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LOAD CUSHION

ABSTRACT

A load cushion for a vehicle suspension system, the load cushion comprising: an end plate; an elastomeric cushion portion extending from the end plate to a top of the cushion portion; wherein a cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and curvilinear; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion.

LOAD CUSHION

CROSS-REFERENCE TO RELATED APPLICATION

This application claims priority to U.S. Patent Application No. 14/500,106 filed September 29, 2014, and is a divisional application of Australian Application No 2014407484 filed on 15 October 2014, which are hereby incorporated by reference in its entirety.

BACKGROUND

The present invention generally relates to vehicle suspensions. More particularly, the present invention relates to vehicle suspensions using bolster springs. Examples of vehicle suspensions having bolster springs are disclosed in U.S. Patent No. 6,585,286 entitled "Vehicle Suspension" that issued on July 1, 2003, herein incorporated by reference in its entirety. The present application includes improvements and advancements over the vehicle suspensions disclosed in the '286 patent noted above.

SUMMARY

In accordance with an aspect, there is provided a load cushion for a vehicle suspension system, the load cushion comprising: an end plate; an elastomeric cushion portion extending from the end plate to a top of the cushion portion; wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion; wherein a vertical cross section of the cushion portion from a left side to a right side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the left side and the right side is positioned at a midpoint between the end plate and the top of the cushion portion; and wherein a horizontal cross section of the elastomeric cushion portion taken at the midpoint is square with rounded comers.

In accordance with an aspect, there is provided a load cushion for a vehicle suspension system, the load cushion comprising: an end plate; an elastomeric cushion portion extending from the end plate to a top of the cushion portion; wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges;

wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion; wherein the vertical cross-section of the elastomeric cushion has a negative Gaussian curvature; and wherein when the load cushion undergoes compression, the vertical cross-section changes through a zero Gaussian curvature.

In accordance with an aspect, there is provided a load cushion for a vehicle suspension system, the load cushion comprising: an end plate; an elastomeric cushion portion extending from the end plate to a top of the cushion portion; wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion; wherein the vertical cross-section of the cushion portion of the elastomeric cushion has a negative Gaussian curvature; and wherein when the load cushion undergoes 50% compression, the vertical cross-section changes to a zero or slightly positive Gaussian curvature.

A vehicle suspension is provided having a frame attachment portion attached to a saddle, first and second bolster springs mounted to spring mounts on an outboard side of the saddle and mounted on walls of a spring mount on an outboard side of an equalizing beam, and third and fourth bolster springs mounted to walls of a spring mount on an inboard side of the saddle and mounted to spring mounts on an inboard side of the equalizing beam. Upwardly extending flanges on the bottom of the first and second bolster springs are mounted to each other with common fasteners, and wherein upwardly extending flanges on the bottom of the third and fourth bolster springs are mounted to each other with common fasteners. The mechanical joints provide retention integrity allowing for the use of fewer and smaller fasteners resulting in a lighter, more optimized design. In addition, an apex angle between the bolster springs has been reduced allowing them to operate more in shear thereby providing for a decrease in the primary and secondary suspension spring rates, as well as reduced axle translation during braking and acceleration. In addition, the reduced apex angle and direct mounting of the bolster springs provides for additional clearance for vehicle tires.

Disclosed herein is a suspension for supporting a longitudinally extending vehicle frame rail above an axle is provided including a frame attachment portion adapted for connection to a vehicle frame rail, a saddle having a top portion attached to the frame attachment portion, a first bolster spring mount extending from an outboard side of a lower portion of the saddle, a second

bolster spring mount extending from the outboard side of the lower portion of the saddle, an equalizing beam having a first end adapted for attachment to a first axle and a second end adapted for attachment to a second axle, a third bolster spring mount extending from an outboard side of the equalizing beam, a first bolster spring having a top attached to the first bolster spring mount and a bottom attached to a first wall of the third bolster spring mount, a second bolster spring having a top attached to the second bolster spring mount and a bottom attached to a second wall of the third bolster spring mount, a fourth bolster spring mount extending from an inboard side of the lower portion of the saddle, a fifth bolster spring mount extending from the inboard side of the lower portion of the saddle, a sixth bolster spring mount extending from an inboard side of the equalizing beam, a third bolster spring having a top attached to the fourth bolster spring mount and a bottom attached to a first wall of the sixth bolster spring mount, a fourth bolster spring having a top attached to the fifth bolster spring mount and a bottom attached to a second wall of the sixth bolster spring mount, a first apex angle between the bottom of the first bolster spring and the bottom of the second bolster spring that is between 30-45 degrees, and a second apex angle between the bottom of the third bolster spring and the bottom of the fourth bolster spring that is between 30-45 degrees.

Disclosed herein is a suspension for supporting a longitudinally extending vehicle frame rail above an axle is provided including a frame attachment portion adapted for connection to a vehicle frame rail, a saddle having a top portion attached to the frame attachment portion, a first bolster spring mount extending from an outboard side of a lower portion of the saddle, a second bolster spring mount extending from the outboard side of the lower portion of the saddle, an equalizing beam having a first end adapted for attachment to a first axle and a second end adapted for attachment to a second axle, a third bolster spring mount extending from an outboard side of the equalizing beam, a first bolster spring having a top attached to the first bolster spring mount and a bottom attached to a first wall of the third bolster spring mount, a second bolster spring having a top attached to the second bolster spring mount and a bottom attached to a second wall of the third bolster spring mount, a fourth bolster spring mount extending from an inboard side of the lower portion of the saddle, a fifth bolster spring mount extending from the inboard side of the lower portion of the saddle, a sixth bolster spring mount extending from an inboard side of the equalizing beam, a third bolster spring having a top attached to the fourth bolster spring mount and a bottom attached to a first wall of the sixth bolster spring mount, a fourth bolster spring having a top attached to the fifth bolster spring mount and a bottom attached to a second wall of the sixth bolster spring mount, wherein a first

flange upwardly extends from the bottom of the first bolster spring and a second flange upwardly extends from the bottom of the second bolster spring, wherein the first flange of the first bolster spring is mounted to the second flange of the second bolster spring with a common fastener, wherein a third flange upwardly extends from the bottom of the third bolster spring and a fourth flange upwardly extends from the bottom of the fourth bolster spring, wherein the third flange of the third bolster spring is mounted to the fourth flange of the fourth bolster spring with a common fastener.

Disclosed herein is a bolster spring for a vehicle suspension is provided including a base plate, a top plate, elastomeric material positioned between the base plate and the top plate, a first flange having a bottom mounting surface upwardly extending from a first end of the base plate at an angle $1/2\alpha$, and one or more mounting holes positioned in the flange adapted for attachment to an upwardly extending flange on a second bolster spring.

Disclosed herein is a load cushion for a suspension system is provided including a base plate, an elastomeric cushion portion extending from the base plate to a top of the cushion portion, wherein a cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and curvilinear; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between a top of the base plate and the top of the cushion portion.

BRIEF DESCRIPTION OF THE DRAWINGS

Exemplary embodiments of the invention are described herein with reference to the drawings provided by way of example only, wherein like parts are designated by like reference numerals, and wherein:

Figure 1A is a perspective view of the outboard side of vehicle suspension 50, according to an example embodiment;

Figure 1B is a perspective view of vehicle suspension 50 shown in Figure 1 and oppositely disposed vehicle suspension 50;

Figure 2 is a front view of the outboard side of vehicle suspension 50 shown in Figures 1A and 1B;

Figure 3 is a perspective view of the inboard side of vehicle suspension 50 shown in Figures 1A, 1B, and 2;

Figure 4 is a rear view of the inboard side of vehicle suspension 50 shown in Figures 2 and 3;

Figure 5 is a bottom view of vehicle suspension 50 shown in Figures 1A-4;

Figure 6 is a top view of vehicle suspension 50 shown in Figures 1A-5;

Figure 7 is a right side view of vehicle suspension 50 shown in Figures 1A-6;

Figure 8 is a left side view of vehicle suspension 50 shown in Figures 1A-7;

Figure 9 is a close up front view of vehicle suspension 50 showing bolster springs 70 and 72, and load cushion 9;

Figure 10 is a close up front perspective view of vehicle suspension 50 shown in Figure 9;

Figure 11 is a perspective view of bolster spring 200, according to an example embodiment;

5 Figure 12 is a perspective bottom view of bolster spring 200 shown in Figure 11;

Figure 13 is a left side view of bolster spring 200 shown in Figures 11 and 12;

Figure 14 is a right side view of bolster spring 200 shown in Figures 11-13;

Figure 15 is a top view of bolster spring 200 shown in Figures 11-14;

10 Figure 16A is a perspective top view of load cushion 300, according to an example embodiment;

Figure 16B is a perspective bottom view of load cushion 300 shown in Figure 16A;

Figure 17 is a right side view of load cushion 300 shown in Figures 16A-16B;

Figure 18 is front view load cushion 300 shown in Figures 16A-17;

Figure 19 is a bottom view of load cushion 300 shown in Figures 16A-18;

15 Figure 20 is a top view of load cushion 300 shown in Figures 16A-19;

Figure 21A is a cross-sectional, perspective view of the inboard side of vehicle suspension 50, taken along line 21A-21A in Figure 4;

Figure 21B is a cross-sectional, perspective view of the outboard of vehicle suspension 50, taken along line 21B-21B in Figure 2;

20 Figure 22A is a cross-sectional, perspective view of the inboard side of vehicle suspension 50, taken along line 22A-22A in Figure 4;

Figure 22B is a cross-sectional, perspective view of the outboard of vehicle suspension 50, taken along line 22B-22B in Figure 2;

25 Figure 23A is a cross-sectional, perspective view of the inboard side of vehicle suspension 50, taken along line 23A-23A in Figure 4;

Figure 23B is a cross-sectional, perspective view of the outboard of vehicle suspension 50, taken along line 23B-23B in Figure 2;

Figure 24 is a perspective view of equalizing beam 100 of vehicle suspension 50 shown in Figures 1A-10; according to an example embodiment;

5 Figure 25 is a top view of equalizing beam 100 shown in Figure 24; and

Figure 26 is a close up view showing how bolster springs 70 and 72 may be mounted to each other with a common fastener.

DETAILED DESCRIPTION OF THE INVENTION

Figures 1A-10 provide various views of vehicle suspension 50. Vehicle suspension
10 50 is designed to support longitudinally extending vehicle frame rails (not shown) which can be of various types that are positioned above laterally extending vehicle axles. As will be appreciated by those skilled in the art, components of vehicle suspension 50 are duplicated on each side of the vehicle as shown in Figure 1B. It will also be appreciated that vehicle wheels may be mounted to the ends of the vehicle axles in a known manner. Further, it will
15 be appreciated that the vehicle frame rails may be connected by one or more vehicle frame cross members.

Those skilled in the art will further understand that a suspension, arranged in accordance with the suspension 50 and the components thereof, alternatively may be attached to frame rails of a trailer (for example, a trailer that connects to a semi-tractor). The frame
20 rails of a trailer may comprise frame rails such as those described above or another type of frame rail.

For purposes of this description, unless specifically described otherwise, hereinafter, “vehicle” refers to a vehicle or a trailer. In this way, for example, a vehicle frame refers to a vehicle frame or a trailer frame. Furthermore, for purposes of this description, the left side of
25 a vehicle refers to a side of the vehicle on an observer’s left-hand side when the observer

faces the back of the vehicle, and the right side of the vehicle refers to a side of the vehicle on an observer's right-hand side when the observer faces the back of the vehicle. Furthermore still, for purposes of this description, "outboard" refers to a position further away from a center line, running from the front to the back of a vehicle, relative to "inboard" which refers
5 to a position closer to that same center line.

Figure 1A is a perspective view of an outboard side of vehicle suspension 50 having a frame attachment portion 62 that is adapted for attachment to a vehicle frame or frame rail with a plurality of mounting holes 63. Frame attachment portion 62 includes outer gussets 66 and 68 and central flange 64 that provide additional strength and rigidity to the vehicle
10 suspension 50. Frame attachment portion 62 is attached to saddle 60. Bolster springs 70 and 72 are provided that each have a top attached bolster spring mounts 170 and 172 extending from an outboard side of saddle 60 and a bottom attached to walls of bolster spring mount 107b positioned on equalizing beam 100. Equalizing beam 100 has a beam hub 102 on a first end and a beam hub 104 on a second end. Beam hub 102 includes a bar pin 110 adapted for
15 attachment to a first axle (not shown) and beam hub 104 includes a bar pin 112 adapted for attachment to a second axle (not shown).

A pair of shock absorbers 120 and 122 each have one end mounted to the equalizing beam 100 and another end mounted to saddle 60 on the inboard side of vehicle suspension 50. In some applications, shock absorbers may not be used. A load cushion 90 is mounted to
20 load cushion mount 94 extending from saddle 60 and load cushion 90 is positioned beneath saddle 60 and positioned inwardly from and generally above bolster springs 70 and 72. A first rebound strap 80 is mounted to load cushion mount 94, and a second rebound strap is mounted to load cushion mount 92 (shown in Figure 3). A bracket 191 having U-shaped ends that are used to mount rebound straps 80 may be positioned between the load cushion
25 and the load cushion mounts 92 and 94. In addition, shims of varying thickness may

positioned between the load cushion 90 and bracket 191 to change the ride characteristics of the vehicle suspension 50.

Figure 1B includes a second vehicle suspension 50a that is a mirror image of vehicle suspension 50, and may be positioned on an opposite side of a vehicle frame. Accordingly, Figure 1B provides a perspective view of the inboard side of vehicle suspension 50a. Vehicle suspension 50a includes a frame attachment portion 62a that is adapted for attachment to a vehicle frame or frame rail with a plurality of mounting holes 63a. Frame attachment 62a further includes outer gussets 66a and 68a that along with a central flange provide additional strength and rigidity to the vehicle suspension 50a. Frame attachment portion 62a is attached to saddle 60a. Bolster springs 71a and 73a are provided that each have a top attached to bolster spring spring mounts extending from the inboard side of saddle 60a and a bottom attached to bolster spring mount 107a positioned on equalizing beam 100a. Equalizing beam 100a has a beam hub 102a on a first end and a beam hub 104a on a second end. Beam hub 102a includes a bar pin 110a adapted for attachment to a second axle (not shown) and beam hub 104a includes a bar pin 112a adapted for attachment to a first axle (not shown).

A pair of shock absorbers 120a and 122a each have one end mounted to the inboard side of equalizing beam 100a and another end mounted to the inboard side of saddle 60a. A load cushion is mounted to load cushion mount 92a extending from saddle 60a. A rebound strap 80a is mounted to load cushion mount 92a.

Figure 2 provides a front view of the outboard side of vehicle suspension 50 and Figures 3 and 4 provide views of the inboard side of vehicle suspension 50. In Figure 2, load cushion 90 is shown mounted to load cushion mount 94 extending from saddle 60. Bolster springs 70 and 72 are mounted to bolster springs mounts 170 and 172 outwardly extending from outboard wall 65 of saddle 60, and also to bolster spring mount 107b on the outboard side of the equalizing beam 100. As shown in Figure 3, bolster springs 71 and 73 are

mounted to bolster spring mounts 171 and 173 extending from inboard wall 67 of saddle 60 and to walls of bolster spring mount 107a positioned on the inboard side of the equalizing beam 100. The configuration of bolster springs 70-73 results in a balanced, split bolster spring arrangement where one pair of bolster springs 70 and 72 is positioned on the outboard side of equalizing beam 100 and one pair of bolster springs 71 and 73 is positioned on the inboard side of equalizing beam 100.

As shown in Figure 3, shock absorber 120 has a first end secured to mount 108 positioned on equaling beam 100 and a second end secured to mount 69 positioned on saddle 60, and shock absorber 122 has a first end secured to mount 106 positioned on equalizing beam 100 and a second end secured to mount 13 positioned on saddle 60. In other embodiments, the second ends of shock absorbers 120 and 122 could also be mounted to a vehicle frame or frame rail, or not used at all.

Prior vehicle suspensions employing bolster springs typically provided an acute angle, or apex angle, between the bottoms of the bolster springs of 53 degrees, which has become a de facto industry standard. However, as best shown in Figures 2 and 9, vehicle suspension significantly departs from the de facto apex angle standard of 53 degrees. In particular, an apex angle α is provided that is significantly less than 53 degrees. In the embodiments shown in Figures 1-10, the apex angle α between the bottom of bolster springs 72 and 70 (and the apex angle between bolster springs 71 and 73) is 37 degrees. While an apex angle of 37 degrees is preferred, the apex angle α may range between 34-40 degrees, or from 30-45 degrees, all lower than a standard apex angle of 53 degrees.

By reducing the apex angle α to 37 degrees, a number of important advantages are achieved. For example, the reduced apex angle α allows the springs to be positioned closer together, and thereby taking up less space longitudinally. In turn, a greater clearance between the vehicle tires and the bolster spring arrangement is provided, which may provide greater

tire chain clearance or allow for the use of larger tires. In addition, by reducing the apex angle α , the bolster springs are put more into a shear, rather than compression. As a result, a lower primary vehicle spring rate may be achieved, while at the same time providing for increased longitudinal stiffness. The present configuration of the bolster springs with an apex angle α of 37 degrees has increased the longitudinal stiffness of the suspension resulting in a corresponding decrease in the longitudinal deflection to less than an inch. As a result, the reduced apex angle α has resulted in reduced axle translation along the SAE X-Axis during braking and acceleration.

Reducing the apex angle α between the bolster springs has advantageously resulted in a reduction in the primary suspension spring rate to 1.5-2.0 kN/mm depending upon the elastomer used to create the bolster springs. Furthermore, a secondary spring rate of the vehicle suspension when the load cushion is engaged measured at 1.0g ranges from 2.0-3.5 kN/mm depending upon the elastomers chosen for both the bolster springs and initial gap between the load cushion and its reaction plate. These primary and second vehicle suspension spring rates are orders of magnitude lower than traditional elastomeric suspensions and are on the same order of magnitude as parabolic 6-rod suspensions.

Additionally, as discussed in more detail below with respect to Figure 26, in addition to reducing the apex angle α between the bolster springs 70 and 72, and 71 and 73, vehicle suspension 50 also incorporates a unique bolster spring mounting arrangement wherein an angled flange 230 on the bottom plate 220 of bolster spring 70 is directly mounted to a corresponding angled flange 230 on bottom plate 220 of bolster spring 72 using a pair of common fasteners for retention. Bolster springs 71 and 73 are also directly mounted to each other using a pair of common fasteners in the same manner. As used herein, the term "directly mounted" means that the flanges are mounted together using a common fastener without a portion of the equalizing beam or bolster spring mount positioned therebetween,

although a gasket or spacer, or portion of a spring saddle, could be positioned therebetween and the flanges would still be “directly mounted” to each other.

Directly mounting bolster springs 70 and 72 to each other, and directly mounting bolster springs 71 and 73 to each other using common fasteners provides a number of advantages. In particular, the bolster springs may be able to be positioned even closer together because there is no portion of the equalizing beam or a bolster spring mount extending between the flanges of the bolster springs. Furthermore, using common fasteners allows the positioning of the bolster springs to be closer together than if independent fasteners were used for each bolster spring. The closer positioning of the bolster springs allows even further clearance from the tires, again providing even greater clearance for tire chains or larger tires. The end result of directly mounting the flanges of the bolster springs with common fasteners provides for the use of fewer fasteners, faster assembly, improved clearances to surrounding components (because bolster springs are closer together), as well as the creation of a mechanical joint between the mounted flanges of the bolster springs.

As known to those skilled in the art, a mechanical joint formed between two components improves retention integrity and can permit the use of smaller fasteners compared to typical bolster spring designs. A benefit of smaller fasteners is improved clearances to surrounding packages, a more weight optimized design, and improved serviceability because smaller fasteners require less torque to achieve design load as a percent of proof load. Therefore, smaller fasteners are more easily and likely to be tightened appropriately.

Figure 5 is a bottom view of vehicle suspension 50. From this view, the equalizing beam 100 is shown with beam hub 104 having inboard side 104a on one end with bar pin 112 and with beam hub 102 having inboard side 102a with bar pin 110. A center-plane 100c of equalizing beam 100 is shown offset towards inboard side 104a and inboard side 102a a

distance d from a center-plane of beam hubs 104 and 102. In this embodiment, the center-plane is offset a distance d of 11 millimeters. Providing such an offset on the equalizing beam has the effect of moving the vehicle suspension towards the inboard side of the vehicle frame, thereby advantageously providing additional clearance on the outboard side of the vehicle suspension.

In Figure 5, there is a clear view of bolster spring 70 and bolster spring 72 mounted to opposing walls of bolster spring mount 107b extending from an outboard side the vehicle suspension 50, as well as of bolster spring 71 and bolster spring 73 mounted to opposing walls of bolster spring mount 107a extending from the inboard side of vehicle suspension 50.

Figure 6 shows a top view of vehicle suspension 50. In Figures 5 and 6, shock absorbers 120 and 122 can be seen secured to the inboard side of saddle using shock absorber mounts 106, 108, 13, and 69. In addition, a gap 105 is shown on the surface of beam hubs 104 and 102 as a result of the offset d of center-plane 100c. In Figure 6, load cushion mount 94 is shown extending from an outboard side of saddle 60 and load cushion mount 92 is shown extending from an inboard side of saddle 60. In addition, central flange 64 is shown positioned on top surface 91 of saddle 60 attached to frame attachment portion 62.

Figure 7 is a right side view of vehicle suspension 50 and Figure 8 is a left side view of vehicle suspension 50. Beam hub 102 is shown with bar pin 110 adapted for attachment to a first axle (not shown) and beam hub 104 is shown with bar pin 112 adapted for attachment for a second axle (not shown). Frame attachment portion 62 with gussets 68 and 66 are shown extending above outboard wall 65 and inboard wall 67 of the saddle and load cushion mount 94 is shown extending from the outboard side of vehicle suspension 50. Shock absorber 122 is shown mounted to shock absorber mount 13 and shock absorber 120 is shown mounted to shock absorber mount 69. In addition, a pair of rebound straps 80 are shown extending from inboard and outboards sides of the vehicle suspension 50. Rebound straps 80

serve to prevent bolster springs 70-73 from being overstretched and overstressed when vehicle suspension 50 is placed in hang or rebound, such as when a vehicle is lifted with an outrigger, hits a large pothole, or during a sudden drop when going over a steep drop in the road.

5 Figure 9 is a close up front view of, and Figure 10 is a close up perspective view of, the bolster springs 70 and 72 and load cushion 90 on the outboard side of vehicle suspension 50. Bolster spring 70 is attached to bolster spring mount 170 on saddle 60 using fasteners 270b and 270c, and also attached to bolster spring mount 107a on the equalizing beam 100 using fastener 270a. Similarly, bolster spring 72 is attached to bolster spring mount 172 on
10 saddle 60 using fasteners 272b and 272c, and also attached to bolster spring mount 107a on the equalizing beam 100 using fastener 272a. As illustrated in Figure 26, upwardly extending flange 230 of bolster spring 70 is directly mounted to a corresponding upwardly extending flange 230 of bolster spring 72 using common fasteners, with a portion of spring saddle 193 positioned therebetween. In other embodiments, the bolster springs flanges 230 may be
15 directly mounted to each other using common fasteners without a portion of a spring saddle positioned between them. As discussed above, apex angle α is formed between the bottom plates of bolster springs 70 and 72.

To further strengthen the bolster spring assembly, a tie-bar 130 is used to tie outboard bolster spring 70 to inboard bolster spring 71 (shown in Figure 3 and Figure 5) and tie-bar
20 132 is used to tie inboard bolster spring 72 to inboard bolster spring 73 (shown in Figure 3 and Figure 5). In this embodiment, the tie-bar is mounted in an intermediate plate located at a midpoint between the top plate and bottom plate of the bolster spring. The midpoint is the point most susceptible to buckling, bulging, or splaying. Therefore, the tie-bar serves to react the inboard and outboard bolster springs to prevent buckling or bulging at the most

vulnerable point on the bolster spring. The tie-bar therefore provides greater rigidity and strength to the bolster spring assembly.

Furthermore, by directly mounting bolster spring 70 to bolster spring 72 with common fasteners and directly mounting bolster spring 71 to bolster spring 73 with common fasteners, and by connecting bolster spring 70 to bolster spring 71 using tie-bar 130 and by connecting bolster spring 72 to bolster spring 73 using tie-bar 132, all four bolster springs 70, 71, 72, and 73 are interconnected. As a result, the present embodiments provide a unified, interconnected assembly of bolster springs that is more rigid and stable than if the bolster springs were not connected.

In addition, as shown in Figures 9 and 10, load cushion 90 is secured to outboard load cushion mount 94 (and to inboard load cushion mount 92 shown in Figure 4), and is positioned above reaction plate 190. Rebound strap 80 is attached to rebound strap flange 80a and to rebound strap flange 80b. The reaction plate 190 is secured via attachment to rebound strap flange 80b. In this embodiment, a bottom surface of the load cushion 90 is positioned a distance D above the reaction plate 190. Distance D may preferably be 19 mm. Therefore, a primary spring rate is based on the bolster springs, and when the load cushion 90 engages the reaction plate 190, a secondary spring rate that includes the load cushion 90 is provided. In this embodiment, a hard stop has been included at 68 mm of travel to protect the bolster springs and load cushion from becoming overcompressed.

The hard stop feature is best shown in Figures 22A and 22B, where fasteners 290a used to mount the load cushion 90 downwardly extend towards the reaction plate 190. Sleeves 291 are positioned around the fasteners 290a and in this embodiment fasteners 290a have a head 293 extending from the end of sleeves 291. When load cushion 90 is significantly compressed, e.g. at 50% compression, the heads 293 of fasteners 290a that contact the reaction plate 190 to provide a hard stop and prevent further compression of the

load cushion 90. In other embodiments, the bottom of sleeves 291 may be counterbored to enclose head 293 so that the head 293 does not extend from the bottom of the sleeve 291 and instead the bottom of the sleeve 291 contacts the reaction plate 190 to provide the hard stop. The bottom of the sleeve 291 has a greater surface area than head 293 of fasteners 290a to spread the forces upon impact with the reaction plate 190. As a result of the hard stop, there is a ceiling on the amount of strain that will experienced by the bolster springs and load cushion. In this embodiment, the rebound strap 80 is comprised of woven material that is advantageously removable to allow for easy repair or replacement of the rebound strap 80. It should be noted that depending upon the application, the disclosed vehicle suspensions may be used without a load cushion.

The components of the vehicle suspension 50 shown in Figures 1-10 may comprise cast or fabricated metal or composite material, including iron, steel, or aluminum. Frame attachment portion 62 and saddle 60, and equalizing beam 100 could also be cast with any suitable castable material. Similarly, the saddle 60 may comprise cast or fabricated metal or composite material. Depending on the application, the metal may, for example, be nodular ductile iron (or more simply, ductile iron), steel, such as a high strength low alloy steel, or aluminum. Typically, high strength low alloy steels are a preferred material to use for the frame hanger and the saddle, although aluminum is often desired when weight considerations are of greater importance.

Figures 11-15 are views of a bolster spring 200. Bolster springs 70, 71, 72, and 73 may be configured as bolster spring 200. As shown in Figures 11-14, bolster spring 200 includes a base plate 220 and a top plate 210. Bolster spring 200 includes an elastomeric section 260 between base plate 220 and intermediate plate 250, an elastomeric section 262 between intermediate plate 250 and intermediate plate 252, an elastomeric section 264 between intermediate plate 252 and intermediate plate 254, and an elastomeric section 266

between intermediate plate 254 and top plate 210. It should be noted that in other embodiments a greater or lesser number of intermediate plates can be used, including no intermediate plates.

Top plate 210 includes mounting holes 212 and 214 that are positioned on flanges of the top plate that extend beyond the elastomer zone with mounting hole 212 located on a flange on a first end of top plate 210 and mounting hole 214 located on a flange on a second end of top plate 210. Such a mounting hole arrangement allows for mounting to a bolster spring mount without using studs extending from the elastomer zone. Bottom plate 220 includes mounting hole 222 that is positioned on a flange on a first end of bottom plate 220 that is also beyond the elastomer zone. An angled flange 230 extends from a second end of bottom plate 220. Angled flange 230 includes a pair of spaced mounting holes 232 and 234 positioned beyond the elastomer zone that are adapted to be directly mounted to a corresponding angled flange of an adjacent bolster spring, as illustrated in Figure 26. Top plate 210 and bottom plate 220 advantageously extend beyond the elastomer zone, and may be formed complementary in shape with the mounting surface of a bolster spring mount to provide a larger mounting surface area, which forms a stronger mechanical joint.

As shown in Figures 13 and 14, angled flange 230 may extend at an angle that is one half of apex angle α , so that when directly mounted to the angled flange of an adjacent bolster spring having the same configuration, an apex angle α is formed between the bottom surfaces of the directly connected bolster springs. In addition, a tie-bar mounting extension 240 having a through hole 241 through which a tie-bar may extend is shown extending from center intermediate plate 252.

Figure 15 is a top view of bolster spring 200. As can be seen, mounting hole 222 of the bottom plate 220 extends beyond the elastomer zone. In addition, mounting holes 232 and 234 on angled flange 230 extend outwardly from the bottom plate 220 and have a spacing

that is wider than the width of the bottom plate 220 and the top plate 210. This wide spacing of the mounting holes 232 and 234 on angled flange 230 advantageously provides for greater contact between the angled flange surfaces when mounted as shown in Figure 26, resulting in a stronger mechanical joint being formed between the angled flanges of the bolster springs.

5 The particular configuration of the base plate 220, top plate 210, and intermediate plates 250, 252, and 254 of bolster spring 200 is illustrative only, and these components may have a variety of geometries and configurations. Thus, the bolster spring 200 is not required to have, but may have, the geometry shown in Figures 9-15. Furthermore, the use of a tie-bar may be, but is not required to be, included.

10 A bolster spring is typically constructed from relatively flat first and second end plates with an elastomer connected between them. This spring will then have compressive and shear rates corresponding to the chosen material, cross-section, and thickness of elastomer. In accordance with the disclosed embodiments, bolster spring 200 may be constructed of elastomeric sections 260, 262, 264, and 266 bonded to one or more of plates
15 210, 250, 252, 254, and 220. Elastomeric sections 260, 262, 264, and 266 may comprise an elastomeric material (i.e., an elastomer) such as natural rubber, synthetic rubber, styrene butadiene, synthetic polyisoprene, butyl rubber, nitrile rubber, ethylene propylene rubber, polyacrylic rubber, high-density polyethylene, thermoplastic elastomer, a thermoplastic olefin (TPO), urethane, polyurethane, a thermoplastic polyurethane (TPU), or some other type of
20 elastomer. In this regard and in particular, elastomeric sections 260, 262, 264, and 266 may comprise an elastomer defined as American Society of Testing and Materials (ASTM) D2000 M4AA 7I7 A13 B13 C12 F17 K11 Z1 Z2. In this case, Z1 represents natural rubber and Z2 represents a durometer selected to achieve a desired shear rate. The selected durometer may be based on a given predefined scale, such as the Shore A scale, the ASTM D2240 type A
25 scale, or the ASTM D2240 type D scale. In a preferred embodiment, in accordance with the

Shore A scale, Z2, for example, is preferably 70 ± 5 . In another embodiment, in accordance with the Shore A scale, Z2 is, for example, within the range of 50 to 80. Other examples of Z2 and ranges for Z2 are also possible.

In another respect, elastomeric sections 260, 262, 264, and 266 may comprise a viscoelastomeric material that (i) has elastic characteristics when the bolster spring 200 is under a load within a given range and when that load is removed, and (ii) has non-elastic characteristics (for example, does not return to an original non-loaded shape) if the applied load exceeds the greatest load of the given range. The given range may extend from no load to a maximum expected load plus a given threshold. The given threshold accounts for possible overloading of bolster spring 200. As an example, the viscoelastomeric material may comprise amorphous polymers, semi-crystalline polymers, and biopolymers. Other examples of the viscoelastomeric material are also possible.

In accordance with the example embodiments, elastomeric sections 260, 262, 264, and 266 may also comprise one or more fillers. The filler(s) may optimize performance of elastomeric sections 260, 262, 264, and 266. The fillers may include, but are not limited to, wax, oil, curing agents, and/or carbon black. Such fillers may optimize performance by improving durability and/or tuning the elastomeric sections for a given shear load and/or a given compressive load applied to the elastomeric sections. Improving durability through the use of fillers may include, for example, minimizing a temperature rise versus loading characteristic of the elastomeric sections and/or maximizing shape retention of the elastomeric sections.

Bolster spring 200 may be formed, for example, by inserting the plates 210, 250, 252, 254, and 220 into a mold (not shown). The plates may each be coated with a coating material. As an example, the coating material may comprise a material comprising zinc and phosphate, modified with calcium. The coating material may have a coating weight of 200-

400 milligrams per square foot. Other examples of the coating material are also possible. A bonding agent may be applied to the coated plates for bonding the plates to the elastomeric sections. As an example, the bonding agent may comprise Chemlok® manufactured by the Lord Corporation, Cary, North Carolina, USA. Other examples of the bonding agent are also possible. Applying the coating material and/or applying the bonding agent may occur prior to, during, and/or after insertion of the plates into the mold. After applying the coating material and the bonding agent, the elastomeric material (while in a pourable form) may be inserted into the mold to form the elastomeric sections.

In a preferred embodiment, any exposed portion of the plates (for example, a portion of the plates not covered by the elastomeric material) is protected against corrosion by a means other than the elastomeric material. In other embodiments, some exposed portions of the plates (e.g., the edges of the plates) may not be protected against corrosion, whereas any other exposed portions of the plates are protected against corrosion.

The plates 210, 250, 252, 254, and 220 can be made of any of a variety of suitable materials, including, but not limited to, iron, steel, aluminum, plastic, a composite material, or some other material. The plates may be fully, or at least substantially, encapsulated in elastomer to further enhance their corrosion resistance and friction at the mating suspension members. As an example, plates 210, 250, 252, 254, and 220 can comprise plates having a thickness between a range of 0.188 inches (3.00 mm) to 0.25 inches (6.35 mm), or more.

Figures 16A and 16B are perspective views of an example load cushion 300 for use in vehicle suspension 50. Figure 17 is a side view, Figure 18 is a front view, Figure 19 is a bottom view, and Figure 20 is a top view of load cushion 300. Load cushion 90 shown in vehicle suspension 50 in Figures 1-10 may be arranged as load cushion 300.

As shown in one or more of Figures 16A-20, load cushion 300 includes a top plate 310, a bottom plate 320, and a load cushion portion 330. Top plate 310 includes mounting

flange 312 with mounting hole 312a and mounting flange 314 with mounting hole 314a adapted for mounting to load cushion mounts 92 and 94 (shown in Figures 2 and 4) of vehicle suspension 50. In this embodiment, a horizontal cross section of the cushion portion 330 is generally square with rounded corners, although it could also be generally circular, rectangular, or conic. As shown in Figure 16B and 19, the bottom plate 320 includes holes 322 that are used during the molding process to provide a passage for the elastomeric material that forms the cushion portion 330.

As shown in Figure 17, the load cushion portion 330 has a unique symmetrical shape that includes curvilinear front and rear outer surfaces 332 and 334 that taper towards the center at the midpoint between the top plate 310 and bottom plate 320 such that the narrowest thickness of the load cushion 330 occurs at the midpoint. Similarly, as shown in Figure 18, the load cushion portion 330 has a unique symmetrical shape that includes curvilinear left and right outer surfaces 336 and 338 that taper towards the center at the midpoint between the top plate 310 and bottom plate 320 such that the narrowest thickness of the load cushion 330 occurs at the midpoint.

Load cushion 330 may have a cross section where front and rear outer surfaces 332 and 334 have a negative Gaussian curvature, and similarly load cushion 330 may have a cross section where left and right outer surfaces 336 and 338 have a negative Gaussian curvature. In addition, load cushion portion 330 may be shaped as a hyperboloid. The curved outer surfaces of the load cushion portion result in a much lower elastomeric strain on the load cushion for the same deflection as compared to a linearly reduced cross-section.

The load cushion 90 may undergo 50% compression at full jounce, or when the hard stop discussed above is reached. At this point, the cross-section of the load cushion portion 330 changes from a negative Gaussian curvature to a 0 or slightly positive Gaussian curvature. As used herein the term, 0 Gaussian curvature means that the outer surfaces of the

cross-section are parallel, and a “slightly positive Gaussian curvature” means that the midpoint of the load cushion portion 330 becomes wider than the end sections, by up to 1 cm on each side of the load cushion portion.

It will be appreciated that bottom plate 320 is not required, and the load cushion 330 may have an exposed surface instead of having bottom plate 320. The use of a bottom plate 320 does not affect in any significant way the load cushion load versus deflection curve. However, the bottom plate 320 may be incorporated to protect the active elastomer of the load cushion portion 330 from debris such as rocks that could inadvertently end up on the reaction plate that is positioned beneath the load cushion. Debris could become embedded temporarily or permanently into the elastomer and create an undesirable crack initiation site.

The bottom plate 320 may be encapsulated to provide for improved corrosion resistance, elimination of metal to metal contact resulting in noise reduction upon contact with the reaction plate, improved friction between the load cushion 300 and the reaction plate 190 (shown in Figures 9 and 10) to reduce or minimize wear between the bottom plate 320 and the reaction plate 190 during vehicle motion because relative motion is decreased or eliminated. In addition, encapsulation may be used as a service wear and replacement indicator similar to wear bars found between tire treads.

Load cushion 300 may have a continuously increasing spring rate as an applied load increases and a continuously decreasing spring rate as an applied load decreases, due to its generally conic shape.

The top plate 310 and base plate 320 may be constructed of any of a variety of suitable materials, including, but not limited to, iron, steel, aluminum, plastic, and a composite material. As an example, the base plate can comprise a plate having a thickness between a range of 0.188 inches (3.00 mm) to 0.25 inches (6.35 mm), or more. The plates can be encapsulated in elastomer and/or bonded to the load cushion portion using a bonding

agent. The plate dimensions and shape can be varied to any dimension or shape desired for packaging, weight, and aesthetics. Preferably, the load cushion top plate 310 is dimensioned to (i) match the surface of the load cushion mount described herein, such as load cushion mounts 92 and 94, (ii) locate mounting holes for securing the load cushion 300 to the load
5 cushion mounts 92 and 94, and (iii) minimize overall mass.

The size and dimensions of the elastomer used for the cushion portion 330 of load cushion 300 may be optimized for the vertical spring rate requirements. As noted above, the vertical spring rate for the load cushions 300 may continuously increase with increasing load and continuously decreases with decreasing load, defining a curvilinear shape with no
10 discontinuities on a graph illustrating spring rate as a function of sprung load.

Preferably, load cushion portion 330 has a generally conic shape as it extends towards a midpoint between top plate 310 and bottom plate 320. With this preferred shape, the vertical spring rate for the load cushion 300 linearly increases with increasing load and linearly decreases with decreasing load. In this regard, load cushion 300 is operable as a
15 progressive spring rate load cushion. In one embodiment, the cross section of load cushion portion 330 adjacent top plate 310 and adjacent bottom plate 320 is 110 mm by 110 mm. At the midpoint between the top plate 310 and the bottom plate 320 the load cushion portion 330 the cross section is 88 mm by 88 mm, and the height of load cushion portion 330 is 105 mm not including plates or wear layer encapsulation. Other example dimensions of portions of
20 load cushion 300 are also possible. For a given geometry, the spring rate of load cushion 300 may be optimized by varying the durometer of the elastomer. By varying the durometer, a family of interchangeable progressive spring rate load cushions can be created.

It will further be appreciated that the load cushion 300 may be mounted with the cushion portion 330 extending either above or below the bottom plate 310. Likewise, the
25 load cushion 300 may be mounted such that the top plate 310 extends beneath the bottom

plate 320. Therefore, the use of the terms “top” and “bottom” are used simply to describe the plates 310 and 320 that are attached to the load cushion portion 330, and do not in any way require that the load cushion 300 is mounted in any particular configuration.

Figure 21A is a cross sectional inboard perspective view of vehicle suspension 50 taken along line 21A-21A shown in Figure 4, and Figure 21B is a cross sectional inboard perspective view of vehicle suspension 50 taken along line 21B-21B shown in Figure 2. Frame attachment portion 62 with mounting holes 63 is shown extending upwardly from upper surface 91 of the saddle with central flange 64 and gusset 68. Shock absorber 122 is shown mounted to inboard surface 67 of the saddle and rebound strap 80 is shown extending beneath load cushion mount 92. Bolster springs 70 and 71 are shown mounted to bolster spring mounts 170 and 171 on opposite sides of equalizing beam 100. Similarly, bolster springs 72 and 73 are shown mounted to bolster springs mounts 172 and 173 on opposite sides of equalizing beam 100. In addition, common fastener 71b is shown directly mounting bolster spring 71 to bolster spring 73 and common fastener 70b is shown directly mounting bolster spring 70 to bolster spring 72.

Figure 22A is a cross sectional inboard perspective view of vehicle suspension 50 taken along line 22A-22A shown in Figure 4, and Figure 22B is a cross sectional outboard perspective view of vehicle suspension 50 taken along line 22B-22B shown in Figure 2. Frame attachment portion 62 with mounting holes 63 is shown extending upwardly from upper surface 91 of the saddle with central flange 64 and gusset 68. Shock absorber 122 is shown mounted to inboard surface 67 of the saddle and rebound straps 80 are shown extending on opposite sides of load cushion 90. Load cushion 90 can be seen positioned directly above reaction plate 190. Load cushion 90 is also shown mounted to the load cushion mounts extending from walls 65 and 67 of the saddle using fasteners 290a.

Spring saddle 193 is shown supporting reaction plate 190. Throughhole 70d is positioned in reaction plate 190 to allow a fastener to extend therethrough for mounting together the angled flanges of bolster springs 70 and 72. Similarly, throughhole 71d is positioned in reaction plate 190 to allow a fastener to extend therethrough for mounting together the angled flanges of bolster springs 71 and 73.

In addition, equalizing beam 100 is shown having a U-shaped cross section with opposed walls 100a and 100b. A tie-bolt 101 having a sleeve 103 is used to tie the two walls 100a and 100b together. Tie-bolt 101 is used to relieve stress in the equalizing beam 100 where the bolster springs 70-73 are attached by "pinching" walls 100a and 100b together such that their inner surfaces contact respective end surfaces of sleeve 103.

Figure 23A is a cross sectional inboard perspective view of vehicle suspension 50 taken along line 23A-23A shown in Figure 4, and Figure 23B is an outboard perspective cross sectional view of vehicle suspension 50 taken along line 23B-23B shown in Figure 2. Frame attachment portion 62 with mounting holes 63 is shown extending upwardly from upper surface 91 of the saddle with central flange 64 and gusset 68. Shock absorber 122 is shown mounted to inboard surface 67 of the saddle and rebound straps 80 are shown extending on opposite sides of load cushion 90. Load cushion 90 can be seen positioned directly above reaction plate 190. Load cushion 90 is also shown mounted to the load cushion mounts extending from walls 65 and 67 of the saddle.

Figure 24 is a perspective view of the inboard side of equalizing beam 100 and Figure 25 is a top view of equalizing beam 100. Beam hubs 102 and 104 are located on opposite ends of the equalizing beam 100. Shock absorber mount 106 having mounting hole 106a and shock absorber mount 108 having mounting hole 108a are shown positioned on the inboard side of the equalizing beam 100. Bolster spring mounts 107a and 107b extend from opposite sides of the center of equalizing beam 100. On the inboard side, the walls of bolster spring

mount 107a include mounting holes 109a and 109b that are used to mount bolster springs 71 and 73 (shown in Figure 3), and on the outboard side, the walls of bolster spring mount 107b include mounting holes 108b and 108a that are used to mount bolster springs 70 and 72 (shown in Figure 2).

5 The equalizing beam 100 is shown in an illustrative configuration. However, equalizing beam 100 may be constructed in any of a variety of arrangements and with a variety of configurations and/or materials.

 Figure 26 provides an illustration showing how bolster springs 70 and 72 may be directly mounted to each other using common fasteners. In particular, flanges 230 of bolster
10 springs 70 and 72 are positioned together as shown, with spring saddle 193 extending therebetween, wherein a pair of common fasteners may be used to directly mount the bolster springs 70 and 72 together. Spring saddle 193 may be formed from a pair of bent plates having a thickness of 6mm, such that the flanges 230 are positioned 12 mm apart. In addition, apex angle α is shown between the bottom surfaces of bottom plates 220 of bolster
15 springs 70 and 72.

 Example embodiments of the present invention have been described above. Those skilled in the art will understand that changes and modifications may be made to the described embodiments without departing from the true scope and spirit of the present invention, which is defined by the claims.

20

CLAIMS

1. A load cushion for a vehicle suspension system, the load cushion comprising:
 - an end plate;
 - an elastomeric cushion portion extending from the end plate to a top of the cushion portion;
 - wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion;
 - wherein a vertical cross section of the cushion portion from a left side to a right side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the left side and the right side is positioned at a midpoint between the end plate and the top of the cushion portion; and
 - wherein a horizontal cross section of the elastomeric cushion portion taken at the midpoint is square with rounded comers.
2. The load cushion of claim 1, wherein each horizontal cross section of the elastomeric cushion portion between the end plate and the top of the cushion portion is square with rounded corners.
3. The load cushion of claim 1, wherein an end plate is positioned over the top of the cushion portion.
4. The load cushion of claim 1, wherein the end plate has two oppositely disposed flanges each having a mounting hole adapted for attachment to a load cushion spring mount.
5. The load cushion of claim 1, wherein a vertical cross-section of the elastomeric cushion has a negative Gaussian curvature.
6. A load cushion for a vehicle suspension system, the load cushion comprising:
 - an end plate;
 - an elastomeric cushion portion extending from the end plate to a top of the cushion portion;

wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges;

wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion;

wherein the vertical cross-section of the elastomeric cushion has a negative Gaussian curvature; and

wherein when the load cushion undergoes compression, the vertical cross-section changes through a zero Gaussian curvature.

7. A load cushion for a vehicle suspension system, the load cushion comprising:

an end plate;

an elastomeric cushion portion extending from the end plate to a top of the cushion portion;

wherein a vertical cross-section of the cushion portion from a front side to a rear side of the cushion is symmetrical and has curvilinear edges; and wherein a narrowest width between the front side and the rear side is positioned at a midpoint between the end plate and the top of the cushion portion;

wherein the vertical cross-section of the cushion portion of the elastomeric cushion has a negative Gaussian curvature; and

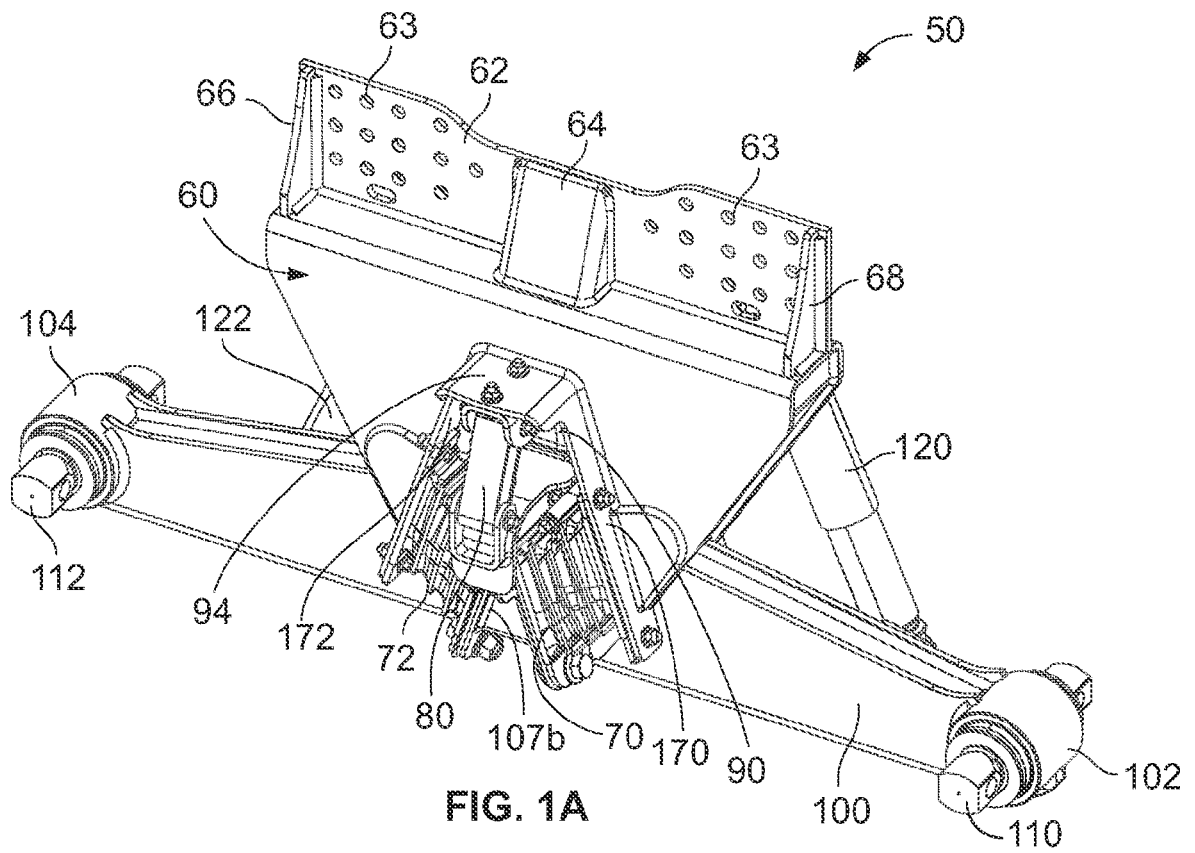
wherein when the load cushion undergoes 50% compression, the vertical cross-section changes to a zero or slightly positive Gaussian curvature.

8. The load cushion of claim 3, wherein an exposed surface of the end plate is encapsulated.

Hendrickson USA, L.L.C.

Patent Attorneys for the Applicant/Nominated Person

SPRUSON & FERGUSON



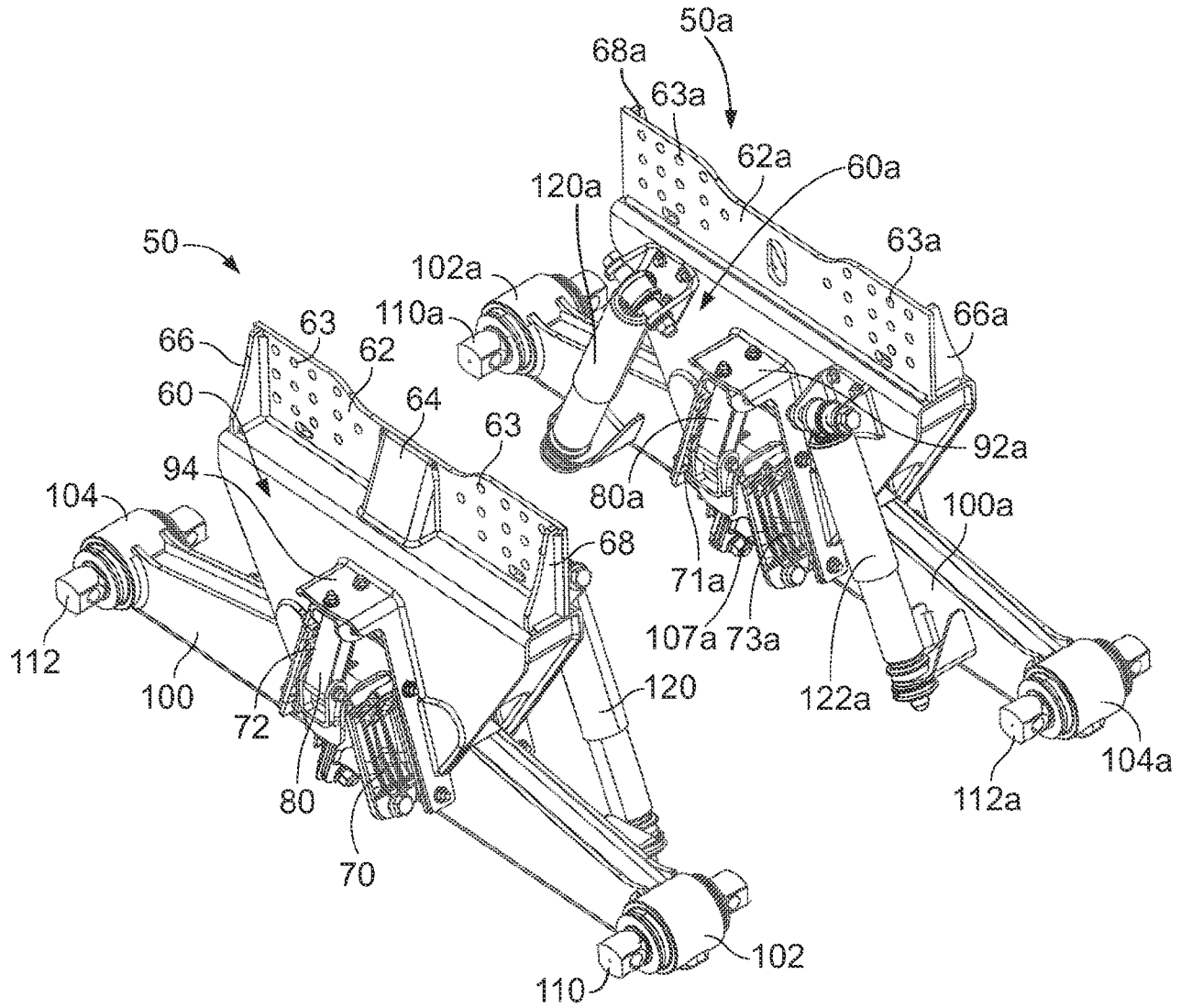


FIG. 1B

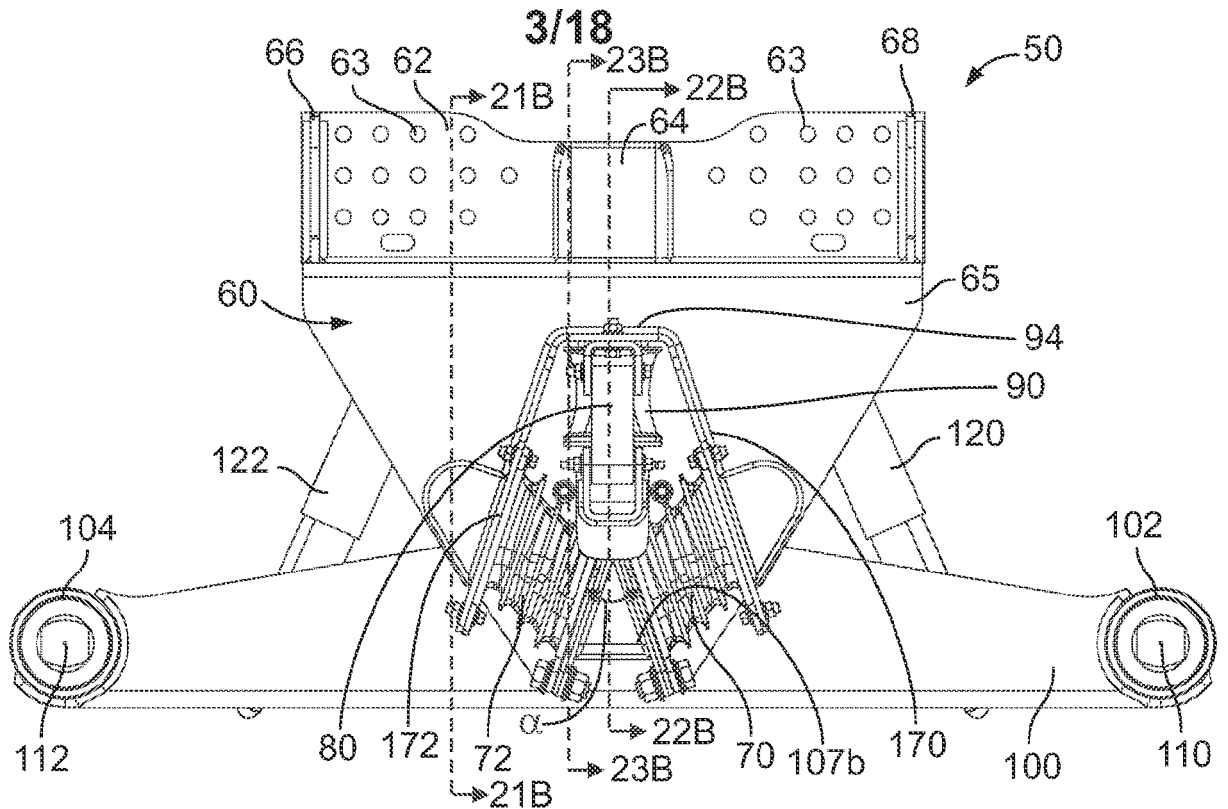


FIG. 2

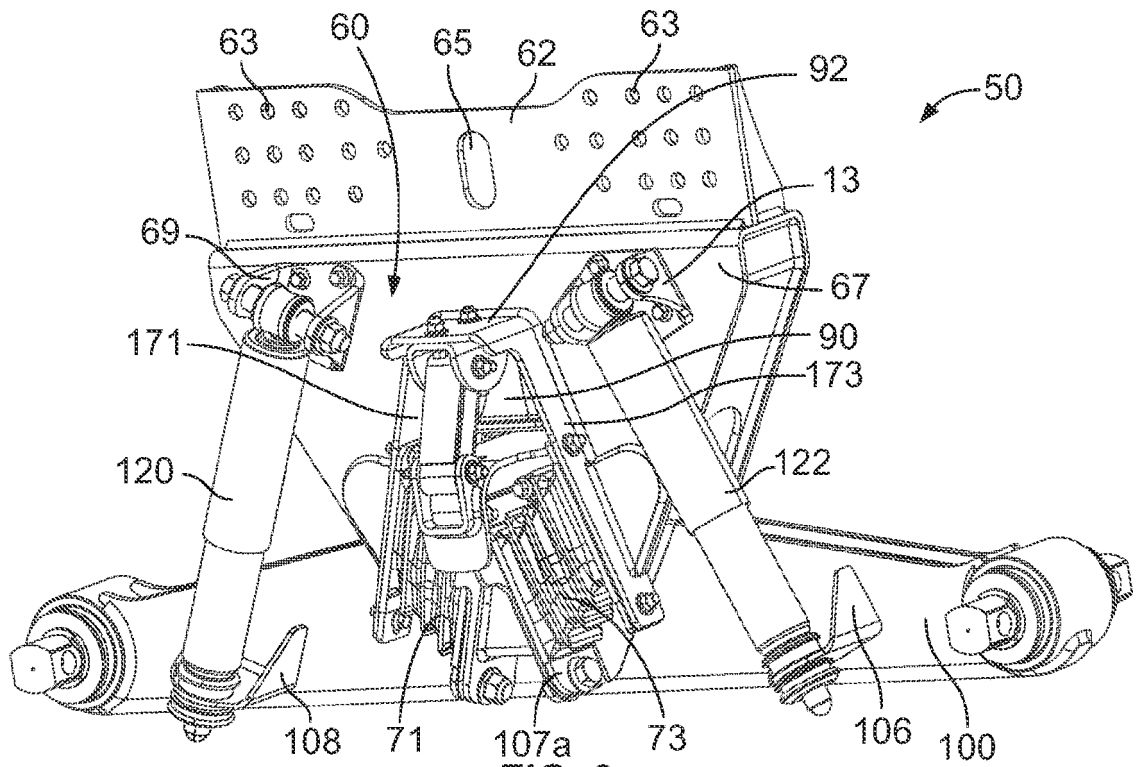


FIG. 3

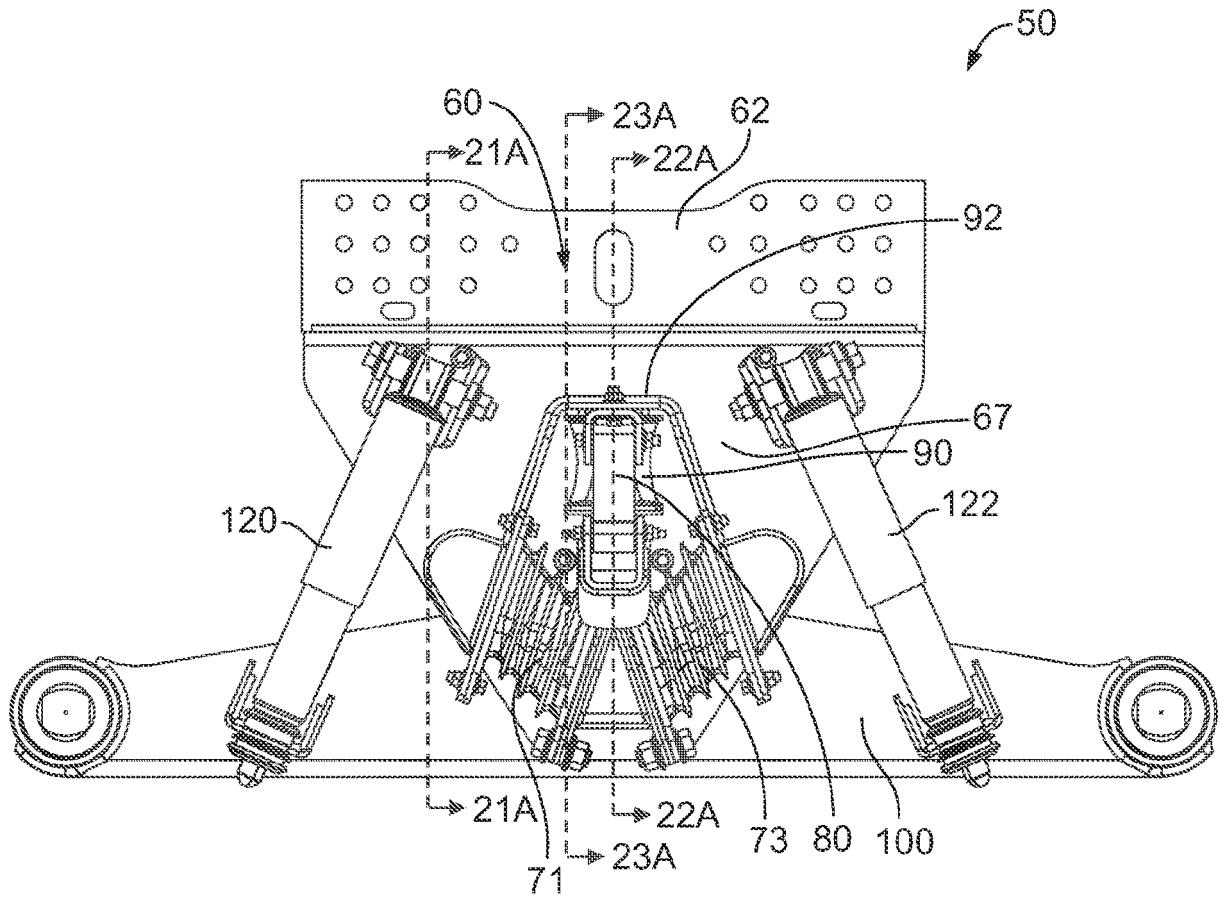


FIG. 4

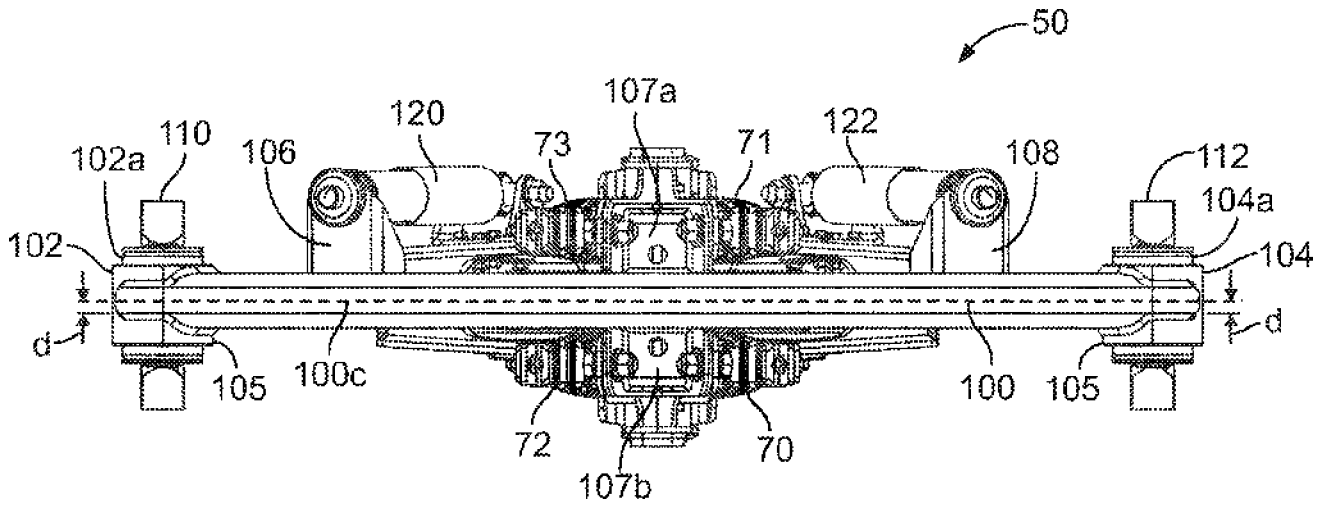


FIG. 5

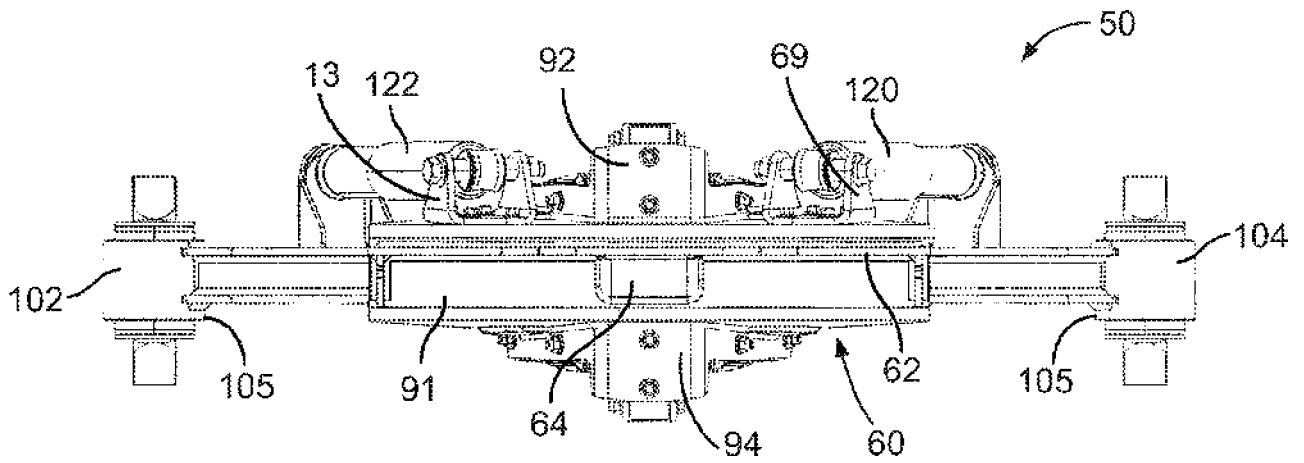
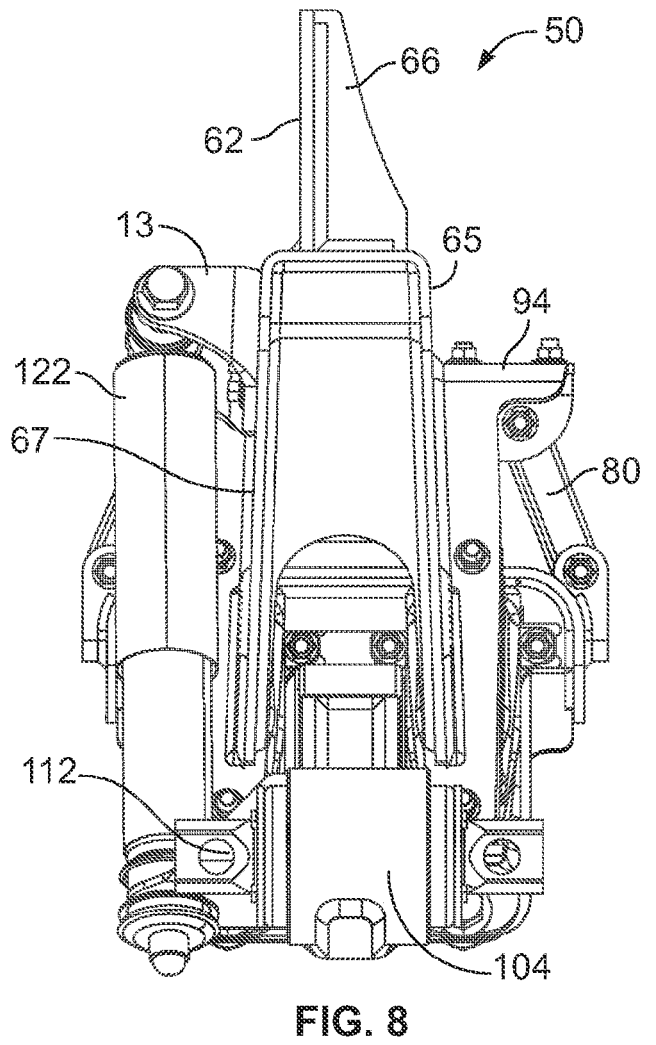
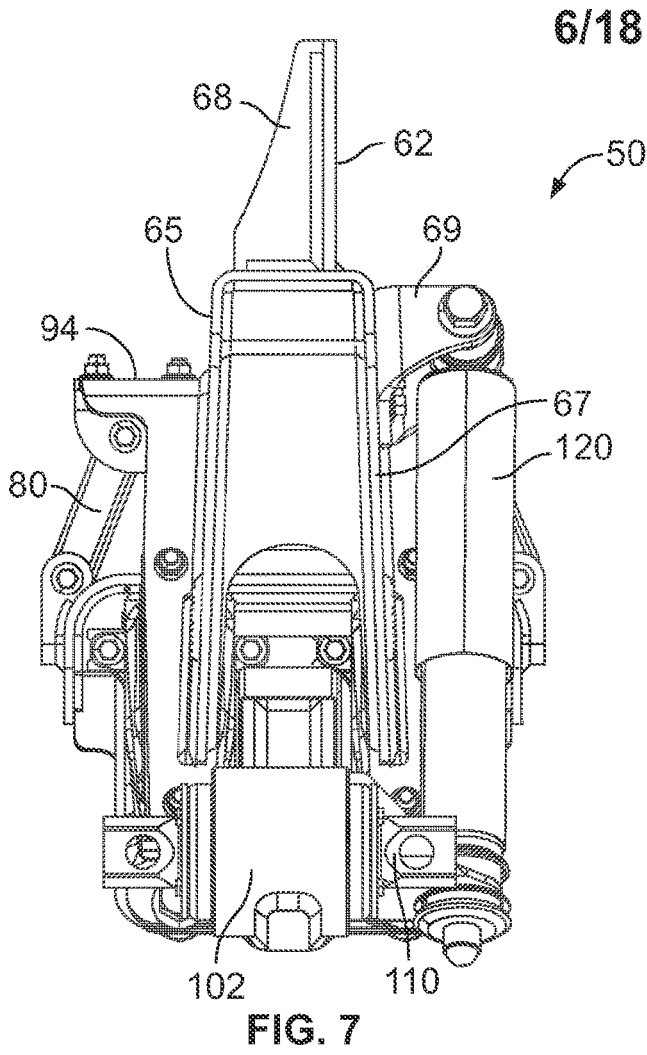


FIG. 6



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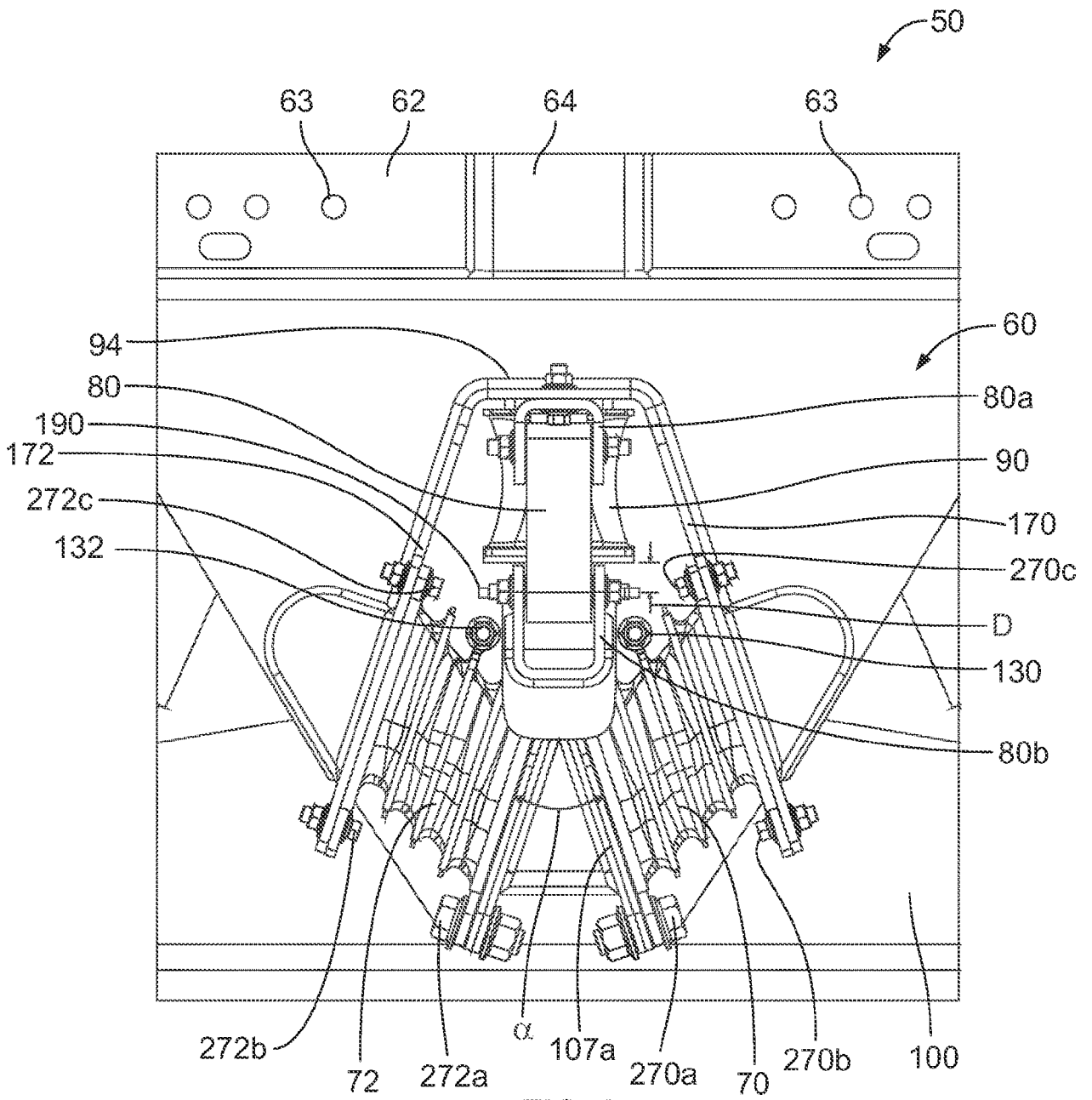


FIG. 9

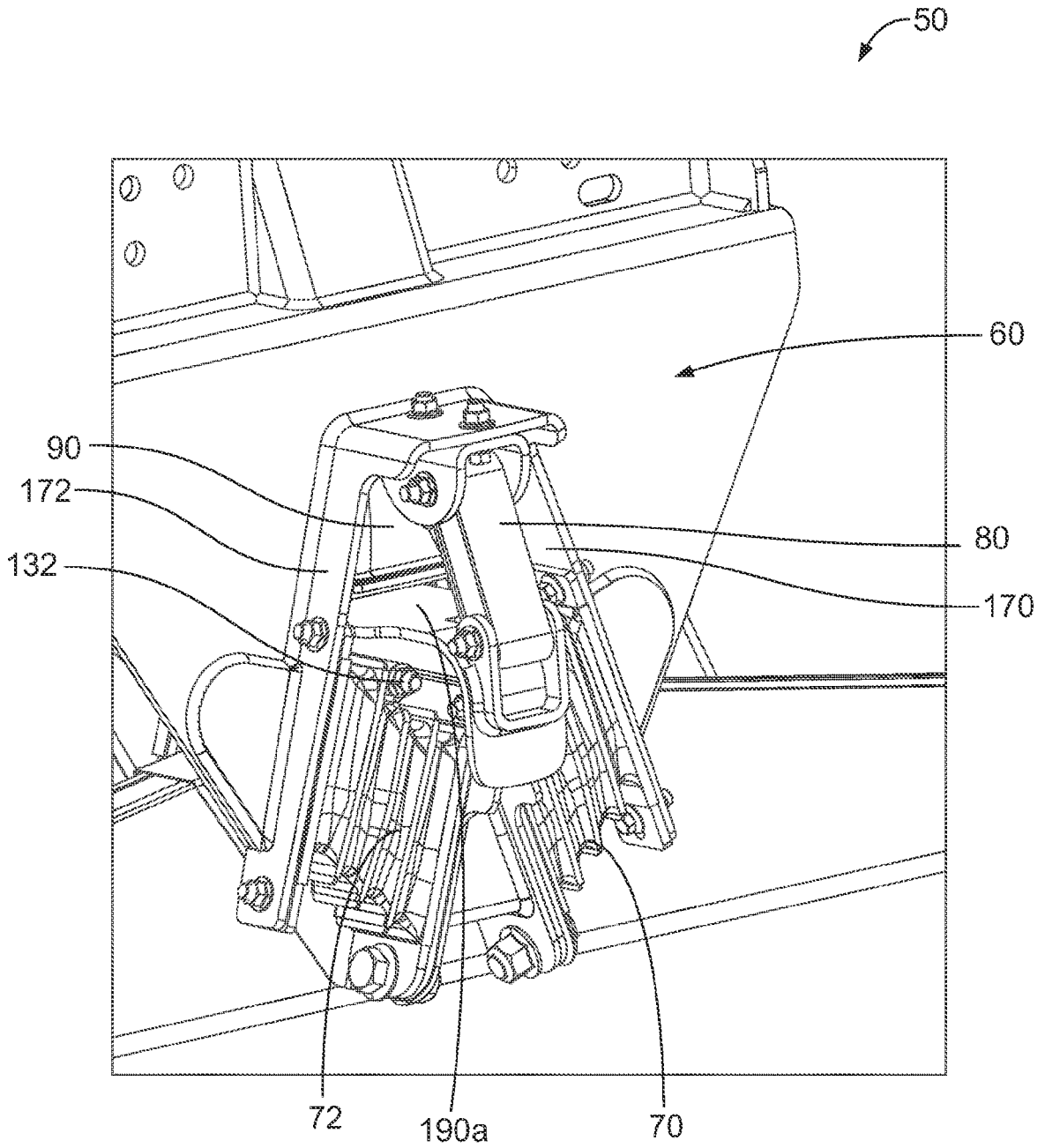


FIG. 10

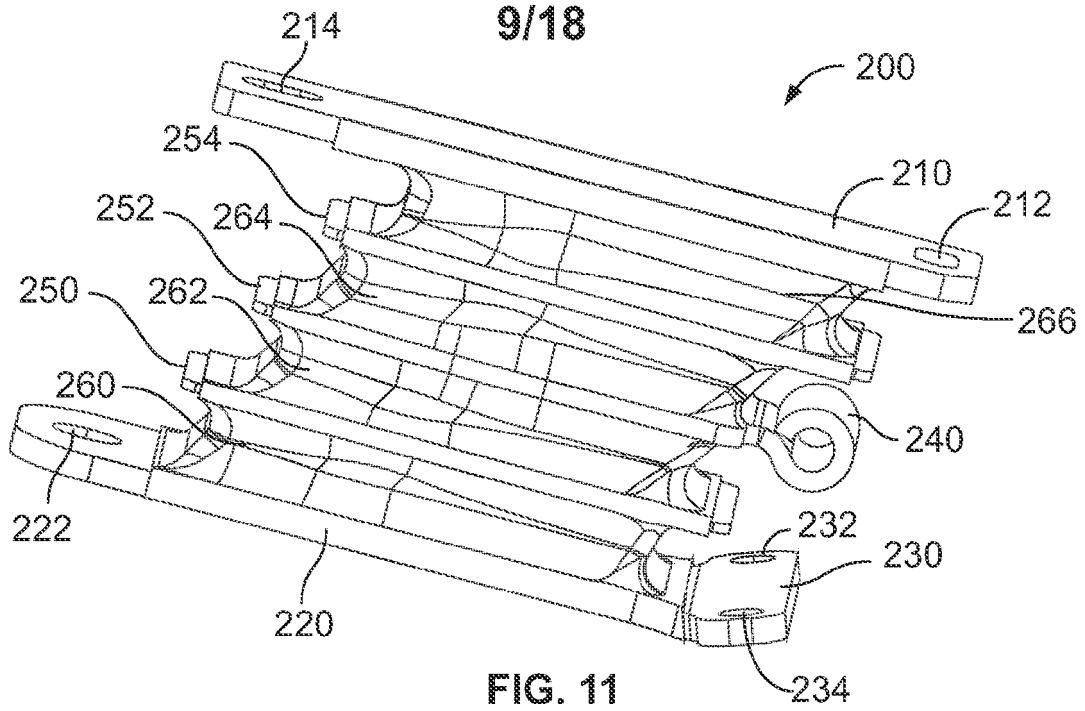


FIG. 11

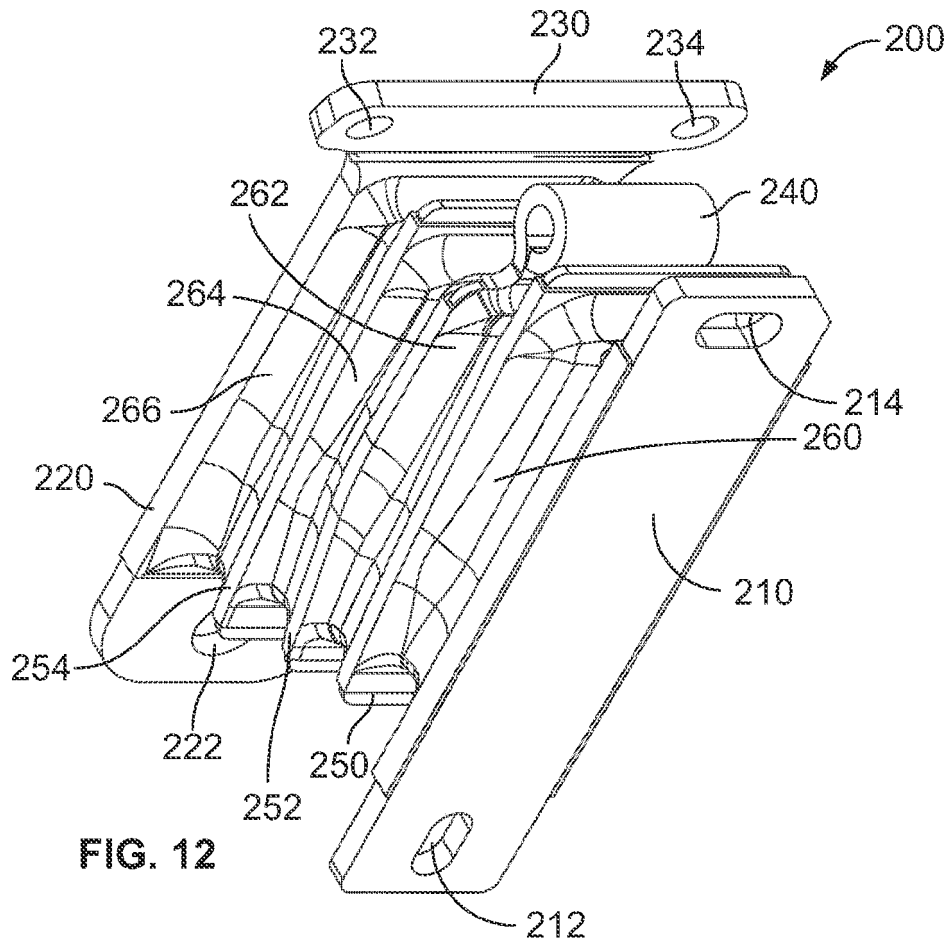


FIG. 12

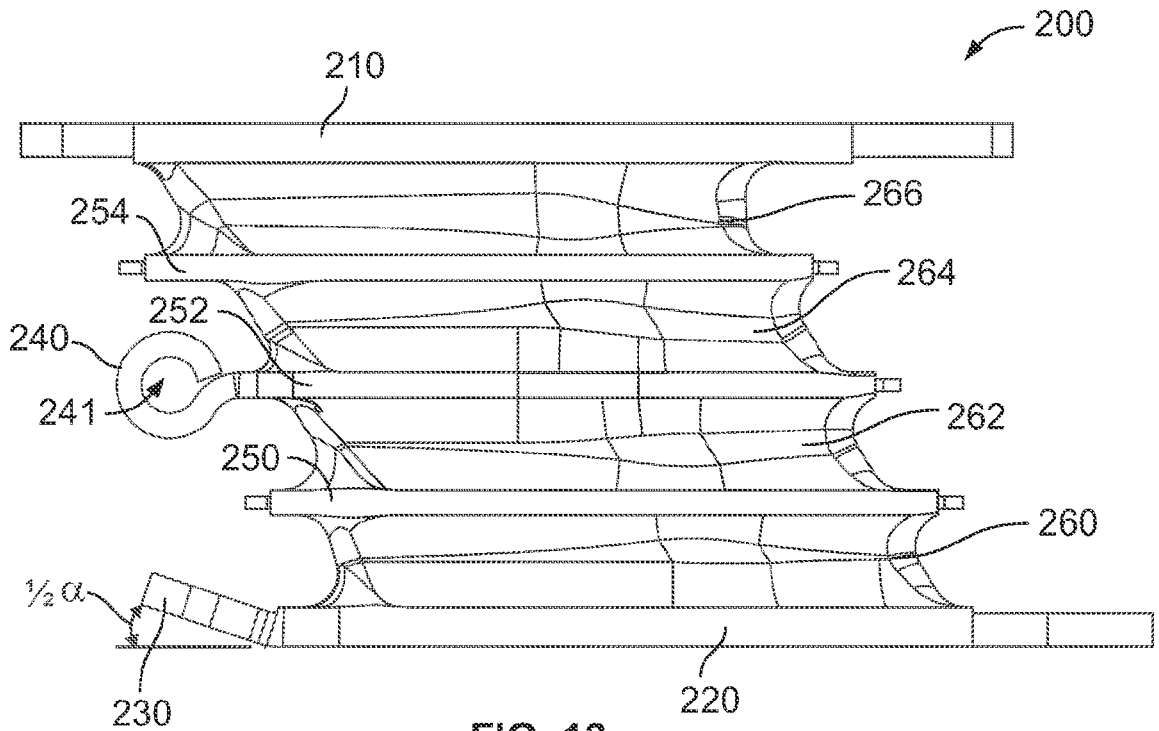


FIG. 13

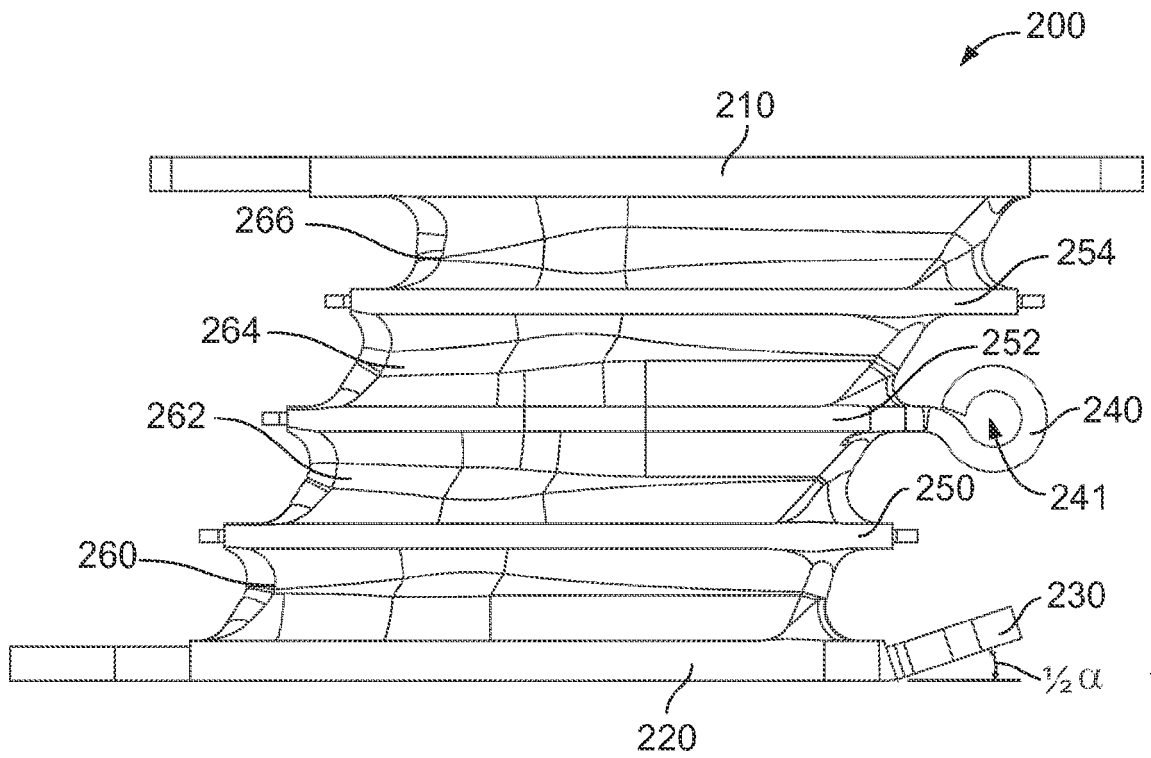


FIG. 14

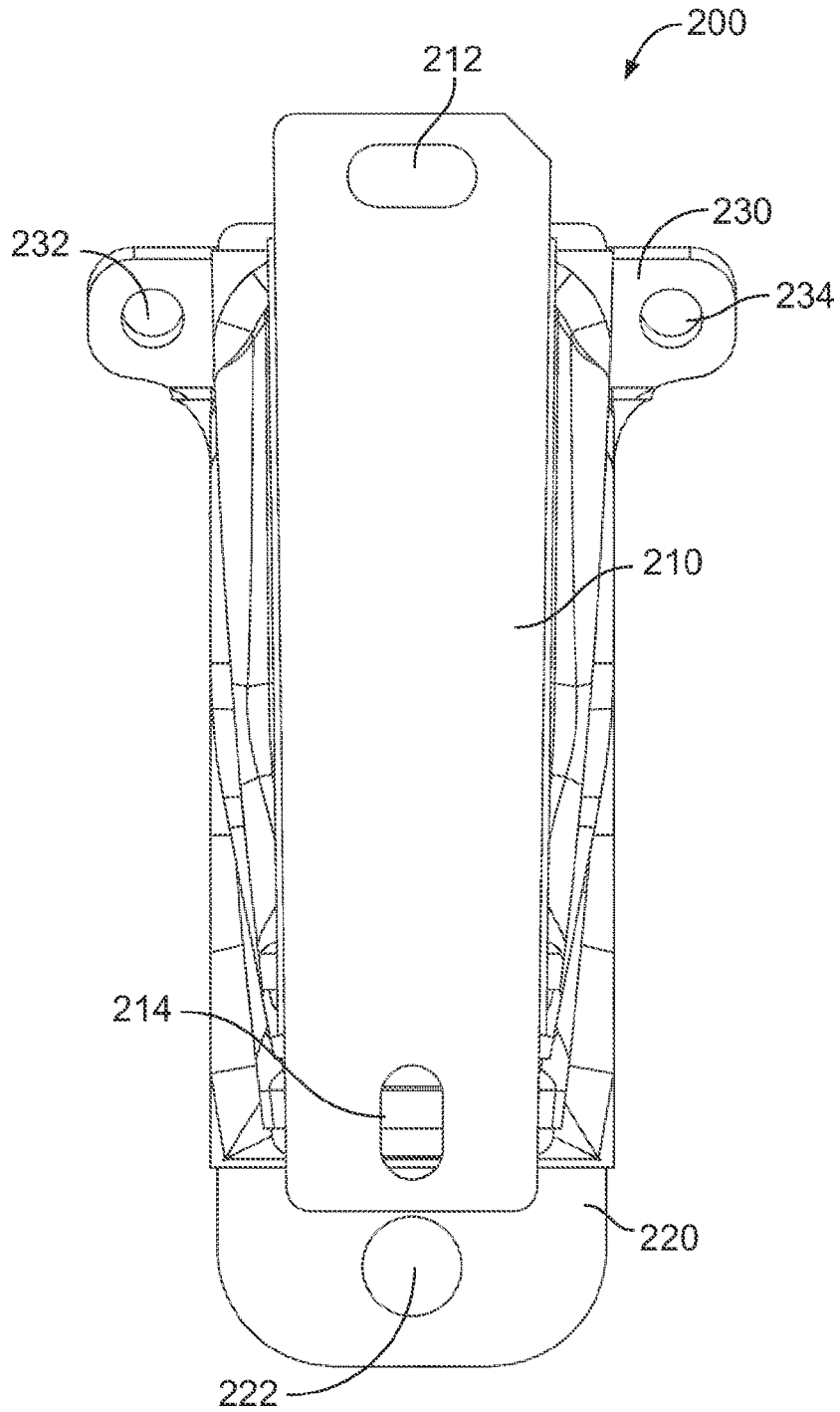


FIG. 15

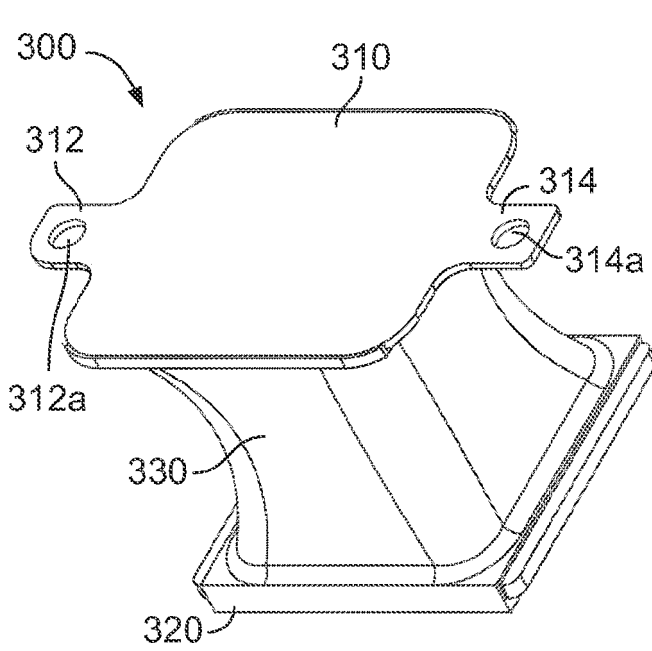


FIG. 16A

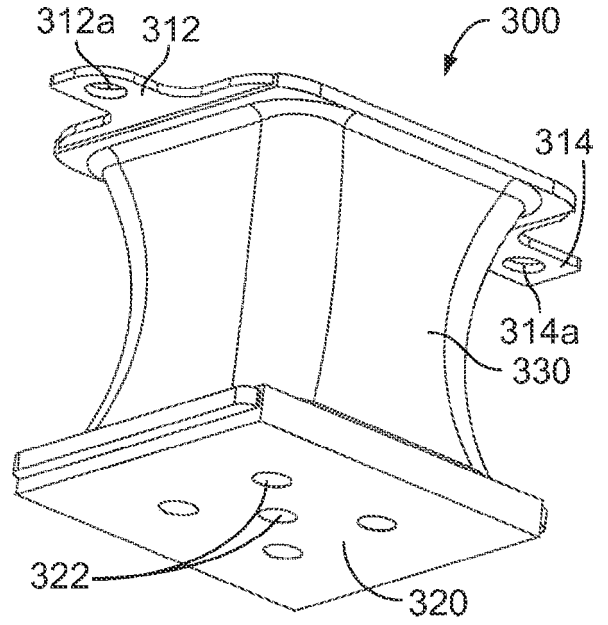


FIG. 16B

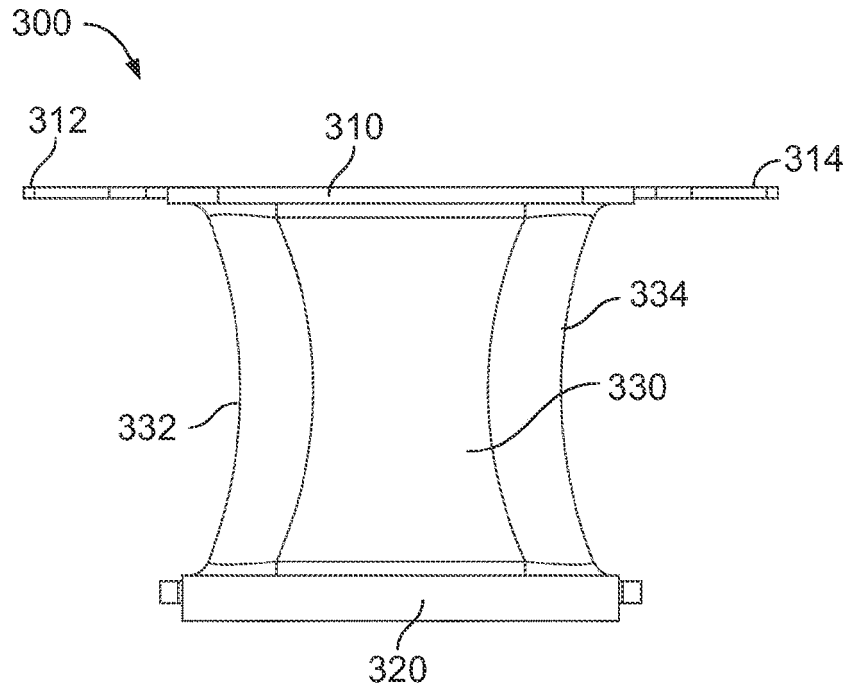


FIG. 17

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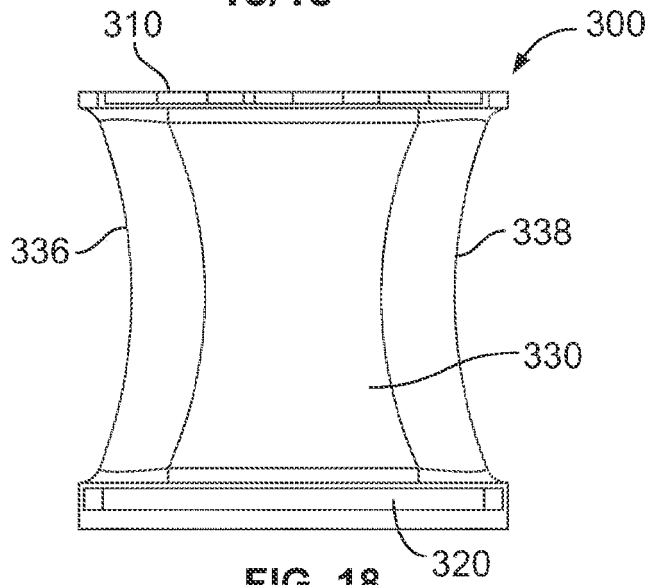


FIG. 18

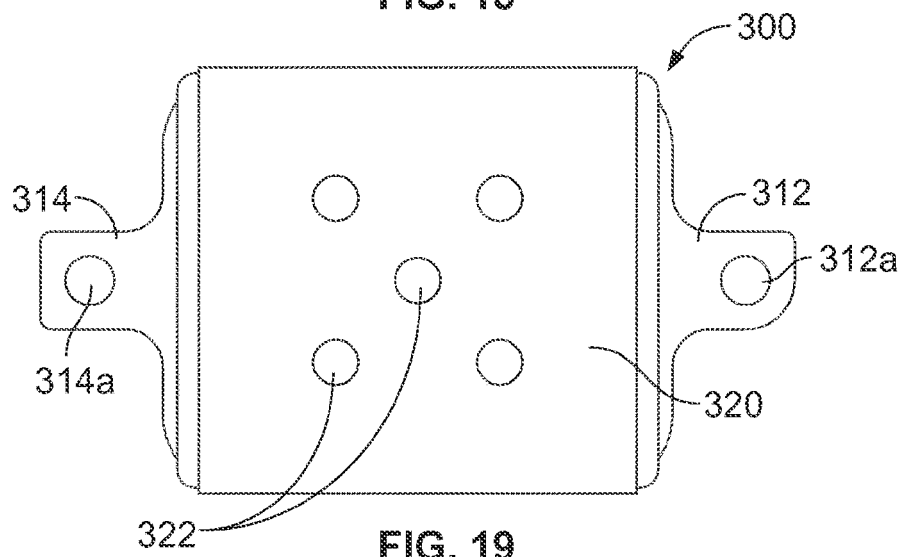


FIG. 19

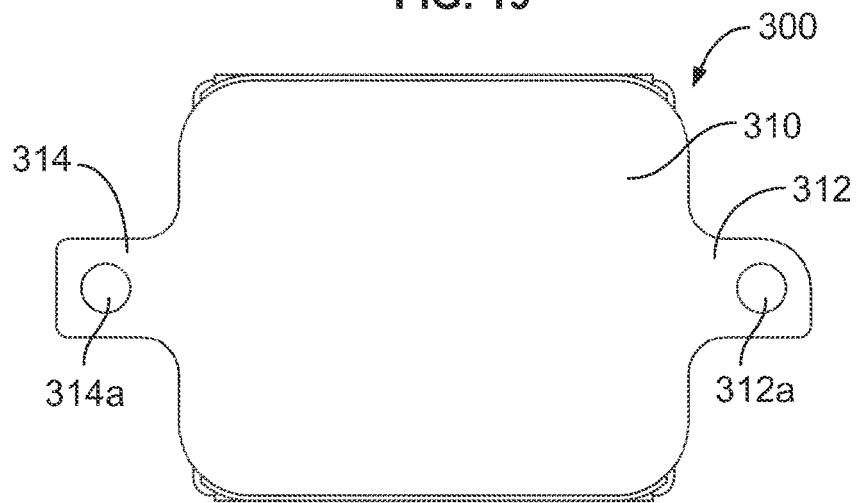


FIG. 20

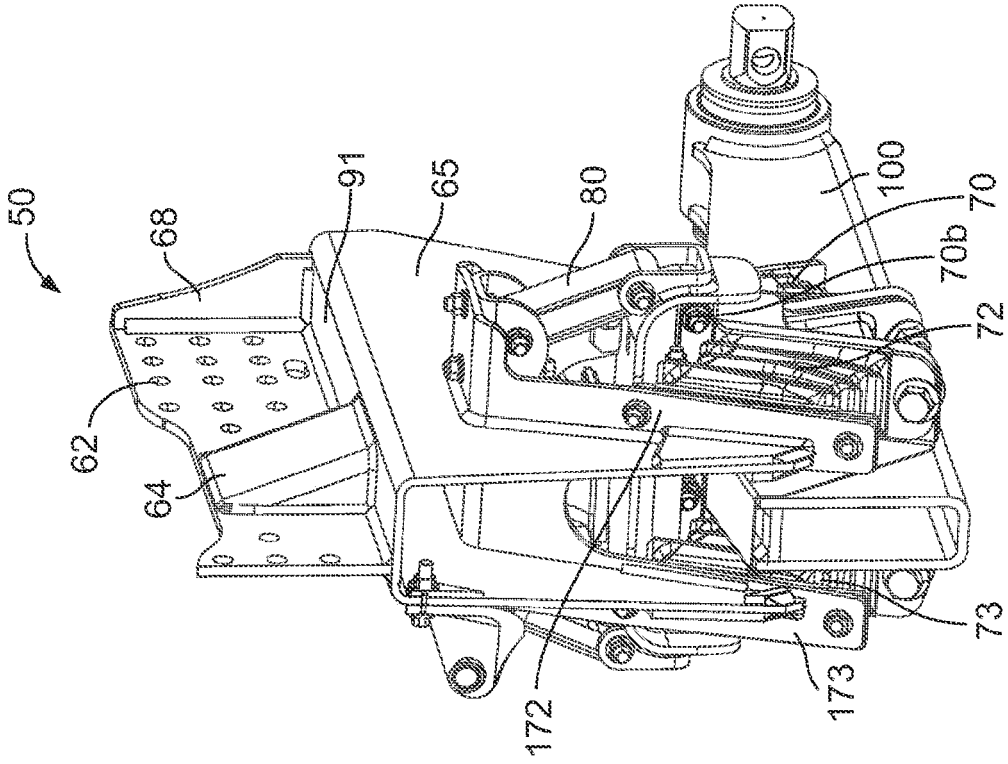


FIG. 21B

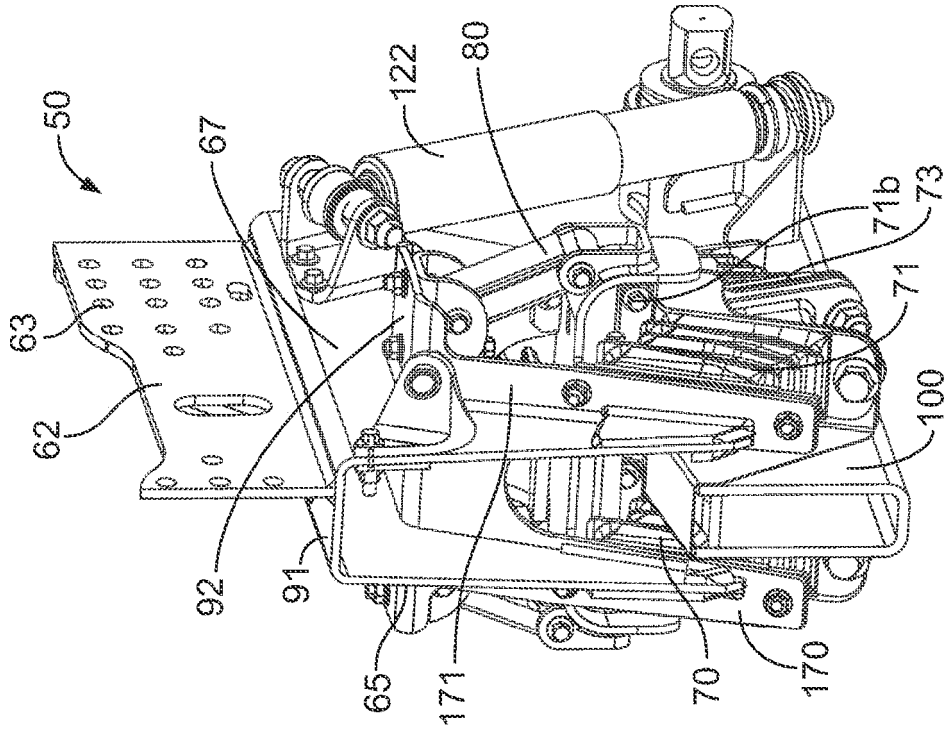


FIG. 21A

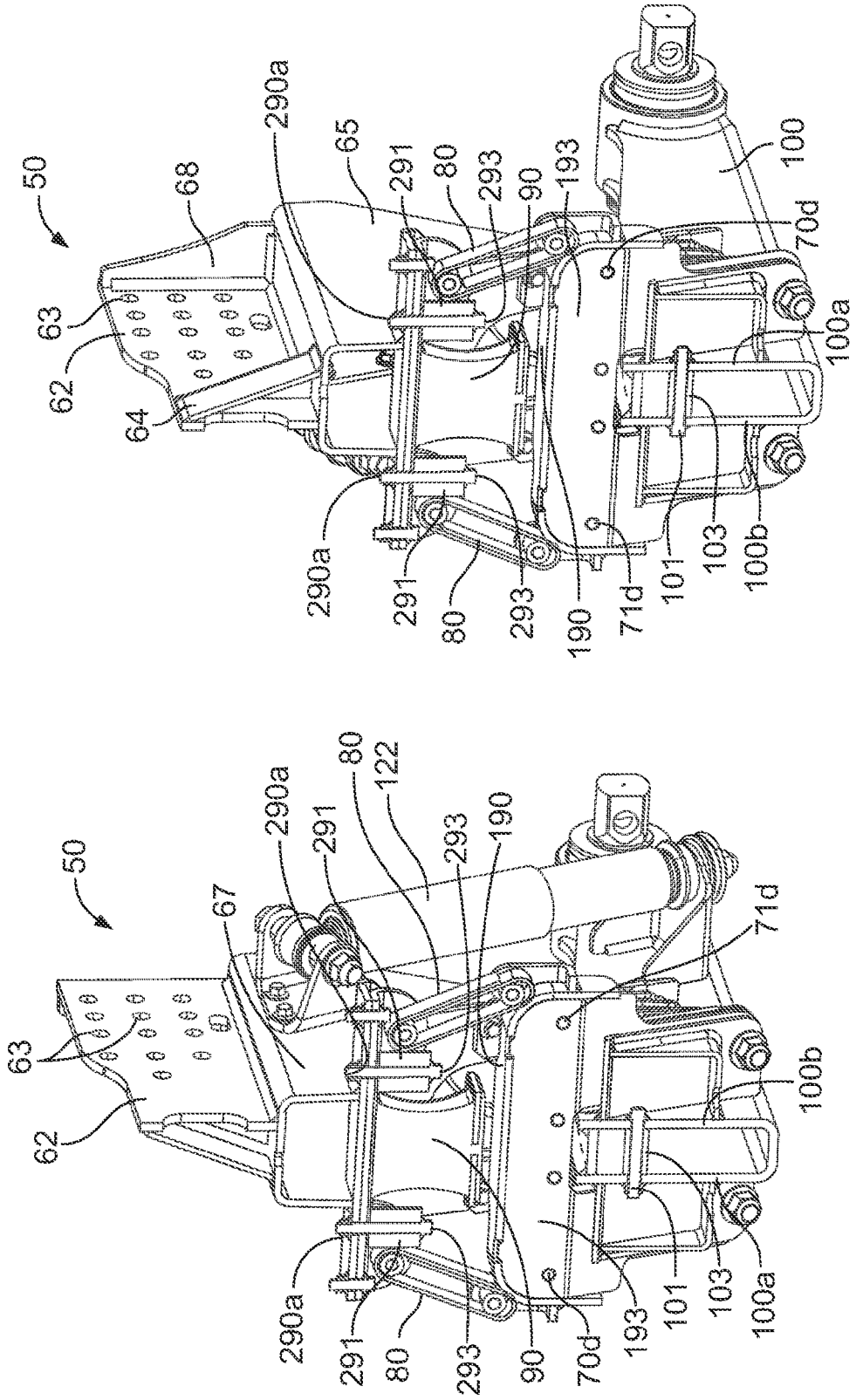


FIG. 22B

FIG. 22A

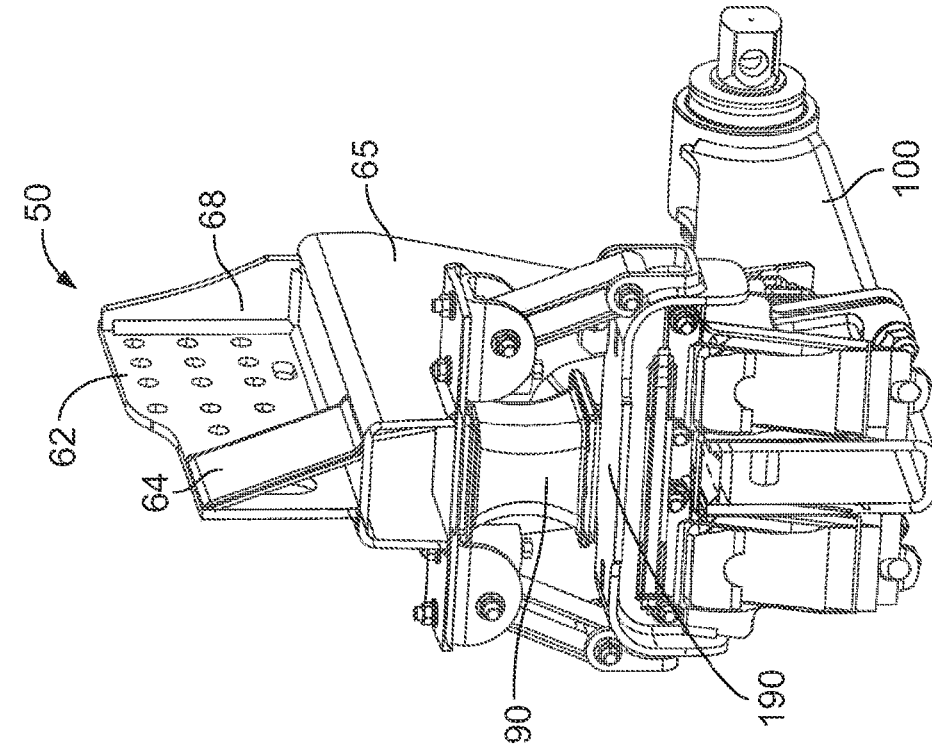


FIG. 23A

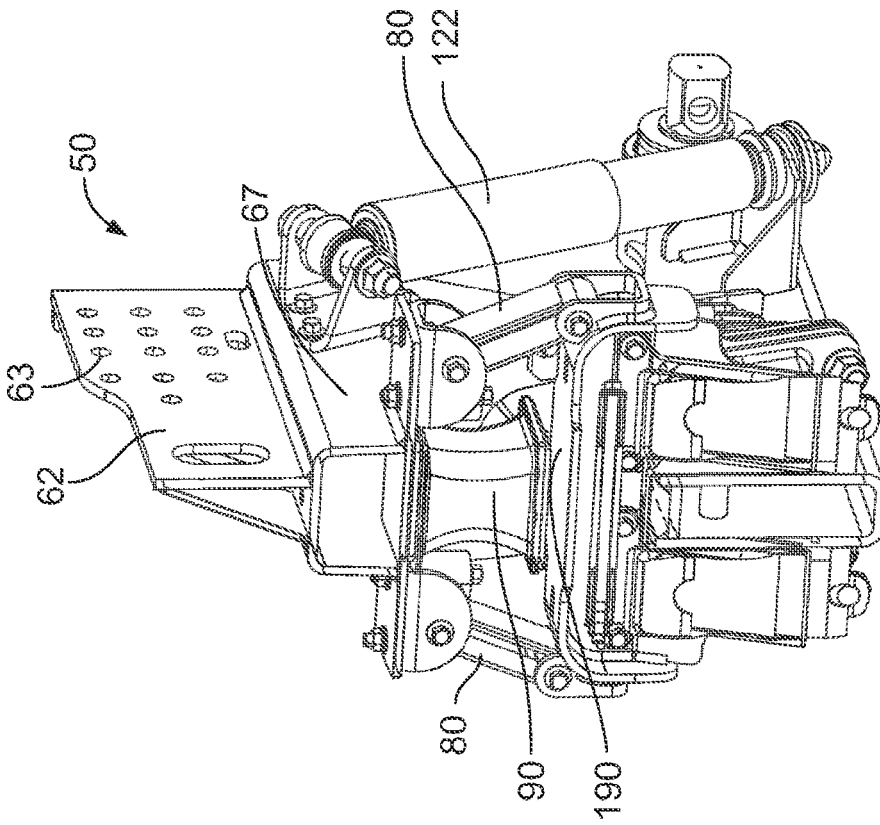


FIG. 23B

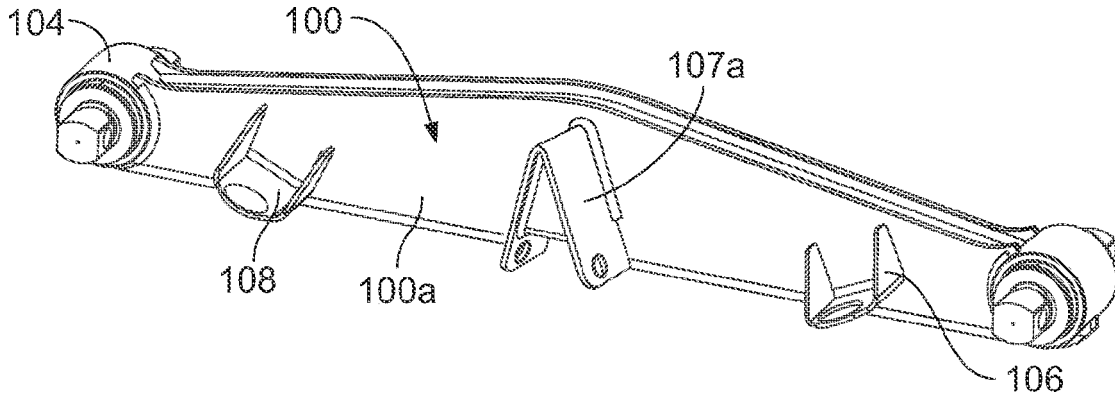


FIG. 24

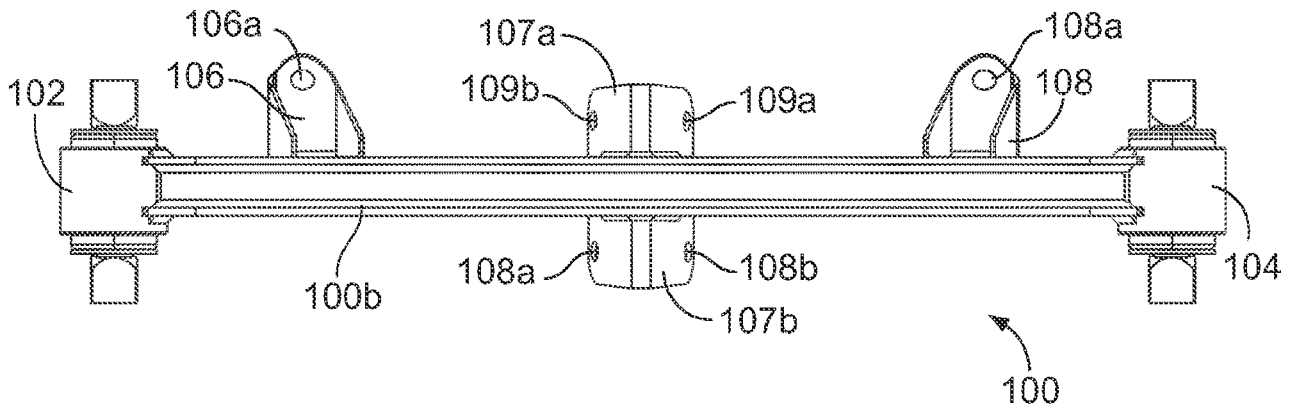


FIG. 25

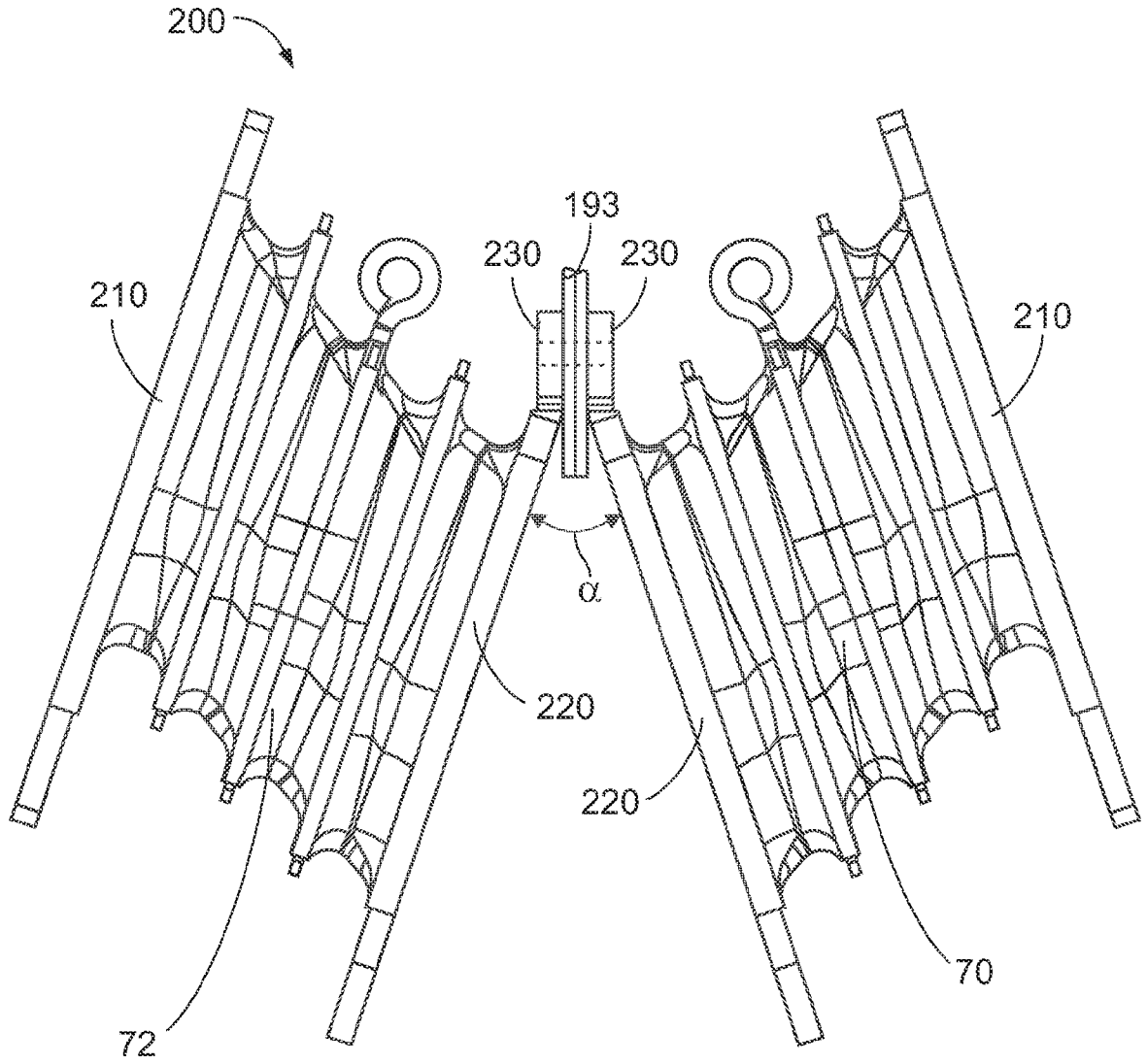


FIG. 26