Sept. 10, 1968 R. G. POWELL 3,400,833

RAILWAY CAR CUSHIONING DEVICE

Filed April 22, 1966 3 Sheets-Sheet 1

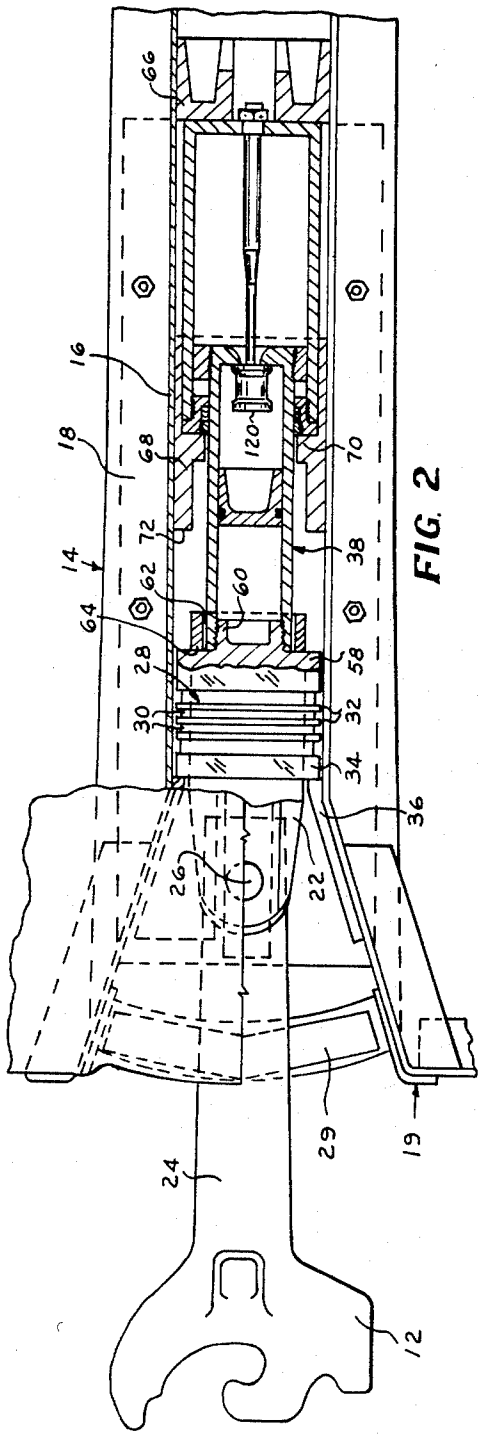


FIG. 2

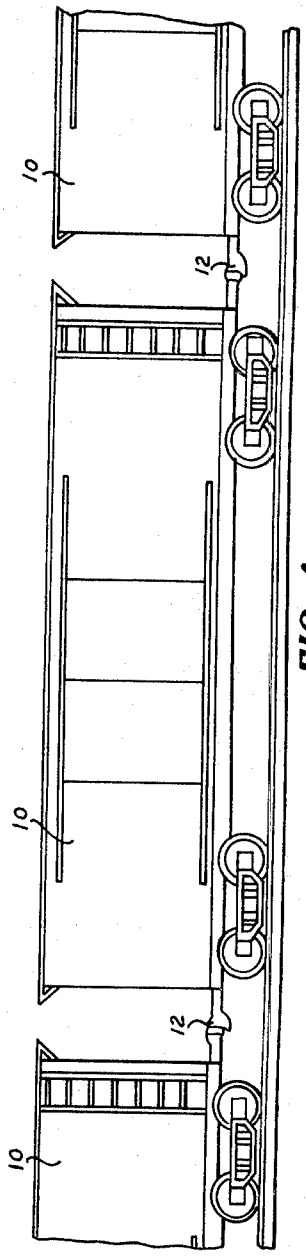


FIG. 1

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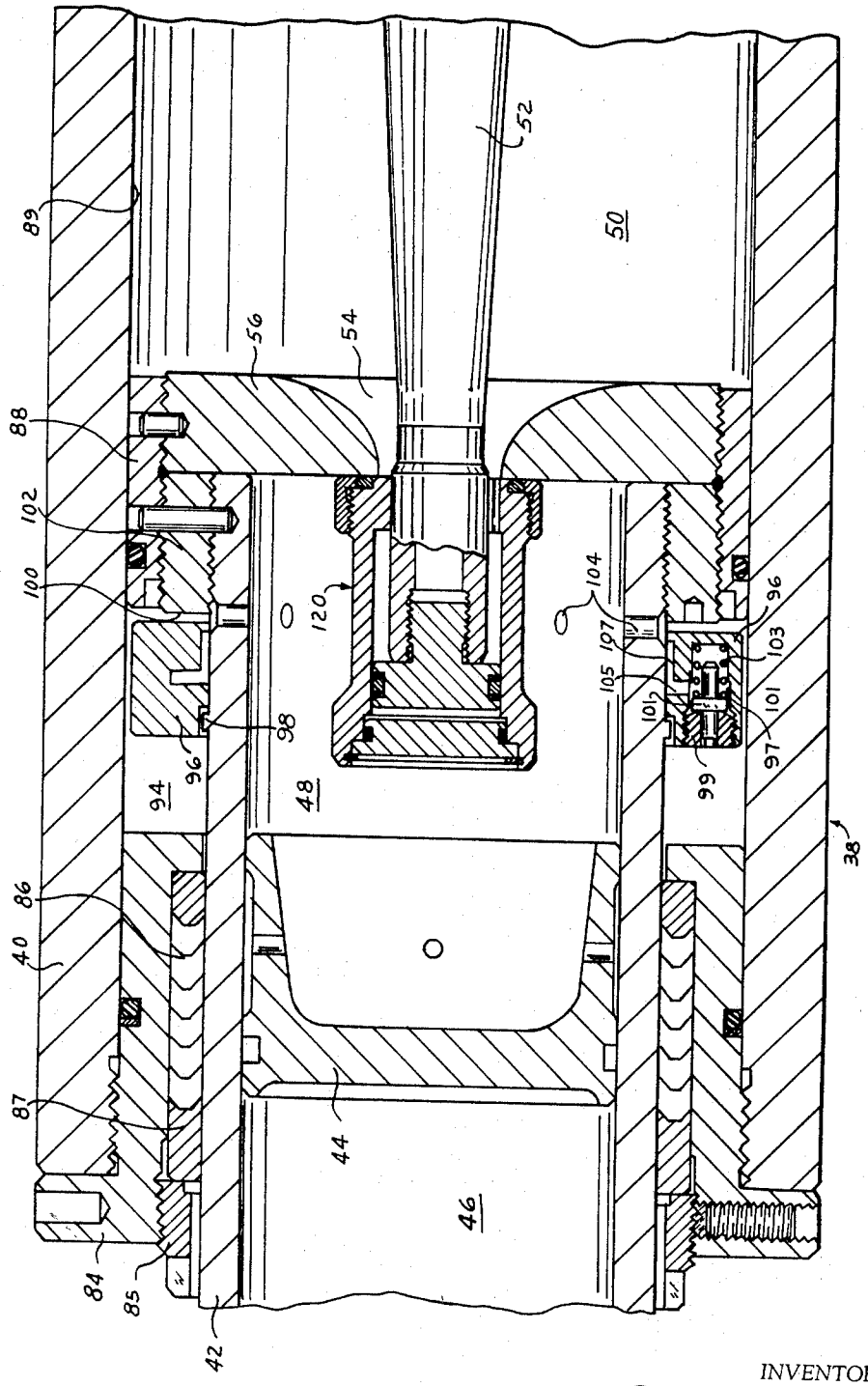


FIG. 3

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Filed April 22, 1966 3 Sheets-Sheet 3

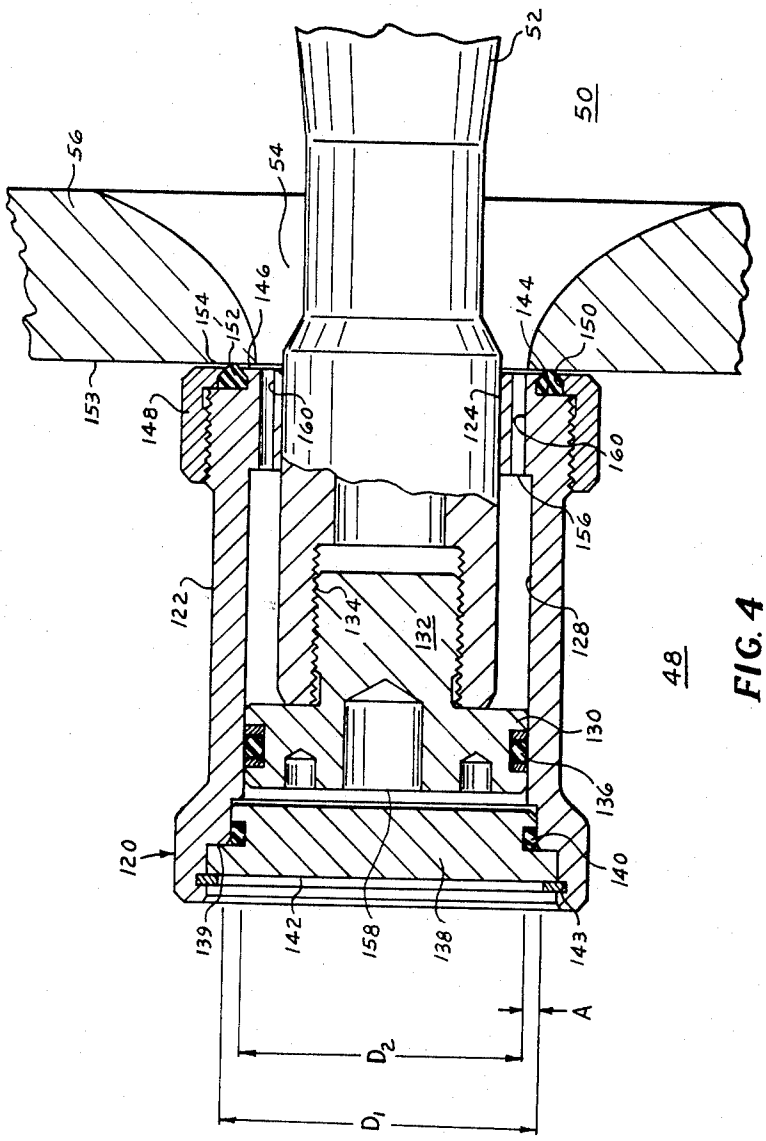


FIG. 4

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3,400,833

RAILWAY CAR CUSHIONING DEVICE

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Filed Apr. 22, 1966, Ser. No. 544,548
10 Claims. (Cl. 213—8)

ABSTRACT OF THE DISCLOSURE

An oleo-pneumatic cushioning device for railway cars which collapses and achieves internal metered transfer of hydraulic fluid between a pair of hydraulic chambers in addition to achieving compression of a compressible fluid within the unit to dissipate energy applied thereto. The unit includes a lockup device which is operative to prevent the interchange of hydraulic fluid between the hydraulic chambers upon the application to the unit of forces of low magnitude, but which is responsive to a predetermined increase in fluid pressure within one of the hydraulic chambers developed by the application of forces in excess of a predetermined magnitude to unlock and allow the interchange of hydraulic fluid between the hydraulic chambers thereby allowing the unit to collapse and dissipate energy.

This invention relates to cushioning devices for railway cars and more particularly to cushioning devices adapted to be operatively connected to a coupler adjacent the end of a railway car for absorbing impact forces exerted against the coupler.

The present invention is especially directed to a cushioning system or shock absorbing system in which the metering of a fluid is employed as an energy dissipative device. The majority of cushioning systems used for end-of-car cushioning employ the principal of hydraulic fluid metering to dissipate the energy. This method of energy dissipation is velocity sensitive in that the amount of energy dissipation is directly proportional to the velocity of fluid flowing through the metering orifice of the cushioning device. On downgrade run-ins of the cushioning devices, the velocity of fluid flowing through the metering orifice will be low and the energy dissipated will also be low allowing low loads to collapse the cushioning device.

Because of the velocity sensitive nature of this method of energy dissipation, the cushioning device will be permitted to go solid or completely contract under the application of extremely low velocity impacts such as slack run-in during a downgrade locomotive braking condition. When a sizable number of railway cars are included in a single train, the length of the train may shorten considerably by the sum of the travel of the cushioning units on all of the railway cars equipped with such units. Additionally, slack run-outs of the coupled railway cars without a sufficient energy dissipation in each of the cushioning units create an acceleration which increases with each successive car. Other objections which result from present end-of-car hydraulic fluid metering systems include the gradual lengthening of a train when the brakes are released after a stop, the creeping of cars at loading docks and the inability of some cushioning units to return to neutral or centered position sufficiently rapid after an impact to be ready for a second impact.

Another problem in present end-of-car cushioning units employing fluid metering systems results from the wear of the piston seals caused by the continuous back and forth movement or hunting of the piston which is induced by normal train operation. With the present systems, the train action induces continuous reciprocation or hunting of the piston with many reversals of the piston seals. As a buff load is gradually applied, the train action caused

by the velocity sensitive nature of the cushioning device produces a relatively large drift of the hydraulic cushioning device causing increased wear on the piston seals and the cylinder seals of the cushioning device.

Thus, the two main undesirable features in present end-of-car cushioning units are: (1) The uncontrolled hunting or longitudinal reciprocation induced by train action on the piston or the movable portions of the device, and (2) the slack run-in of the cushioning device upon the application of continuous low velocity impact load with the cushioning device going solid and thereby having insufficient energy dissipating capability when the cushioning unit is subject to a draft force when in the full buff position.

It is therefore a primary object of the present invention to provide a novel cushioning device of the fluid metering type for an end-of-car cushioning system which eliminates the uncontrolled hunting or longitudinal back and forth movement of the piston which is induced by normal train action.

A further object of this invention is the provision of a novel fluid metering type of cushion device which retains its full energy dissipating capability under train operating conditions involving draft and buff forces which are below a predetermined minimum.

A further object of this invention includes the provision of a resilient draft gear in combination with such a fluid metering type of hydraulic cushioning device to supplement and complement the cushioning action of the hydraulic cushioning device.

The present invention comprises a hydraulic-pneumatic cushioning device of the fluid metering type for dissipating energy applied to the coupler of the railway cars. In operation, the invention is effective to maintain in positive resistance to impact forces until a predetermined fluid pressure is reached within specific areas of the cushioning device. Basically, the cushioning device comprises inner and outer cylinders which cooperate in telescoping relationship to define an inner variable volume hydraulic fluid chamber and an outer variable volume hydraulic chamber separated by an orifice plate. An elongated metering member is fixed to one of the cylinders and is received within a metering orifice formed in the orifice plate. The metering member cooperates with the orifice to vary the effective size thereof in response to the relative positioning of the inner and outer cylinders. A free floating piston disposed within the inner cylinder in sealed relation with the inner cylindrical wall thereof forms a movable partition separating the inner hydraulic chamber from a variable volume pneumatic chamber. The pneumatic chamber is filled with a compressible fluid such as compressed air or nitrogen and is maintained under a predetermined pressure. The fluid pressure within the pneumatic chamber is operative under conditions of no load to maintain the cushioning device in its extended or neutral position. This is accomplished by causing the floating piston to force a majority of the hydraulic fluid from the inner hydraulic chamber to the outer hydraulic chamber. Upon receiving a buff force or impact load, the cushioning device will contract causing the hydraulic fluid within the high pressure outer hydraulic chamber to be forced through the metering orifice into the low pressure inner hydraulic chamber causing movement of the free floating piston and increasing the pressure within the pneumatic chamber. As this occurs, a large amount of the energy applied to the cushioning device is dissipated.

The cushioning device includes means which prevent contracting thereof under low impact loads until such time as a predetermined fluid pressure is reached within the high pressure outer hydraulic fluid chamber, thereby allowing the low force impacts to be absorbed by a resilient cushioning mechanism which is provided for simul-

taneous cooperation with the hydraulic-pneumatic cushioning device both in draft and buff.

As the predetermined unlocking pressure is reached within the outer or high pressure hydraulic chamber, the hydraulic cushioning device will unlock and contract absorbing a substantial amount of the energy applied thereto. As the hydraulic-pneumatic cushioning device is contracted, it cooperates with the resilient cushioning mechanism to achieve a relatively smooth transition of forces, thereby precluding any transmission of jarring from the coupler of the railway car to the underframe thereof.

The cushioning device is provided with a variable auxiliary draft cushioning chamber defined between the inner and outer cylinders which is in fluid communication with the inner hydraulic chamber. Means is provided for control of the ingress and egress of hydraulic fluid between the inner hydraulic chamber and the auxiliary draft cushioning chamber to provide cushioning in draft as well as in buff.

While this application is primarily directed toward the cushioning of railway cars, it is not intended that the scope of the application be limited thereto. It is deemed apparent that the invention will have application in dock facilities for ocean going vessels. The dock facility may have movable bumper portions which are hydraulically cushioned to prevent damage from impacts. The invention may also be employed in the landing gear systems of aircraft to protect the aircraft against damage which might occur as a result of severe landing impacts.

Other and further objects of this invention will be obvious upon understanding of the illustrated embodiment about to be described or will be indicated in the appended claims, and various advantages not referred to herein will occur to one skilled in the art upon employment of the invention in practice. A preferred embodiment of the invention has been chosen for the purpose of illustration and description and is shown in the accompanying drawings forming a part of the specification wherein:

FIGURE 1 is a side elevational view of a plurality of railway cars coupled to each other;

FIGURE 2 is a plan view having portions thereof broken away showing the hydraulic cushioning device comprising this invention mounted within a thick sill adjacent the rear end of a draft gear and coupler structure, the cushioning device being illustrated in a neutral position;

FIGURE 3 is a fragmentary sectional view of the cushioning device of FIGURE 2 illustrating the cushioning unit in the neutral position thereof;

FIGURE 4 is a fragmentary sectional view of the cushioning device of FIGURE 3 illustrating the lockup mechanism of the cushioning unit in the neutral position thereof.

Referring now to the drawings for a better understanding of this invention, railway cars **10** illustrated in FIGURE 1 are interconnected by means of couplers **12**. With reference to FIGURE 2, assuming that the railway cars **10** are of the cushioned underframe type, an end-of-car cushioning system will be arranged inwardly of each of the couplers **12**. A fixed center sill generally designated **14** within which the cushioning system is arranged is a hat-shaped sill having side webs **16** with bottom legs **18**. The center sill construction is provided with an open outer end **19** which is flared to receive the coupler member **12** and to allow swinging of the coupler member relative to the center sill construction. Connection between the coupler member **12** and the center sill **14** of the railway car is achieved by means of a yoke member **22** disposed within the center sill and transmitting forces from the coupler to the draft gear and hydraulic cushioning unit structure as described in detail hereinbelow. A shank portion **24** of the coupler **12** is connected to the yoke **22** by means of a pivot pin **26**. A coupler carrier **29** is located at the flared portion of the center sill structure and supports the shank portion **24** of the coupler **12**. While the coupler **12** has been shown as a type F coupler (Association of American Railroads' designation) mounted about a vertical pin, it is to be understood that the present invention may, if desired, be employed with a type E coupler connected to a horizontal key. Mounted within the yoke **22** is a resilient draft gear generally designated **28** and comprising a plurality of resilient pads **30** separated by metal plates **32**. A front follower block **34** in engagement with the draft gear **28** is adapted to engage front stops **36** secured to the inner surface of the fixed sill structure **14** to define an outward limit and to transmit draft forces to the center sill structure.

As illustrated in FIGURES 2 and 3 forming an important part of this invention, a hydraulic-pneumatic cushioning unit generally designated **38** is positioned rearwardly of the draft gear **28**. The hydraulic cushioning unit **38** (which is frequently referred to in the trade as an ole-pneumatic cushioning unit) comprises an outer cylinder **40** and an inner cylinder **42** movably cooperating in telescoping relationship. Disposed within the inner cylinder **42** is a floating piston **44** dividing the inner cylinder into a pneumatic chamber **46** and an inner hydraulic chamber **48**. An outer hydraulic fluid chamber **50** formed within the outer cylinder **40** contains a relatively incompressible liquid such as hydraulic fluid. The pneumatic fluid chamber **46** formed within the inner cylinder **42** contains a compressible fluid such as air or dry nitrogen gas which is maintained under a static pressure. A metering pin **52** is secured to the closed end or base of the outer cylinder **40** and has a free end thereof received within a metering orifice **54** in an orifice plate **56** forming a closure for the inner end of the inner cylinder **42**. The orifice **54** provides fluid communication between the inner hydraulic chamber **48** and the outer hydraulic chamber **50**, thereby providing for the interchange of hydraulic fluid between the outer and inner hydraulic chambers upon relative movement of the inner and outer hydraulic cylinders. The metering pin **52** is tapered as illustrated in FIGURE 3 and cooperates with the metering orifice **54** to define a velocity sensitive control for controlling the interchange of hydraulic fluid between the inner and outer hydraulic chambers. The metering pin **52** varies the effective size of the metering orifice **54** in response to relative positioning of the inner and outer hydraulic cylinders **40** and **42**.

Referring now to FIGURE 2 the inner cylinder **42** has an end cap **58** thereon which serves the double function of a rear follower block. The end cap **58** has an externally threaded inner extension **60** engaging internal threads within the inner cylinder **42** as illustrated in FIGURE 3. The inner cylinder **42** extends through an opening **62** in the yoke **22** and the end cap **58** engages a peripheral shoulder or rim **64** defined inwardly of the yoke **22**. Thus, upon forward movement of the yoke **22** which would occur upon the application of a draft force, the shoulder **64** in engagement with the end cap **58** causes extending movement of the inner cylinder **42**. The end cap **58** in addition to further extending the inner cylinder **42** of the cushioning unit **38** will cause compression of the resilient draft gear **28** and through the follower plate **34** will transmit the energy applied to the draft gear to the center sill structure **14**. The outer cylinder **40** abuts a rear support **66** secured to the fixed sill structure **14**. An end cap stop **68** is provided with a lug **70** which engages a portion of the front face of the outer cylinder **40** as illustrated in FIGURE 2 to maintain the outer cylinder in its proper position. The rear abutment surfaces **72** on the end cap stop **68** engage the end cap **58** to limit the rearward travel of the inner cylinder **42** upon exertion of impact forces against the coupler **12** as shown in FIGURE 2.

With reference now to FIGURE 3 illustrating an important part of this invention, a packing gland adapter **84** is retained at one extremity of the outer cylinder **40**

and carries a packing member **86** therein for the establishment of a fluid tight seal between the outer cylinder **40** and the inner cylinder **42**. A retainer **85** is threadedly engaged within the gland adapter **84** and retains the packing **86** and an outer bearing **87** in assembly between the inner and outer cylinders **42** and **40**. An inner annular bearing member **88** is retained at one extremity of the inner cylinder **42** for the establishment of bearing engagement between the inner cylinder and the inner wall **89** of the outer cylinder **40**.

The packing assembly **86** and the bearing member **88** cooperate to define a variable volume annular draft cushioning chamber **94** between the inner and outer cylinders. An annular pressure responsive fluid flow control mechanism **96** is disposed about the outer circumference of the inner cylinder **42** and is axially movable relative to the inner cylinder **42** and a stop surface **100** formed on a spacer member **102**.

With reference now particularly to FIGURE 3, the flow control mechanism **96** includes a generally ring-like body member having at least one and preferably a series of poppet valve constructions which prevent the flow of hydraulic fluid in one direction and allow the flow of hydraulic fluid in the opposite direction in response to predetermined fluid pressure differential across the poppet valve structure. Each of the pressure responsive flow control or poppet structures comprises a stepped valve housing bore **97** which is internally threaded at its outer extremity to receive a valve seat structure **99**. The valve seat **99** is provided with an axial bore and a frusto-conical seat surface. A poppet **101** is disposed within the bore and is maintained in engagement with the valve seat under a predetermined mechanical bias by a spring **103**.

The poppet **101** is a generally rectangular member and when disposed within the cylindrical valve housing bore defines a plurality of flow passages between the poppet edges and the bore. An annular groove **105** interconnects the inner periphery of the flow control mechanism with each of the bores **97** and a recess **107** formed at the inner periphery of the flow control mechanism provides a flow passage from the groove **105** to the ports **104** in the inner cylinder **42**. The annular recess **107** obviates the need for precise radial alignment of the flow control mechanism relative to the ports **104**. An annular recess **109** is formed at the inner periphery of the flow control mechanism **96** defining an annular shoulder for engagement with the stop **98** on the inner cylinder **42**. The draft cushioning mechanism being loosely received by the inner cylinder **42** is movable axially of the inner cylinder within limits defined by the shoulder **100** and the stop **98**. As illustrated in FIGURE 3, a clearance exists between the outer circumference of the flow control mechanism **96** and the inner cylindrical wall **89** of the outer cylinder **40**.

Hydraulic fluid flowing from the inner hydraulic chamber **48** into the draft cushioning chamber **94** passes through the ports **104** and around the flow control mechanism **96** by means of the clearance between the flow control mechanism **96** and the inner cylindrical wall **89** of the outer cylinder. During flow of hydraulic fluid into the draft cushioning chamber **94**, the flow control mechanism **96** is maintained in engagement with the stop **98** by the hydraulic fluid allowing the flow of hydraulic fluid between the surface **100** and the flow control mechanism **96** and around the exterior periphery of the flow control mechanism into the draft cushioning chamber **94**. Upon reversal of the flow of hydraulic fluid from the draft cushioning chamber **94** to the low pressure inner hydraulic chamber **48**, which occurs during extension of the cushioning device to its normally extended or static position, the flow control mechanism **96** is forced by the flowing fluid to move into intimate engagement with annular stop surface **100** of the spacer member **102** to prevent the flow of hydraulic fluid around the outer periphery of the flow control mechanism **96** in the return direction. The flow control mechanism **96** in its relation with the inner cylinder **42** defines a calibrated fixed flow control orifice or fluid flow passage which restricts the flow of hydraulic fluid being exhausted from the draft cushioning chamber **94**, thereby causing substantial energy dissipation as the cushioning unit is extended by its internal pressure after being collapsed by an impact or buff force. The fluid flow control mechanism therefore provides a snubbing action preventing excessively rapid extension of the cushioning unit thereby controlling slamming or rebounding of the same. The draft cushioning mechanism will also dissipate energy applied to the inner cylinder in draft when the inner cylinder is in its collapsed position or neutral position as will be explained in detail hereinbelow.

Referring now to FIGURES 3 and 4 forming an important part of this invention, the hydraulic cushioning unit **38** is particularly adapted to permit contraction of the outer and inner cylinders **40** and **42** thereof only when a predetermined fluid pressure is reached within the high pressure hydraulic chamber **50** of the outer cylinder **42**. The hydraulic cushioning unit will therefore remain in its neutral position until a buff force is applied thereto of sufficient magnitude to raise the fluid pressure within the outer hydraulic chamber to a predetermined level. As pointed out above, the outer hydraulic chamber **50** contains a relatively incompressible fluid, such as hydraulic fluid, which is metered through an orifice **54** in an orifice plate **56** fixed at the inner end of the inner cylinder **42** under control of a metering pin **52** secured to the closed end of the outer cylinder **40**. The metering pin **52** cooperates with the metering orifice **54** to define a velocity sensitive control for the cushioning unit responsive to relative positioning of the inner and outer cylinders **40** and **42**. The free floating piston **44** is disposed for movement within the inner cylinder **42** and separates the compressible fluid such as air, nitrogen, or other suitable gasiform fluid from the hydraulic fluid within the inner hydraulic chamber **48**. In order for the cushioning device **36** to be collapsed under buff forces, hydraulic fluid must flow from the outer hydraulic chamber **50** through the metering orifice **54** and into the inner hydraulic chamber **48**. In conventional oleopneumatic type cushioning units, employing the orifice plate-metering pin arrangement of velocity sensitive control, the interchange of hydraulic fluid between the inner and outer hydraulic chambers is accomplished merely by applying a force on the cushioning unit of sufficient magnitude to cause the pressure in the outer chamber to exceed the inner chamber fluid pressure. The instant invention, however, contemplates the prevention of fluid interchange between the inner and outer hydraulic chambers until a predetermined hydraulic fluid pressure differential is established therebetween. This allows the cushioning unit **38** to remain in a neutral position during conditions of low impact and to allow contraction of the unit only under large magnitude force application.

To prevent the flow of hydraulic fluid from the outer chamber **50** to the inner hydraulic chamber **48** under condition of low impact force applied to the hydraulic cushioning device **38**, a lockup member generally designated **120** is disposed within the inner hydraulic chamber **48** and is adapted to establish a fluid tight seal with the orifice plate **56** about the metering orifice **54**. The lockup member **120** comprises a generally cylindrical body portion **122** formed with a bearing aperture **124** receiving a generally cylindrical enlarged outer extremity of the metering pin **52** in bearing engagement therewith. The body portion **122** is provided with a generally cylindrical bore **128** of larger diameter than the bearing aperture **124**. A piston **130** is provided with an externally threaded extension **132** which is received within an internally threaded bore **134** formed within the metering pin **52**. The piston **130** is provided with an annular sealing element **136** for the establishment of a fluid tight seal with the cylindrical bore **128**. A closure member **138** is re-

ceived within an enlargement formed at the outer extremity of the bore **128** and cooperates with an annular seal member **140** to form a sealed closure for one extremity of the lockup member **120**. The closure member **138** bears against an annular shoulder **139** defined by the enlargement to prevent movement of the closure in one direction. A snap ring **142**, retained within a groove **143** formed at one extremity of the lockup member **120**, retains the closure member **138** against excessive movement in the opposite direction.

The body portion **122** of the lockup member **120** is provided with an annular shoulder **144** and an axial extension defining an annular seal retaining lip **146**. An annular retainer member **148** is threadedly received by the body portion **122** and cooperates with the annular shoulder **144** and annular lip **146** to define a groove **150** for containing a sealing member **152**. The sealing member **152** is preferably formed of an elastomeric material which is impervious to the fluid contained within the hydraulic cushioning unit. The retainer member **148** is provided with an annular lip **154** which cooperates with the lip **146** to define an outer restriction for the annular groove **150**. The lips **146** and **154** bear against the sealing member **152** maintaining the sealing member within the groove **150**, thereby positively preventing the seal **152** from being displaced from the groove by hydraulic fluid flowing under high pressure.

Inadvertent separation of the lockup member **120** from the metering pin **52** is prevented by cooperation of the piston **130** with a shoulder **156** defined by the intersection of the bore **128** and the bearing aperture **124**. Engagement between the piston **130** and the shoulder **156**, however, will occur infrequently due to the pressure depression or partial vacuum developed in the area **158** of the bore **128** defined between the piston **130** and the closure **138** upon movement of the piston within the bore **128**. The seal **136** of the piston **130** maintains a positive fluid tight seal between the piston and the bore **128**, thereby maintaining the area **158** in a sealed condition.

Fluid pressure from the outer hydraulic chamber **50** is communicated to the bore **128** of the lockup member **120** by a series of fluid passages **160** and bears on the piston **130**, thereby maintaining the lockup member **120** in the condition illustrated in FIGURE 4.

The operational concept of the invention depends upon the difference in the effective area of the lockup member **120** which is exposed to the hydraulic fluid pressure within the inner hydraulic chamber **48** as compared to the effective area of the lockup member exposed to fluid pressure within the outer hydraulic chamber **50** and, also, the fluid pressures within the inner and outer hydraulic chambers. For example, the compressible fluid chamber **46** of the inner cylinder may be charged with a gasiform fluid under a static pressure in the range of 260 p.s.i., which acting through the floating piston **44** exerts the same pressure on the hydraulic fluid within the inner hydraulic chamber **48**. The hydraulic fluid within the inner hydraulic chamber therefore acts upon an effective area represented by the diameter D_1 of the sealing member **152** when the sealing member is in sealing engagement with a sealing surface **153** formed on the orifice plate **56**. The hydraulic fluid within the outer hydraulic chamber **50** acts through a smaller effective area A defined by the difference in area defined by diameters D_1 representing the area of sealing contact between the sealing member **152** and the sealing surface of the orifice plate and D_2, representing the diameter of the cylindrical bore **128**. Fluid pressure within the bore **128** acting on the interior shoulder **156** of the lockup member will balance a portion of the force applied by the hydraulic fluid against the exterior surface area of the lockup member defined by the sealing member **152** thereby defining the resultant effective annular area A. It is therefore apparent that due to the construction of the lockup member **120**, the differential or effective areas in communication with the inner and outer fluid chambers effectively provide for automatic pressure responsive control of the lockup member **120**. Since the small effective area A of the lockup member **120** is in communication with fluid within the outer hydraulic chamber **50**, it is apparent that, in order to break the seal between the sealing member **152** and the surface **153** of the orifice plate **56** thereby unlocking the lockup member **120**, the pressure of the hydraulic fluid within the chamber **50** must rise to a level considerably above the pressure of the inner hydraulic chamber **48**. Assuming, for example, that the inner hydraulic chamber **48** is maintained under a static pressure of **260** p.s.i. by the compressible fluid in the chamber **46** and that the effective area A, represented by the difference in diameters D_1 and D_2, is 1.5 square inches, a fluid pressure of 1810 p.s.i. is required in the outer hydraulic chamber to effect unlocking of the lockup member. A hydraulic cushioning unit embodying the invention may be so designed that a buff force in the order of 100,000 lbs. is required to effect a fluid pressure of 1810 p.s.i. in the outer hydraulic chamber. It is apparent therefore that in accordance with such an embodiment of the invention a buff force below 100,000 lbs. will not unlock member **120** and the cushioning unit **38** will remain in its neutral and locked position.

With the hydraulic unit **38** in its neutral locked condition, the lockup member **120** will effectively prevent the flow of hydraulic fluid between the inner and outer hydraulic chambers when the unit is subjected to a low level buff force or a draft force. The cushioning unit **38** when subjected to a draft force in its neutral locked position will extend to its fully extended position without any transfer of hydraulic fluid between the inner and outer hydraulic chambers. This extension will be small, for example less than two inches in a unit having a 30 inch stroke, and will cause a vacuum to be formed within the outer hydraulic fluid chamber **50** developing a void or vapor bubble therein. Hydraulic fluid within the draft cushioning chamber **94** will be metered into the inner hydraulic chamber **48** by the flow control mechanism **96** in the manner described above. The additional volume of hydraulic fluid forced into the inner hydraulic chamber **48** will force the piston **44** toward the outer extremity of the inner cylinder thereby further compressing the compressible fluid within the chamber **46**.

Upon dissipation of the draft force the cushioning unit **38** will be forced to return to its neutral position by the compressed gas within the chamber **46** which acts on the piston and forces a volume of hydraulic fluid to flow from the inner hydraulic chamber **48** into the draft cushioning chamber **94**. At the same time the void or vapor bubble will dissipate and the cushioning unit **38** will have returned to its neutral position.

The resilient draft gear **28** will act simultaneously with the hydraulic cushioning unit **38** in draft between the neutral and fully extended positions to effect a smooth force transition in the same manner discussed above.

Operation

When the coupler structures **12** of the railway cars are impacted with a force which is less than that required for unlocking the lockup member **120** with the cushioning unit in its neutral locked condition, the cushioning unit because of the lockup member will remain in its neutral position as illustrated in FIGURE 3 and will transmit the force directly from the resilient draft gear to the underframe of the railway car. The resilient draft gear under low impact loads (buff loads under 100,000 lbs., for example) will therefore be effective acting alone to dissipate sufficient energy to afford ample protection to the railway car lading under such low impact conditions. Assuming that the coupler structure **12** of the railway car has received a buff impact which is above the force required to unlock the cushioning unit **38**, the pressure of

the hydraulic fluid within the outer hydraulic chamber **50** will rise to a sufficient magnitude that acting through the small effective annular area A represented by the difference in the areas defined by diameters D_1 and D_2, the force produced on the lockup member will be greater than the force on the lockup member developed by the fluid pressure within the inner hydraulic chamber **48** acting through the area defined by the diameter D_1. Under this condition the lockup member **120** will break the sealing engagement between the sealing member **152** and the sealing surface **153** of the orifice plate **56**, thereby allowing the flow of hydraulic fluid from the outer hydraulic chamber **50** to the inner hydraulic chamber **48** causing contraction of the hydraulic cushioning unit **38**. Contraction of the cushioning unit **38**, after the unlocking operation has taken place, is therefore controlled solely by the cooperative relationship between the metering pin **52** and the orifice **54** thereby causing the cushioning unit to be velocity sensitive in its contraction. Fluid flowing from the outer hydraulic chamber **50** to the inner hydraulic chamber **48** subsequent to the unlocking operation is under control of the metering pin **52** causes energy dissipation controlled by the effective size of the metering orifice. Hydraulic fluid flowing into the inner hydraulic chamber **48** causes the free floating piston **44** to move toward the outer extremity of the inner cylinder **42**, thereby compressing the compressible fluid within the pneumatic chamber **46**. During a maximum buff impact the fluid pressure of the compressible gasiform fluid within the chamber **46** may be raised from its precharged pressure of 260 p.s.i. to a pressure range of 1500 p.s.i., for example.

After the hydraulic cushioning unit **38** has been fully compressed by the buff force and the buff force has been dissipated, the fluid pressure within the pneumatic chamber **46** will force hydraulic fluid disposed within the chamber **48** through the metering orifice **54** into the outer hydraulic chamber **50**. This will allow the hydraulic cushioning unit **38** to expand toward its neutral or FIGURE 3 position.

As was explained hereinabove, upon contraction of the cushioning unit **38** hydraulic fluid will be forced through the ports **104** and into the draft cushioning chamber **94**. Return of the hydraulic unit to its neutral position therefore will be determined by the calibrated flow passage structure defined by the clearance between the inner cylinder **42** and the inner periphery of the flow control mechanism **96**. Extension of the cushioning unit **38** therefore will be retarded by the flow control mechanism **96** thereby preventing excessively rapid extension of the cushioning unit which could produce undesirable results such as slamming the cushioning unit to its fully extended position or rebounding of the unit. Rebounding of the cushioning unit is undesirable oscillation produced by excessively rapid over extension of the cushioning unit.

Assuming that the hydraulic cushioning unit **38** is subjected to a draft force while the unit is in a contracted condition, the fluid flow control mechanism **96** will be moved by the hydraulic fluid axially into engagement with the annular surface **100** of the spacer member **102**, thereby preventing the flow of hydraulic fluid around the exterior periphery of the flow control mechanism in the return direction. Since under this condition fluid may not flow around the exterior periphery of the flow control mechanism and since the calibrated orifice or flow passage structure is too small to afford sufficient fluid flow from the draft cushioning chamber **94** to the inner hydraulic chamber **48** to dissipate draft forces, it is obvious that fluid pressure within the draft cushioning chamber **94** will build rapidly to a high level. The fluid under pressure will therefore act upon the poppet structures **101** in the flow control mechanism **96** and when reaching a predetermined pressure differential relative to the pressure within the inner hydraulic chamber will cause actuation of the poppet structures, allowing a sufficient flow of fluid through the flow control mechanism to the inner hydraulic chamber **48** from the draft cushioning chamber **94** and resulting in substantial dissipation of energy. The flow control mechanism **96** is therefore responsive to the fluid pressure differential between the draft cushioning chamber **94** and the inner hydraulic chamber **48** to control extension of the cushioning unit and to cause dissipation of energy in draft. The draft cushioning chamber **94** and the flow control mechanism **96** are so designed that anticipated draft forces are readily dissipated. It is pointed out also that the draft cushioning mechanism cooperates with the resilient draft gear **28** allowing a smooth dissipation of draft forces thereby protecting the lading within the railway car. For example, the hydraulic cushioning unit will allow draft extension thereof a predetermined distance beyond the neutral position thereof. As pointed out above, with the cushioning unit **38** in its locked condition, full extension of the cushioning unit will occur without interchange of fluid between the inner and outer hydraulic chambers. When the hydraulic cushioning unit has been fully extended, further dissipation of draft energy occurs only in the resilient draft gear **28**. The resilient draft gear effectively prevents subjection of the cushioning unit to excessive draft forces and effects a smooth transition of forces between the coupler and underframe structures of the railway car.

As the hydraulic cushioning unit **38** moves from a contracted condition to its neutral position as illustrated in FIGURE 3, the sealing member **152** will again engage the sealing surface **153** of the orifice plate **56** thereby effecting a seal and again defining the area differential defined by the difference in areas represented by the diameters D_1 and D_2 as indicated hereinabove. The hydraulic cushioning unit upon reaching its neutral position therefore will again become locked and subsequent contraction of the cushioning unit will occur only upon reaching sufficient fluid pressure within the fluid chamber **50** to unlock the cushioning unit in the manner described above.

It will be evident from the foregoing that I have provided a unique oleopneumatic cushioning unit which effectively prevents downgrade run-ins of the hydraulic cushioning unit due to train operation. Due to the unique construction of the instant invention, the cushioning device will not be permitted to go solid or to be completely contracted under the application of low velocity impacts such as slack run-in during a downgrade locomotive braking condition, for example. The hydraulic cushioning unit incorporating the instant invention will be maintained locked in a neutral position until such time as buff forces developed by impact become sufficiently high to cause an increase in hydraulic pressure within the outer hydraulic chamber of the cushioning unit to cause unlocking of the lockup member of the respective cushioning units. Since the hydraulic cushioning unit incorporating the invention will normally be retained in its neutral position, continuous back and forth movement of the free piston due to train action is effectively prevented and piston seals therefore experience little wear thereby prolonging the life of the cushioning unit. Subsequent to impact forces and unlocking of the lockup member **120** and dissipation of the impact forces, slack run-outs of the coupled railway cars will be effectively prevented by the draft cushioning mechanism set forth hereinabove. Excessively rapid extension of the cushioning unit as well as rebounding of the same is effectively prevented by the draft cushioning mechanism, thereby reducing any tendency of damage to the railway car lading by slamming or rebounding of the cushioning unit. The hydraulic cushioning unit of the invention cooperates with the resilient draft gear structure both in buff and draft, thereby producing a cushioning effect thereby producing smooth force transition thereby giving maximum protection to the railway car lading. Therefore it is seen that this invention is one well adapted to obtain all of the objects hereinabove set forth together with other

advantages which become obvious and inherent from the description of the apparatus itself.

It will be understood that certain combinations and subcombinations are of utility and may be employed without reference to other features and subcombinations.

What is claimed is:

1. A railway car having an underframe structure, a coupler device movably carried by said underframe structure, an oleopneumatic cushioning unit interposed between said underframe structure and said coupler device and adapted to dissipate buff forces applied to the coupler device to protect the lading within the railway car from excessive shock, said cushioning unit comprising inner and outer tubular members disposed in telescoping relation and being movable one relative to the other between collapsed, neutral and fully extended positions, means closing the outer extremities of said tubular members, an orifice plate closing the inner extremity of the inner tubular member and defining an inner variable volume low pressure hydraulic chamber within said inner tubular member and an outer variable volume high pressure hydraulic chamber within said outer cylinder, hydraulic fluid filling said inner and outer hydraulic chambers, means maintaining the hydraulic fluid within said inner hydraulic chamber under a static pressure, a metering orifice formed in said orifice plate allowing the interchange of hydraulic fluid between said inner and outer hydraulic chambers, a metering pin cooperating with said metering orifice to define a velocity sensitive control for the interchange of hydraulic fluid between the inner and outer hydraulic chambers, said orifice plate defining a substantially planar sealing surface, a lockup member disposed within said low pressure hydraulic chamber of said cushioning unit and adapted in the neutral to fully extended positions of the cushioning unit to form a seal with said sealing surface to prevent the interchange of hydraulic fluid between said hydraulic chambers, said lockup member when in sealing engagement with said sealing surface defining a first effective surface area of said lockup member through which said static pressure within said inner low pressure hydraulic chamber acts to maintain said sealing engagement, said seal defining a second surface area of said lockup member through which fluid in said outer hydraulic chamber acts, said lockup member providing an inner surface area through which said fluid pressure in said outer hydraulic chamber acts to balance a portion of the force acting on said second surface area thereby defining an effective third surface area which is less than said first surface area.

2. A railway car as set forth in claim 1, said lockup member comprising a body portion having a bearing aperture formed therein, said metering pin having a generally cylindrical end portion received within said bearing aperture, said body portion having a bore formed therein of larger size than said bearing aperture and defining a pressure balancing chamber, at least one fluid passage communicating fluid pressure exteriorly of said lockup member to said pressure balancing chamber, whereby in the locked position of said lockup member fluid pressure within said pressure balancing chamber develops a force balancing a portion of the force applied to said lockup member by the fluid within said outer hydraulic chamber.

3. A cushioning unit comprising an outer tubular member, an inner tubular member slidably disposed in radial spaced relation within said outer tubular member, base means closing the outer ends of said members, sealing means secured to one of said members and closing said radial space between the members to form a main variable volume chamber enclosed by said members, an orifice plate secured to the inner open end of said inner tubular member and dividing said main chamber into an inner low pressure hydraulic fluid chamber and an outer high pressure hydraulic fluid chamber, a fixed area opening in said orifice plate, a metering pin carried by the base of said outer tubular member and movable through the opening in said orifice plate to vary the effective area thereof, hydraulic fluid in said inner and outer chambers, a movable piston disposed within said inner tubular member and defining a compressible fluid chamber, gas under pressure disposed within said compressible fluid chamber and acting on said piston to pressurize said hydraulic fluid within said inner hydraulic fluid chamber, and lockup valve means carried by said metering pin within said inner chamber, said lockup valve means having an annular sealing element establishing an annular seal with said orifice plate about said orifice plate opening thereby isolating said outer chamber from the effect of said gas pressure, said hydraulic fluid in said inner low pressure hydraulic chamber acting against said lockup valve under the influence of said gas pressure and causing said lockup valve to maintain said annular seal with said orifice plate thereby preventing the interchange of hydraulic fluid between said hydraulic chambers, the seal between said lockup valve and said orifice plate being broken by a force differential developed by a predetermined increase in hydraulic fluid pressure within said high pressure outer hydraulic chamber which acting through a small area of said lockup valve in communication with the hydraulic fluid in said high pressure outer hydraulic chamber is effective to overcome the force created by the hydraulic fluid within said inner hydraulic chamber acting under pressure influence of said gas pressure against a large area of said lockup valve encompassed by said annular seal, thereby allowing the occurrence of metered interchange of hydraulic fluid between the inner and outer hydraulic chambers to achieve dissipation of energy only when hydraulic fluid pressure in said high pressure hydraulic chamber reaches a predetermined level.

4. The structure of claim 3 characterized in that said valve means is mounted on said metering pin in interengagement therewith and for axial movement relative thereto, vacuum means normally maintaining said valve means in full interengagement with said metering pin and permitting movement of said valve means relative to said metering pin upon the application of external force to said tubular members.

5. The structure of claim 4 characterized in that said lockup valve means is formed with a generally cylindrical bore having a closed end, the free end of said metering pin extending into said bore in sealing engagement therewith, upon movement of said lockup valve means relative to said metering pin a vacuum developing within said bore and biasing said lockup valve means toward full interengagement with the free end of said metering pin.

6. The structure of claim 4, characterized in that said lockup valve means is formed with a generally cylindrical bearing aperture adapted for a sliding fit with the free extremity of said metering pin, a generally cylindrical bore formed in said lockup valve means of larger diameter than said bearing aperture and coaxial therewith, a piston fixed to the free extremity of said metering pin and having a fluid tight fit with said bore, upon movement of said lockup valve means relative to said metering pin, a vacuum developing within said bore and biasing said lockup valve means toward full interengagement with said metering pin.

7. A hydraulic cushioning unit for use in railway cars comprising inner and outer tubular members disposed in telescoping relation and being relatively movable between collapsed neutral and fully extended positions, means closing the outer ends of said tubular members, said inner member having an orifice plate fixed at its inner extremity defining a barrier separating the inner and outer variable volume hydraulic chambers, hydraulic fluid disposed within said hydraulic chambers, said orifice plate having a metering orifice formed therein for allowing the interchange of hydraulic fluid between the inner and outer hydraulic chambers, means defining a variable volume gas chamber within said inner tubular member,

pressurized gas disposed within said gas chamber and maintaining the hydraulic fluid within the inner hydraulic chamber under pressure, a metering pin connected to one of said tubular members and cooperating with said metering orifice to control the interchange of hydraulic fluid between said inner and outer hydraulic chambers upon relative movement of said tubular members, a lockup member disposed within said inner hydraulic chamber and supported by said metering pin, said lockup member in the neutral position of said cushioning unit establishing a single annular seal with said orifice plate about said metering orifice thereby preventing the interchange of hydraulic fluid between said inner and outer hydraulic chambers, said lockup member when in sealing engagement with said orifice plate presenting a large effective area thereof to the pressurized hydraulic fluid in said inner hydraulic chamber and a small effective area thereof to hydraulic fluid within said outer hydraulic chamber, whereby a substantially greater pressure is required within said outer hydraulic chamber than in said inner hydraulic chamber to break said sealing engagement.

8. A hydraulic cushioning unit as set forth in claim **7**, said lockup member having an axial bore extending partially therethrough, cylindrical means on said metering pin in fluid tight sealing engagement within said bore, said cylindrical means and bore cooperating to develop

a vacuum within said bore upon relative movement of said lockup member and said metering pin whereby said lockup member is biased toward its fully interengaged relation with said metering pin.

9. A hydraulic cushioning unit as set forth in claim **8**, said cylindrical means comprising a piston fixed to the free extremity of said metering pin, a sealing member carried by said piston and establishing sealing engagement between said piston and the wall defining said bore.

10. A hydraulic cushioning unit as set forth in claim **8**, a retainer member received by said lockup member and cooperating with said lockup member to define an annular groove having an outer restriction, an annular sealing member of larger dimension than said restriction received and retained in said groove, a sealing portion of said sealing member extending beyond said lockup member and adapted for sealing engagement with said orifice plate.

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