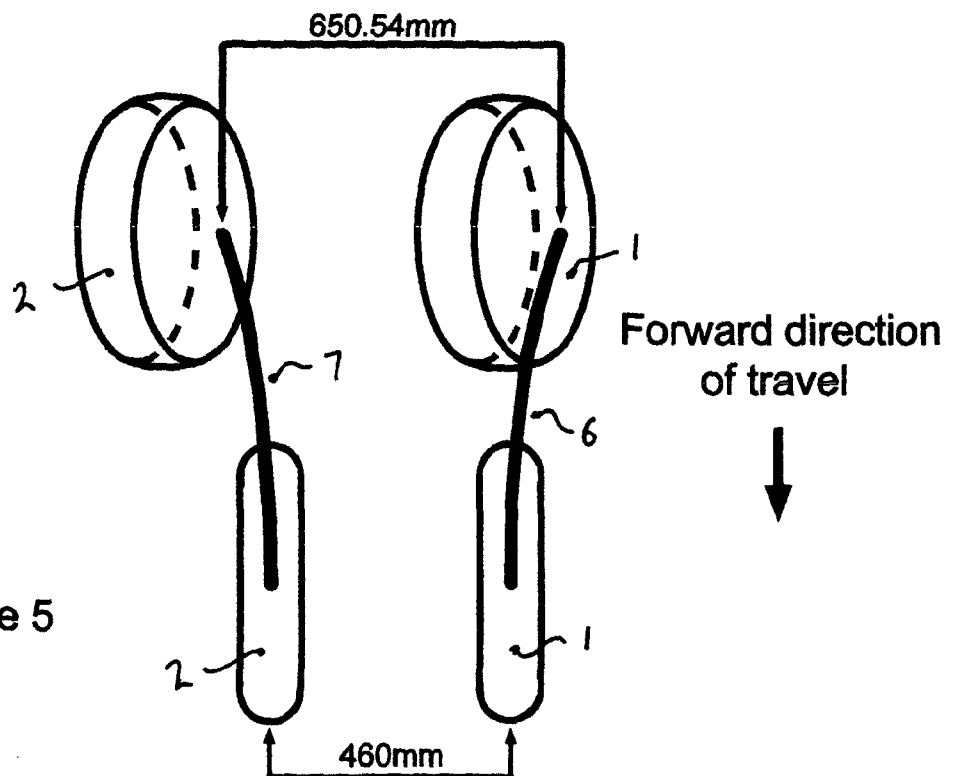


Figure 6

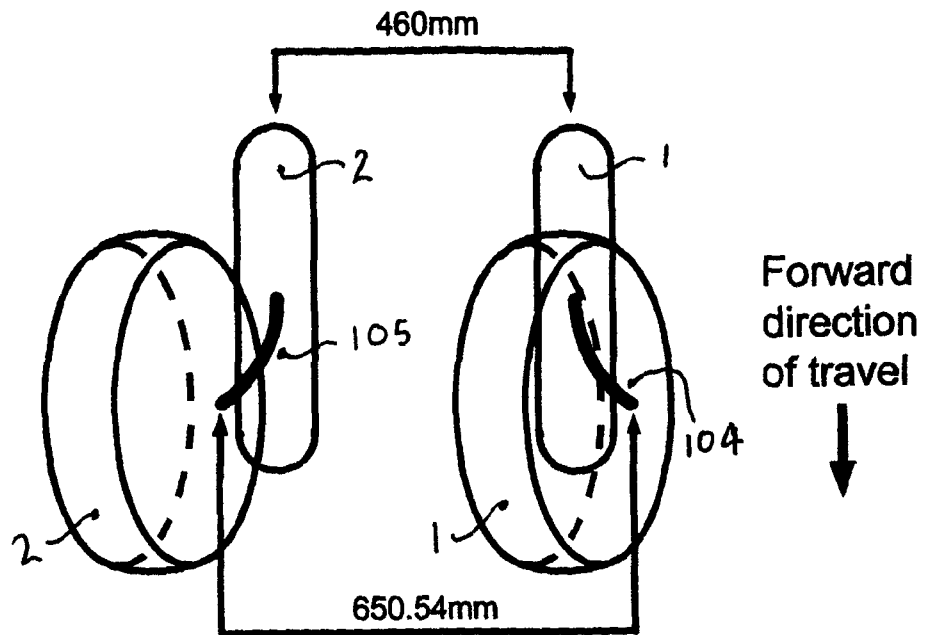
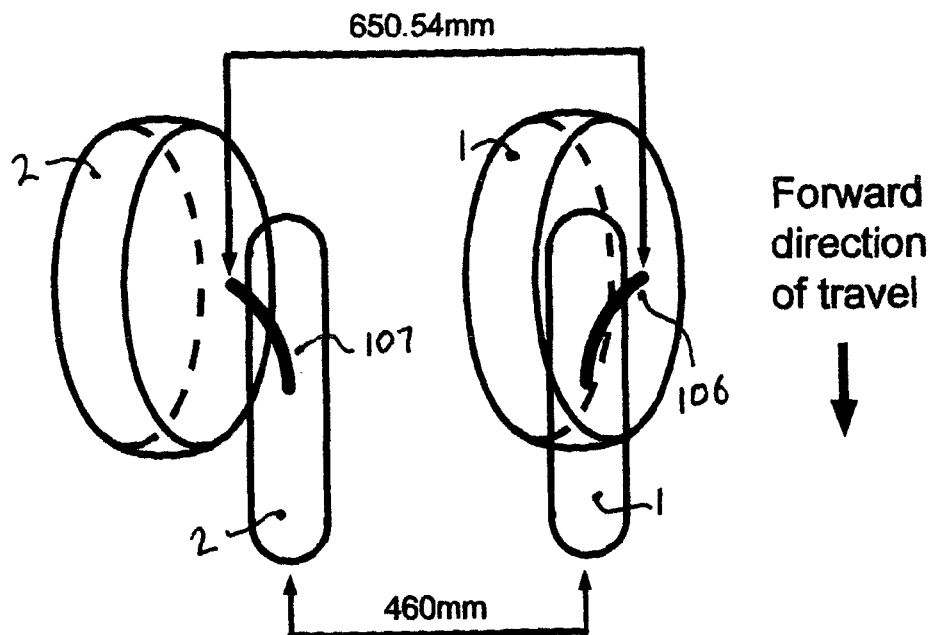
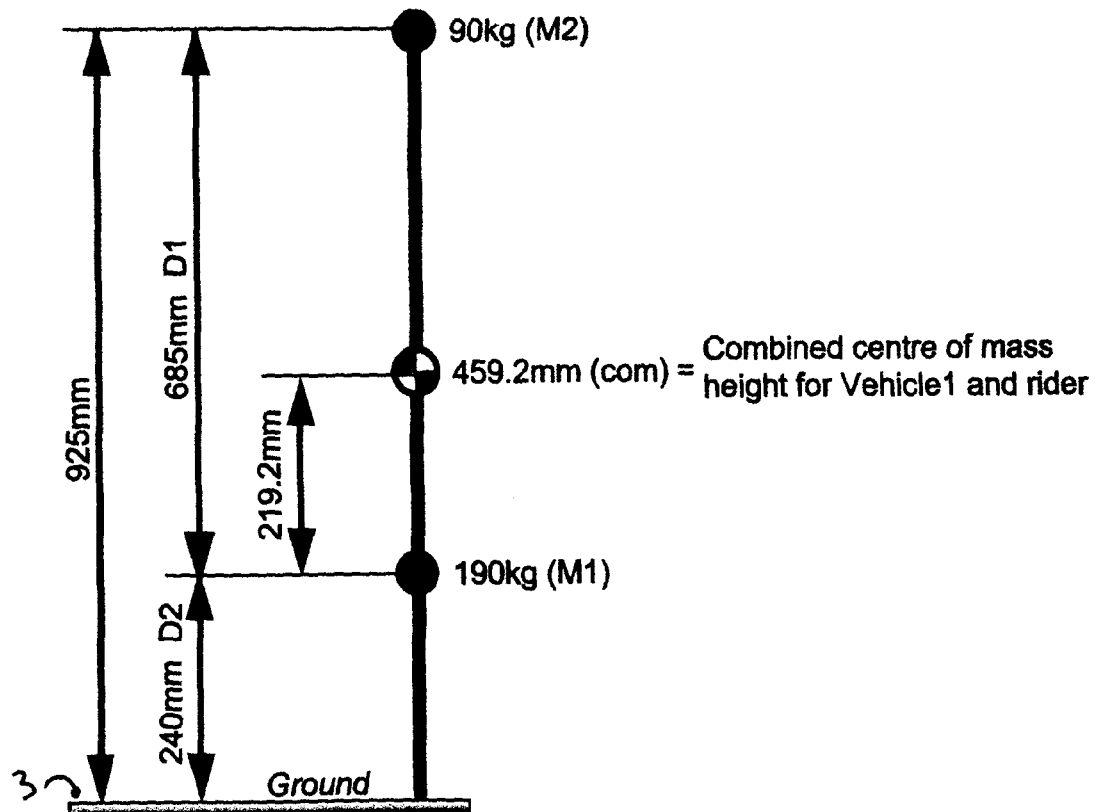


Figure 7





$$\text{Combined centre of mass height for Vehicle1 and rider} = \frac{M2}{M1 + M2} \times D1 + D2 =$$

$$\frac{90}{190\text{kg} + 90\text{kg}} = 0.32 \times 685\text{mm} = 219.2\text{mm} + 240\text{mm} = 459.2\text{mm}$$

Figure 8

Figure 9

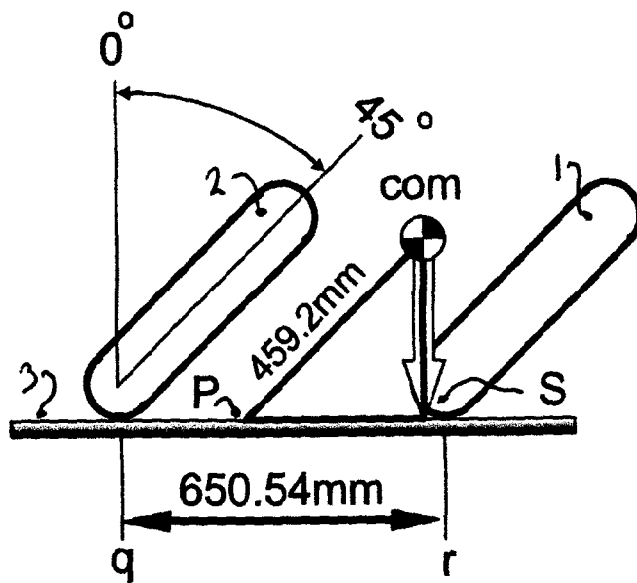
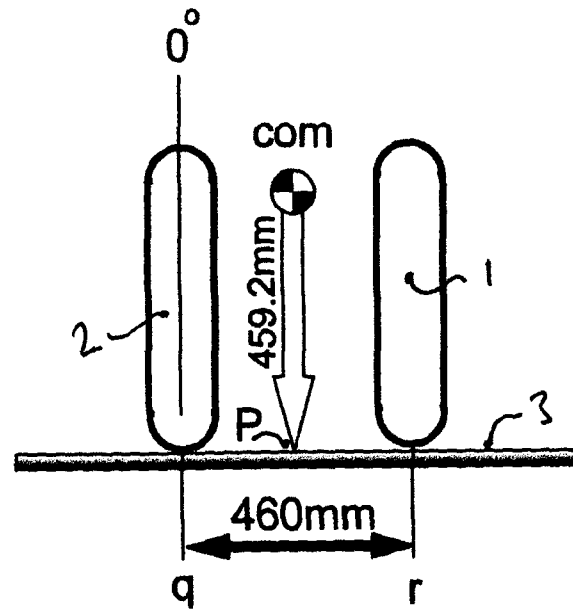


Figure 10

Figure 11

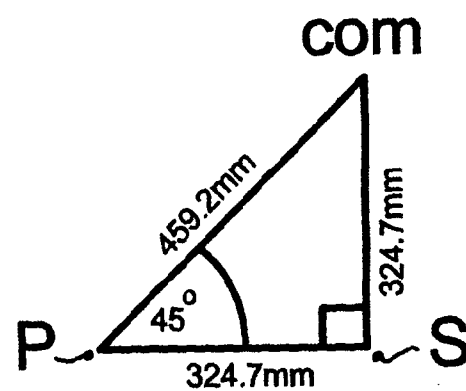


Figure 12

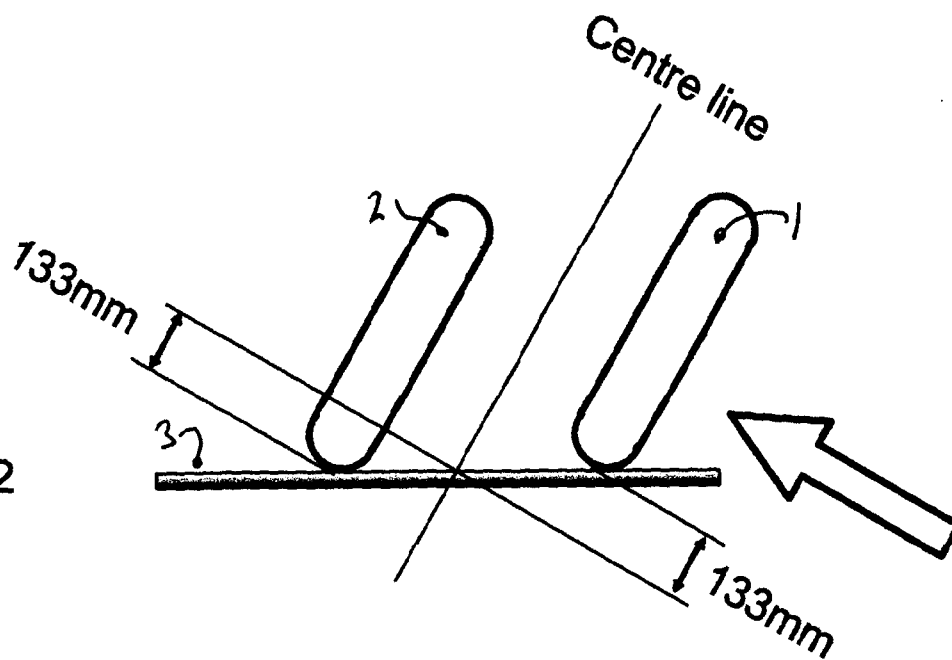
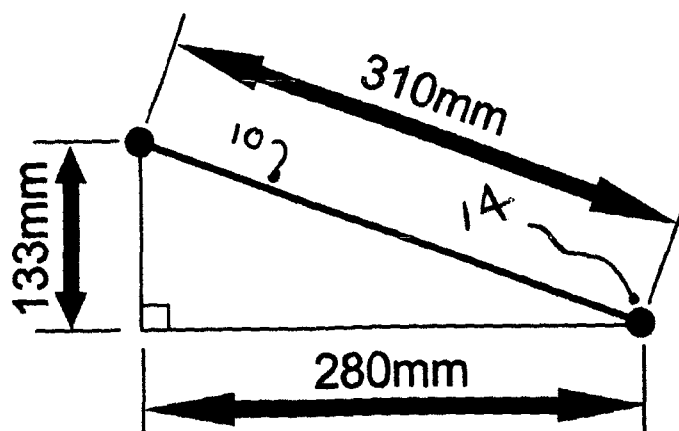


Figure 14



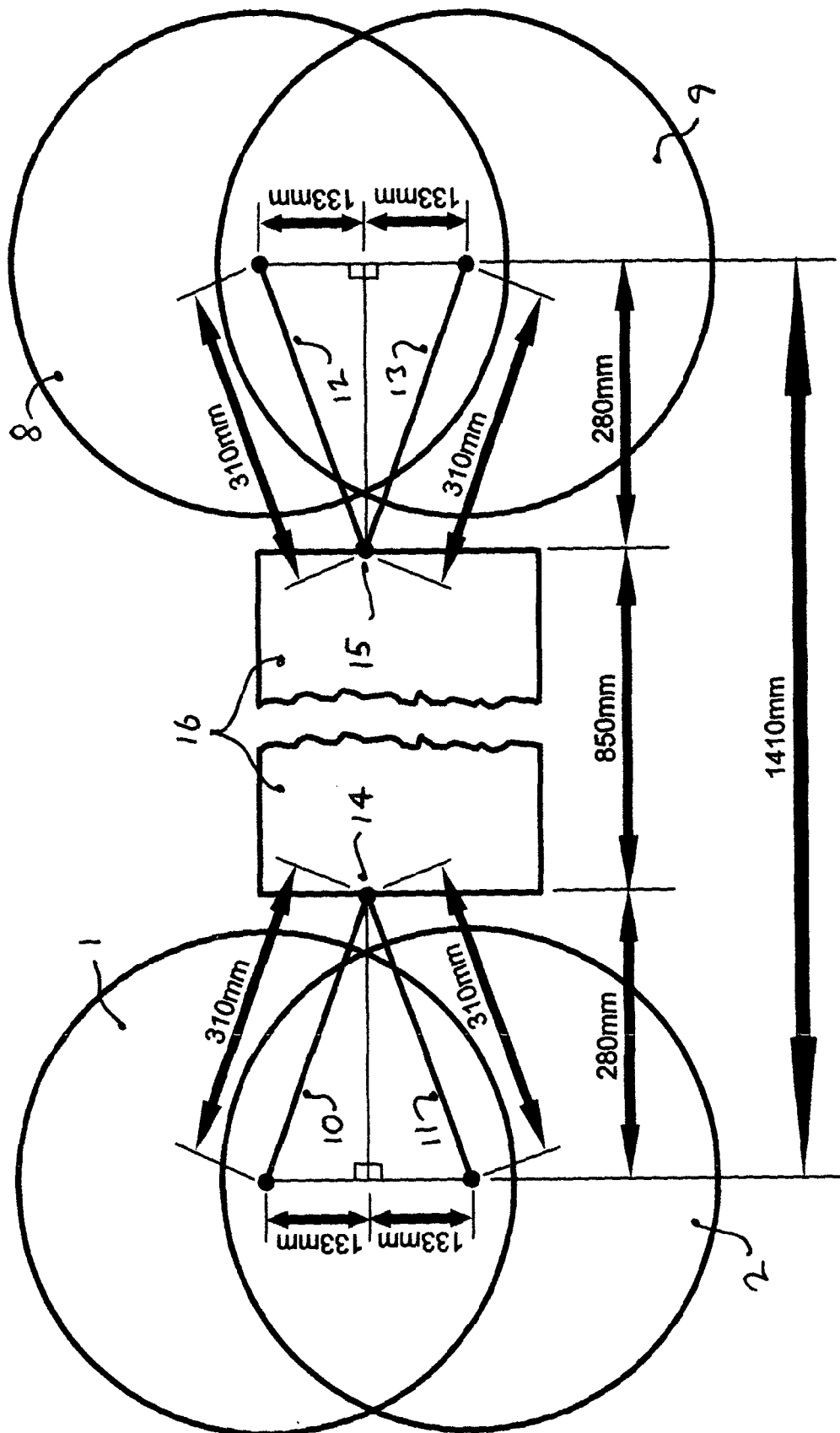


Figure 13

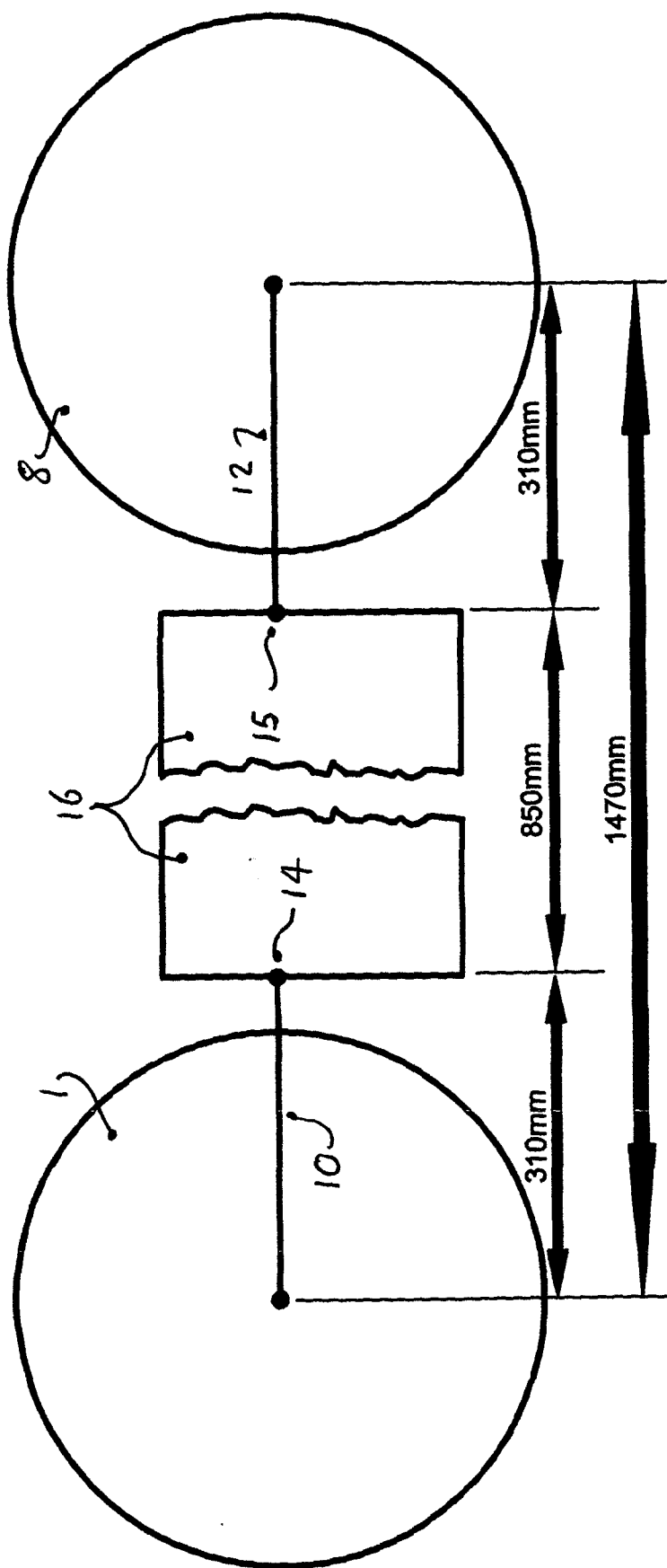


Figure 15

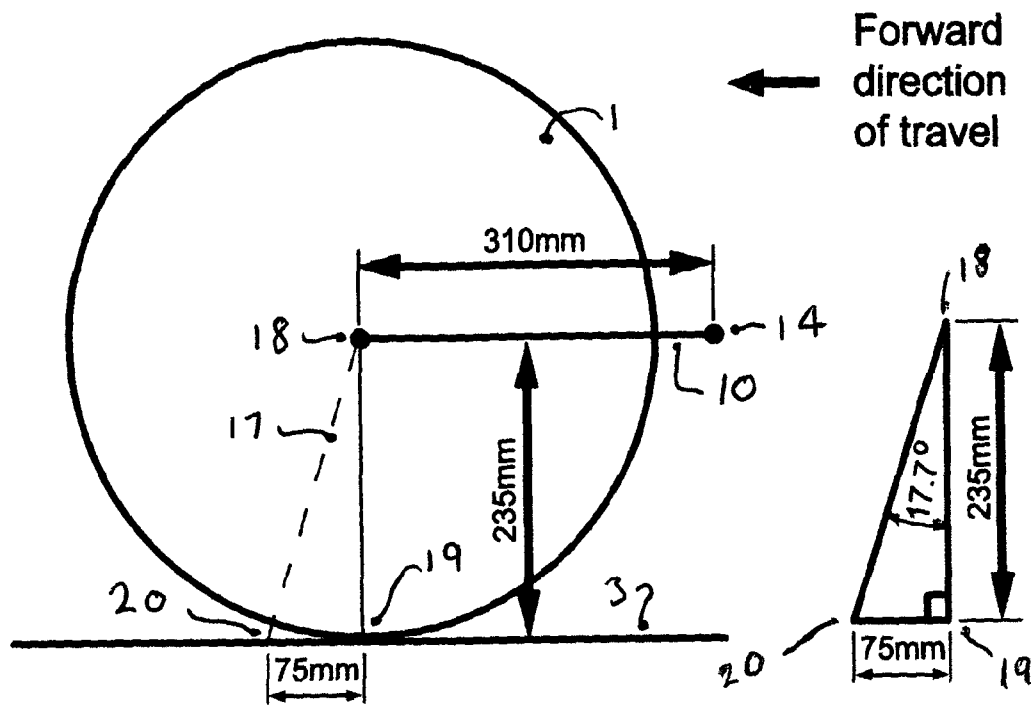


Figure 16

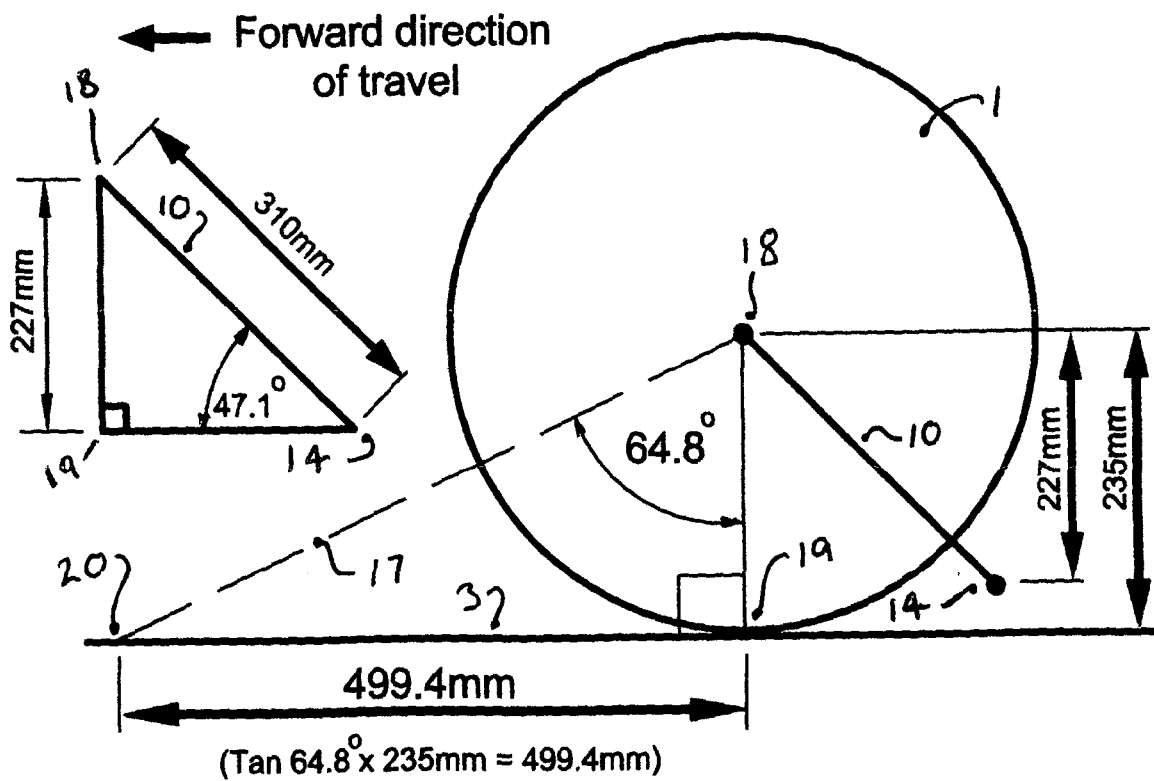


Figure 17

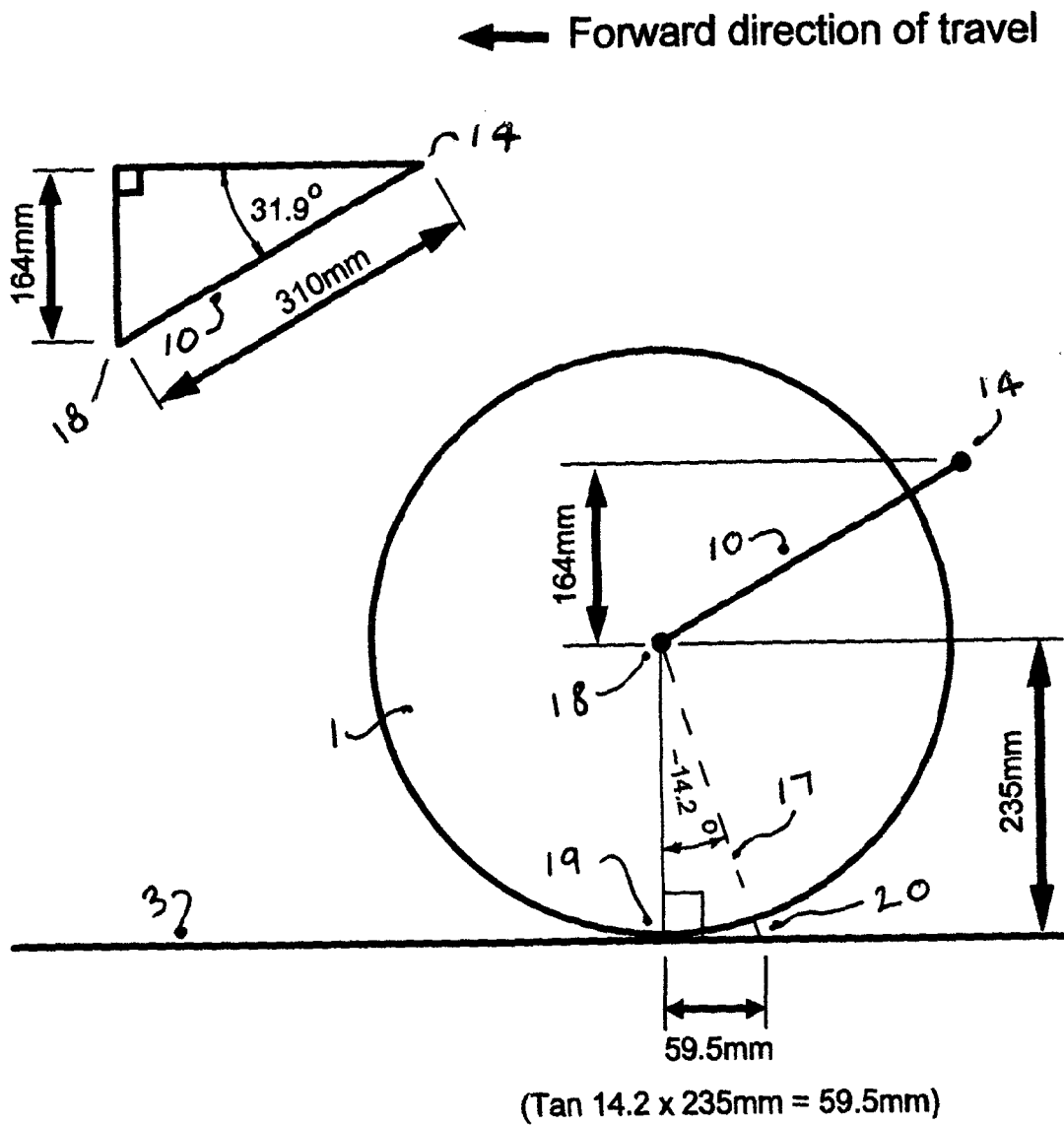
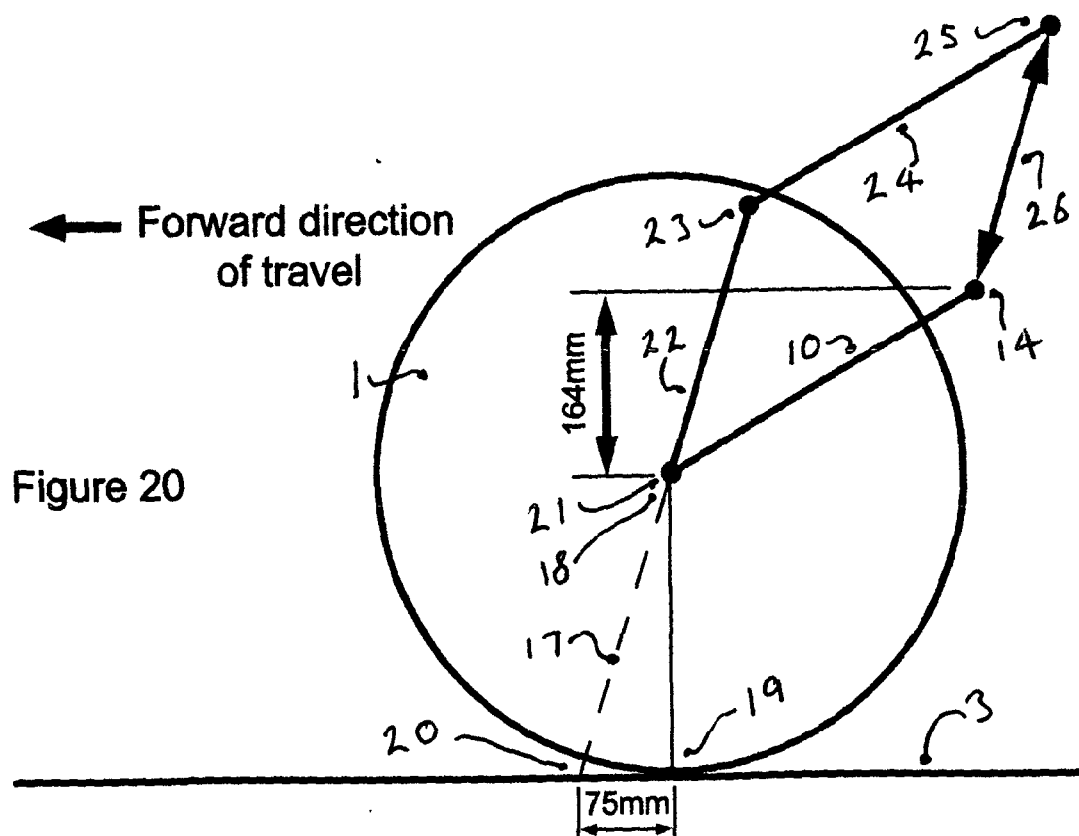
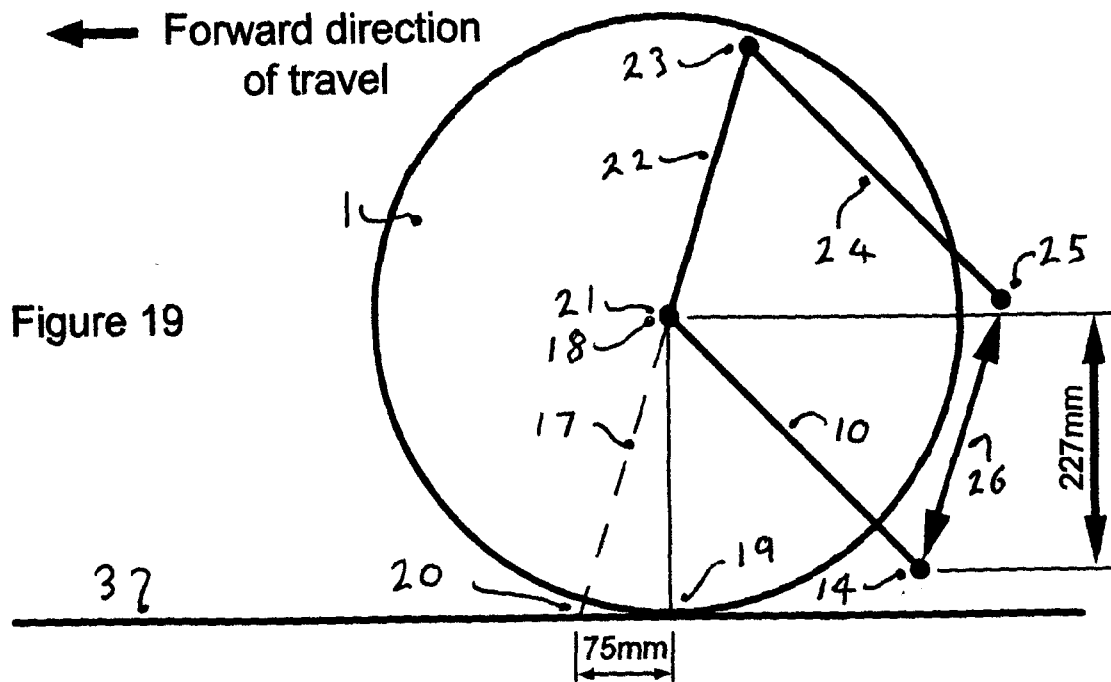
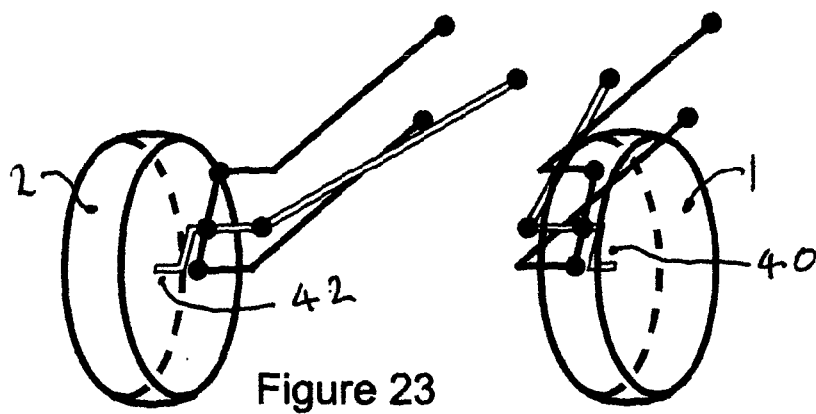
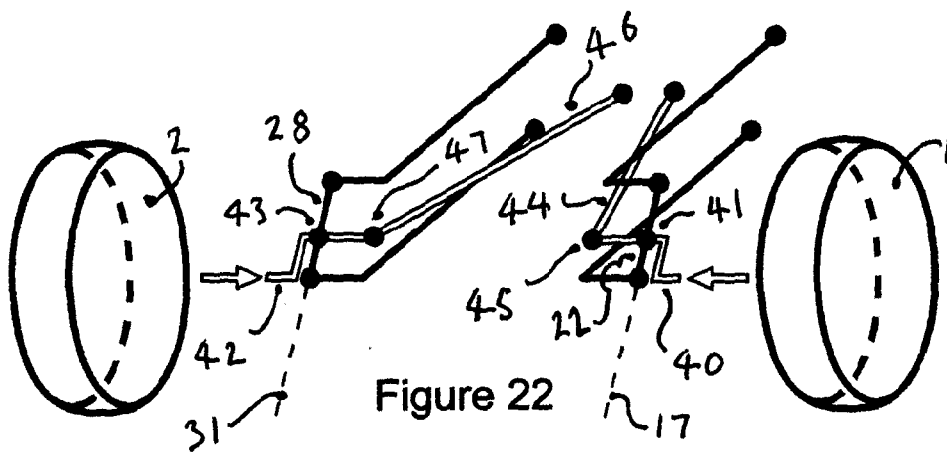
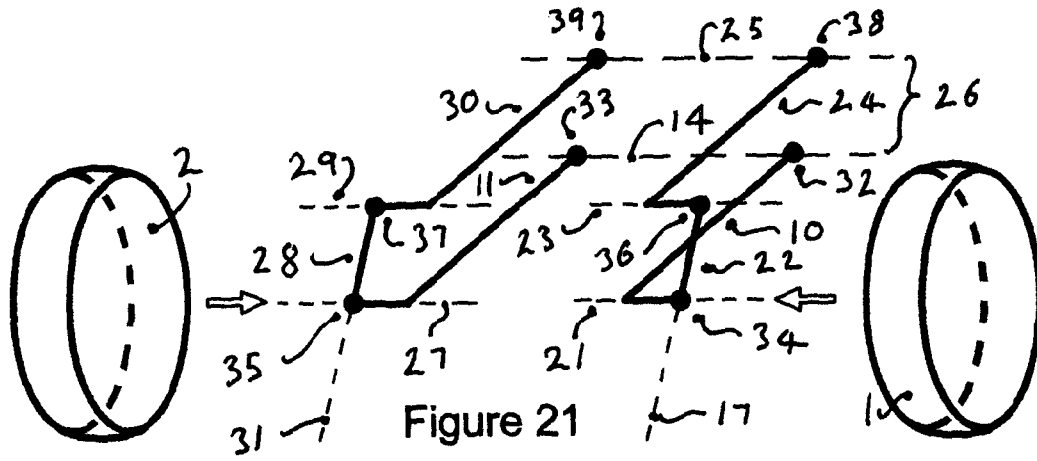


Figure 18





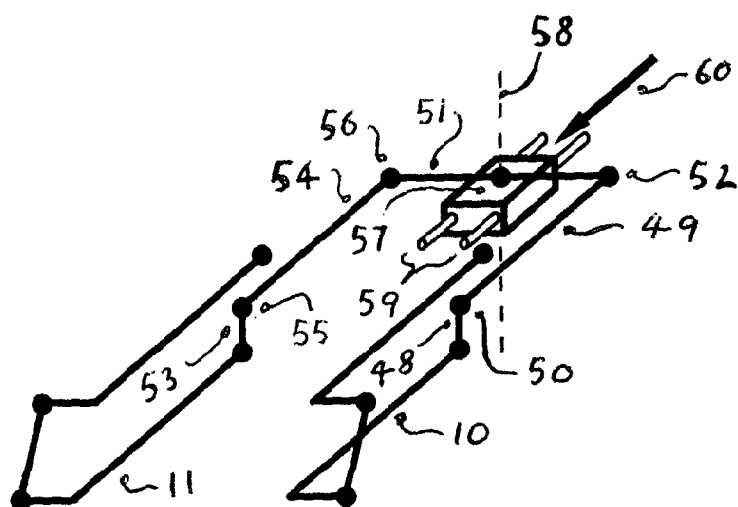


Figure 24

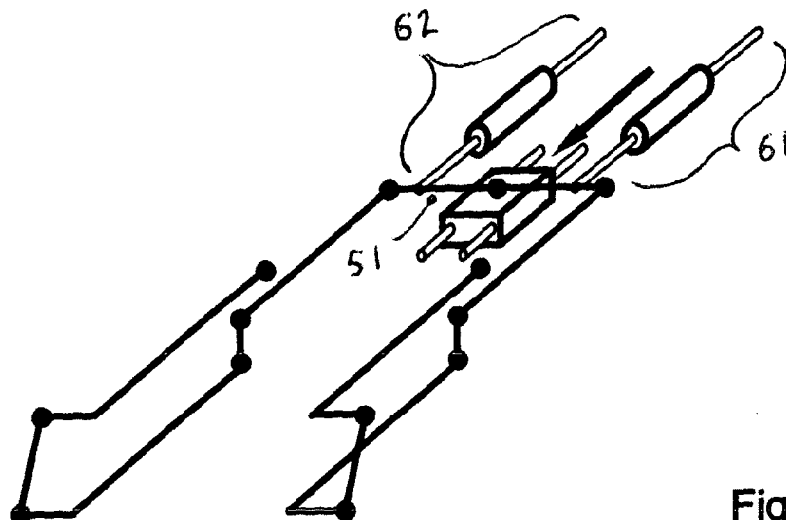
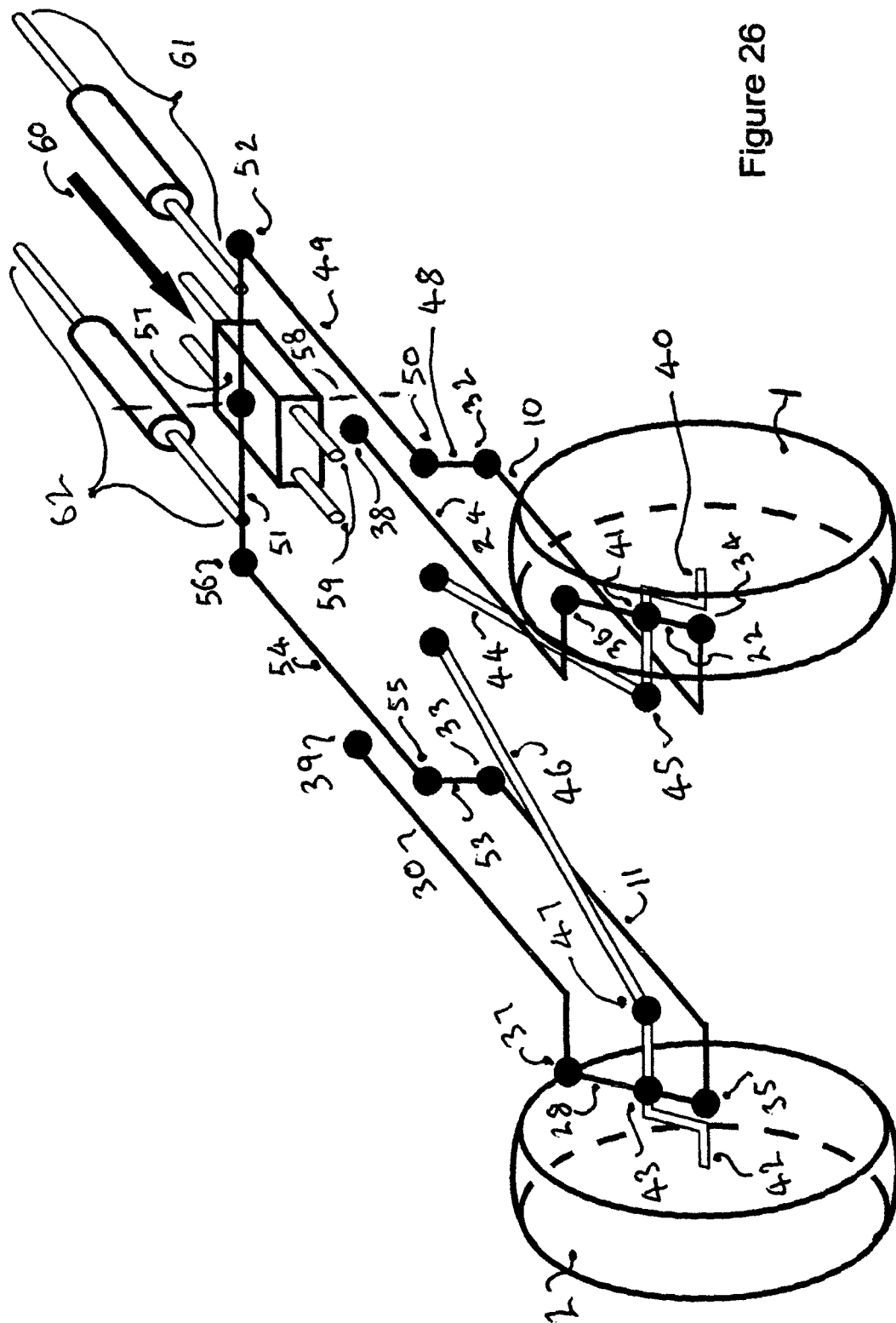


Figure 25



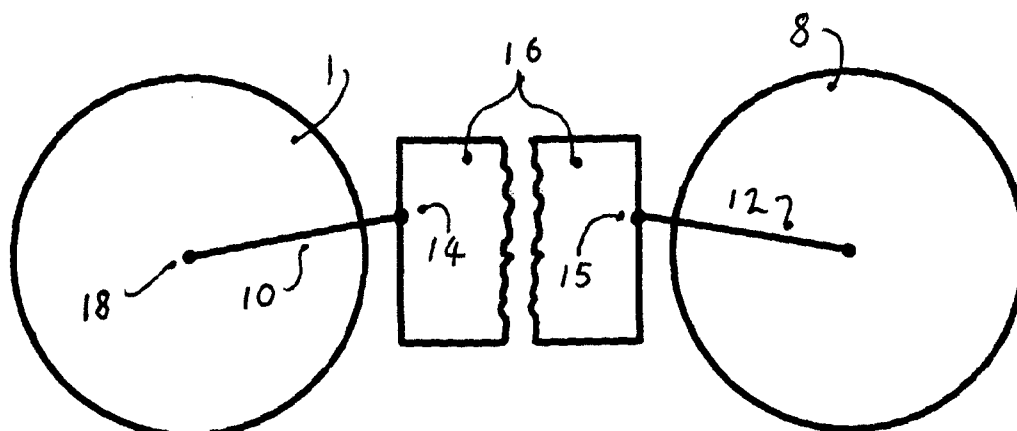


Figure 27

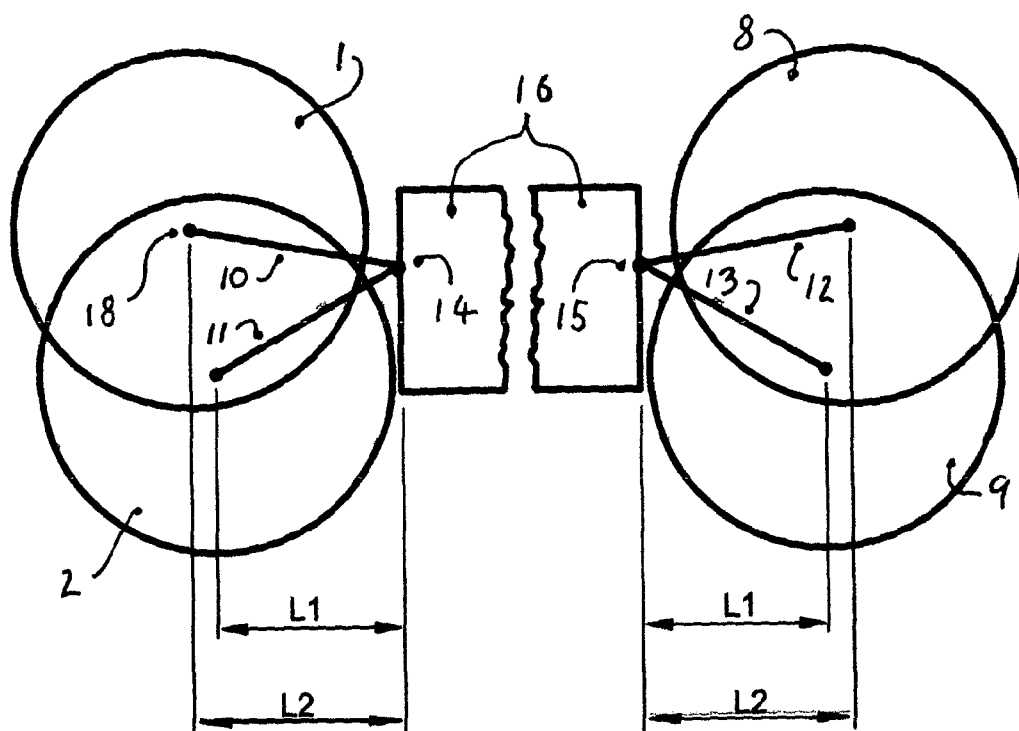
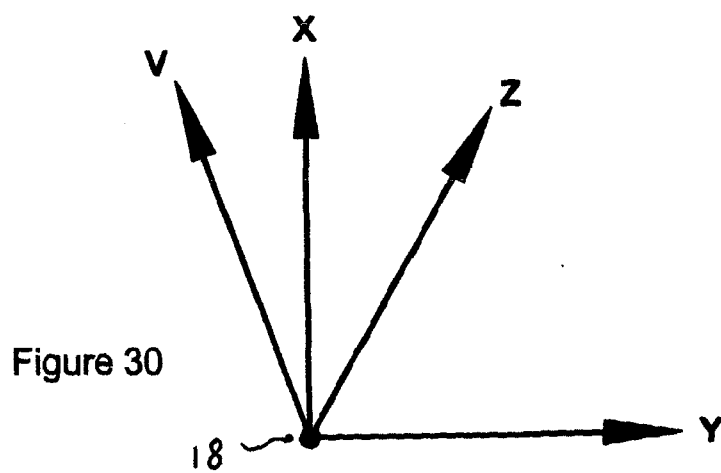
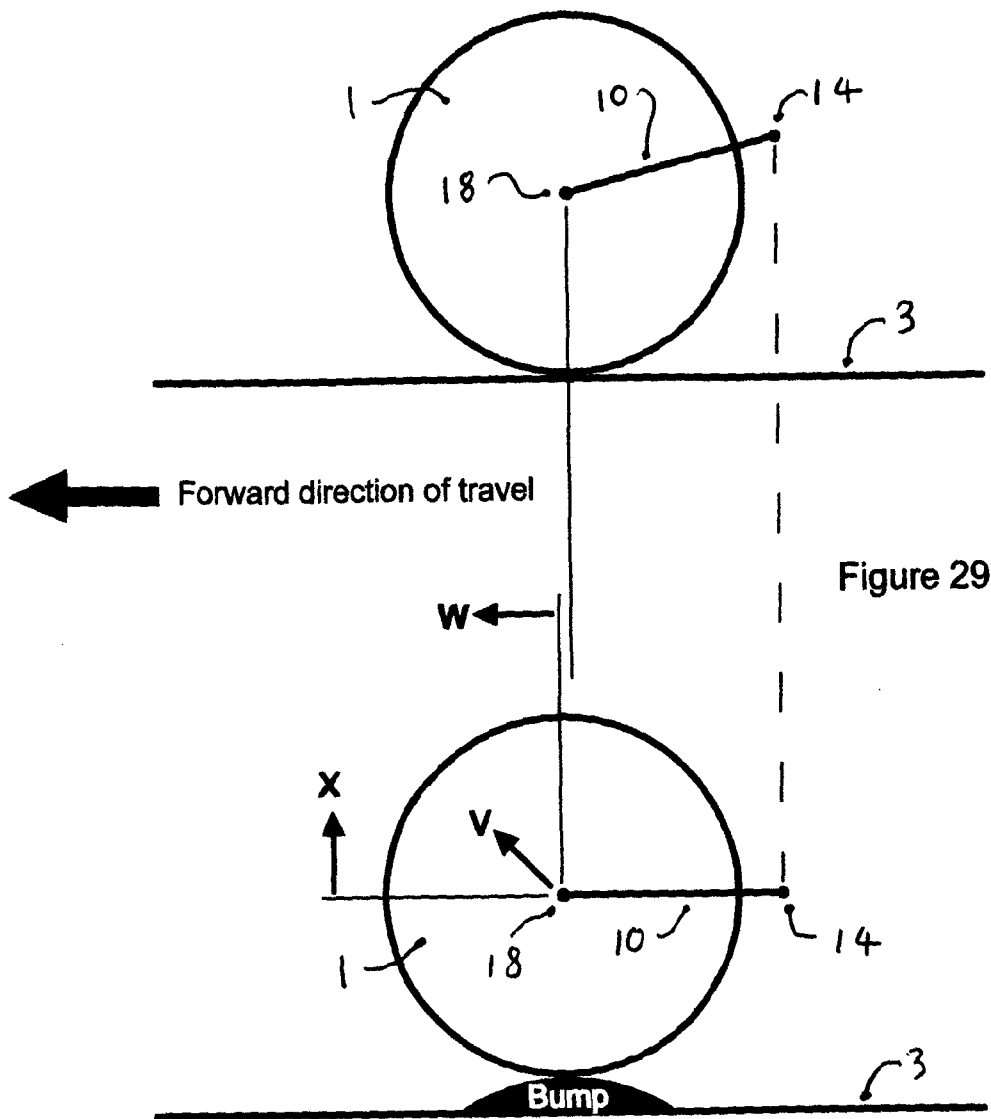


Figure 28



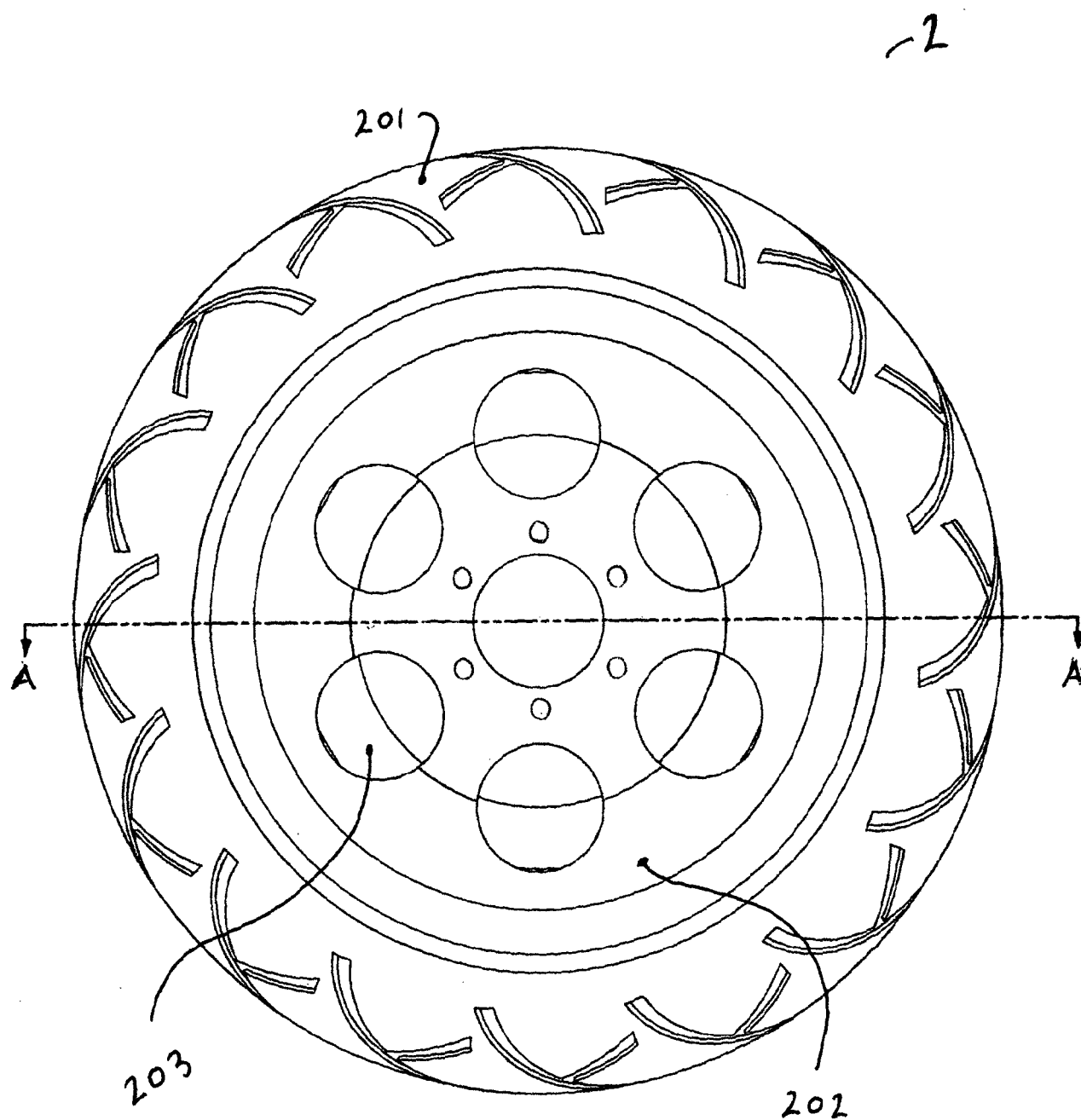
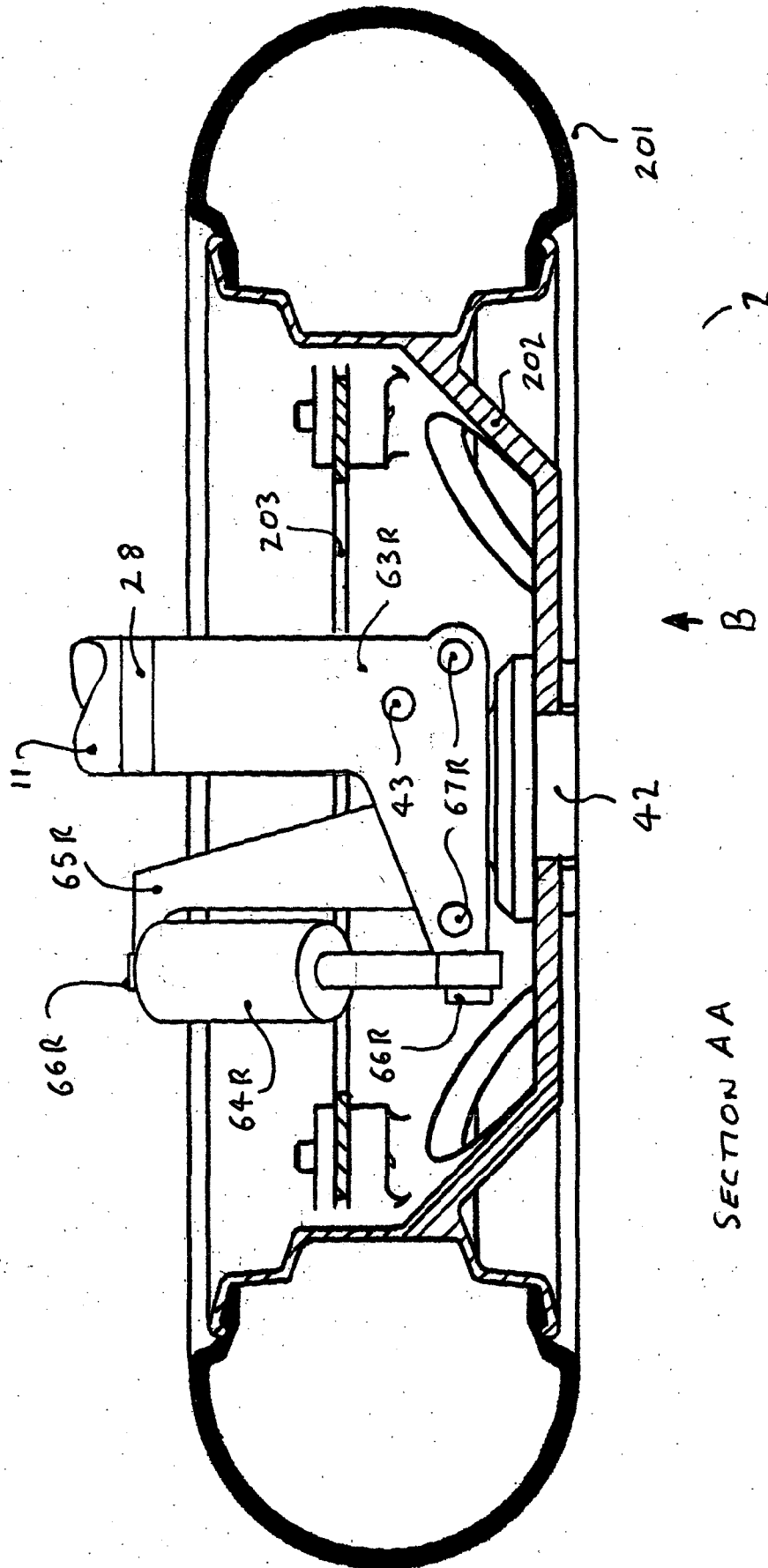


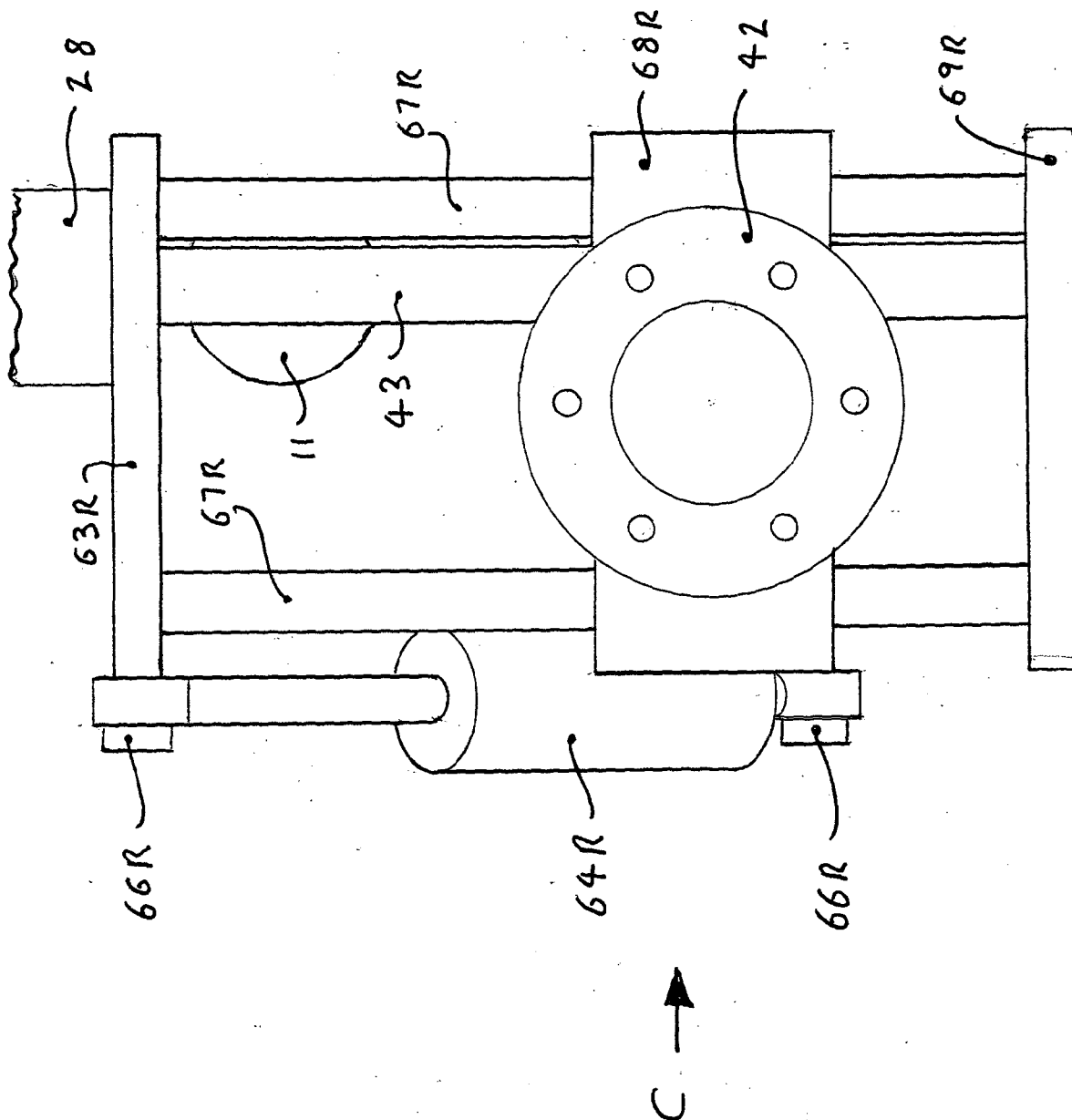
FIGURE 31



SECTION AA

FIGURE 32

FIGURE 33



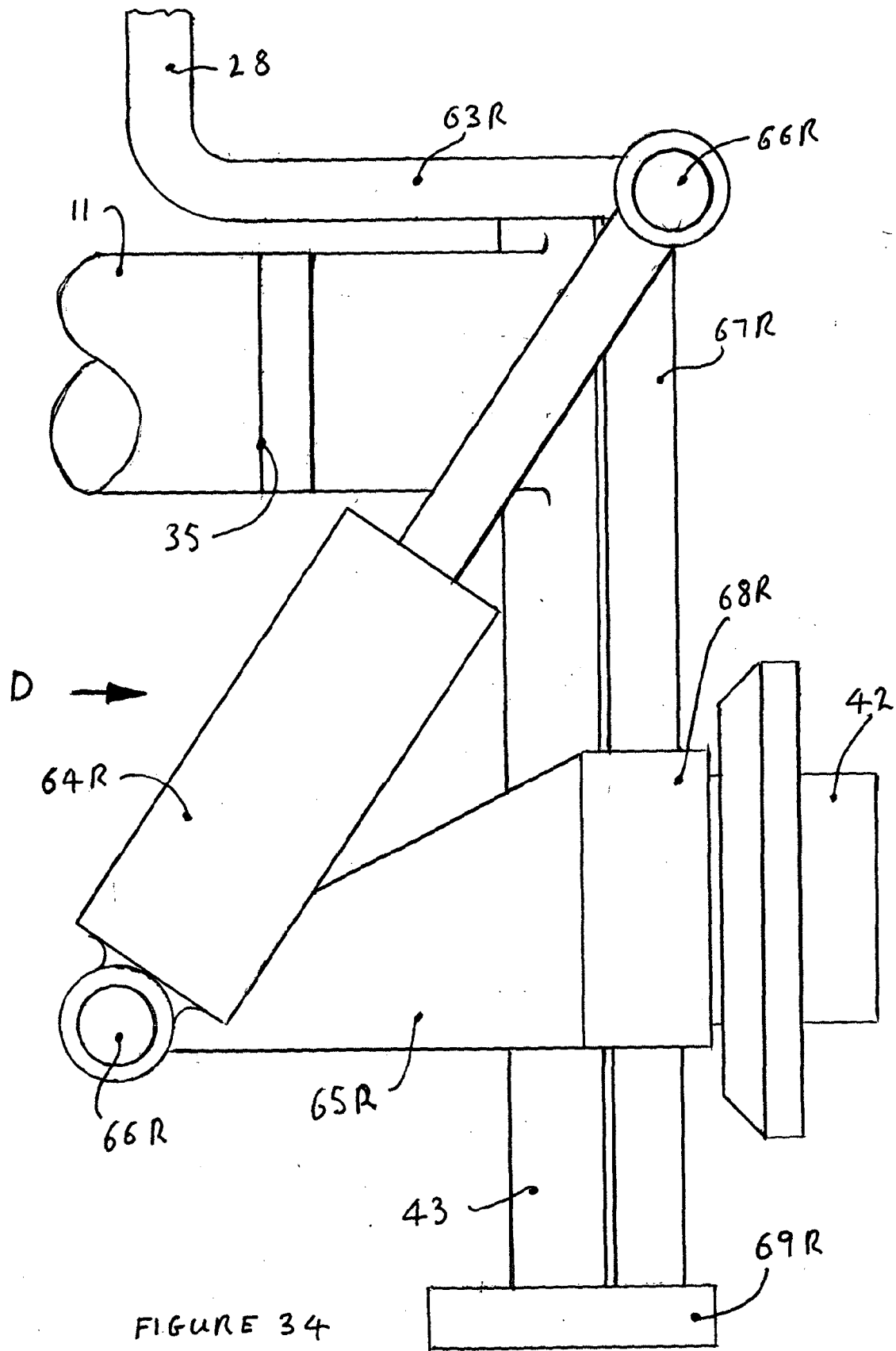


FIGURE 34

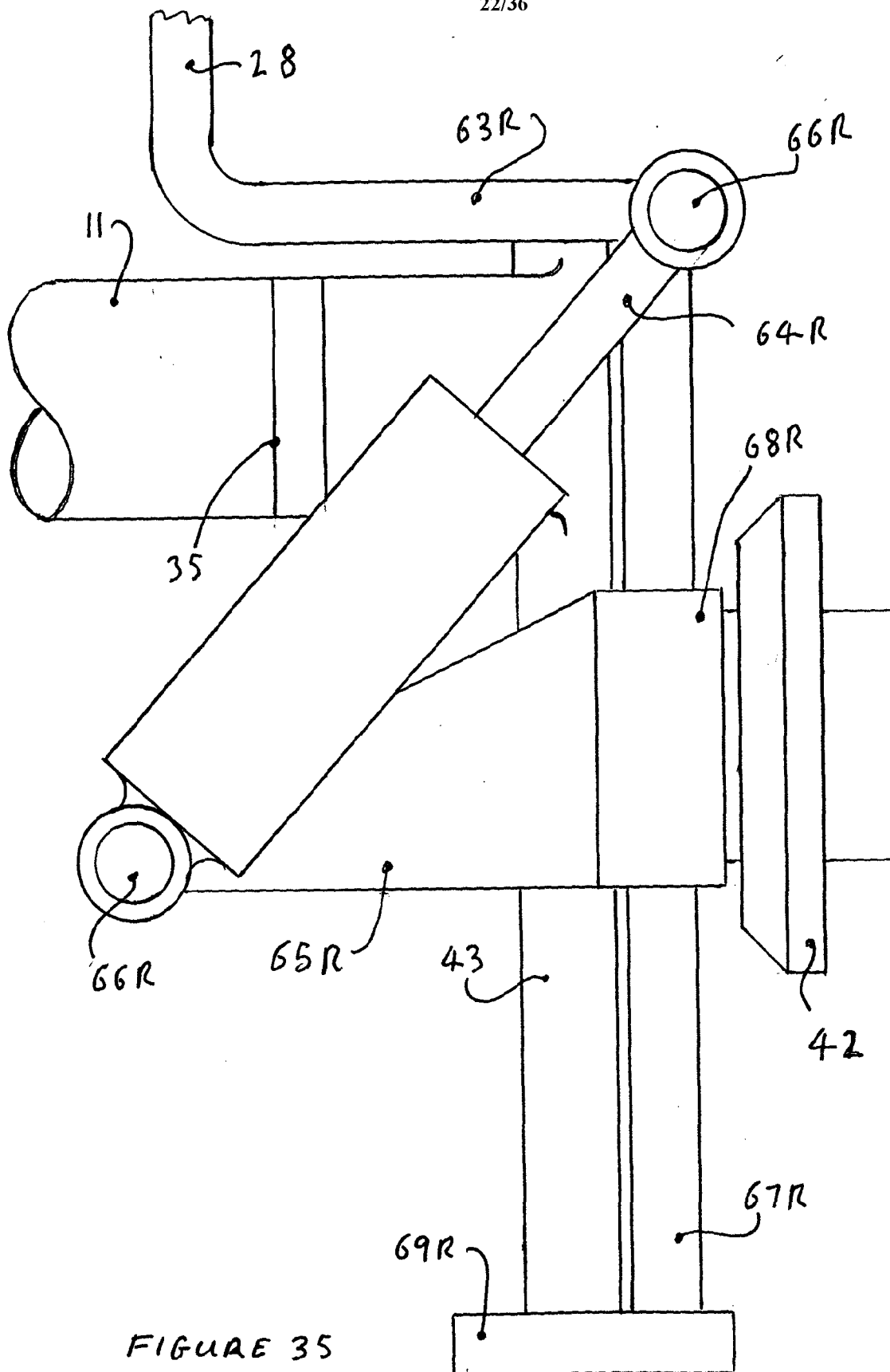


FIGURE 35

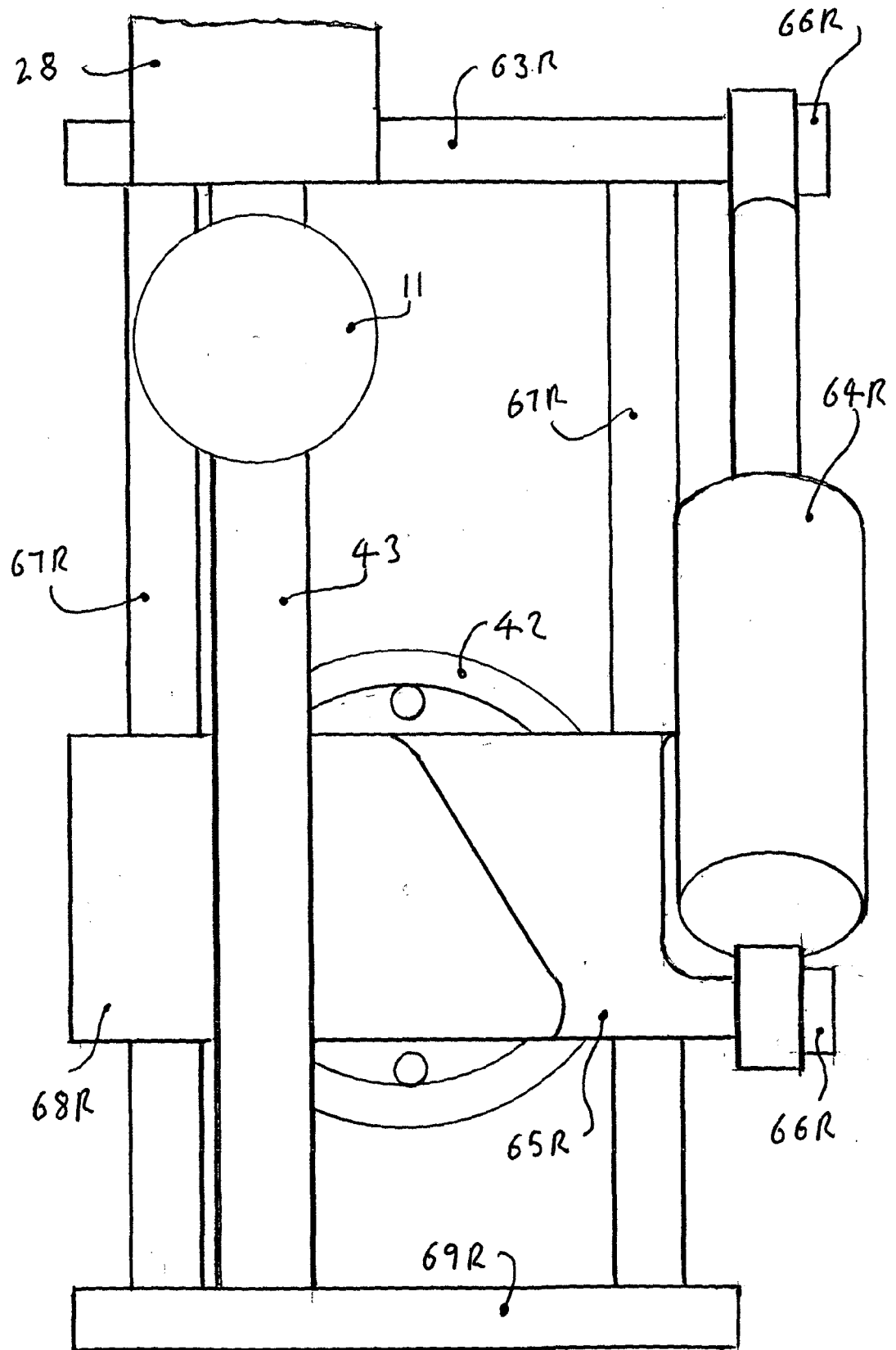


FIGURE 36

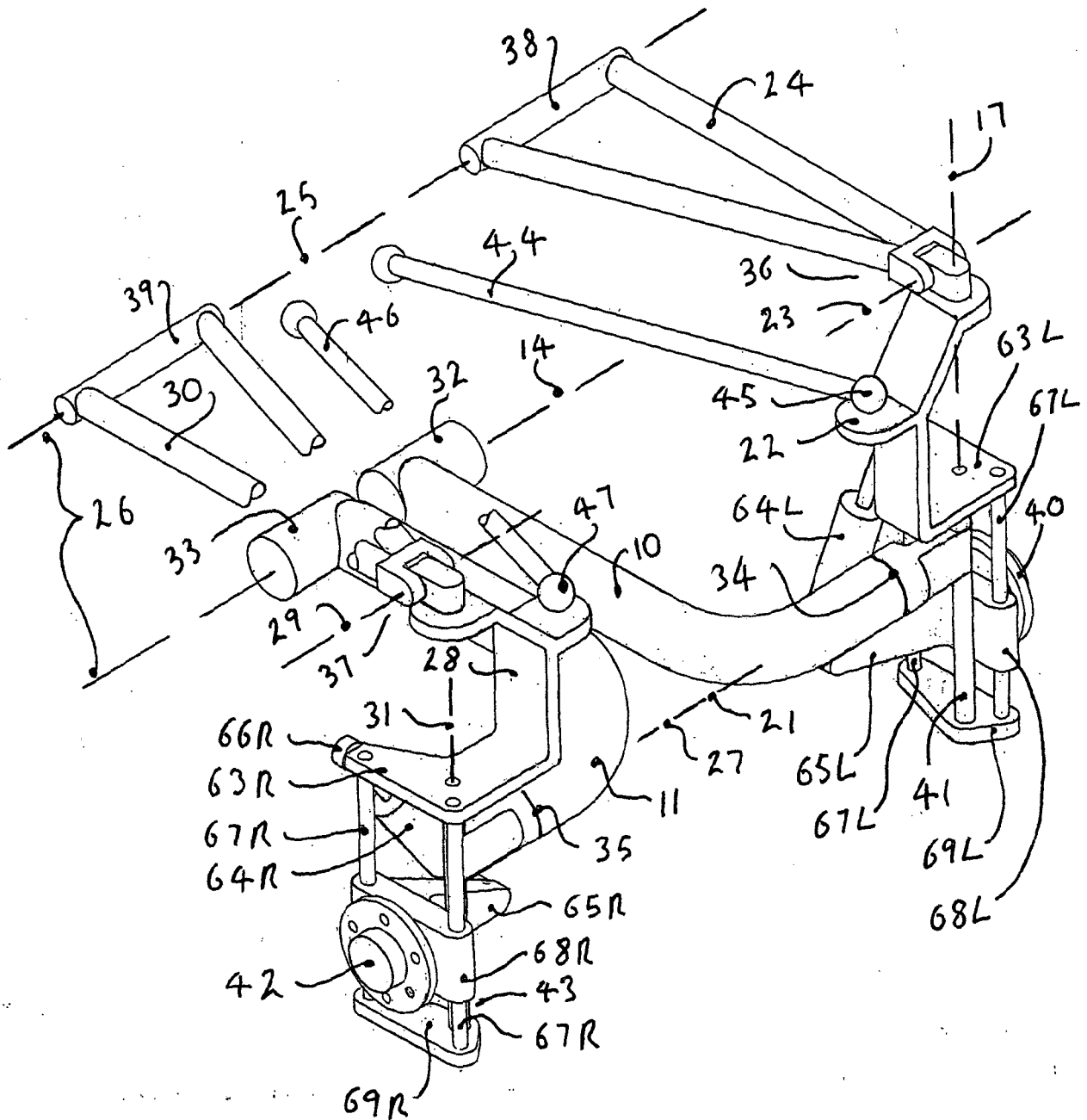


FIGURE 37

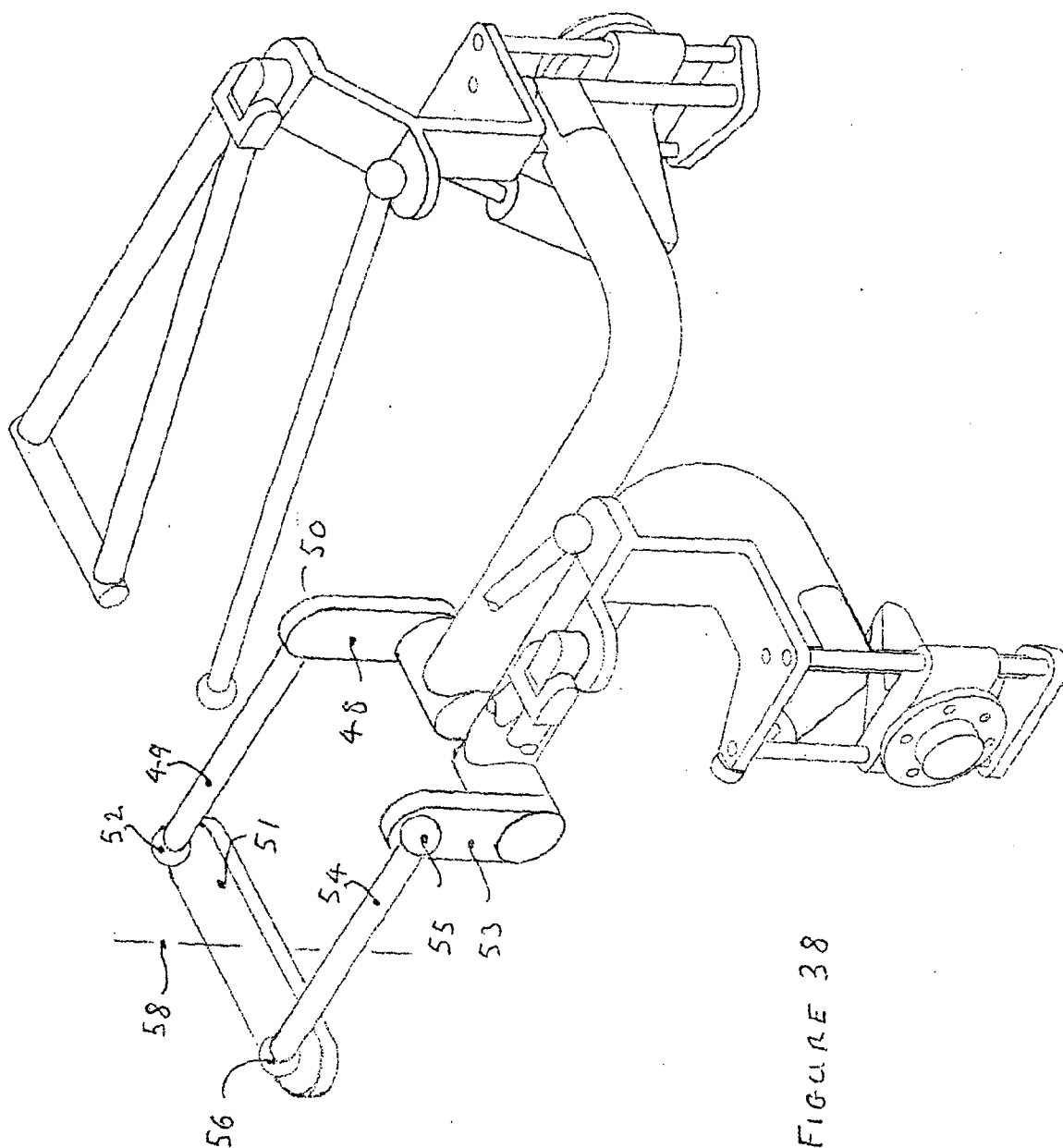


FIGURE 38

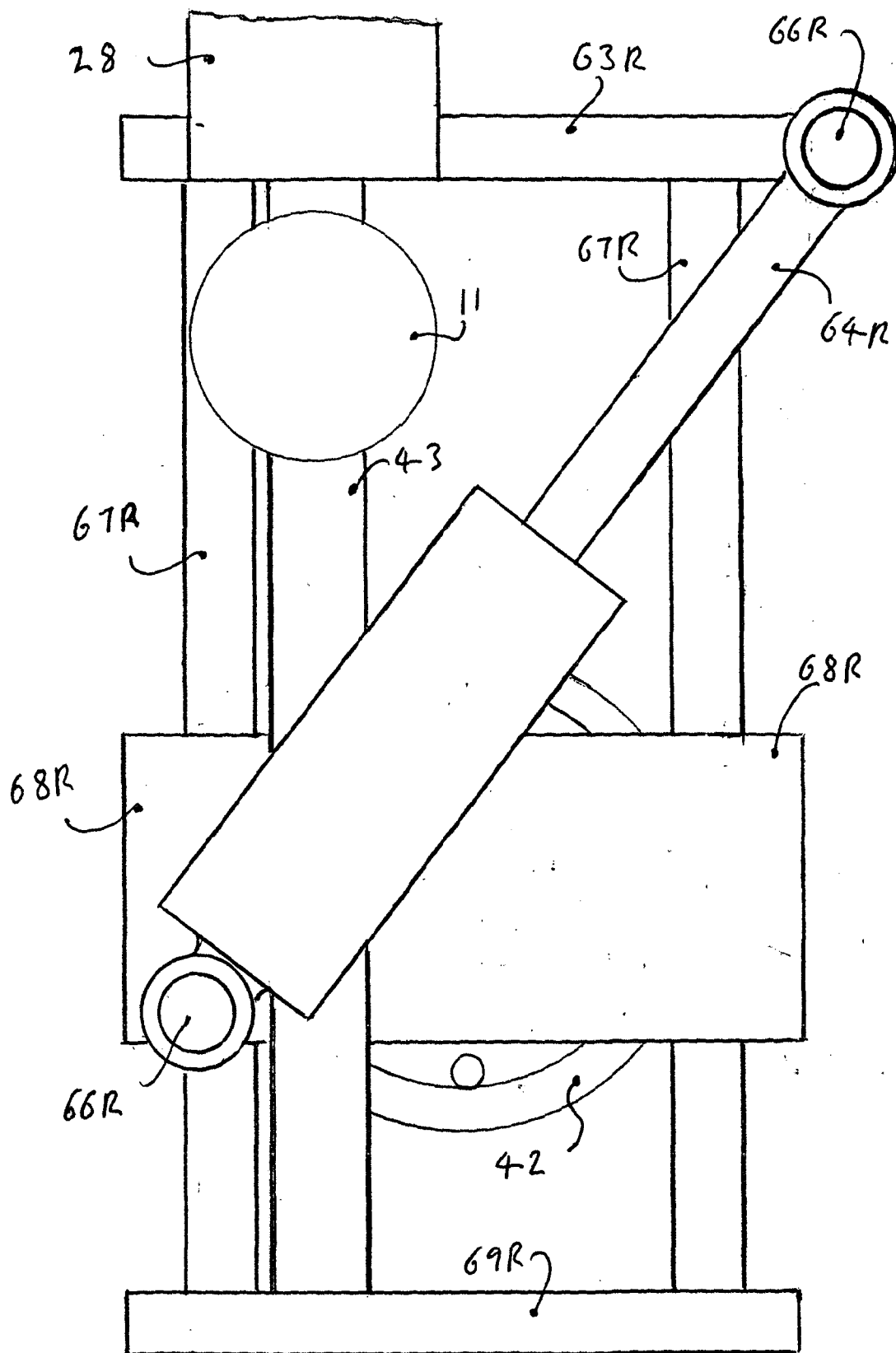


FIGURE 39

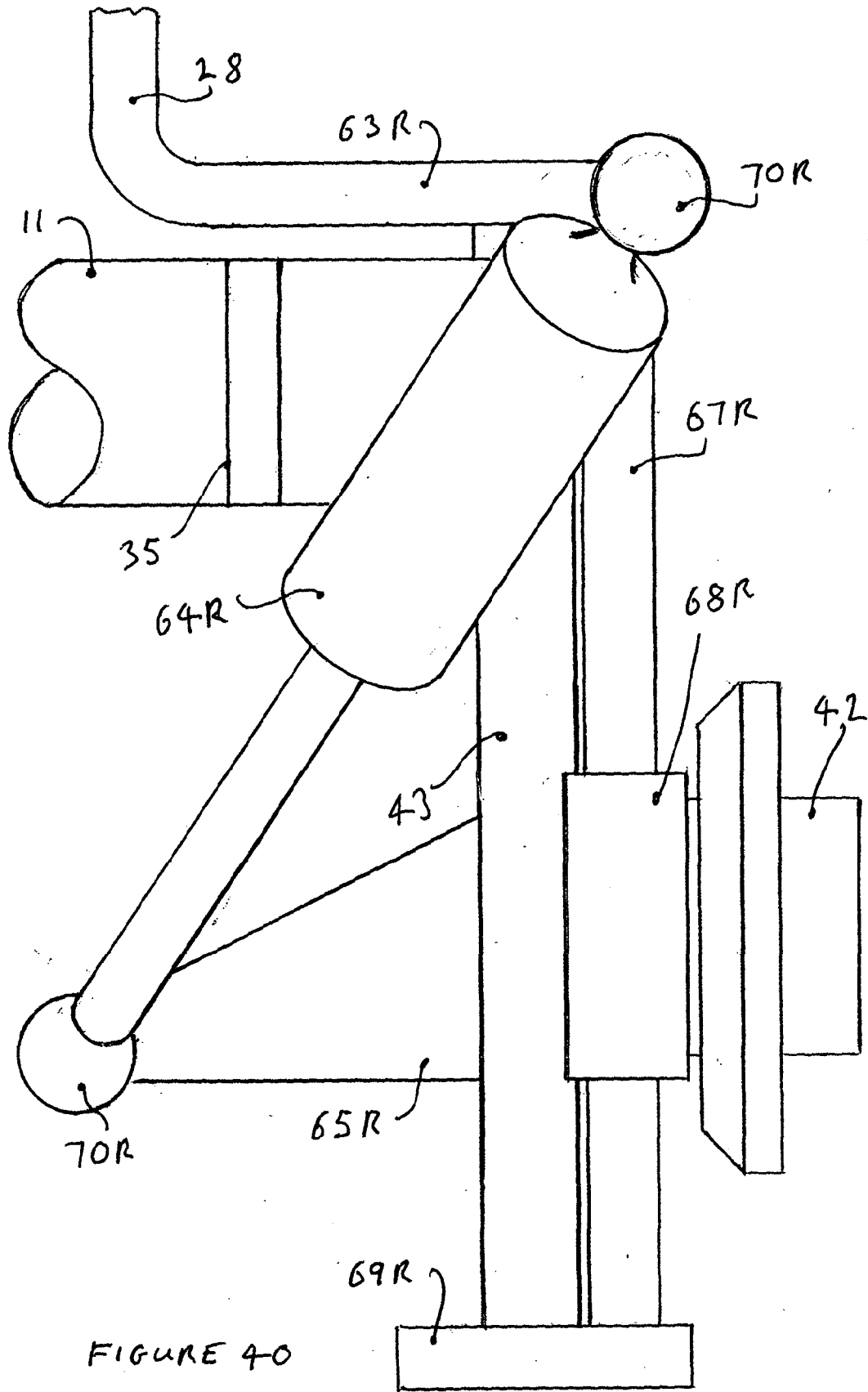


Figure 41

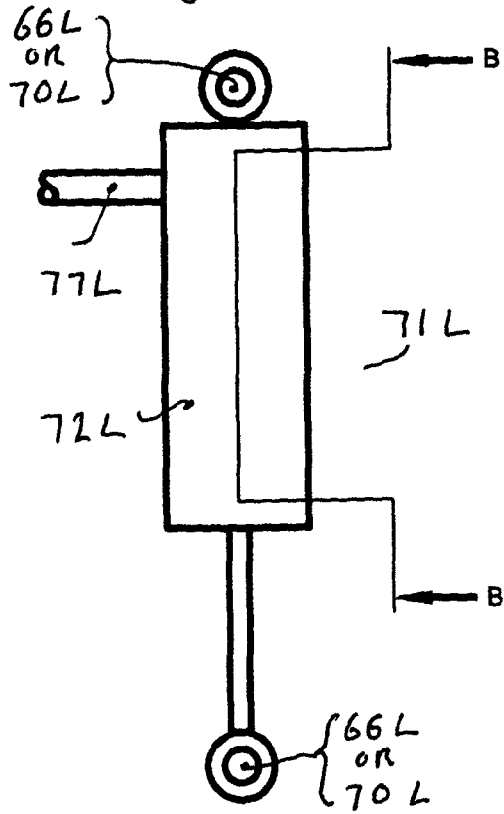


Figure 42

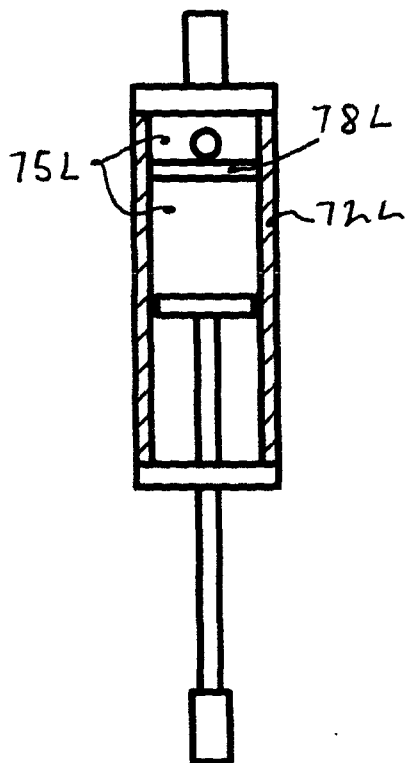
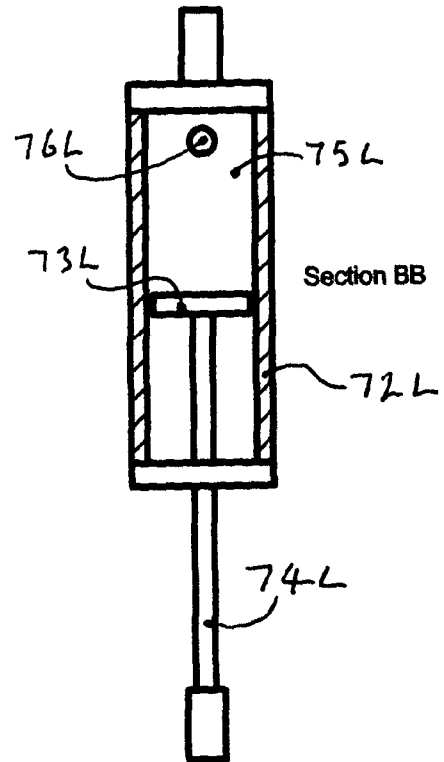


Figure 43

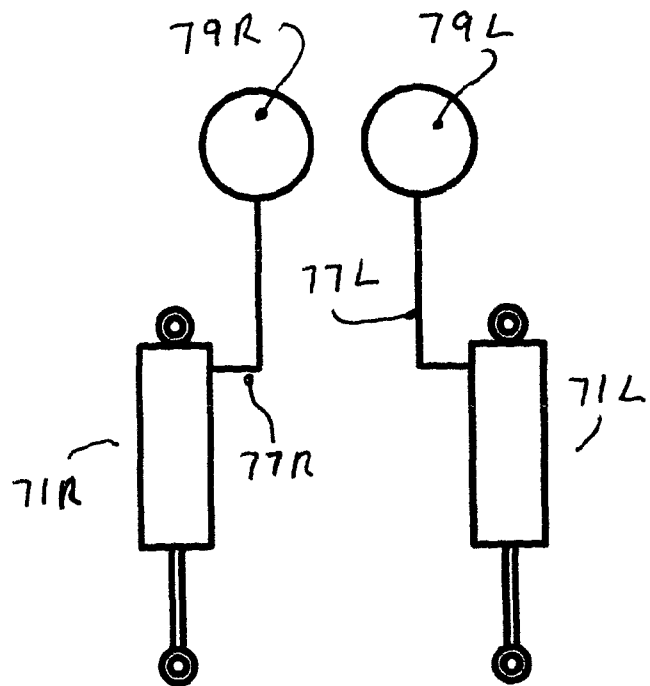


Figure 44

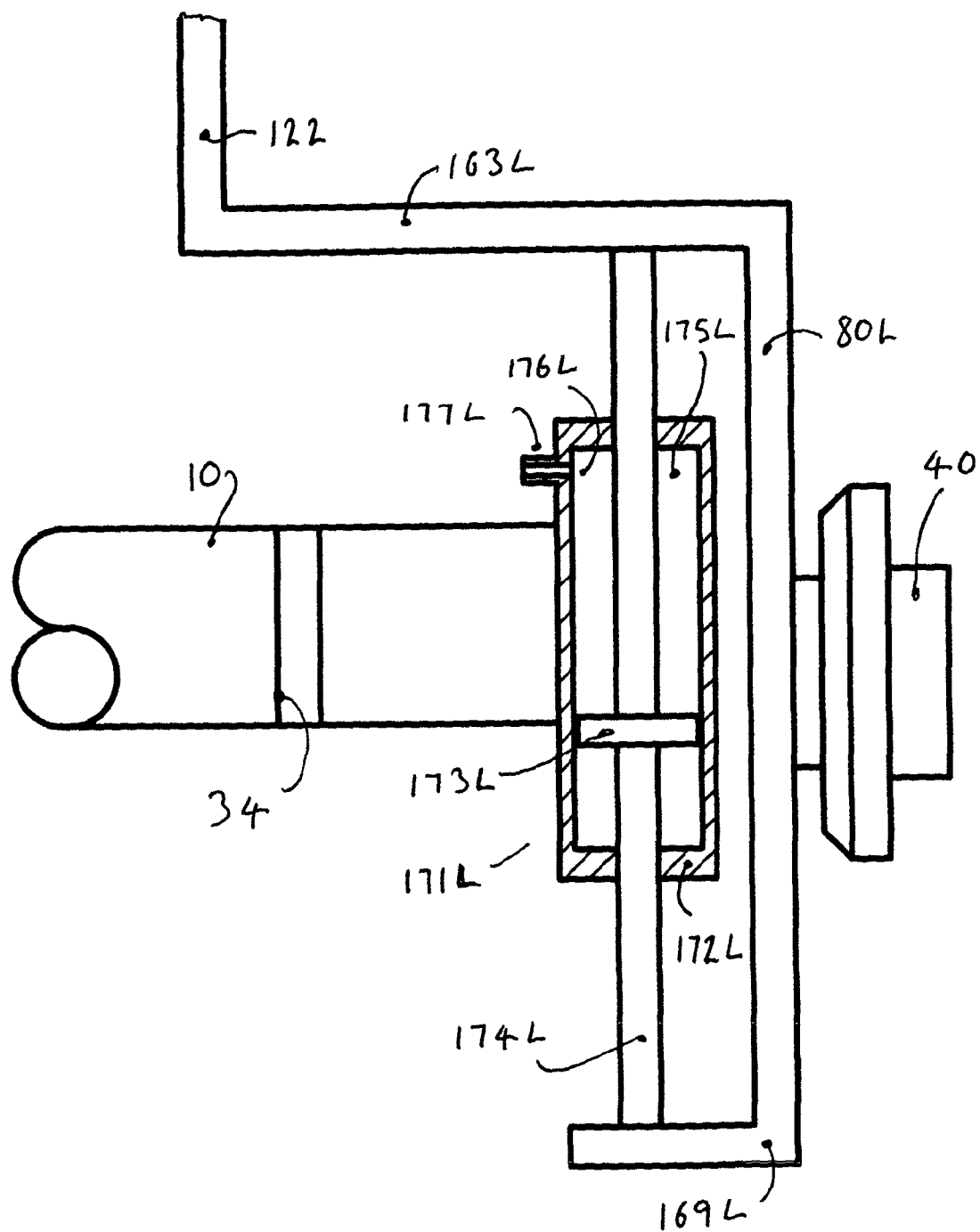


Figure 45

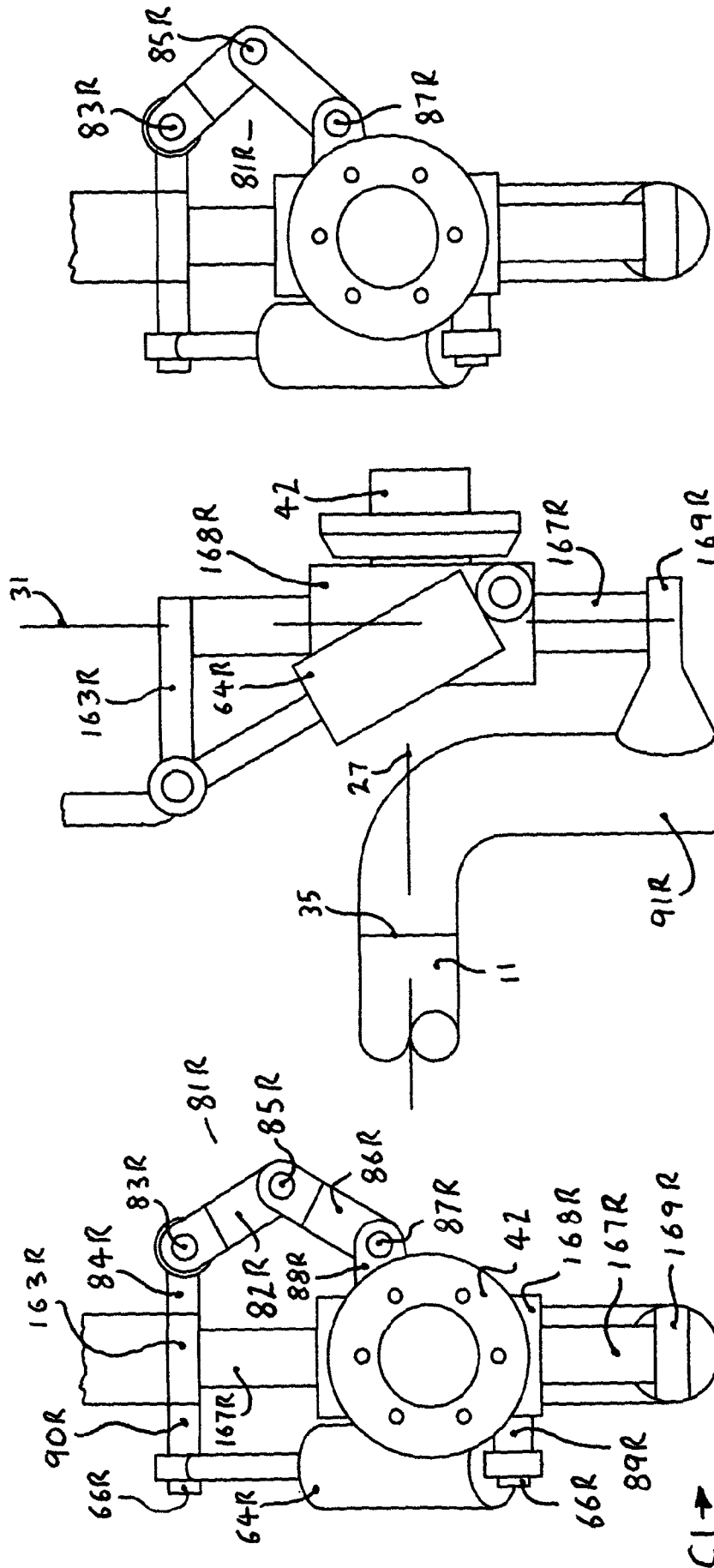


Figure 48

Figure 47

Figure 46

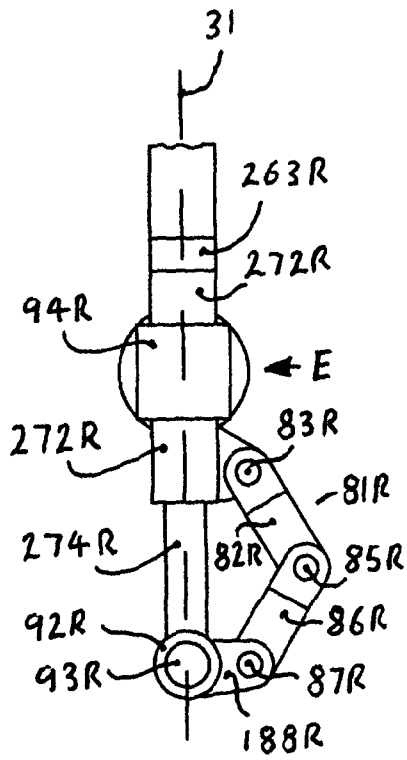


Figure 49

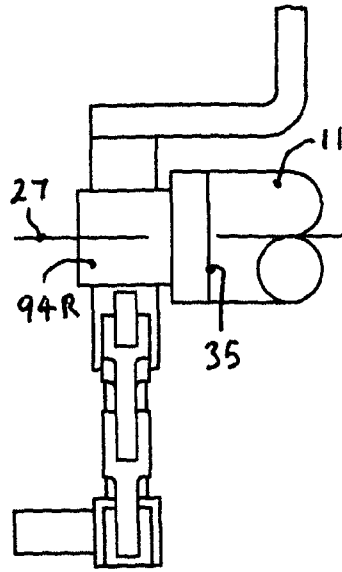


Figure 50

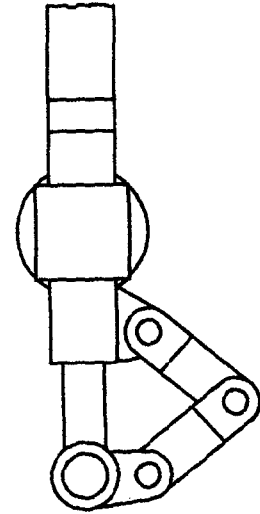


Figure 51

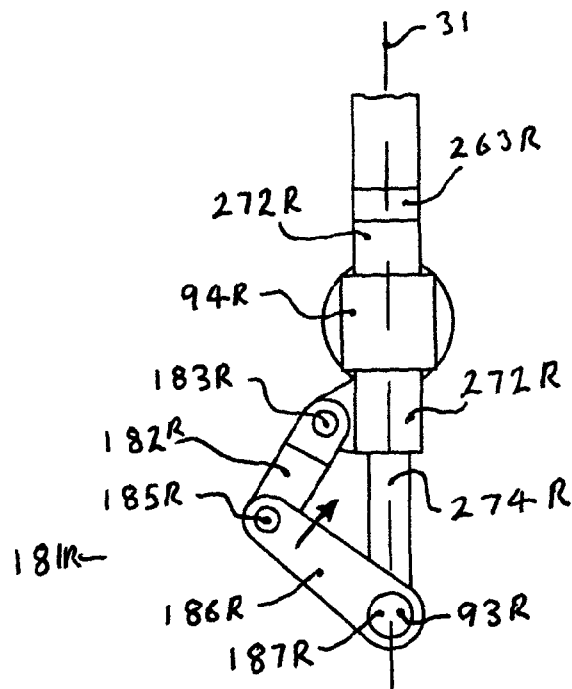


Figure 52

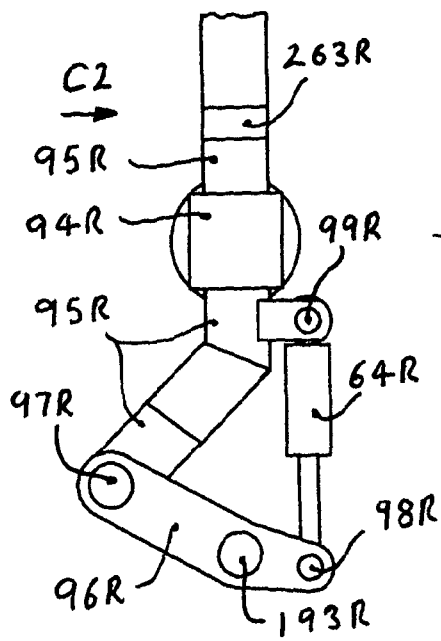


Figure 53

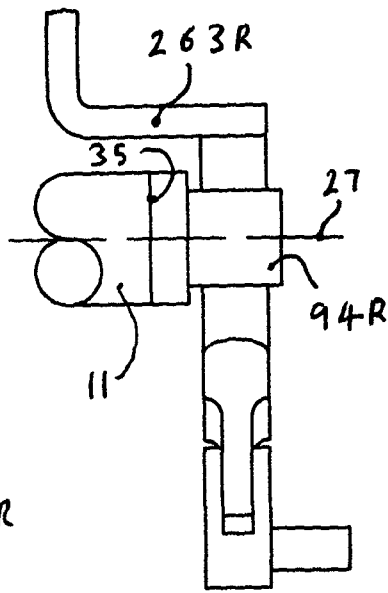


Figure 54

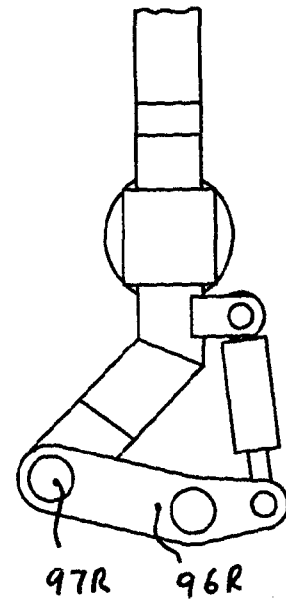


Figure 55

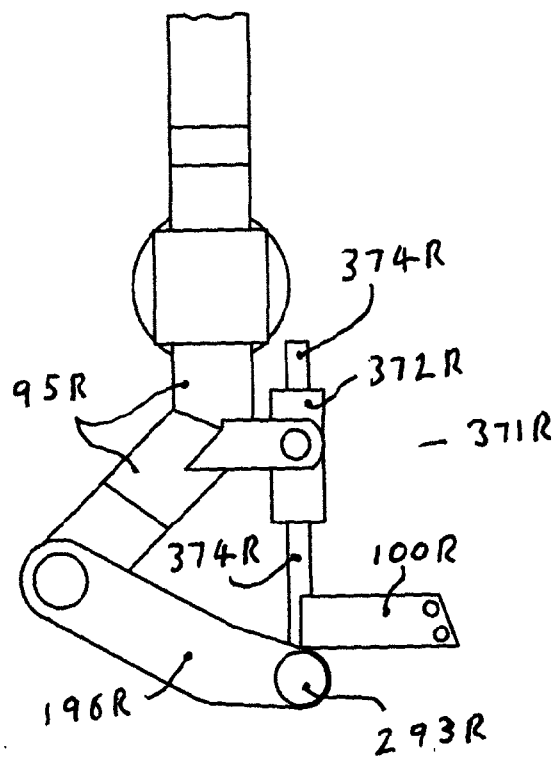


Figure 56

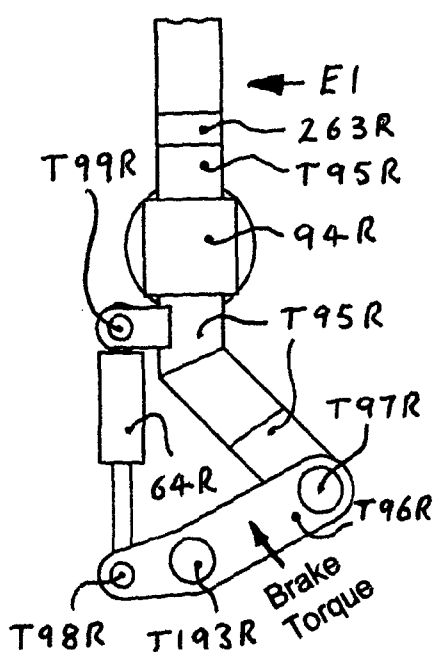


Figure 57

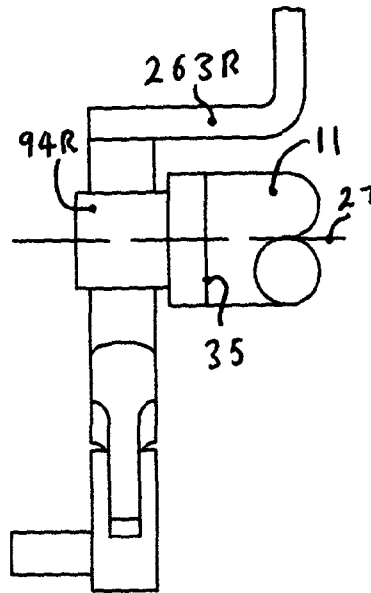


Figure 58

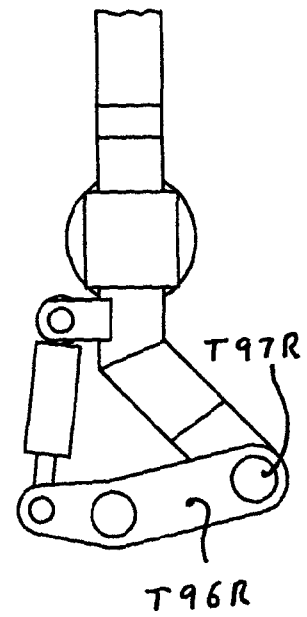


Figure 59

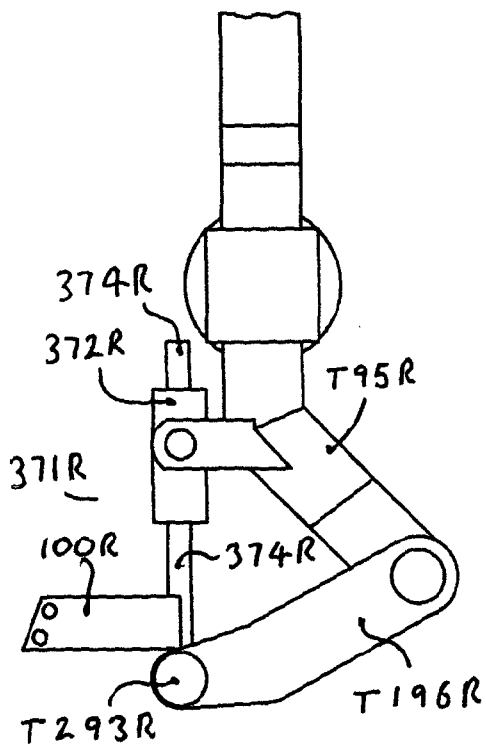


Figure 60

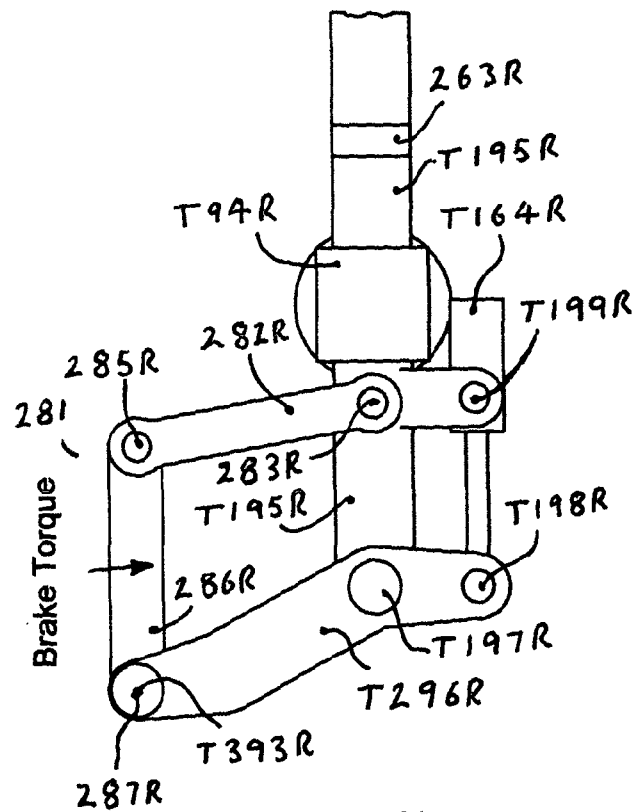


Figure 61

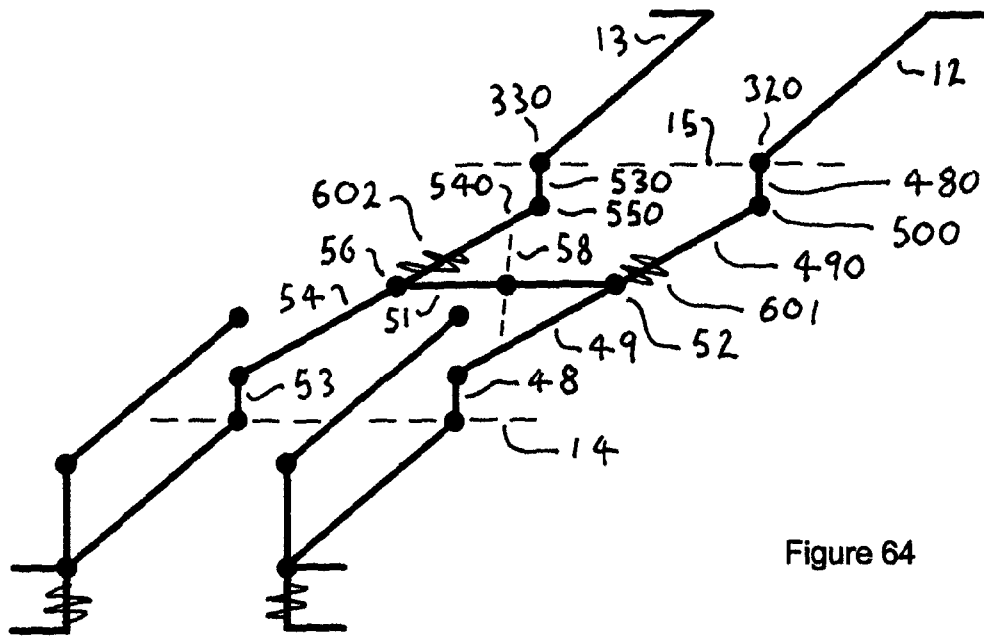


Figure 64

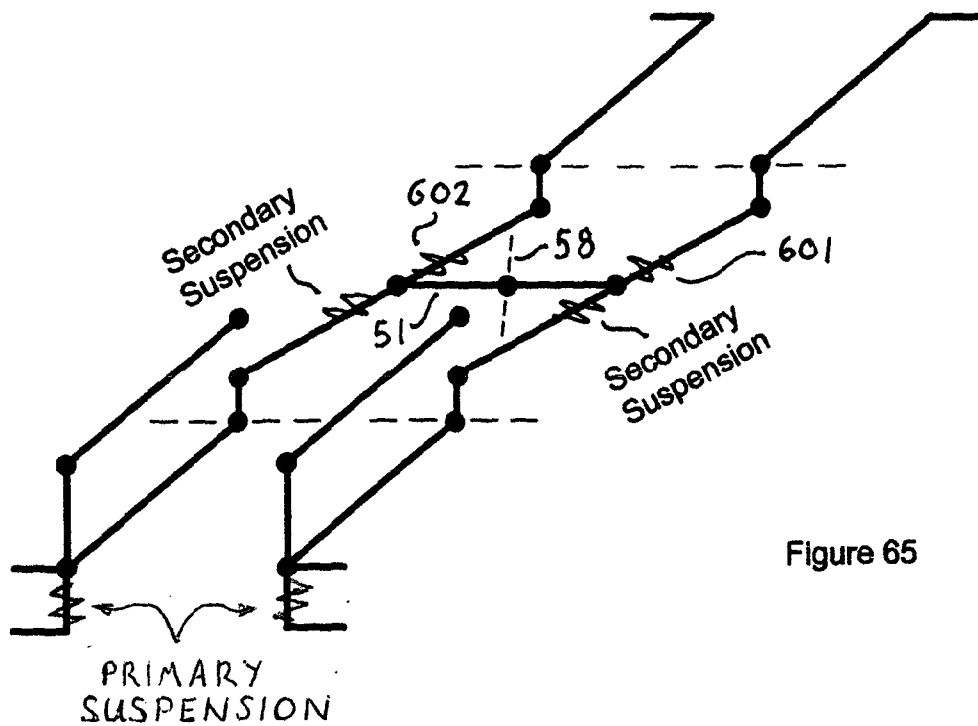


Figure 65

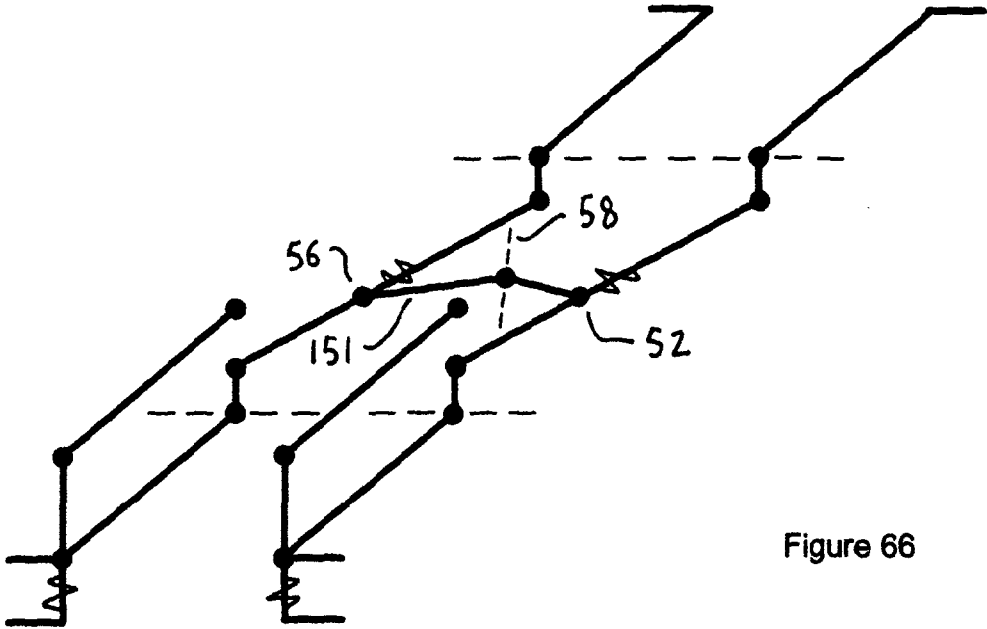


Figure 66