Sept. 29, 1964 W. H. PETERSON 3,150,866

MAXIMUM ENERGY ABSORBING HYDRAULIC CUSHION

Filed Dec. 13, 1962 2 Sheets-Sheet 1

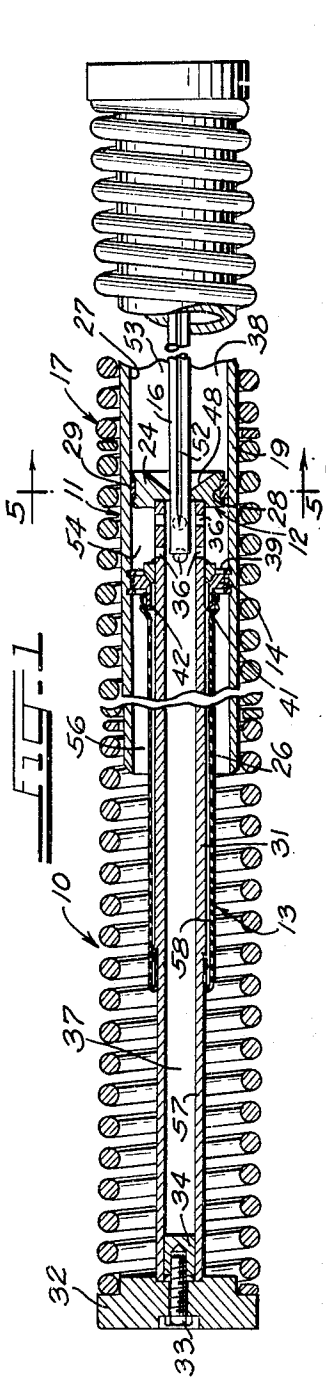


FIG. 1

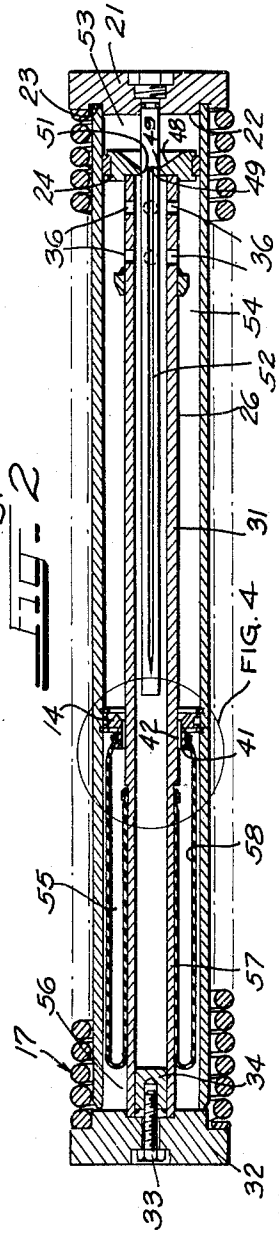


FIG. 2

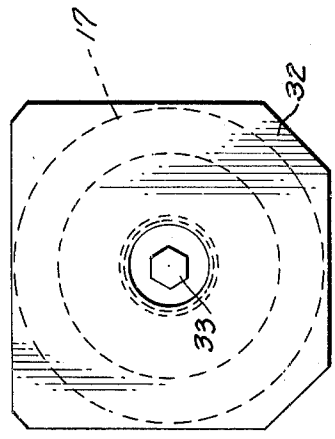


FIG. 3

FIG. 4

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FIG. 4

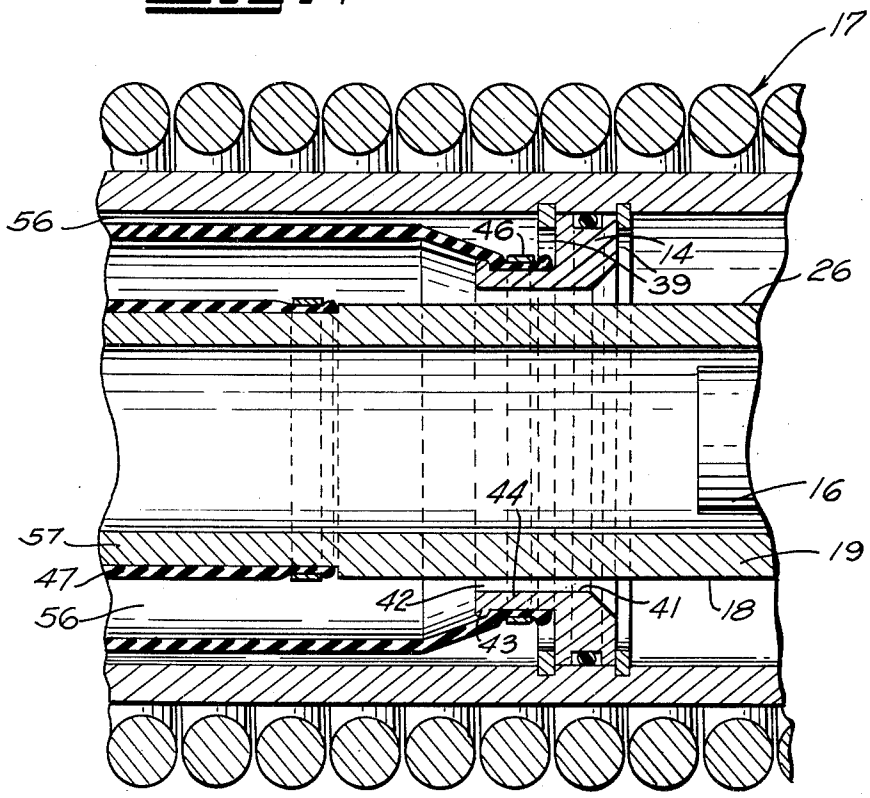
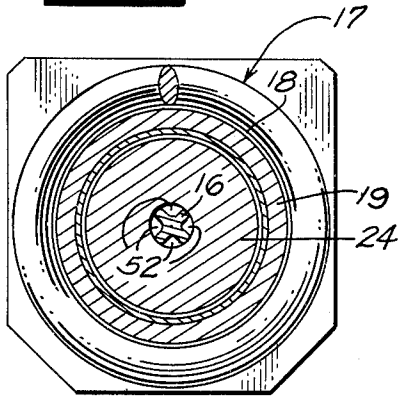


FIG. 5



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MAXIMUM ENERGY ABSORBING HYDRAULIC CUSHION

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4 Claims. (Cl. 267—1)

The present invention relates to hydraulic energy absorbing devices and more particularly to long travel hydraulic energy absorbing devices of the type employed in railway cars for the purpose of protecting lading carried thereon from damage.

Railway cars in which cushioning devices of the above type are installed are generally of the cushion underframe type employing a sliding sill, and of the cushioned rack type employing a lading carrying rack slidably mounted on the car underframe. The cushioning devices employed in these cars preferably are arranged to provide the long travel cushioning protection described in my Patent No. 3,003,436, in order to be capable of protecting both rigid and frangible lading from damage, and thus should have a length of travel in the range of from about 20 to about 40 inches, and the cushion closure characteristics disclosed in said patent, which patent also discloses a preferred form of double acting cushioning device that is arranged to provide the indicated travel in either direction and have sufficient energy absorbing capacity to withstand shock impacts of 12 m.p.h. and up.

Cushioning devices of the type disclosed in my Patent No. 3,003,436 have been found to be particularly suitable in their simplicity of construction, trouble-free operation and their economy of construction. These devices are hydraulic units of the type more specifically described in my U.S. Patents Nos. 3,035,714 and 3,035,827.

Hydraulic units of the type mentioned above generally comprise a hydraulically filled cylinder having telescopically mounted therein a piston member which is constructed and arranged to displace fluid from one side of the piston member to the other during the cushioning stroke to provide the energy dissipating function of the cushion. To avoid the use of sliding or dynamic seals this unit employs a rubber boot which also serves as a reservoir for the hydraulic fluid as the unit retracts and expands during its operation. This unit in addition is constructed so as to have substantially constant force travel characteristics over its full length of travel in either direction, i.e., substantially each increment of travel thereof provides a substantially constant resisting force or substantially constant energy absorbing rate as more fully described in the aforementioned patents.

In the typical cushion underframe and cushion rack car, adequate space in which to install a hydraulic unit of the above type is oftentimes not readily available.

Accordingly it is an object of the present invention to provide a cushioning unit of the above described hydraulic type for use in railway cars wherein the over-all size of the unit including the length and diameter are maintained at a minimum while at the same time maintaining the cushion travel requisites disclosed in my Patent No. 3,003,436, thereby providing a device that will prevent damage to the lading at impact speeds of 12 m.p.h. and up.

A further object is to provide a cushioning unit of the above described hydraulic type for use in railway cars wherein a specially designed piston member of the unit in combination with the attachment of the expandable boot-type hydraulic fluid reservoir therewith permits increased reservoir capacity without exceeding operational design limitations of said unit.

Further objects and features will hereinafter appear.

FIG. 1 is a view partly in cross-section of the cushioning device embodying the present invention showing the device in its fully expanded position;

FIG. 2 is a cross-sectional view of the cushioning device shown in its contracted position;

FIG. 3 is right end view of the cushioning device;

FIG. 4 is an enlarged fragmentary view of a portion of the device of FIG. 2 included in the inset therein;

FIG. 5 is a cross-sectional view taken generally along the line 5—5 of FIG. 1.

Referring now to the drawings, the cushion device comprises generally a hydraulic fluid filled unit 10 including a cylinder 11, a piston member 12 telescopically reciprocable within the cylinder 11, a boot reservoir 13 connected to an intermediate cylinder head 14 and the piston member 12, a metering pin 16 for regulating the rate of flow of the hydraulic fluid from one side of the piston member 12 to the other and thereby to control the energy absorbing rate of the cushioning device, and a resilient means 17 for returning the cushioning device components to their original position after the shock of impact has been dissipated.

As heretofore discussed the cushion device 10 of the present invention is constructed for use in railway cars in accordance with the invention of my said Patent No. 3,003,436 for the purpose of protecting the lading carried by the cars during bumping and switching. While efforts are made to maintain the impact speeds at a minimum it can be expected that speeds as high as 12 m.p.h. or more will be encountered. As disclosed in said Patent No. 3,003,436 substantially all classes of lading, including both rigid and fragile, may be adequately protected by employing cushion travels in the range of from about 20 to about 40 inches, and preferably 30 inches in either direction, and having the closure characteristics therein disclosed. Cushioning devices having a travel in this range are known as long travel cushions, and provide a cushion travel that insures that the lading will be fully compacted before the cushion device has completed its length of travel. Although the majority of impacts do not exceed 12 m.p.h., and therefore it is not practicable to insure full protection for all forms of lading in excess of this speed, there is a small percentage of impacts which are in excess of 12 m.p.h. It is therefore necessary in order to provide equipment durable enough to withstand impacts in excess of 12 m.p.h. and experience has shown that cushions should have energy absorbing capacities on the order of 1,000,000 ft. lbs. to keep equipment functional on a prolonged period without failure.

It should be apparent that the structural components of the cushioning unit of the type generally described above must be capable at the high energy absorbing capacity required to withstand the relatively high unit stresses associated therewith. Accordingly the high unit stresses imparted to the unit during its operation tend to be a size limiting factor and this should be kept in mind in the following description of the cushioning unit of the present invention.

The cylinder 11 may be formed from a steel tube 18 or the like of which the wall thickness 19 is sufficient to withstand the high unit pressures encountered during operation. Fixed to one end of the cylinder 11 is a closure or follower plate 21 which is engageable with a set of stops (not shown) fixed to the stationary and movable components of the cushion railway car structure. The inner wall 22 of the closure plate 21 is formed with a recess 23 receiving the end of the cylindrical tube 18 which is fixed therein as by welding.

Telescopically disposed within the cylinder 11 is the piston member 12 including a piston head 24 fixed as by welding to one end of the hollow piston rod 26. The piston head 24 is formed from metal, preferably alloy

steel. To prevent galling between the piston head 24 and the inner cylinder wall 27 and also to provide a guiding and sealing surface there is seated within a peripheral groove 28 a raised guiding ring 29 formed preferably from a laminated phenolic resin.

The piston rod 26 is formed from a steel tube having fixed to the free end thereof a closure or follower plate 32 similar to closure plate 21. The closure plate 32 is engageable with a second set of stops or keys also fixed to movable and stationary car components of the cushion railway car. The closure plate 32 is fixed to the rod 31 by way of a bolt 33 threaded into a plug 34 welded in the end of the rod 31. An array of ports 36 is formed in the rod 26 adjacent the piston head 24 to provide communication between the piston rod bore 37 and the cylinder bore 38.

Primarily for the purpose of securing the boot 13 within the cylinder 11, the intermediate cylinder head 14, which may be formed from a suitable metal, is seated within the bore 38 of the cylinder and fixed against axial movement by way of a pair of snap rings 39. Formed in the head 14 is an axial opening 41 sized so as to form an annular opening 42 about the piston rod 26 extending therethrough and to permit low pressure flow of the hydraulic fluid therethrough into the boot 13.

The boot or flexible fluid reservoir 13 is preferably made from an elastomer characterized by its high tear resistance in any direction, its ability not to permanently set, its low temperature flexibility, and its resistance to attack by industrial atmospheres and compatability with fluid. As shown, the boot 13 is fixed at one end within a rim 43 formed on a boss 44 extending from the cylinder head 14 by way of a ring or adjustable strap clamp 46. At its other end the boot is reversely folded and the reversely folded end 47 is suitably clamped by a second ring clamp 46 to the piston rod 26 inwardly of the closure plate such that it lies closely adjacent the piston rod 26 in the unit extended position shown in FIG. 1. In this connection it should be mentioned that in the cushioning devices constructed as heretofore wherein a piston rod 26 of constant outer diameter is employed, the inner wall of the boot 13 in its extended position is spaced from the outer periphery of the tubular piston rod 26 a distance which is about equal to the thickness of the reversed end of the boot so that this space is occupied by fluid.

Referring again to the piston head 24 the outboard face thereof is formed with an inwardly converging frustoconical face which terminates at its inner end with a rim 48 of an axial opening 51 defining an orifice. Slidably disposed within the orifice 48 of the piston head 24 is the metering pin 16 which serves to meter the rate of flow of the hydraulic fluid through the orifice.

One end of the metering pin 16, which may be formed of a mild steel, is suitably fixed to the closure plate 21 as by threading. To assure a fluid tight connection a sealing compound may be applied at the threaded connection.

For metering the rate of flow of the hydraulic fluid to obtain a substantially constant force or energy absorbing rate for each increment of travel of the cushioning device, a plurality of longitudinally tapering flutes 52 is formed on the metering pin periphery. The flutes are formed to cooperate with the orifice 48 in the piston head 24 so as to maintain the pressure substantially constant upon relative movement of the latter with respect to the cylinder by varying the effective orifice area at the rim 48 in accordance with the total cross-sectional area of the flutes encompassed by the piston head.

In operation and when the above described cushioning device is applied in a cushion railway car in the normal extended position thereof the cushion device 10 is under very little pressure, say for example, about 2 p.s.i. When the car is subjected to shock impact at either end of the car the tubular cylinder 11 and the piston member 12 commence to move relatively to each other in a direction

to compress the hydraulic fluid in the high pressure chamber 53 on one side of the piston head 24 to provide a force opposing or resisting the shock impact. As the cylinder 11 and piston member 12 commence movement the hydraulic fluid is displaced from the high pressure chamber 53 through the orifice 48 and past the flutes 52 and into the piston rod bore 37 thence radially outwardly of the piston rod bore 37 through the array of ports 36 into the low pressure chamber 54. Concurrently the hydraulic fluid displaced by the metering pin 16 also flows through the ports 36.

The hydraulic liquid flow through ports 36 is at a relatively high velocity which creates considerable turbulence in the low pressure chamber 54. The turbulence is caused at least in part by the radially directed flow of hydraulic fluid impinging directly against the inner wall 14 of the cylinder 11 so as to dissipate much of the kinetic energy of the shock impact in the form of heat.

Upon further contraction of the cushioning device the high pressure volume in chamber 53 is further reduced by the advancement of the piston head 24 toward the closure plate 21. During this contraction of the device pressures of about 13,000 p.s.i. may be developed in the high pressure chamber 53. The hydraulic fluid flowing through orifice 48 as controlled by the metering pin 16 fills the low pressure chamber 54 while a volume of hydraulic fluid equivalent to that displaced by the total entry into the fluid chamber or bore 37 of the piston 26, flows through the annular opening 42 in the volume 55 provided by the boot 13 which expands and rolls to fill the boot chamber 56 in the open end of the cylinder 11. The pressure within the boot 13 is relatively low compared to the pressures in the chamber, by way of example on the order of about 4 to 10 p.s.i.

When the shock of impact has fully dissipated, the resilient means which may be in the form of a plurality of springs 17 acting in tandem are operative to return the hydraulic cushion components to the initial extended position shown in FIG. 1. During this movement, the hydraulic fluid flow is reversed and the boot 13 returns to its initial position so that the displaced hydraulic fluid is returned to the piston rod bore 37 and high pressure chamber 53 and thereby is in an operative position to again receive the shock of impact and absorb a portion of the energy thereof.

From the foregoing description of the structure and function of the hydraulic cushioning device it should be readily apparent that the long travel cushioning device must be arranged to resist the operating pressures that will be occasioned in employing the invention of my Patent No. 3,003,436 to protect lading from damage at impact speeds at 12 m.p.h. and up, and thus must be constructed so as to be capable of withstanding the relatively high stresses that will be imparted thereto without failure. These stresses pose a problem when space limitations are such that a cushioning device of the above type is required to be reduced in its diametral dimensions to be accommodated within the space while at the same time achieving the lading protection results desired.

In constructing a cushion device of the above type such factors as gross weight of the railway car, the maximum impact speeds for which the cushioning device is to provide lading protection, and the length of travel of the cushioning are established and known. The gross weight may range from 50 tons to 90 tons, the impact speed between 10 m.p.h. and up and the travel in the range of between about 20 to about 40 inches, preferably 30 inches, in either direction.

With the above factors fixed and known the total operating pressures the cylinder will have to resist, and the corresponding cushion resistance forces, may be calculated in accordance with well established methods. Under the conditions indicated and assuming a cushion travel of about 30 inches, the total or basic resistance force to be considered should probably be about 350,000 to 400,-

000 pounds to prevent cushion failure in a car having an average gross weight.

When the total force to be considered is determined, the requisite minimum wall thickness of the cylinder is determined in accordance with the limitations as to outer diameter of the tube as dictated by the cross-section area available for the installation of the cushioning device. In determining the tube outer diameter the particular spring employed to return the cushioning device and clearances must also be considered in deciding on the outer diameter. With the outer diameter, the total force, and yield strength of the tube material known, it is a simple matter to calculate the minimum thickness required to withstand the pressures within the cylinder.

In accordance with past experience of the construction of the heretofore described cushion device it has been determined the outer diameter of the hollow piston rod 26 should preferably be about 45 to 50 percent of the diameter of the bore of the cylinder to adequately accommodate the expanded volume of the boot without extending outwardly of the open end thereof or being restricted in its expansion within the boot chamber. Any constriction outside the boot chamber or limitation of the boot expansion could cause expansion in a location where it might be ruptured and consequently result in failure of the cushion device. If greater ratios are used, the boot chamber must be made excessively long, creating less resistance of piston rod to buckling, so much less in fact that poor utilization of the compressive strength of rod 26 results.

It is, of course, also apparent that the wall thickness of the piston rod must also be adequate to resist the compressive forces exerted thereon during contraction. To this end the minimum wall thickness is calculated so that the net cross-sectional area is adequate to resist such compressive forces. In this connection, it is to be noted that the net cross-sectional area containing the array of ports must be of sufficient wall thickness to withstand the compressive forces.

In a long travel cushion of the preferable 30 inch travel type the piston rod is of such length that it may be subject to buckling. Thus when the piston rod minimum wall thickness has been calculated the rod must also be checked against buckling. Such a check may be made by employing the radius of gyration to tube length ratio method to determine if the maximum column force is capable of withstanding the total or basic force exerted thereon.

The boot as previously described has an optimum thickness of say about 3/16 inch so as to provide a maximum volume capacity commensurate with its strength characteristics.

From the foregoing it should be readily apparent that a reduction in size, namely the outer diameter of the cushioning device, presents a problem in the matter of achieving the maximum capacity in a minimum size cylinder bore.

In this matter the involved magnitude of the total resisting force or total energy absorbing capacity in protecting lading at the anticipated impact speeds are such that any reduction in the outer diameter of the cylinder 11 also results in a reduction in the diameter of the cylinder bore 38 because of the necessity of maintaining the minimum wall thickness. Such a decrease in the cylinder bore size also correspondingly decreases the volume of the boot chamber 56 so that it unduly restricts the volume into which the boot 13 may expand. Such reduction in boot volume 55 as heretofore discussed results in adversely affecting the operation of the cushioning device.

With the structure employed heretofore it would appear to be a simple matter to overcome this deficiency in the boot volume 55 by merely elongating the same. However, such elongation of the boot chamber 55 also requires the cylinder 11 also be extended at its free end to accommodate the lengthwise increased boot 13. It follows that if the free end of the cylinder 11 is extended

it is also necessary to correspondingly increase the length of the piston rod 26 to permit the selected full length of travel. In this connection it is to be noted that the free end of the cylinder 11 abuts the closure plate 32 in the contracted position and provides a stop at the end of travel of the piston rod 26.

Increasing the length of the piston rod 26 is not satisfactory for the reason that any increase in length at a predetermined minimum wall thickness withstanding the compressive forces results in a change of radius of gyration to length ratio such that the rod is subject to buckling. To overcome the buckling would require greater wall thickness which again would decrease the size of the boot chamber such that further rod elongation would be necessary. Thus there does not appear to be any solution to providing adequate boot volume 55 for a minimum bore cylinder having the requisite operating pressure resistance to provide the lading protection that is described above. Even if the required boot volume could be achieved by mere elongation of the piston rod and the cylinder the resulting over-all length of the cushion device may be such as to fall outside of the length limitations dictated by the installation space in the railway car.

In accordance with the present invention the piston rod is constructed such that it may be readily employed in a minimum size cylinder bore to provide an adequate boot chamber accommodating the required boot volume while at the same time maintaining the over-all length of the cushion device substantially unchanged or at a minimum.

This is accomplished as shown in particular in FIG. 4 by reducing the diameter of the end portion 57 of the piston rod 26 adjacent the follower plate 32 so that the inner wall of the reversely turned end of the boot 13 when clamped thereon has the same or lesser diameter than the major outer diameter of the remainder of piston rod 26. The piston rod 26 thus formed is selected as heretofore to have the necessary strength characteristics to withstand the compression and column buckling forces. However, it should be readily apparent that when wall thickness of the cylinder 11 is maintained at a minimum any reduction in the outer diameter of the cylinder 11 to a minimum also results in a decrease in the diameter of the cylinder bore 38. Furthermore, to insure that the cushioning will withstand resistance forces of the above indicated levels, the piston rod dimensions must be such that they are capable of withstanding the compressive and column buckling forces associated therewith. This results in the outer diameter of the piston rod exceeding the desired 45 to 50 percent of the bore diameter and results in a restricted boot chamber.

While such a percentage of outer diameter of piston rod 26 is still employed as the goal in accordance with the present invention the major diameter of piston rod 26 is determined in accordance with its compressive strength and column buckling strength. However, since column bending moments are greatest at the mid-point of the piston rod, a larger diameter is necessary at that point. Also at the portion just behind the piston containing ports 36, a larger diameter is required to resist compressive stresses due to the material removed in providing ports 36. Although a stepdown in diameter symmetrical about the center of the rod length at both ends of the rod would be indicated from only the buckling considerations, because of the increased diameter necessary at the port location for compressive strength, the larger diameter is continued throughout that portion of the rod further rigidifying the piston rod and simplifying manufacture.

To achieve the requisite boot chamber volume 55 the relationship of the minimum diameter of the rod and the major diameter is such that the annular thickness of the reversed end of the boot clamped to the closure end is substantially the same, or less than, the difference of the outer diameters of the rod sections. In this manner as shown in particular in FIG. 1 the inner wall 58 of the boot 13 in the extended position of the cushion de-

vice lies closely adjacent to the outer periphery of the piston rod 26 so that the boot is substantially devoid of hydraulic fluid in the extended position. Hence, as the cushioning device contracts under the force of impact substantially the entire boot volume is employed as a reservoir for the displaced hydraulic fluid. In this connection reference is made to the construction of the cushion devices employed heretofore (see the above identified patents) wherein the boot volume in its capacity to receive the displaced fluid by the existence of the quantity of fluid in the extended position of the device has not been reduced by the use of a relatively greater diameter piston rod in the present device.

What is claimed is:

1. A hydraulic cushion device for use in railway cars and adapted for providing protection for lading carried thereon from impacts of 12 m.p.h. and up comprising a cylinder having a closed end and an open end, a piston member telescopically disposed in said open end of said cylinder and including a hollow piston rod and a piston head fixed to one end thereof, an intermediate cylinder head disposed in said cylinder between said closed and open ends thereof, said cylinder head having an axial opening through which said piston rod extends and defines therewith an annular opening, an orifice in said piston head, metering pin means fixed to said cylinder closed end and extending through said orifice into said hollow piston rod, a port means in said piston rod adjacent said piston head, a flexible boot reservoir fixed at one end to said cylinder head and having a reversely turned portion at its other end fixed to said piston rod outboard of said cylinder head, said boot being in communication with said cylinder by way of said port means, annular opening and orifice and being expandable to serve as a reservoir for receiving fluid displaced from said piston rod bore upon relative telescopic movement of said piston member and said cylinder, resilient means disposed between said piston member and said cylinder, and hydraulic fluid filling said cushion device, said cylinder having a cylinder bore of minimum diameter and a wall of minimum thickness for achieving the requisite lading protection, and said hollow piston rod being formed with the terminal end portion thereof remote from said piston head with an outer diameter less than the outer diameter of the central portion of said rod and the end portion thereof adjacent said piston head, said difference in said outer diameter of said terminal end portion and the outer diameter of said central and said end portion adjacent said piston head being equal to or exceeding the thickness of said reversely turned portion of said flexible boot which is attached to said terminal end portion.

2. A hydraulic cushion device for use in railway cars and adapted for providing protection for lading carried thereon from impacts of 12 m.p.h. and up comprising a cylinder having a closed end and an open end, a piston member telescopically disposed in said open end of said cylinder and including a hollow piston rod and a piston head fixed to one end thereof, an intermediate cylinder head disposed in said cylinder between said closed and open ends thereof, said cylinder head having an axial opening through which said piston rod extends and defines therewith an annular opening, an orifice in said piston head, metering pin means fixed to said cylinder closed end and extending through said orifice into said hollow piston rod, port means in said piston rod adjacent said piston head, a flexible boot reservoir fixed at one end to said cylinder head and having a reversely turned portion at its other end fixed to said piston rod outboard of said cylinder head, said boot being in communication with said cylinder by way of said port means, annular opening and orifice and being expandable to serve as a reservoir for receiving fluid displaced from said piston rod bore upon relative telescopic movement of said piston member and said cylinder, resilient means disposed between said piston member and said cylinder, and hydraulic fluid filling said cushion device, said cylinder having a cylinder bore of minimum diameter and a wall of minimum thickness for achieving the requisite lading protection, and said hollow piston rod being formed with the terminal end portion thereof remote from said piston head with an outer diameter less than the outer diameter of the remainder of said rod, said difference in said outer diameter of said terminal end portion and said outer diameter of said remainder of said rod being equal to or exceeding the thickness of said reversely turned portion of said flexible boot, and said reversely turned portion of said boot being clamped to said piston rod along said terminal end portion having the lesser diameter.

3. A hydraulic cushion device for use in railway cars and adapted for providing protection for lading carried thereon from impacts of 12 m.p.h. and up comprising a cylinder having a closed end and an open end, a piston member telescopically disposed in said open end of said cylinder and including a hollow piston rod and a piston head fixed to one end thereof, an intermediate cylinder head disposed in said cylinder between said closed and open ends thereof, said cylinder head having an axial opening through which said piston rod extends and defines therewith an annular opening, an orifice in said piston head, metering pin means fixed to said cylinder closed end and extending through said orifice into said hollow piston rod, port means in said piston rod adjacent said piston head, a flexible boot reservoir fixed at one end to said cylinder head and having a reversely turned portion at its other end fixed to the terminal end portion of said piston rod outboard of said piston head, said boot being in communication with said cylinder by way of said port means, annular opening and orifice and being expandable to serve as a reservoir for receiving fluid displaced from said piston rod upon relative telescopic movement of said piston member and said cylinder, resilient means disposed between said piston member and said cylinder, and hydraulic fluid filling said cushion device, said cylinder having a cylinder bore of minimum diameter and a wall of minimum thickness for achieving the requisite lading protection, said hollow piston rod including a bore having substantially constant diameter along the length of said rod and an outer diameter adjacent the end to which said reversely turned end portion of said flexible boot is fixed and which provides a minimum wall thickness to withstand the maximum compressive forces present in said device during operation thereof and of which said outer diameter of said terminal end portion is less than the outer diameter of the remainder of said rod to an extent at least substantially equal to the thickness of said flexible boot, said outer diameter of the remainder of said rod being such as to impart to said rod the necessary column strength characteristics and provide adequate wall thickness at the piston head portion of said rod containing said ports to withstand the compressive forces.

4. A long travel hydraulic cushion device having a travel between 20 to 40 inches in either direction for use in railway cars and thereby being adapted for providing protection for lading carried thereon from impacts at 12 m.p.h. and up comprising a cylinder having a closed end and an open end, a piston member telescopically disposed in said open end of said cylinder and including a hollow piston rod and a piston head fixed to one end thereof, an intermediate cylinder head disposed on said cylinder between said closed and open ends thereof, said cylinder head having an axial opening through which said piston rod extends and defines therewith an annular opening, an orifice in said piston head, metering pin means fixed to said cylinder closed end and extending through said orifice into said hollow piston rod, port means in said piston rod adjacent said piston head, a flexible boot reservoir fixed at one end to said cylinder head and having a reversely turned portion at its other end fixed to the terminal end portion of said piston rod remote from said piston head, said boot being in communication with said cylinder by way of said port means, annular opening and orifice and being expandable to serve as a reservoir for receiving fluid

displaced from said piston rod bore upon relative telescopic movement of said piston member and said cylinder, resilient means disposed between said piston member and said cylinder, and hydraulic fluid filling said cushion device, said cushion device when in a fully extend position being of a minimum length to accommodate the full length of travel thereof in either direction and said cylinder having a cylinder bore of minimum diameter and a wall of minimum thickness for achieving the requisite lading protection, said piston rod being formed so as to withstand the column buckling forces and compression forces imparted thereto and when so formed having the wall thickness which is reduced to a minimum along a length contiguous to said end portion remote from said piston head end to

withstand the compressive forces, said piston rod having an outer diameter along said remote end portion lesser than the diameter of the remainder of said rod to provide said minimum thickness, and said reversely turned end portion of said boot being clamped along said remote end portion.

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