Aug. 19, 1969 E. H. HARTEL 3,462,024

DRAFT GEAR

Filed May 16, 1967 2 Sheets-Sheet 1

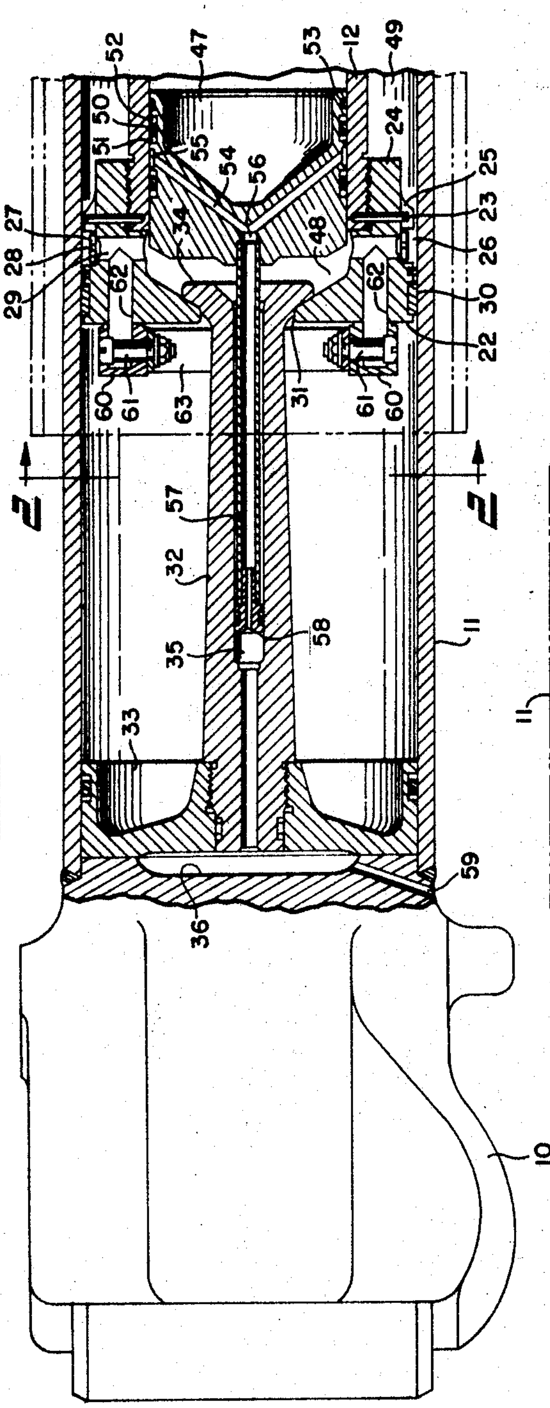

FIG. 1

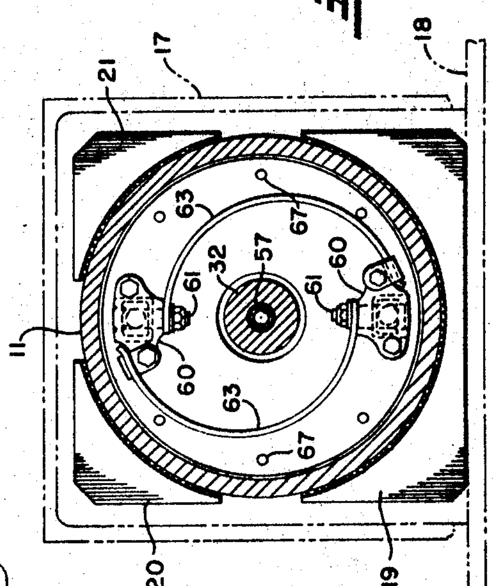

FIG. 2

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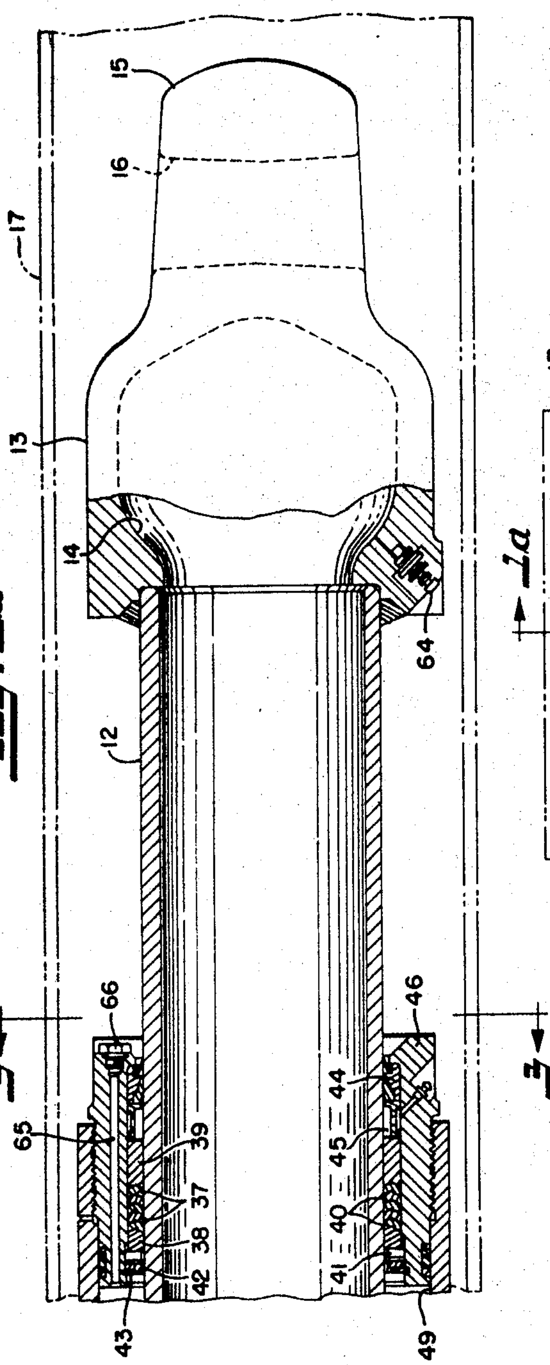

Fig. 1a

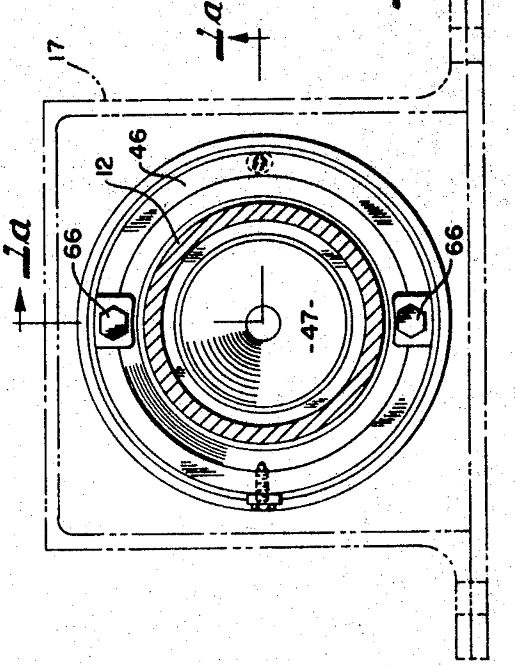

Fig. 3

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DRAFT GEAR

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U.S. Cl. 213—8 12 Claims

ABSTRACT OF THE DISCLOSURE

A draft gear in the form of a hydropneumatic shock absorber providing end-of-car cushioning in draft as well as in buff. The gear dissipates energy through dynamic damping during buff action by forcing oil at high velocity through a continuously varying orifice, and an air charge provides a preload in both directions and serves also to return the unit to neutral after a stroke in either direction. During such return, the oil flows oppositely to the flow during buff at a valve controlled rate to prevent sudden extension. A special relief-valve arrangement affords overload protection in the event of harder than normal impact during buff.

This invention relates to hydropneumatic draft gear for railroad cars.

It is a primary object of the invention to provide an improved gear for hydropneumatic cushioning at high efficiency in a unit that will fit the standard freight car draft gear pocket. Another object is to provide such draft gear which is fully hydropneumatic, to the exclusion of mechanical springs and thus distinguishable from known impact energy absorbing mechanisms for railway vehicles in which such springs are employed in conjunction with liquid cushioning means.

It is another object to provide such hydropneumatic end-of-car cushioning in both buff and draft, thereby greatly to minimize problems of train dynamics.

It is an additional object to provide such draft gear in which the energy of impact is dissipated by forcing of the oil at high velocity through a continuously varying orifice against an air spring, with valve control of the flow of oil in the opposite direction providing a predetermined snubbing action in the return of the unit to the neutral position after the impact has been absorbed and in the operation of the gear in draft. Rapid extension if the unit is thus prevented in these conditions which might produce unwanted reaction forces on the car. Such operating mode in draft provides virtually "solid-train" operation, thereby minimizing the increase in train length which has been experienced and found objectionable in many conventional end-of-car cushioning devices.

Another object is to provide such hydropneumatic draft gear including improved relief valve protection against overload, for example, in the event of greater than normal impact.

It is also an object of the present invention to provide such draft gear in a simple and self-contained design of high structural integrity and requiring no external connections or accessories.

Other objects and advantages of the present invention will become apparent as the following description proceeds.

To the accomplishment of the foregoing and related ends the invention, then, comprises the features hereinafter fully described and particularly pointed out in the claims, the following description and the annexed drawings setting forth in detail certain illustrative embodiments of the invention, these being indicative, however, of but a few of the various ways in which the principle of the invention may be employed.

In said annexed drawings:

FIGS. 1 and 1*a* are longitudinal sections which together show a full draft gear in accordance with the present invention:

FIG. 2 is a transverse cross-sectional view of the plane of which is shown by the line **2—2** in FIG. 1; and

FIG. 3 is a further transverse section as viewed from the plane of the line **3—3**.

Referring now to the drawings in detail, the illustrative embodiment of the invention is a gear basically comprising a coupler head section **10**, which can be of any known type, a cylinder **11** welded to and extending from the coupler head, a hollow piston **12** within and projecting from the cylinder, and a piston fitting or cap **13**.

The fitting **13** is a forging or casting welded to the outer end of the hollow piston and having a cavity **14** which complements the latter in the sense of forming a closed and extension of the piston interior. This fitting is further formed with a horizontal lug **15** in which there is a vertical aperture **16** for pin connection of the gear to the railway car. The dashed outline **17** represents a standard freight car sill, and it will be appreciated that the gear is simply inserted in such a sill and supported in any suitable manner. A solid plate, such as shown by the further dashed outline **18**, which closes the sill bottom over the full length of the gear is recommended to protect the piston in particular from direct impingement of debris, and it is also preferred that antiturn blocks **19**, **20** and **21** be secured to the exterior of the cylinder to prevent rotation of the coupler. The bottom block **19** engages the hanger plate, while the other blocks **20** and **21** are separate angular projections respectively adjacent the upper corners of the sill.

A piston head **22** is threaded on the inner end of the hollow piston **12** and locked by one or more dowel pins **23** which extend radially through the internally threaded skirt part **24** of the piston head and into the piston. The periphery of this portion **24** is relieved by a multiplicity of circumferentially spaced axial slots **25**, cooperably with the inner surface of the cylinder **11**, and also a clear annular space **26** at the outer side of such slotted part. A ring valve **27** in the form of a split ring of significant axial extent and having a series of spaced apertures or orifices **28** encircles the portion of the piston head at the annular space **26** and communication from the latter to the interior of the piston is provided by a series of radial passages **29** through this portion of the head.

The piston head is supported for relative movement in the cylinder by a split bronze ring bearing **30** seated partially in a peripheral groove in the piston head, and a central metering orifice **31** is provided in the piston head for flow of hydraulic fluid from the cylinder to the interior of the piston. This flow is programmed by a metering pin **32** which extends from a secondary piston head **33** in a direction away from the coupler head **10** or inwardly of the gear with a progressively decreasing external diameter. Such pin, which is in sealed threaded engagement with the secondary piston head as shown, extends through the metering orifice **31** of the piston head and its free end **34** within the piston is flared to engage the head about the orifice and accordingly limit the relative extension of the pin. For a purpose to be later described, the pin is also provided with a through bore **35**, with a relief **36** provided in the face of the coupler head **10** which affords communication with the pin bore **35** even when the secondary piston is in the illustrated condition against such coupler face.

The hollow piston is sealed in the cylinder by a main gland preferably of chevron type and shown as comprising a series of rings 37 of V-shaped cross-section held in compression between machined bronze packing rings 38, 39. These rings or chevrons are interleaved with bronze or Teflon spacers 40 to maintain their shape under load, and the compressive load is applied by a wave spring 41. The assembly is held in place by a retainer washer 42 and a snap ring 43, with positive retention of the latter assured by machining a recess in the end of the retaining washer and thus making it necessary to physically compress wave spring 41 before snap ring 43 can be removed. The gland further includes a conventional wiper and scraper assembly 44 external to the chevron seal unit for removing any foreign material which might adhere to the extended piston, and a grease cavity 45 is incorporated between the chevron unit and the wiper ring to keep the outboard chevrons from drying. This is an added precaution to assure long seal life, since if the inboard chevrons seal effectively, the remaining rings will be found to run almost completely dry. It is preferred for this purpose to employ a light grease to minimize wear and operating friction, and the cavity 45 is fairly large in volume compared to the amount of the grease needed so that servicing at frequent intervals should not be required. The full assembly is contained in a gland nut 46.

A floating piston 47 is disposed in sealed relation within the hollow piston 12 to define a variable volume accumulator chamber 48 therein isolated from the end portion of the piston having communication through the metering orifice 31 with the main interior of the cylinder and also through the passages 29 and slots 25 with the annular chamber 49 about that portion of the hollow cylinder which is inboard of the main gland. As will be more fully explained below, the cylinder will be filled with oil, and a charge of pressurized air will be provided in the hollow piston chamber.

Dual seals are employed for sealing of the floating piston 47, with each preferably comprising a standard O-ring 50 in assembly with dual polytetrafluoroethylene backup rings 51 and a quad ring 52. These quad rings serve not only to preload the O-rings for most efficient sealing, but also provide two added sealing surfaces and thereby three lines of contact instead of the usual single contact in a conventional O-ring installation. The backup rings 51 are recommended to insure long life, since they prevent extrusion of the O-ring into the clearance gap during dynamic operation at high pressures. These backup rings are closely fitted to both the bore and the O-ring groove, with the polytetrafluoroethylene selected for long life and minimum friction. A felt wiper 53 is also used on the air side of the floating piston to maintain a thin oil film in the piston-accumulator bore, with this also helping to enhance long life and leak-free operation.

In addition, the floating piston 47 is provided with one or more passages 54 which extends from a peripheral relief 55 between the two seals commonly to an axial passage 56 from which a hollow pilot tube 57 extends into the main bore 35 of the metering pin 32. The pilot tube, which is of course movable relative to the pin, has a head 58 at its free end sealed to the bore. This tube and passage arrangement provides an atmospheric bleed between the seals completed by vent passage 59 from the coupler head relief 36 to the exterior, with the result that full charge pressure is always experienced by both to avoid the potential difficulty of providing leak-free sealing with essentially zero differential pressure.

The gear is also protected against overload, in the event of greater impact than normally expected, by a relief valve arrangement mounted on the piston head 22. This assembly comprises two valve blocks 60 against the end face of the head and spaced 180° apart. Each block contains a small stepped piston 61 controlling a large relief passage 62 extending to a radial passage 29 and thus in bypass relation to the head metering orifice 31. The pistons 61 are loaded by circular spring segments 63 designed to provide a specific load stroke curve over the required deflection, and the step dimensions on the pistons are established by the spring load and by the desired pressure for the relief. The springs 63 will be seen normally to maintain the pistons 61 in extended outward relation, with radial inward movement against the spring load resulting in the uncovering of the relief ports. The latter will be sized to accept the high flow rates that will be encountered at impacts in excess of the normal design value.

This new draft gear is filled and charged in the following manner. The gland nut 46 is backed off from contact with the cylinder end and a small air charge is introduced at a fitting 64 in the piston end 13, so that the accumulator will hold the unit in the neutral position during filling. The gland nut has two fill passages 65 closed by plugs 66, and the latter are removed, with oil supplied through one with the unit vertical and interior air displaced through the other until overflow occurs, the plugs then being reinstalled. The piston head can be provided with a plurality of same holes 67 to facilitate the filling operation.

The gland nut is then tightened against the cylinder end, and this action forces some of the oil through the metering orifice 31 and into the piston-accumulator, displacing the floating piston 47 from its original seated position against the primary piston head 22 and slightly compressing the air charge. This volume of oil serves not only to prevent physical contact of the pistons, but allows for thermal expansion and contraction during varying ambient temperature conditions and, moreover, provides a positive reservoir of oil to offset any small leakage that might occur. Until this excess volume of the oil is depleted, no amount of leakage will have any effect on the performance of the gear, and the reservoir volume is designed to be large enough so that frequent servicing will be unnecessary.

After this filling and the tightening of the gland nut 46, the unit is then charged to the full air pressure. However, if desired for handling, the unit can be retracted by reducing the charge and subsequently recharged for extension to the neutral position.

The operation of the gear in buff will now be described. When the coupler head 10 is struck, the cylinder 11 is forced to the right as shown in the drawings and this will of course be into the car sill. The cylinder will carry the secondary piston 33 and metering pin 32 in this movement, while the primary piston 12 and piston head 22 remain fixed relative to the car. Accordingly, all oil displaced by the motion of the secondary piston 33 must flow either through the small fixed fill orifices 67 in the piston head 22 or through the central controlled orifice 31. The contour of the metering pin 32 is of course originally established by calculation to provide high efficiency in energy absorption.

The displaced oil, after passing through the piston head 22, enters the end of the hollow main piston 12 and forces the floating piston 47 inwardly to compress the air charge. Part of the oil also flows outwardly through the relatively large radial passages 29, thereby forcing open the ring valve 27, so that the flow continues through the slots 25 around the head and increases the volume in the annular cavity 49. When the impact has been absorbed or the motion compression stopped by bottoming of the gland nut 46 against the piston end 13, the air pressure built up in the accumulator will extend the unit to the neutral position. During this return stroke, the oil flows from cavity 49 through the same orifices, but in the opposite direction, and snubbing action is provided by the ring valve 27 which will be held tightly by this flow against the bottom of the groove in which it is seated as a result of its inherent spring action and also the pressure of the oil flowing from the cavity. The orifices 28 in the ring valve are sized to provide a controlled rate for this return which will preclude any sudden extension causing unwanted reaction forces. This snubbing action in such proper degree largely eliminates the problems in train dynamics which can occur when the train has a number of spring loaded cushioning devices in series.

It should be noted that the draft gear is designed so that the gland nut **46** actually bottoms on the piston end or fitting **13** before any of the internal parts come into contact. In this manner, the load is transferred directly into the outer cylinder eliminating external bump stops required by most other cushioning devices. The operation of the relief mechanism has already been described.

With respect to the draft operation, as the coupler **10** is pulled out, it carries the cylinder **11** with it while the metering pin **32** and secondary piston **33** are, however, restrained by the primary piston **12** and do not move relative to the car frame. The volume of oil between the heads of the two pistons accordingly does not change, but the volume of annular cavity **49** between the primary piston and the gland nut is reduced. The only path for the oil displaced by this reduction is through the snubbing holes **28** in the ring valve **27**, and again this action prevents the unit from being extended rapidly. At the same time, the air charge pressure acts on the secondary piston area to provide a very high pullout force which may, for example, be on the order of about 63,000 lbs. and increase about 2,000 lbs. at full extension because of the compression of the charge. At full extension, the primary piston **12** bottoms on the gland nut **46**, with nominal bearing pressure regardless of the large area of contact. As a result of the high pullout force, full extension would most likely occur only on the forward cars of a train during starting or while pulling up grade and under most operating conditions there will be virtually solid train operation with minimal change in length.

By way of further illustration, the new design can provide a stroke of 13" in buff and 3" in draft, which will be seen to provide over-all performance superior to conventional end-of-car cushioning devices. This unit can provide protection for a car weighting 222,000 lbs. to at least 10 miles per hour, within the standard coupler-force specification of 500,000 lbs. when bumped into a car equipped with conventional friction type draft gear. The pivotal attachment of the new gear to the car is, moreover, shifted inwardly or closer to the truck structure relative to the usual knuckle and rod type of gear, with such shift clearly adding to lateral stability by reducing the swing of the coupler head on turning.

I, therefore, particularly point out and distinctly claim as my invention:

1. In a railroad car draft gear including a cylinder and hollow piston, with a coupler head carried by one and the other adapted to be attached to the car, a floating piston within the hollow piston, the inner end of the latter having an orifice providing communication with the interior of the hollow piston at one side of the floating piston, the cylinder being filled with oil and the hollow piston at the other side of the floating piston containing compressed air, metering pin means movable relatively in said orifice to control the area thereof, said metering pin means having an axial passage, and a pilot member attached to the floating piston and slidably extending into the passage of the metering pin means for stabilizing relative movement of the floating position.

2. The combination set forth in claim **1**, wherein the metering pin means is connected to a secondary piston relatively movable within the cylinder and stop means is provided to limit relative retraction of the pin means from the orifice of the hollow piston.

3. The combination set forth in claim **1**, wherein the floating piston carries two axially spaced seals engaging the interior of the hollow piston, and passage means extends from the exterior thereof between the two seals to an atmospheric vent.

4. The combination set forth in claim **3**, wherein the pilot member is a tube forming part of said passage means, and the pin means passage is another part which extends to the atmospheric vent.

5. Railroad car draft gear comprising a cylinder, a hollow piston, a coupler attached to one of the cylinder and hollow piston, means for pivotally attaching the other of the cylinder and hollow piston to a railroad car in sill structure of the same, a floating piston within the hollow piston dividing the interior thereof in first and second chambers, an orifice in the hollow piston providing communication between the first chamber and the cylinder interior, metering pin means movable relatively in said orifice to vary the area thereof, a secondary piston relatively movable within the cylinder to which the pin means is attached, and stop means for preventing full relative withdrawal of the pin means from the orifice, the first chamber and cylinder being filled with liquid and the second chamber containing a charge of pressurized air which maintains the cylinder and hollow piston in a neutral partially extended condition.

6. Railroad car draft gear as set forth in claim **5**, wherein the hollow piston has further orifices means therethrough in bypassing relation to the first-named orifice, and relief valve means applied to the end face of such hollow piston normally closing said further orifice means and opening the same when the liquid pressure in the cylinder reaches a preselected value, said relief valve means including a valving member and a curved spring segment biasing the same in the closing direction.

7. Railroad car draft gear as set forth in claim **6**, wherein the relief valve means comprises a pair of pistons and curved spring segments loading the same, said pistons being stepped to provide small differential areas to overcome the spring forces.

8. In a railroad car draft gear including a cylinder and hollow piston slidable therein, with one carrying a coupler head and the other adapted for attachment to the car, a floating piston within the hollow piston and dividing the interior thereof into first and second chambers, first orifice means providing communication between the first chamber and the cylinder interior, metering pin means movable relatively in said orifice to vary the area thereof with relative movement of the cylinder and hollow piston, second orifice means in bypassing relation to the first orifice means, and relief valve means applied to the end face of the hollow piston and controlling the second orifice means, said relief valve means comprising a body forming a continuation passage for the second orifice means generally normal to said end face, the passage having a side opening, and the valve means further including a traversely movable differential area piston normally closing said opening and hence the second orifice means, and a curved spring segment acting at one end against said differential area piston for loading the same and being anchored at the other end to the end face of the hollow piston.

9. Railroad car draft gear comprising a cylinder, a hollow piston slidable in said cylinder, a coupler head affixed to one of said cylinder and piston and the other being adapted for attachment to a railroad car, first packing means engaging the inner end portion of the hollow piston and the interior of the cylinder, second packing means engaging the cylinder adjacent its free end and the exterior of the hollow piston, with the latter formed to provide with the cylinder an annular cavity between the first and second packing means, a floating piston movable within the hollow piston to define first and second chambers of variable volume therein, first and second spaced seals carried by the floating piston in engagement with the hollow piston, passage means extending from the exterior of the floating piston between said first and second seals to an atmospheric vent, first orifice means in the hollow piston providing communication between said first chamber and the cylinder interior, piston-position responsive means for controlling the area of said first orifice means, second orifice means between said first chamber and said annular cavity, the cylinder and first chamber being filled with liquid and the second chamber containing a pressure air charge which maintains the cylinder and hollow piston in a neutral position, and snubbing valve means controlling said second orifice means to provide a controlled lower rate of flow of the liquid therethrough from the annular cavity to the first chamber than in the reverse direction.

10. Railroad car draft gear comprising a cylinder, a hollow piston slidable in said cylinder, a coupler head affixed to one of said cylinder and piston and the other being adapted for attachment to a railroad car, first packing means engaging the inner end portion of the hollow piston and the interior of the cylinder, second packing means engaging the cylinder adjacent its free end and the exterior of the hollow piston, with the latter formed to provide with the cylinder an annular cavity between the first and second packing means, a floating piston movable within the hollow piston to define first and second chambers of variable volume therein, first orifice means in the hollow piston providing communication between said first chamber and the cylinder interior, a metering pin carried by a secondary piston relatively movable within the cylinder and through said first orifice means to control the area thereof, said metering pin having stop means for limiting retraction thereof from the first orifice means, second orifice means between said first chamber and said annular cavity, the cylinder and first chamber being filled with liquid and the second

chamber containing a pressure air charge which maintains the cylinder and hollow piston in a neutral position, and snubbing valve means controlling said second orifice means to provide a controlled lower rate of flow of the liquid therethrough from the annular cavity to the first chamber than in the reverse direction.

11. Railroad car draft gear as set forth in claim **10**, wherein the metering pin has an axially extending passage, and the floating piston has a pilot member slidable in said passage.

12. Railroad car draft gear as set forth in claim **11**, wherein said pilot member is a tube and part of passage means extending to the exterior of the floating piston, and the pin passage leads to an atmospheric vent.

References Cited

UNITED STATES PATENTS

2,737,301	3/1956	Thornhill	213—43
2,994,442	8/1961	Frederick	213—43
3,152,699	10/1964	Vickerman	213—8
3,216,592	11/1965	Peterson et al.	213—43
3,378,149	4/1968	Powell	213—43

DAYTON E. HOFFMAN, Primary Examiner

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213—43, 223; 188—88, 96