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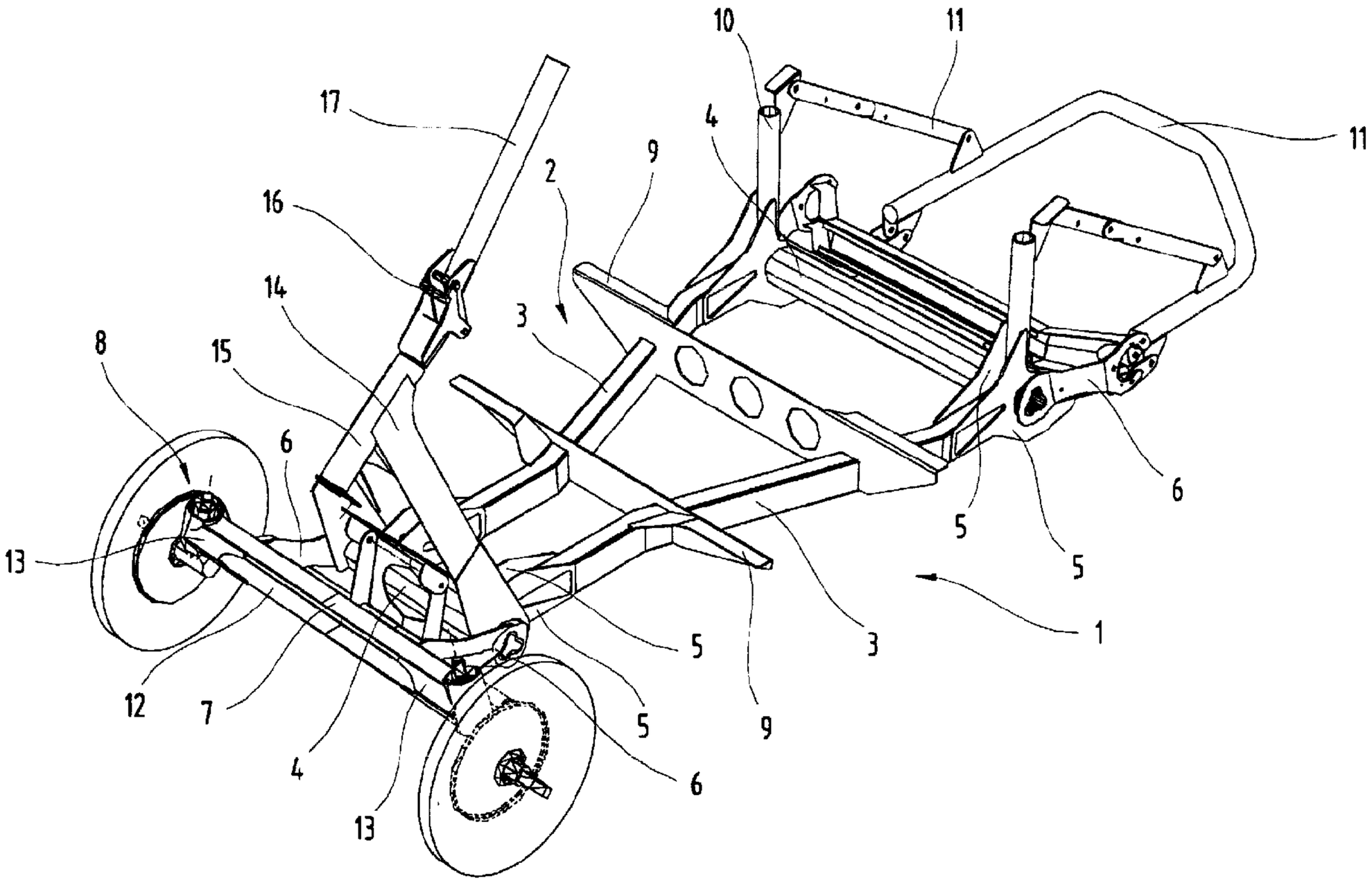
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(54) Titre : CHASSIS DE VEHICULE
(54) Title: VEHICLE UNDERCARRIAGE



(57) Abrégé/Abstract:
The invention relates to a vehicle undercarriage (1) or a chassis for a vehicle, especially for a four-wheeled, driven vehicle for transporting at least one person. At least one main frame (2) is provided, said main frame (2) being configured as an arrangement of closed hollow profiles. The main frame (2) also has diametrically opposed torsion springs (4) whose axis of revolution runs crosswise in relation to the direction of travel. The torsion springs (4) are connected to side hollow profiles (3) of the main frame (2) with gusset plates (5). A flexurally resistant connecting pin (7) is provided on the torsion springs (4) with an oscillating lever (6), the connecting pin having a mounting (8) at its ends for the wheels.

Abstract

The invention relates to a vehicle undercarriage (1) or chassis for a vehicle, especially for a four-wheeled, driven vehicle for transporting at least one person. At least one main frame (2) is provided, said main frame (2) being configured as an arrangement of closed hollow profiles. The main frame (2) also has diametrically opposed torsion springs (4), whose axis of revolution runs crosswise in relation to the direction of travel. The torsion springs (4) are connected to side hollow profiles (3) of the main frame (2) with gusset plates (5). A flexurally resistant connecting pin (7) is provided on the torsion springs (4) with an oscillating lever (6), the connecting pin having a mounting (8) at its ends for the wheels.

Use Fig. 1 with the summary.

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The invention relates to a vehicle undercarriage or chassis for a vehicle as described in Claim 1.

Vehicle carriages for multi-track motor vehicles, whereby the wheel suspension is provided by torsion springs, are well known and widespread. Thereby, a distinction must basically be made between two different embodiments, namely the crank axle and the cam-lever shaft.

In the crank axle, the wheel axle is arranged on an elastically rotatable arm, whereby the arm changes its angle in relation to the axle suspension with changing wheel load. The return torque usually increases with the change in angle, thus providing a rebound force on the wheel axle that acts in the opposite direction of the forces of inertia. The only degree of freedom for movement in such an arrangement is the elastically limited rotation around the torsion spring element, so that with sufficiently stiff design of the arrangement a defined wheel guidance is obtained in the other directions of movement. The torsion spring element can be designed as a metal rod in the axis direction of the free rotational motion, or as an elastomer spring element that is deformed primarily by shearing. Since each wheel is guided by a separate crank guide, independent capability of movement of the wheels is achieved, i.e. independent wheel suspension. The crank axis represented here is used preferably in load-bearing vehicles with relatively stiff suspension. The torsion spring may have high torsion stiffness, so that it has sufficient stiffness as a guiding element for the additional degrees of freedom of wheel movement. The disadvantage of this embodiment is the fact that in passenger vehicles the suspension must be softer for reasons of comfort. However, with a softer suspension the torsion spring can no longer manage the wheel guidance satisfactorily. Such inadmissible bump steers and camber changes under changing wheel loads lead to an unstable behaviour of the vehicle.

In the embodiment with a cam-lever shaft, the spring energy is also stored in a component that counters its twist with the relevant torque. It differs from the crank axle in that further directions of movement can partly be restricted by separate elements. For example, embodiments are known in which only longitudinal guidance of the wheel in the direction of movement is achieved by the spring. Lateral guidance and maintenance of the track and camber angle are provided by the split axle of the wheel drive. With this arrangement, stiff wheel

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guidance in all directions except the compression is easier to achieve in terms of construction. However, it is also possible to guide solely the compression vertically to the tyre contact area using the oscillating lever, whilst defining all the other directions of movement with a suitable wheel suspension. The disadvantage of this design is the high construction cost, which also results in a high degree of failure.

The aim of this invention is to create a vehicle undercarriage of the type mentioned at the beginning, which on the one hand avoids the disadvantages described above and on the other hand makes use of the constructional simplicity of the crank axle, whereby the restriction of lacking wheel guidance in the case of soft suspension is avoided.

The aim of this invention is achieved with the features in Claim 1.

The surprising advantage resulting from the characteristics described in Claim 1 lies in the fact that optimal wheel guidance that guarantees stable driving behaviour of the vehicle is achieved with an extremely simple construction. By designing the main frame with hollow profiles joined together in such a way as to be flexure- and torsion-resistant, the superstructure of the vehicle itself is also advantageously torsion-resistant. Moreover, the arrangement of the torsion springs offers the advantage that in addition to a high degree of material utilisation for the elastic element of the suspension it is also possible to design wheel guidance functions using this suspension. The flexurally stiff connecting pin is provided as wheel guidance element. In addition, this flexurally resistant connecting pin offers the advantage that bump steer and camber change of the wheels are linked more or less rigidly by a connecting pin. Therefore, track width, toe angle and camber angle are invariant towards the plane through the contact points of the wheels on the contact surface.

The embodiment according to Claim 2 is also advantageous, since it allows rational manufacturing, possibly even with the use of welding robots.

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In an advantageous embodiment in accordance with Claim 3 and Claim 4, these torsion springs provide a sufficient longitudinal stiffness in the direction of their axis of rotation and on the plane vertical to this axis.

In the embodiment in accordance with Claim 5, it is possible to achieve optimal flexural and torsion stiffness for the main frame.

The further embodiment in accordance with Claim 6 is also advantageous, since a torsion-resistant connection of the hollow profiles is achieved by the special arrangement of the gusset plates on both sides of the profile. The gusset plates are necessary in order to connect at least two profiles with any connecting angle with each other rigidly by means of welding. The profiles may have the same vertical parting sections. Unlike gusset plates in conventional framework constructions, in the profile rods of which traction and pressure forces usually occur, flexural and torsion forces also have to be transmitted in this construction.

The gusset plates should be arranged on both sides of the profiles so that the welding seam is located in the neutral plane of the main flexural stress of each connecting profile. As a result, expansion stress on the welding seam in such a case of stress can be avoided. This fact has an advantageous effect on the stability of the welding seam.

Due to the bilateral arrangement of the gusset plates on the connecting profiles, however, stress with a general direction, such as e.g. flexion, torsion, traction or pressure, can be transferred without an unfavourable strain on the welding seams. The type and level of the collective stress has a structural impact on the design of the gusset plate. The construction principle itself remains unaffected.

The advantage of this type of gusset connection consists in the fact that separate profile sections, which are easy to manufacture, can be used even for complex framework geometries under any stresses vertical to the profile axis.

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The embodiment in accordance with Claim 7 is also advantageous, since profiles of this type unite a high flexural resistance with low torsion resistance.

With the embodiment in accordance with Claim 8 and further embodiment in accordance with Claim 9, the torsion resistance of the connecting pin can be varied in accordance with the length of reinforcement, and it can be calculated in advance. Since the ratio between flexural stiffness and torsion resistance of a profile rod is variable in many areas with the design of the profile cross-section, the wall thickness, dimensions and/or profile design, it is possible to link the individual compressions of an axle with each other. By increasing the torsion resistance of the connecting pin accordingly, there is stronger coupling of the compression, so that the rolling angle of the vehicle, i.e. rotational movement around the longitudinal axis, can be prevented specifically. This is imperative especially in superstructures that create an elevated vehicle centre of gravity.

An embodiment in accordance with Claims 10, 11 and 12 is also advantageous. A variable compression invariably causes torsional flexing of the connecting pin in relation to the vehicle superstructure. In this case, twisting is forced upon the oscillating levers, which are connected rigidly to the connecting pin on the one hand, and can exclusively perform a rotational movement crosswise to the direction of travel. Therefore, the oscillating levers should advantageously be designed so as to have low torsion resistance along the longitudinal axis, in order to be able to achieve the necessary axle compression. This may be achieved with a thin-walled, flat design of the oscillating levers. Lateral forces such as those acting on the vehicle with every change in direction would cause an inadmissible deformation of the oscillating levers crosswise to the direction of travel.

Thereby, an embodiment in accordance with Claim 13 proves advantageous, since with the arrangement along this line the point of rotation of the oscillating lever is not additionally stressed by torque due to braking forces during a braking manoeuvre using the front brakes, as a result of which there is no spring feedback through the braking forces. Of course this arrangement is suitable both for front and rear axles.

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An embodiment in accordance with Claim 14 is also advantageous, since it guarantees rational manufacturing of components, whereby further processing is not required.

The embodiment in accordance with Claim 15 is particularly advantageous, whereby steering of the vehicle is independent of compression as a result of the construction. Normally, the steering rod is connected to the steering arms of the steering stubs via a ball joint. This allows some degree of swivel movement of the steering rod around it longitudinal axis. A relay lever arranged so as to be capable of swivelling around an axis parallel to the steering rod can balance out spacing changes together with the steering lever. A steering lever is mounted rigidly on the steering rod with a hinge at the upper end, the rotational axis of which is parallel to the steering rod. A relay lever is pivoted to the rotational axis of the steering lever, thus being capable of swivelling around an axis parallel to the steering rod. The relay lever is connected to the eccentric shaft so as to be articulated. This construction transfers movements of the eccentric shaft parallel to the steering rod directly to the joints of the steering stubs, but it can balance out the changeable spacing of the eccentric shaft vertical to the steering rod.

In the following, the invention is explained in more detail based on the embodiments illustrated in the figures.

The figures show the following:

Fig. 1 an oblique view,

Fig. 2 a side view,

Fig. 3 a top view of the vehicle undercarriage in accordance with the invention, and

Fig. 4 part of the steering construction.

By way of introduction, it is noted that in the described embodiment the same parts are allocated the same reference numbers and the same component names, whereby the disclosures

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contained throughout the description can be applied by analogy to the same parts with the same reference numbers or same component names. Furthermore, position details given in the description, e.g. top, bottom, side, etc., relate to the figures being described and illustrated at the time and with a change of position should be transferred accordingly to the new position. Furthermore, individual features of the illustrated embodiment may themselves represent independent solutions in accordance with the invention.

Fig. 1 shows an undercarriage 1 or chassis for a four-wheel motorised vehicle for transportation of at least one person. In its basic construction, this vehicle undercarriage 1 consists of one main frame 2, which is made up of a more or less ring-shaped arrangement of closed hollow profiles. This main frame 2 is made preferably of two lateral hollow profiles 3 and two diametrally opposed torsion springs 4, whereby the rotational axes of the torsion springs 4 are arranged crosswise to the vehicle's direction of travel. With this embodiment of the main frame 2 with hollow profiles, a torsion-resistant superstructure of the vehicle is also possible. This is made possible by the fact that each lateral hollow profile 3 of the main frame 2 is connected to the torsion springs 4 so as to be flexion- and torsion-resistant. This flexion and torsion resistance is achieved by providing the connection preferably with plane gusset plates 5. The gusset plates 5 are mounted bilaterally on the lateral hollow profiles 3, whereby the welding joint is located on the neutral plane of the main flexural stress of each connecting profile, in this case the lateral hollow profiles 3. Extension stresses on the welding joint can be avoided in such a case of stress, thus providing advantages with regard to the strength of the welding joint. In addition, a flexurally stiff connecting pin 7 is provided at the torsion springs 4 via the oscillating lever 6. The connecting pin 7 has a mounting 8 for the wheels of the vehicles at its ends.

The lateral hollow profiles have fixtures 9 for the body of the vehicle arranged crosswise in relation to the direction of travel. Moreover, vertical supports 10 for this body can be arranged on the gusset plates 5 for the rear torsion spring 4. Additional fixtures or bearing frames 11 can, even if they project beyond the rear torsion spring 4, could be mounted on the main frame 2 and/or the supports 10 using the gusset plates 5.

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The connecting pin 7 is designed as a torsion-elastic open profile 12 in the area between the oscillating levers 6. In the area between the oscillating lever 6 and the mounting 7, a flexure- and torsion-resistant reinforcement 13 is provided, so that the type of reinforcement 13 can create a hollow profile. By choosing the length of the reinforcement 13 accordingly, the torsion resistance of the connecting pin 7 can be varied. Since the relation between flexural stiffness and torsion resistance of a profile rod, in this case of connecting pin 7, can be changed by the design of the profile cross-section in many areas, it is possible to link the individual compressions with each other on one axle. High torsion resistance leads to stronger coupling of the compression, thus actively preventing rolling of the vehicle. In addition, this construction also permits controlled torsion of the connecting pin 7.

The control panel 14 for steering the vehicle is mounted on the torsion spring 4, the front torsion spring facing in the direction of travel. On this control panel 14, which can be designed in two parts and each part of which is mounted close to the oscillating lever 6, a camshaft tube 15 with an internal steering tube is mounted. A handlebar 17 is connected to this steering tube 15 via a connecting element 16.

The general advantage of this type of construction is the fact that all the connections between the individual components can be made exclusively using welding joints. Moreover, it is technically optimal and rational in terms of manufacturing that all the hollow profiles, like for example the lateral hollow profile 3, the torsion spring 4 or the connecting pin 7 have vertical parting sections.

In accordance with Fig. 2, the torsion springs 4 are attached to the lateral hollow profiles 3 via the gusset plates 5. Moreover, the oscillating levers 6 are provided on the torsion springs 4 for both the front and the rear connecting pins 7. In addition, the bearing frame 11 is provided on the support 10 or on the gusset plate for the rear rotational axis 4, for example for the cage of the vehicle. Equally, the control panel 14 with the steering tube 15 and handlebar 17 is mounted on the front torsion spring 4.

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The torsion spring 4 consists of two concentrically arranged hollow profiles, whereby the outer hollow profile is a torsion spring tube 18 and the inner hollow profile is a torsion spring axis 19. At least one elastomer element 20 is arranged between the torsion spring tube 18 and the torsion spring axis. With this arrangement of the torsion springs 4, sufficient longitudinal stiffness is achieved in the direction of its rotational axis and the vertical plane in relation to this axis.

The oscillating levers 6 are rigidly attached to the connecting pin 7. in addition, the oscillating levers are designed so as to have low torsion resistance along their longitudinal axis, and they have a thin-walled, flat shape. As already mentioned, varying compression invariably causes torsional flexing of the connecting pin 7 in relation to the vehicle superstructure. In this case, twisting is forced upon the oscillating levers 6, which are connected rigidly to the connecting pin 7, and can exclusively perform a rotational movement crosswise to the direction of travel. Therefore, the oscillating levers 6 are designed so as to have low torsion resistance along the longitudinal axis, in order to be able to achieve the necessary axle compression.

The rotation point 21 of the oscillating lever 6 lies on the front connecting pin 7 (facing in the direction of travel) on the – dotted -- line of efficacy between the overall centre of gravity 22 of the vehicle and the contact point 23 of the tyres 24 on the wheels. Thus, the rotation point 21 of the oscillating lever 6 is not additionally subject to torque due to braking forces during a braking manoeuvre with the brakes, and there is no spring feedback due to the braking forces.

In accordance with Fig. 3 the main frame 2, consisting of the lateral hollow profiles 3 and the torsion springs 4 has fittings 9 for the vehicle body. The bearing frame 11 may project beyond the rear torsion spring 4 facing in the direction of travel.

The connecting pin 7 is also mounted on the rear torsion spring 4 via the oscillating lever 6. The connecting pin 7 can be offset, for example for mounting of the drive.

The front connecting pin 7 with the mounting 8 on both sides, in particular in this case with a steering stub mounting 25 and the steering construction, which is designed as a cassette

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construction, is mounted on the front tension spring 4. The steering construction consists basically of a steering rod 26, which is connected via a ball joint each to the steering levers 27 that can be arranged on both sides. The steering levers in turn are connected with steering stubs 28, which form a unit with the connecting pin 7 via the steering stub mounting 25. Since the steering rod 26 is connected to the steering levers 27 of the steering stub 28 via a ball joint 29, a certain scope of swivel movement of the steering rod 26 around its longitudinal axis is possible.

On the steering stubs 28, wheel hubs 30 are provided for the wheel bearings, whereby these wheel hubs 30 are fitted with braking discs 31.

Fig. 4 shows the steering construction in detail. A steering lever 32 is mounted rigidly on the steering rod 26, whereby a rotating joint 33, the rotational axis 34 of which is parallel to the steering rod 26, is provided at the upper end. On the rotating joint 33, a relay lever 35 is pivoted to the rotational axis 34 of the steering lever 32, thus being capable of swivelling around an axis parallel to the steering rod 26. The relay lever 35 is connected via a crank joint 36 to an eccentric shaft 37 that is attached to the steering tube 15.

This steering construction transmits movements of the eccentric shaft 37 via the steering rod 26 and parallel to it directly to the joints for moving the wheels. In addition, this steering construction can balance out the varying distance from the eccentric shaft 37 vertical to the steering rod 26.

In conclusion, it is noted that for a better understanding of the solution in accordance with the invention, the components are illustrated partly untrue to scale and/or are enlarged or illustrated schematically in the embodiments described above. Moreover, individual parts of the combination of features from the embodiment illustrated and described can represent independent inventive solutions or solutions according to the invention in themselves when combined with other individual features.

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In particular, the individual embodiments illustrated in Figures 1 to 4 can represent individual solutions in accordance with the invention. The relevant aims and solutions in accordance with the invention can be found in the detailed descriptions of these Figures.

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List of References

1 Vehicle undercarriage
2 Main frame
3 Lateral hollow profile
4 Torsion spring
5 Gusset plate
6 Oscillating lever
7 Connecting pin
8 Mounting
9 Fixture
10 Support
11 Bearing frame
12 Profile
13 Reinforcement
14 Control panel
15 Steering tube
16 Connecting element
17 Handlebar
18 Torsion spring tube
19 Torsion spring axis
20 Elastomer element
21 Rotation point
22 Overall centre of gravity
23 Contact point
24 Tyre
25 Steering stub mounting
26 Steering rod
27 Steering lever
28 Steering stub
29 Ball joint
30 Wheel hub
31 Brake disk
32 Steering rod lever
33 Rotating joint
34 Rotational axis
35 Relay lever
36 Crank joint
37 Eccentric shaft

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Patent Claims

1. Vehicle undercarriage or chassis for a four-wheeled, driven vehicle for transporting at least one person, whereby at least one main frame (2) is provided as an arrangement of closed hollow profiles, characterised by the fact that the main frame (2) consists of two lateral hollow profiles (3) and two torsion springs (4), and is configured as a roughly ring-shaped, flexure- and torsion-resistant arrangement, whereby the torsion springs (4) are diametrally opposed with an axis of revolution that runs crosswise in relation to the direction of travel, and the torsion springs (4) have outer tubes (18), whereby the outer tubes (18) are connected to the lateral hollow profiles (3) of the main frame (2) with gusset plates (5), and that a flexurally resistant connecting pin (7) is provided on the torsion springs (4) with an oscillating lever (6), the connecting pin (7) having a mounting (8) at its ends for the wheels, whereby the connecting pin (7) is designed as a profile with low torsion resistance in the area between the oscillating levers (6), also designed with low torsion-resistance along the longitudinal axis.

2. Vehicle undercarriage in accordance with Claim 1, characterised by the fact that all the connections between the individual parts are made exclusively with welding joints.

3. Vehicle undercarriage in accordance with Claim 1 or 2, characterised by the fact that the torsion spring (4) consists of two concentrically arranged hollow profiles, whereby at least one elastomer element (20) is provided between the hollow profiles, preferably in the area of mounting of the oscillating lever (6).

4. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the torsion spring (4) consists of an outer tube (18), an elastomer spring element (20) and a torsion spring axis (19).

5. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the connection between the lateral hollow profiles (3) and other connecting

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profiles is achieved by flat gusset plates (5) arranged on both sides of the lateral hollow profile (3).

6. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the connecting pin (7) is configured as an open profile in the area between the oscillating levers (6).

7. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the connecting pin (7) has a flexure- and torsion-resistant reinforcement (13) in the area between the oscillating lever (6) and the mounting (8), in particular the steering stub mounting (25).

8. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the connecting pin (7) is designed as a hollow profile in the area between the oscillating lever (6) and the mounting (8), in particular the steering stub mounting (25).

9. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the oscillating lever (6) is connected rigidly to the connecting pin (7).

10. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the oscillating lever (6) has a thin-walled, flat shape.

11. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that the rotation axis (21) of the oscillating lever (6) lies on the front connecting pin (7) facing in the direction of travel, on the line of efficacy between the overall centre of gravity (22) of the vehicle and the contact point (23) of the wheels.

12. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that all the hollow profiles have vertical parting sections.

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13. Vehicle undercarriage in accordance with one or more of the above Claims, characterised by the fact that a steering linkage is provided, consisting of a steering rod lever (32) connected rigidly to a steering rod (26), as well as a relay lever (35) capable of swivelling on the steering rod lever (32) around a rotational axis (34) parallel to the steering rod (26) and which is connected to an eccentric shaft (37) by a crank joint (36).

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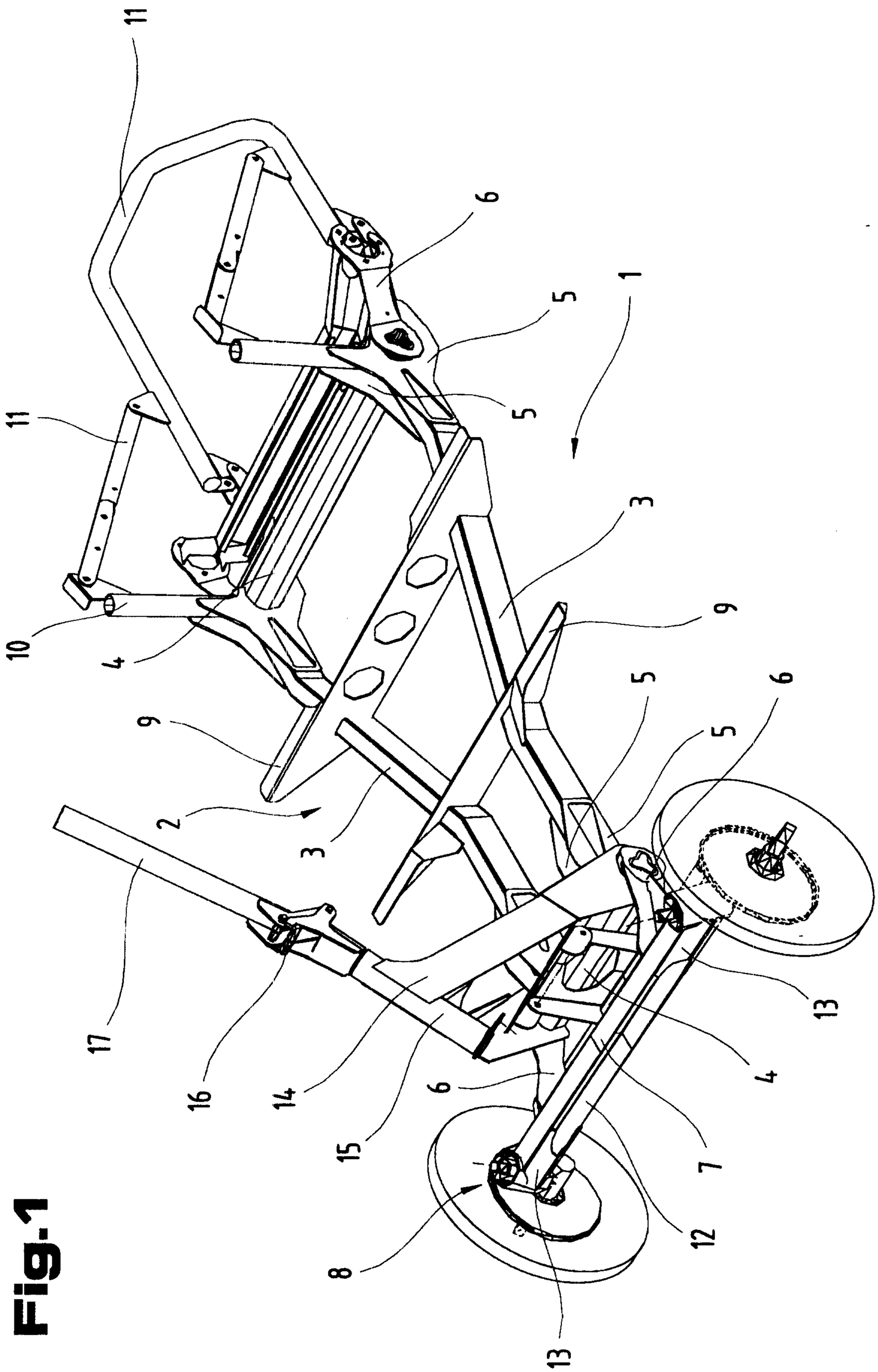


Fig.1

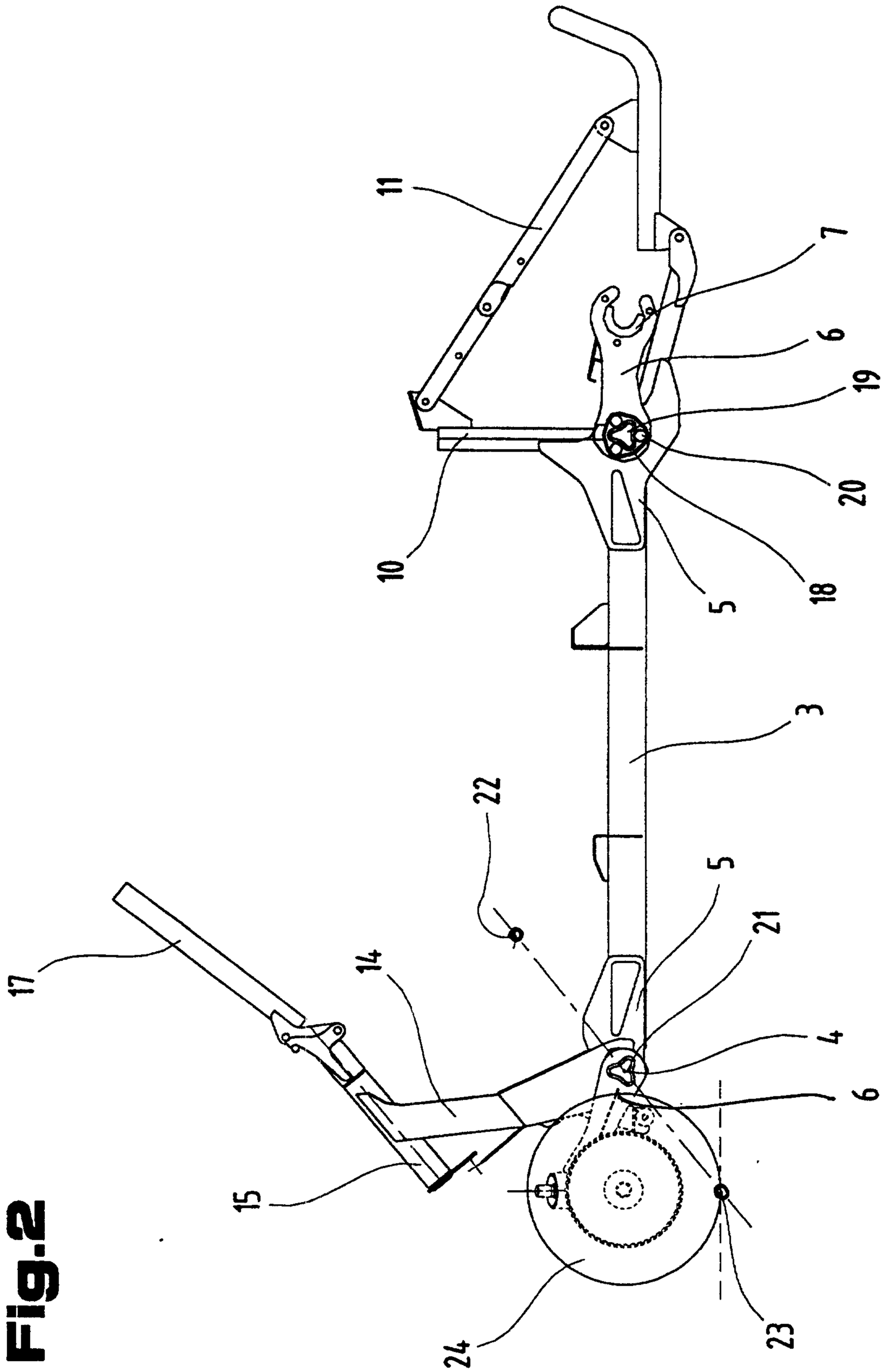


Fig.2

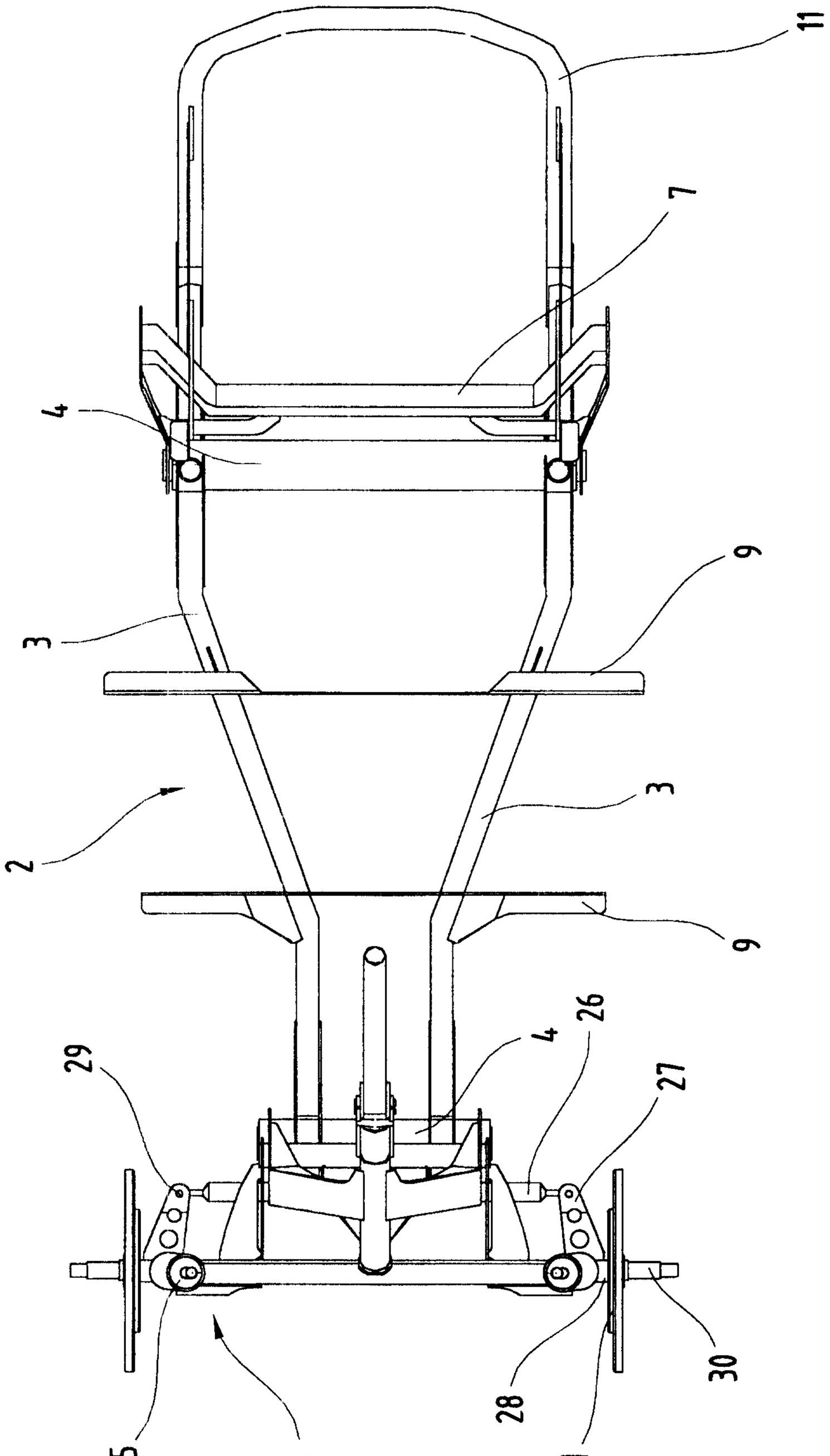
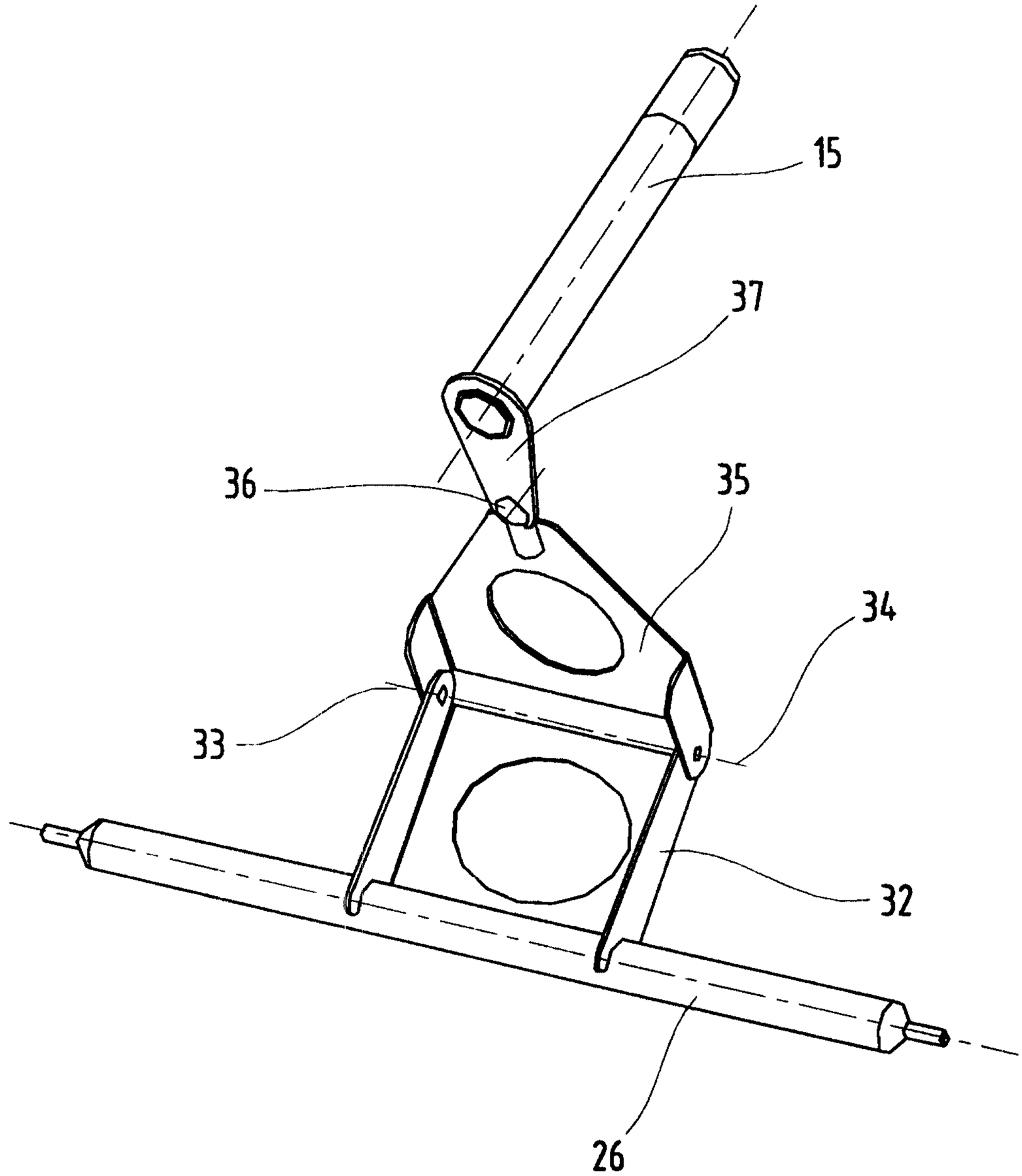
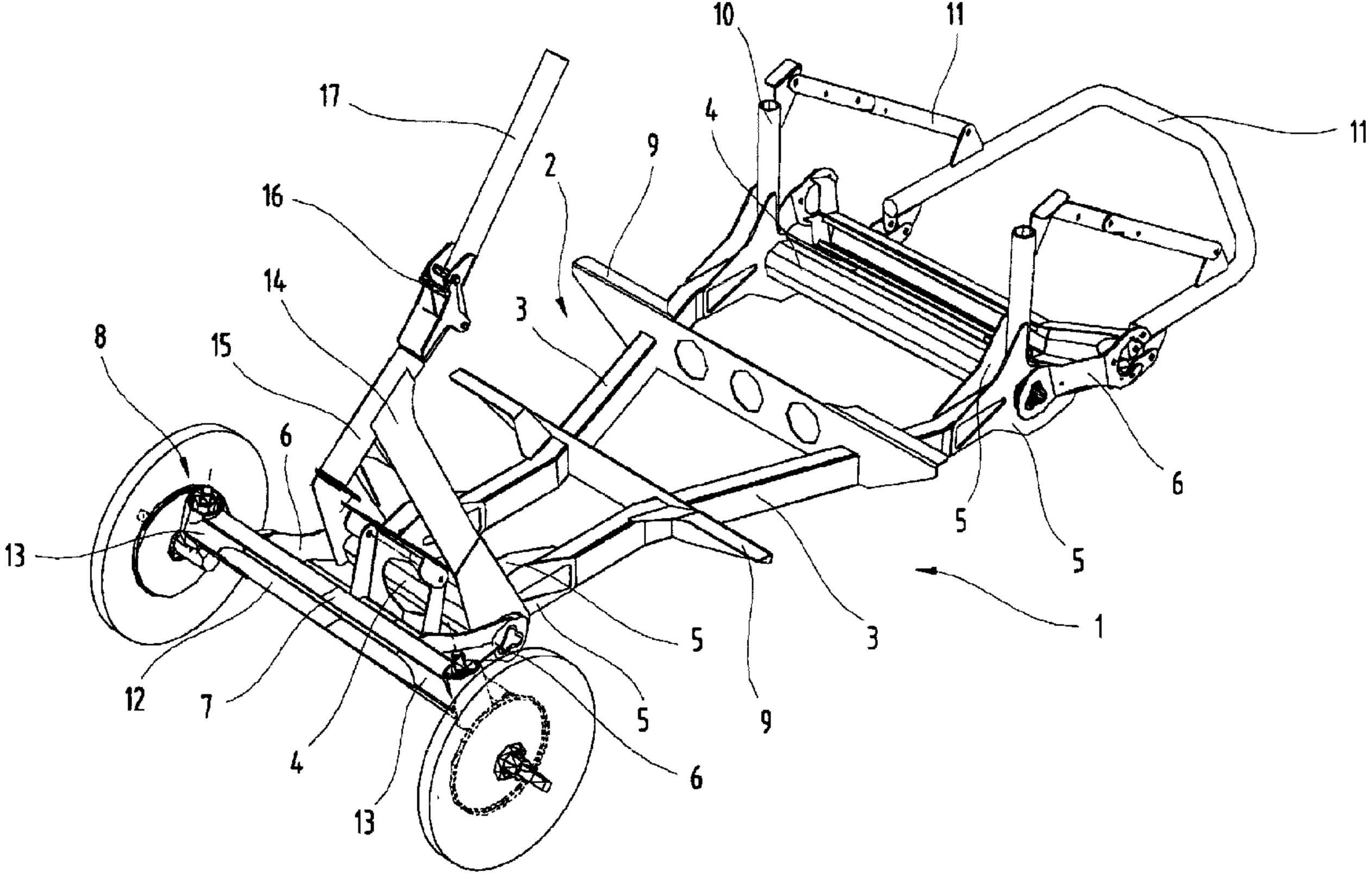


Fig.3

Fig.4




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