Oct. 20, 1959 T. O. HENRIKSON 2,909,292

HYDRAULIC DRAFT GEARS

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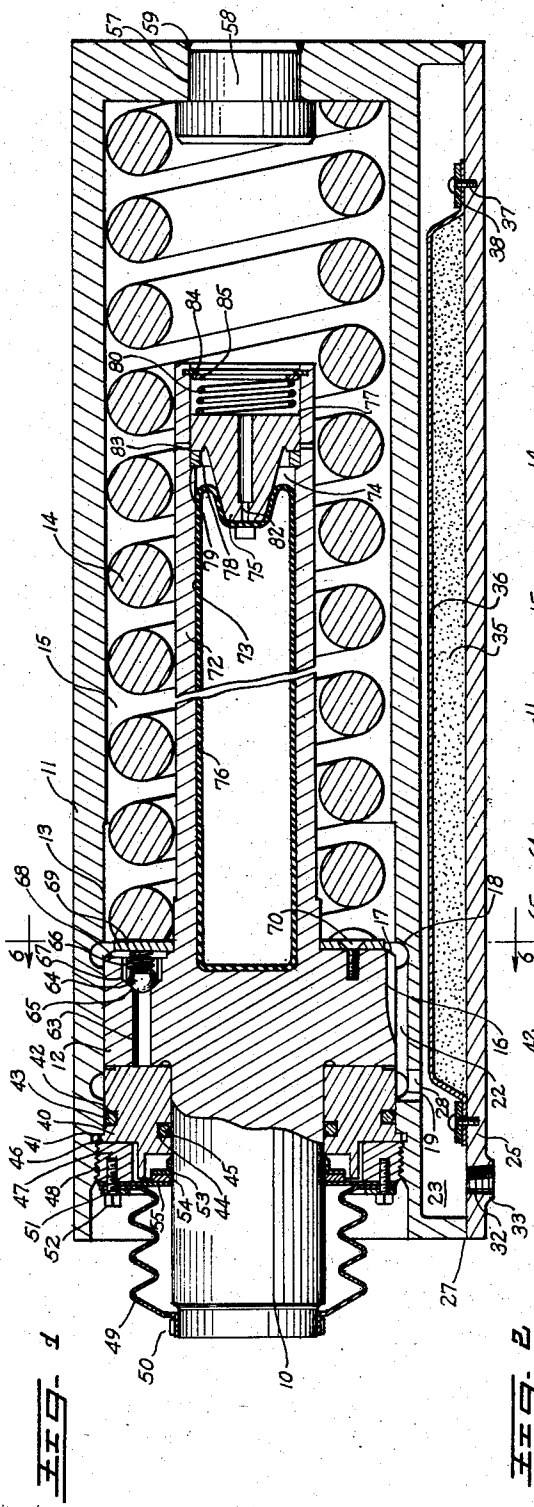

FIG. 1

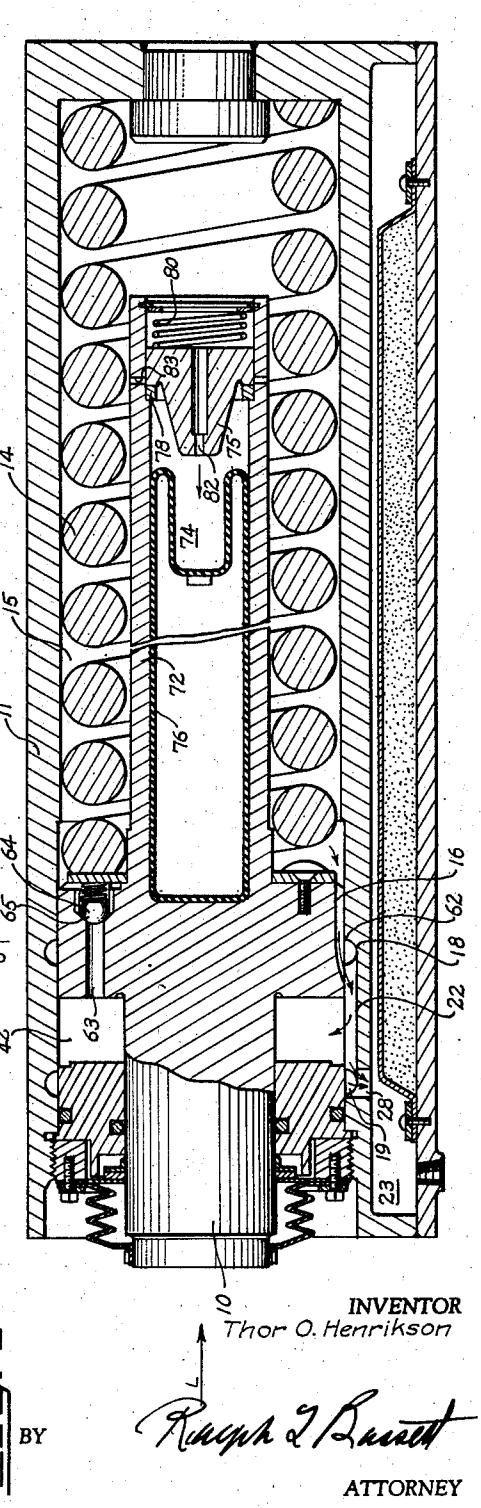

FIG. 2

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HYDRAULIC DRAFT GEARS

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FIG. 3

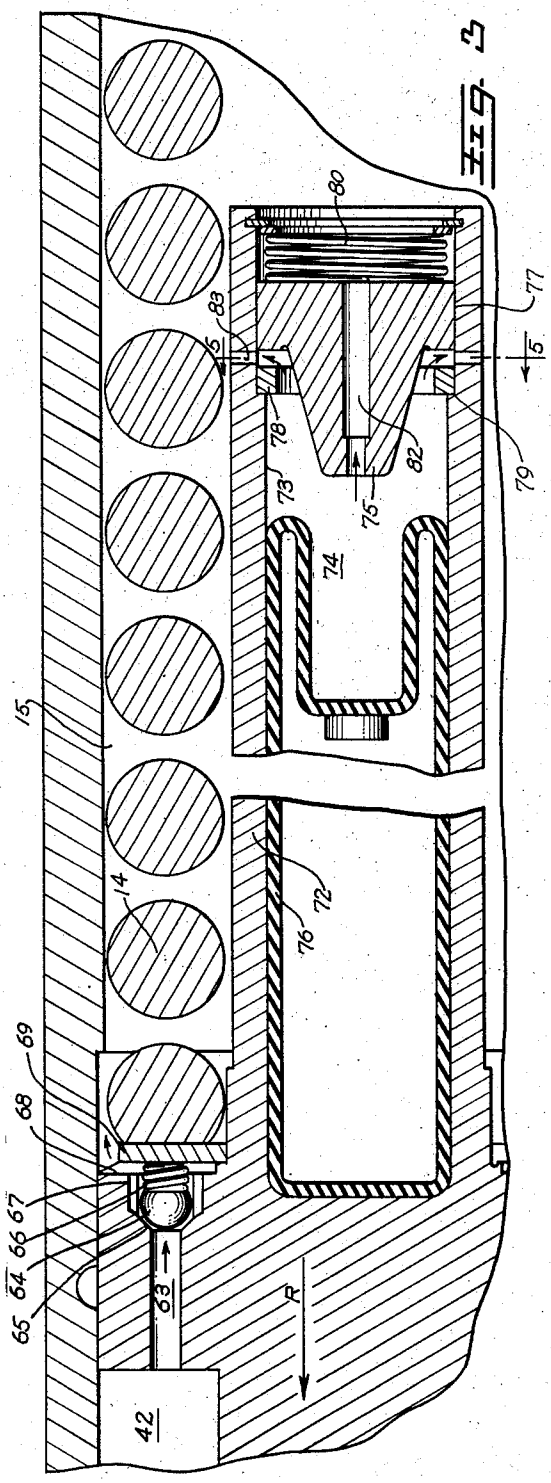


FIG. 5

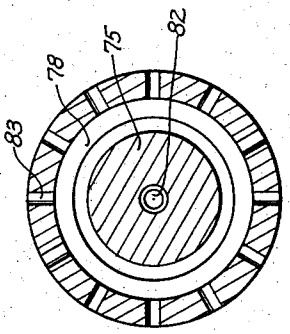
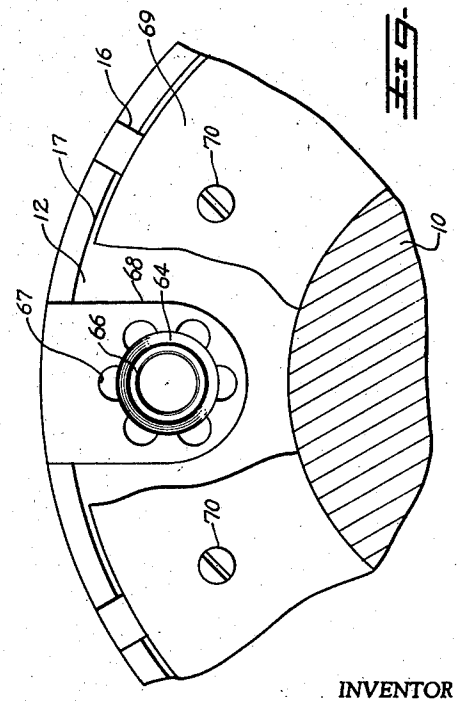


FIG. 4



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HYDRAULIC DRAFT GEARS

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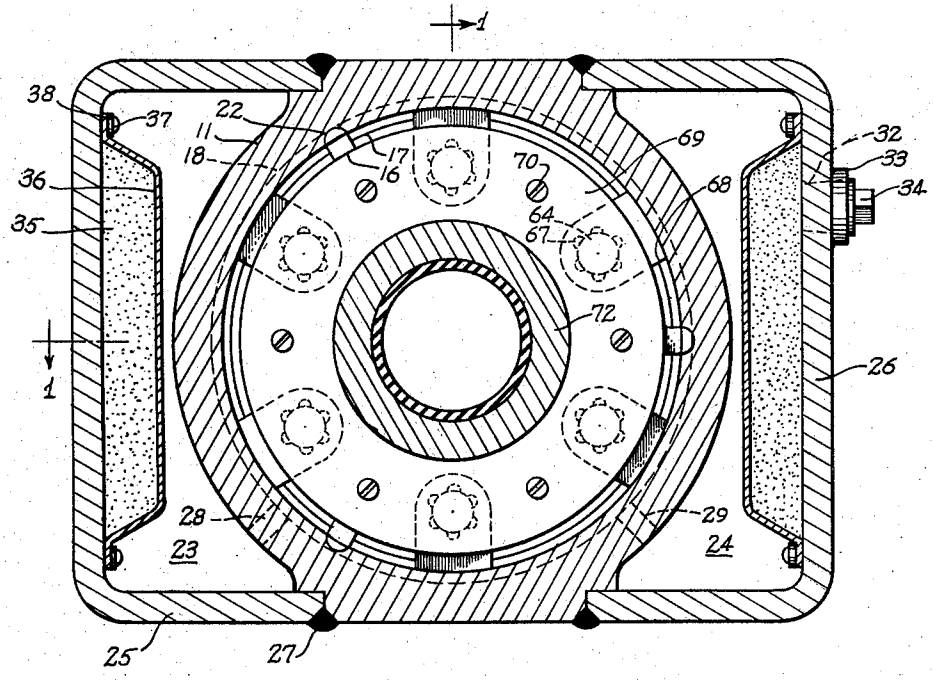

Fig. 6

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FIG. 7

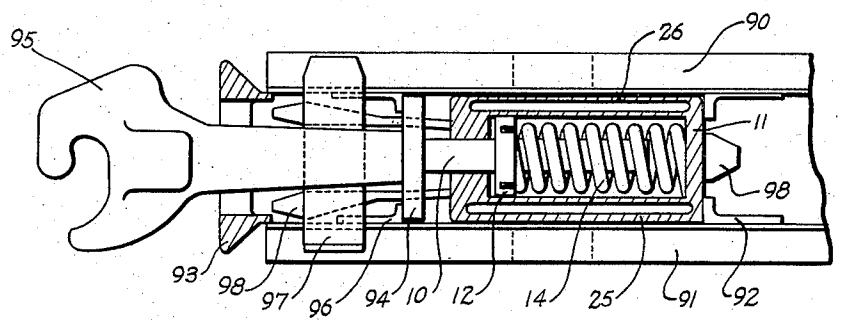
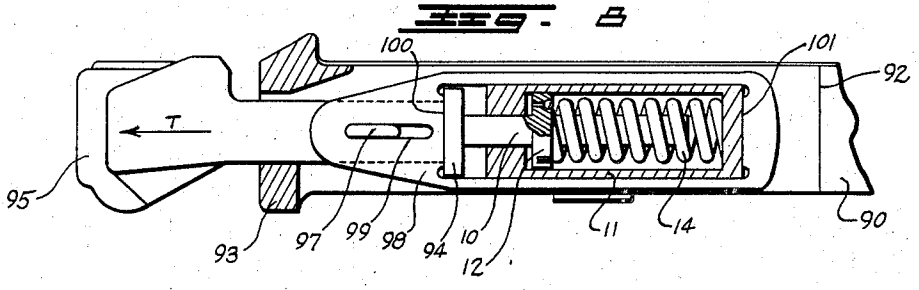


FIG. 8



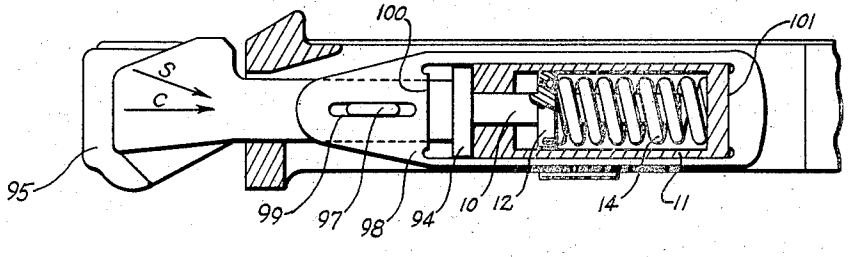


FIG. 9

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FIG. 10

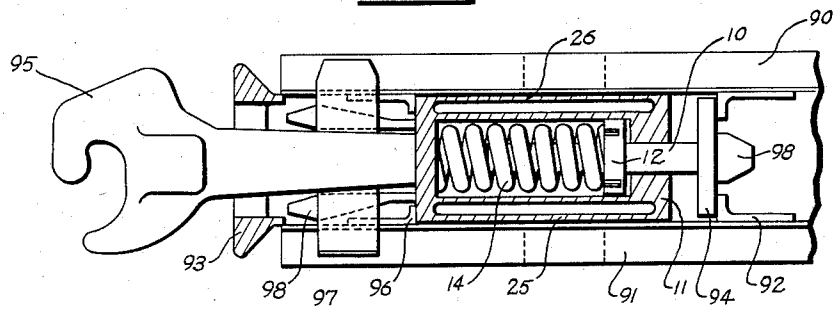
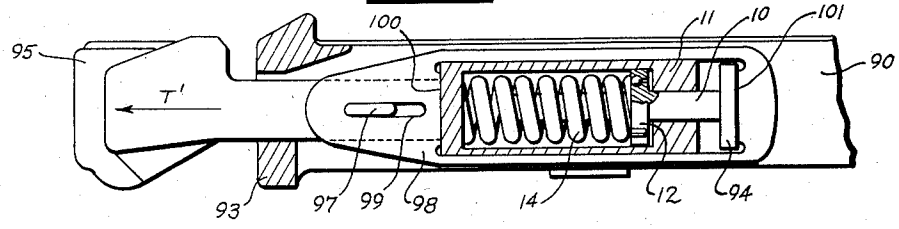


FIG. 11



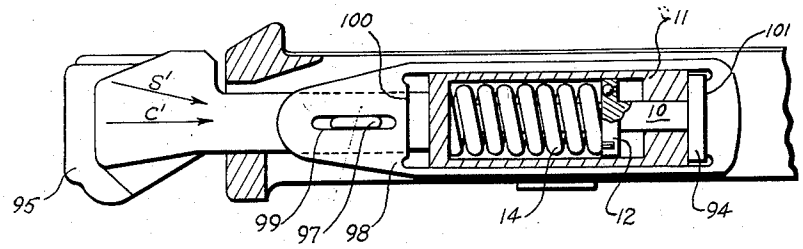


FIG. 12

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HYDRAULIC DRAFT GEARS

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Application June 6, 1956, Serial No. 589,684

7 Claims. (Cl. 213—43)

This invention relates to a hydraulic draft gear, or shock absorber, primarily for use in connection with railroad car couplings or the like and suitable for interchange with standard friction or rubber type draft gears, although it is of general utility as a heavy-duty shock absorber in many applications. This invention is particularly directed to a shock-absorber of the type in which shock loads are absorbed in part by throttling of hydraulic fluid under pressure through passageways, and in part by compression of a spring axially disposed in a cylinder so as to oppose axial sliding of a piston.

It is the general object of this invention to provide an improved draft gear of this type combining maximum ability to absorb shock loads with increased rapidity of return of the parts to a normal position after removal of an applied load; and to provide an improved construction and combination of parts affording improved performance with greater protection against accidental injury.

In a shock-absorber of this type, fluid throttled through passageways formed in the piston passes into a chamber formed by that portion of the cylinder bore lying behind the piston. This rear chamber also receives a piston-rod axially. Since a portion of the rear chamber is occupied by the piston-rod, its volume increases in proportion to the movement of the piston and the difference in cross-sectional area of the bore and the piston-rod; while the volume of the chamber in the cylinder lying ahead of the piston decreases in proportion to the entire cross-sectional area of the bore and the movement of the piston. The difference in change of volume between the two chambers may be accommodated by provision of suitable reservoirs for receiving the surplus fluid. Such reservoirs have previously been provided in the form of a free piston cooperating with a cylinder containing pressurized gas, such that the surplus fluid was accommodated by a movement of the free piston further compressing the confined gas, as in the patent to Welch, No. 880,257. A reservoir of this type is shown in the Welch patent, formed within the piston of a draft gear with a fluid entry port formed in the piston face confronting the compressed fluid lying ahead of the piston. In order to permit rapid return flow of the fluid contained in the reservoir upon removal of a load from the piston-rod, and consequent rapid return of the piston to an unloaded position, the fluid entry port has been made of a sufficiently large diameter in proportion to the piston diameter as to prevent the application of a throttling effect to fluid entering the reservoir which would materially impede the piston stroke, and throttling has been primarily confined to the passageways extending through the piston and joining the two fluid chambers formed by the cylinder and the piston.

It is an object of this invention to provide restricted orifice means for throttling fluid entering an expansible excess-fluid reservoir in a hydraulic draft gear during a shock-absorbing stroke of the piston, combined with valved port means to permit rapid escape of the fluid from the reservoir upon release of the load from the draft gear, and a consequently rapid return stroke of the piston to its original position.

Throttling passageways previously provided in draft gears of this type, extending through the piston and connecting the cylinder chambers separated by the piston, have been made of such a small diameter in proportion to the piston diameter as to secure a substantial throttling effect, and have not permitted rapid return flow of the fluid during the return stroke because of this relatively small size and concomitant throttling effect. Consequently, the return stroke, as well as the shock-absorbing stroke, has been impeded by throttling of the fluid through these passages.

It is a further object of this invention to provide return-flow ports in the piston of a hydraulic draft gear of this type, of such a large diameter relative to the piston diameter as to produce little throttling effect and a rapid return flow during the return stroke, together with one-way valve means for the return-flow ports, and relatively small throttle passageways producing a more greatly throttled and consequently slower flow during the shock-absorbing stroke than during the return stroke.

Draft gears of the present type have been provided with a main hydraulic fluid reservoir comprising a double-walled chamber partially surrounding the working cylinder circumferentially, and connected by ports with the cylinder chamber, as in the patent to Cotton No. 1,614,657. A screw-capped filler hole may be provided in the external wall of this reservoir for the purpose of filling the fluid chambers of the draft gear. This filler hole is generally so placed in the outer wall as to prevent complete filling of the reservoir, so that some of the surplus fluid passing into the rear chamber of the cylinder may be accommodated therein, and so that excessive pressures occurring through mechanical failures or other accidental factors will not produce destructive stresses on the walls of the main reservoir, but will be absorbed by compression of the air entrapped in the main reservoir. However, human error may nevertheless result in complete filling of the main reservoir with hydraulic fluid, with consequent danger of destructive pressures occurring.

It is accordingly a further object of this invention to provide compressible means for absorption of excessive fluid pressures accidentally occurring within the main reservoir of a draft gear. These means comprise compressible pads of cellular solid material with a fluid-impervious surface covering, placed within the main reservoir to absorb excess fluid pressure caused by overfilling or other accidental factors.

It is a still further object of this invention to provide an improved mounting means in combination with a draft gear of this type, having improved resistance to cocking and galling of the piston within the cylinder under the influence of shock loads applied in a non-axial or skew direction by a coupler associated with the draft gear.

Referring to the drawings:

Fig. 1 is a longitudinal section of a recommended embodiment of the present invention, with parts shown in normal or no-load position, taken along line 1—1 of Fig. 6;

Fig. 2 is a similar longitudinal section of the same device, with parts shown in their relative positions when the device is subjected to a load;

Fig. 3 is an enlarged view of a partial section of the same device, with parts shown in their relative positions during a return stroke from the position of Fig. 2 to that of Fig. 1;

Fig. 4 is a partial sectional cutaway view of the piston member taken along line 6—6 of Fig. 1;

Fig. 5 is an axial section taken along line 5—5 of Fig. 3;

Fig. 6 is an axial section taken along line 6—6 of Fig. 1;

Fig. 7 is a schematic plan view of a first mounting means for the draft gear, constructed according to this invention, with parts shown in normal or no-load position;

Fig. 8 is a schematic elevation of the mounting means of Fig. 7;

Fig. 9 is a schematic elevation similar to Fig. 8, with parts shown in loaded position;

Fig. 10 is a schematic plan view of an alternative improved mounting means for the draft gear, with parts shown in normal or no-load position;

Fig. 11 is a schematic elevation of the mounting means of Fig. 10; and

Fig. 12 is a schematic elevation similar to Fig. 11, with parts shown in loaded position.

In Fig. 1, piston 10 having annular flange 12 is placed in axially slidable relationship within bore 13 of cylinder 11, and is adapted to be driven by a shock-load in the direction shown by arrow L of Fig. 2, against the resilient bias of compression spring 14 axially disposed within forward chamber 15 of cylinder 11.

A pair of main fluid reservoirs 23, 24 are formed about cylinder 11 by means of channel members 25, 26, welded to cylinder 11 as at 27. Threaded filler holes formed in walls 25, 26 and bosses 33 are placed intermediate the extremities of the reservoirs 23, 24 to permit only partial filling of the draft gear with hydraulic fluid, and are provided with threaded caps 34.

To furnish means for absorption of excess fluid pressures which may accidentally occur in the reservoirs 23, 24 through overfilling or other factors, pads 35 of cellular material, preferably neoprene foam, are placed therein. Covers 36 of a suitable fluid-impervious material are placed over pads 35 and secured to walls 25, 26 by means of screws 37 and gasket 38 to prevent saturation of pads 35 with hydraulic fluid. In the event that excess pressure should develop, pads 35 will be compressed sufficiently to prevent disruptive stresses in reservoirs 23, 24.

An annular cap 40 is placed within the open end of cylinder 11 and about piston 10, enclosing a rear chamber 42 and rendering it fluid-tight. Cap 40 is provided with annular grooves 42, 44 and annular seals 43, 45 cooperating with the surfaces of cylinder 11 and piston 10 in fluid-sealing relationship. Cap 40 is secured in place by means of threaded ring 47 engaging threads 48 formed in cylinder 11 and bearing against flange 46 of cap 40, which abuts on shoulder 41 formed in cylinder 11.

A collapsible bellows seal 49 is placed over cap 40 to prevent the entry of dirt and dust into the draft gear. Bellows seal 49 is secured to piston 10 by means of clamping ring 50, and to ring 47 by means of gasket ring 51 and screws 52 assembled in tapped holes formed in ring 47. An annular oil seal 53, 54 may be placed about piston 10 and axially secured by means of annular plate 55, secured to ring 47 by means of screws 52, gasket ring 51, and bellows seal 49.

Cylinder 11 may be initially formed with an axial hole 57 at its closed end, for convenience in manufacture. Hole 57 is closed and sealed by means of plug 58, welded as at 59 to cylinder 11.

Flange 12 of piston 10 is provided with a series of circumferentially spaced longitudinally disposed tapered passageways 16 and may be further provided with chamfer 17. A pair of longitudinally spaced annular grooves 18, 19 and a group of circumferentially spaced longitudinally disposed grooves 22 are formed in the walls of bore 13. Grooves 18 and 22 cooperate with groove 16 to provide fluid passageways of combined cross-sectional area. A pair of radial ports 28, 29, as shown in Fig. 6, are formed in cylinder 11 and provide communication between main reservoirs 23, 24 and grooves 19.

Upon application of a shock load to piston 10, it moves in the direction of arrow L in Fig. 2. Fluid pressure in

chamber 15 causes a throttling flow through tapered passageways 16, and further flow through grooves 22 into rear chamber 42, as shown by arrows in Fig. 2. This throttling flow absorbs the shock load in conjunction with spring 14, and applies it slowly and uniformly to cylinder 11. A portion of the throttled fluid flows from grooves 22 through groove 19 and ports 28, 29 into main reservoirs 23, 24, compressing any air trapped therein. As piston 10 moves toward the position of Fig. 2, and flange 12 slidably engages bore 13, groove 18 is covered thereby, so that fluid is thereafter throttled through a minimum orifice equal to the cross-sectional area of that portion of tapered passageways 16 adjacent shoulder 62, formed by bore 13 and groove 18. Tapered passageways 16 may be so formed that their effective cross-sectional areas at any position of piston 10 render the pressure obtaining in chamber 15 constant at all positions; thus, the force absorbed by cylinder 11 will remain substantially constant throughout the stroke. A constant application of force minimizes the peak force absorbed, and thus represents an ideal condition, as is well known in the art.

In order to assist rapid return of piston 10 in the position of Fig. 1 after the removal of a shock load, a series of return-flow ports 63 of large cross-sectional area relative to passageways 16 are formed in flange 12, connecting chambers 42 and 15, and are circumferentially interspersed between grooves 16, as shown by Fig. 6. One-way valve members are placed in these ports to prevent unthrottled flow therethrough during the shock-absorbing stroke, while permitting free flow during the return stroke. These valves comprise balls 64, yieldingly urged into engagement with conical seats 65, formed about ports 63, by means of compression springs 66. As best shown in Fig. 4, a series of grooves 67 of circular section are formed circumferentially about ports 63, affording communication between ports 63 and chamber 15 when balls 64 are displaced from seats 65. An annular plate 69 is secured to the forward face of flange 12 by means of screws 70, in order to provide a seat for compression springs 66. U-shaped grooves 68 are formed in the forward face of flange 12 about the ends of ports 63, to provide communication between ports 63 and chamber 15 behind plate 69.

As a shock load is applied to piston 10, tending to drive it in the direction of arrow L in Fig. 2, balls 64 are forcibly seated upon conical seats 65 by the increased fluid pressure obtaining in chamber 15. Thus, during the shock-absorbing stroke, fluid flow from chamber 15 is confined to throttling passageways 16 and the grooves connected therewith, and does not occur in ports 63.

Upon removal of the load from piston 10, potential energy stored in compressed spring 14 tends to return piston 10 in the direction of arrow R in Fig. 3. As the pressure obtaining in rear chamber 42 approaches and then surpasses that obtaining in chamber 15, balls 64 are lifted from seats 65 against the bias of spring 66, and a rapid return flow occurs from chamber 42 through ports 63, grooves 67, and grooves 68 to chamber 15, as shown by arrows in Fig. 3, as well as through grooves 22 and 28 and throttling passageways 16.

As previously discussed, fluid displaced from chamber 15 during the shock-absorbing stroke of piston cannot be entirely accommodated by chamber 42, because the decrease in volume of chamber 15 is proportional to the cross-sectional area of bore 13, while the corresponding increase in volume of chamber 42 is in proportion only to the difference in cross-sectional areas of bore 13 and piston 10. Only a small portion of the excess fluid displaced is accommodated within main reservoir 23, by compression of air entrapped therein.

In order to accommodate the remaining excess fluid displaced, an annular axial extension 72 having internal bore 73 is formed integrally with piston 10, internally

containing expansible excess-fluid reservoir **74**. A compressible accumulator bag **76**, flexible and impervious to hydraulic fluid, is placed within bore **73** and filled with gas to a predetermined pressure. Fluid received within excess-fluid reservoir **74** is accommodated by compression of accumulator bag **76**. Reservoir **74** is enclosed by means of valve member **75**, received in axially slidable relationship within enlarged bore **77** at the open end of bore **73**. Valve member **75** bears upon an annular ring **78** seated upon shoulder **79** formed by the junction of bores **73** and **77**, and is resiliently biased into engagement therewith by means of compression spring **80**, for a purpose further to be described. Compression spring **80** is retained within bore **77** by means of an annular ring **84** and snap-ring **85**, which cooperates with a suitable groove formed in the wall of bore **77**.

A throttling orifice **82** is formed axially through valve member **75**, connecting expansible reservoir **74** with chamber **15**. A series of return-flow ports **83** are radially disposed in the walls of bore **77**, communicating with chamber **15**, and are normally closed by valve member **75** when in the position of Fig. 1.

During the shock-absorbing stroke a portion of fluid displaced from chamber **15** is throttled through port **82** into expansible reservoir **74**, with consequent absorption of force, as shown by arrows in Fig. 2. Accumulator bag **76** is compressed by the fluid entering reservoir **74** until the pressure of the gas confined therein is equal to the pressure of the fluid. Valve member **75** remains seated upon ring **78** during the shock-absorbing stroke, under the bias of spring **80** and the pressure of fluid in chamber **15**, and thus closes ports **83** and prevents any unthrottled flow therethrough.

Upon removal of the applied load from piston **10**, it is driven in the direction of arrow R in Fig. 3 by compressed spring **14** and by the pressure of the fluid in chamber **74**. A rapid return flow of fluid from expansible reservoir **74** to chamber **15** is aided by the compressed gas confined within accumulator bag **76**. Ports **83** are exposed to expansible reservoir **74** during the return stroke by an axial movement of valve member **75** away from ring **78**, resulting when the relatively higher fluid pressure obtaining in reservoir **74** than in chamber **15** is sufficient to overcome the resilient bias of spring **80**. Return flow from reservoir **74** to chamber **15** takes place through ports **83** as well as through orifice **82**, as shown by arrows in Fig. 3. Ports **83** are sufficiently large in number and cross-sectional area, as best seen in Fig. 5, to provide for a highly increased rate of flow relative to that obtainable through orifice **82**, and prevent any materially flow-restraining amount of throttling of the fluid passing into chamber **15** from reservoir **74**. Valve member **75**, orifice **82**, and ports **83**, with their associated means, thus cooperate to provide a slower and more highly throttled flow from chamber **15** to reservoir **74** during the shock-absorbing stroke than during the return stroke, which is characterized by a relatively rapid and unthrottled flow in the reverse direction with a consequent rapidity of return of piston **10** to its unloaded position.

Improved mounting means for the draft gear of this invention are shown in Figs. 7, 8 and 9, and a modification representing a further improvement is shown in Figs. 10, 11 and 12. In Figs. 7 and 8, schematically showing the mounting means in plan and elevation views, respectively, the body of the draft gear comprising cylinder **11** and channel members **25** and **26** abuts members **92**. Flanged members **92** are affixed to car structure beam members **90**, **91**, serving to transfer the load from the draft gear to the car structure. The internal members of the draft gear are shown schematically, including spring **14**, piston **10**, and flange **12** cooperating with cylinder **11** in axially slidable relationship. Piston **10** contacts a rectangular follower plate **94**. A knuckle coupler **95**, or other means subject to shock loading, contacts follower plate **94** and is adapted to transmit applied forces thereto, regardless of direction.

A pair of flanged members **96** are rigidly affixed to beam members **90**, **91** and act as stops for follower plate **94**, which abuts thereon when coupler **95** is subjected to a load in tension, as shown by arrow T in Fig. 8.

A yoke member **98** contacts face **101** at the closed end of cylinder **11**, and is provided with slot **99** cooperating with a key **97** affixed to, and transfixing, coupler **95**. When coupler **95** is subjected to a load in tension, as shown by arrow T in Fig. 8, key **97** is drawn into abutment with the forward end of slot **99**. Force T is transmitted by key **97** through yoke **98**, cylinder **11**, piston **10**, spring **14**, follower **94**, and flanged members **96** to beam members **90**, **91** delivering a shock-absorbing action to the car structure.

Application of a load in compression, as shown by arrow C in Fig. 9, drives coupler **95**, follower plate **94**, piston **10**, flange **12**, and compression spring **14** to the right and produces a shock-absorbing action of the draft gear, as previously described. Absorbed force is transmitted by cylinder **11** through flanged members **92** to beam members **90**, **91**.

An improvement in resistance to cocking of piston **10** and consequent galling or scoring of cylinder **11** resides in the combination and arrangement of parts described in the present draft gear. Under the influence of a skew load, as represented by arrow S in Fig. 9, exerted in any direction other than axially of the draft gear, coupler **95** tends to twist piston **10** and flange **12** in cylinder **11**. Such skew loads commonly occur in practice, as when the vehicles associated with coupler **95** are upon a curved or an irregularly inclined track. The provision and location of follower plate **94**, axially slidable between beam members **90**, **91** and in axially spaced relation with flange **12**, provides improved resistance to cocking, because the moment arm acting to resist torsion is equal to the axial distance between the oppositely disposed faces of follower plate **94** and flange **12**, rather than the relatively small axial thickness of flange **12** alone.

In the arrangement of Figs. 10, 11 and 12, the draft gear is reversed relative to the vehicle undercarriage frame, achieving a still further improvement in resistance to cocking of piston **10** and flange **12** in cylinder **11**. Follower plate **94** contacts flanged members **92**, yoke **98**, and piston **10**. Cylinder **11** and channel members **25**, **26** are placed in axially slidable relationship within beam members **90**, **91**, and abut flanged members **96** under the influence of a load in tension, as shown by arrow T in Fig. 11. Operation of the draft gear and mounting means under the influence of loads in tension or compression, as shown by arrows T' in Fig. 11 and arrow C' in Fig. 12, respectively, is the same as for the arrangements of Figs. 7 and 9, with the exception that cylinder **11** now assumes the former functions of follower plate **94**, and vice versa.

Upon application of a skew load, as represented by arrow S' in Fig. 12, exerted in any direction other than axially of the draft gear, coupler **95** tends to twist cylinder **11** relative to piston **10** and flange **12**. The association of cylinder **11** in rigid assembly with coupler **95** results in application of the twisting moment through the walls of cylinder **11** to beam members **90**, **91**, with which cylinder **11** is associated in axially slidable relationship. Because the moment arm acting to resist torsion is now equal to the entire axial length of cylinder **11**, which length may be greater than the moment arm of the previous arrangement, a still further improvement in resistance to cocking of cylinder **11** relative to piston **10** and flange **12** is obtainable.

What I claim is:

1. In a hydraulic draft gear comprising a piston and a cylinder assembled for relative axial movement, a piston rod projecting from the front face of the piston through the cylinder wall and movable with the piston

within the cylinder, a spring in said cylinder abutting the inner opposite face of the piston and normally urging said piston towards the front end of the cylinder, front and rear fluid confining chamber formed in said cylinder and separated by said piston, ports extending through said piston and connecting said chambers, normally seated valve means for preventing flow of fluid through said ports upon movement of the piston toward said front face under shock absorbing strokes but permitting free flow of fluid during return stroke, tapering throttling passages formed in the peripheral surface of said piston in communication with the cylinder, grooves formed in the wall of said cylinder communicating with said rear chamber and with said throttling passageways to permit fluid flow through said tapering throttling passageways and said groves to be varied by relative axial movement of the piston and the cylinder, a recess formed within the inner face of said piston to provide a fluid reservoir within the piston, a passageway forming an entrance to said reservoir, a valve normally closing said passageway during the power stroke of the piston regardless of pressure, and an orifice opening into said fluid reservoir to allow the flow of surplus fluid into said reservoir in said piston in proportion to the displacement of said piston rod as the latter enters the front cylinder upon the return stroke of said piston.

2. The structure of claim 1 characterized in that the fluid reservoir within the piston is formed in an annular extension projecting from the rear face thereof.

3. The structure of claim 1 characterized in that the fluid reservoir within the piston is formed in an annular extension projecting from the rear face thereof, said reservoir containing an accumulator bag for compression upon entry of fluid under pressure into the reservoir.

4. The structure of claim 1 characterized in that the

valve normally closing said passageway during the power stroke of the piston regardless of pressure is of cylindrical form and is slidably associated with ports formed in the lateral wall defining the reservoir within the piston.

5. The structure of claim 1 characterized in that the valve normally closing said passageway during the power stroke of the piston regardless of pressure is spring seated and of cylindrical form and is slidably associated with ports formed in the lateral wall defining the reservoir within the piston.

6. The structure of claim 1 characterized in that the orifices opening into the fluid reservoir to allow the flow of surplus fluid into the reservoir in the piston is formed in the valve normally closing the passageway during the power stroke.

7. The structure of claim 1 characterized in that a main reservoir is provided exteriorly of the cylinder assembly and is in communication with the front fluid confining chamber formed in the cylinder, said reservoir containing absorption pads for accommodating excess fluid pressures.

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