Aug. 26, 1969 W. T. BLAKE 3,463,328

DRAFT GEAR TRAIN ACTION CONTROL VALVE

Filed Jan. 26, 1967 3 Sheets-Sheet 1

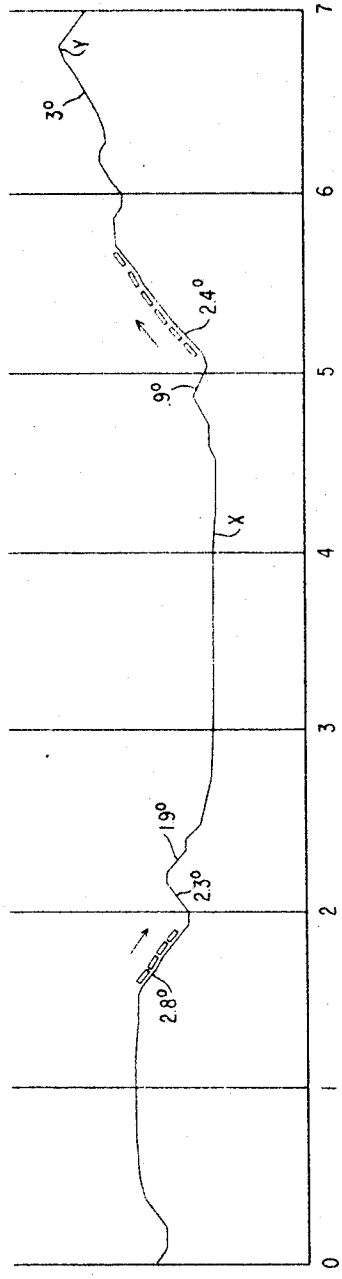
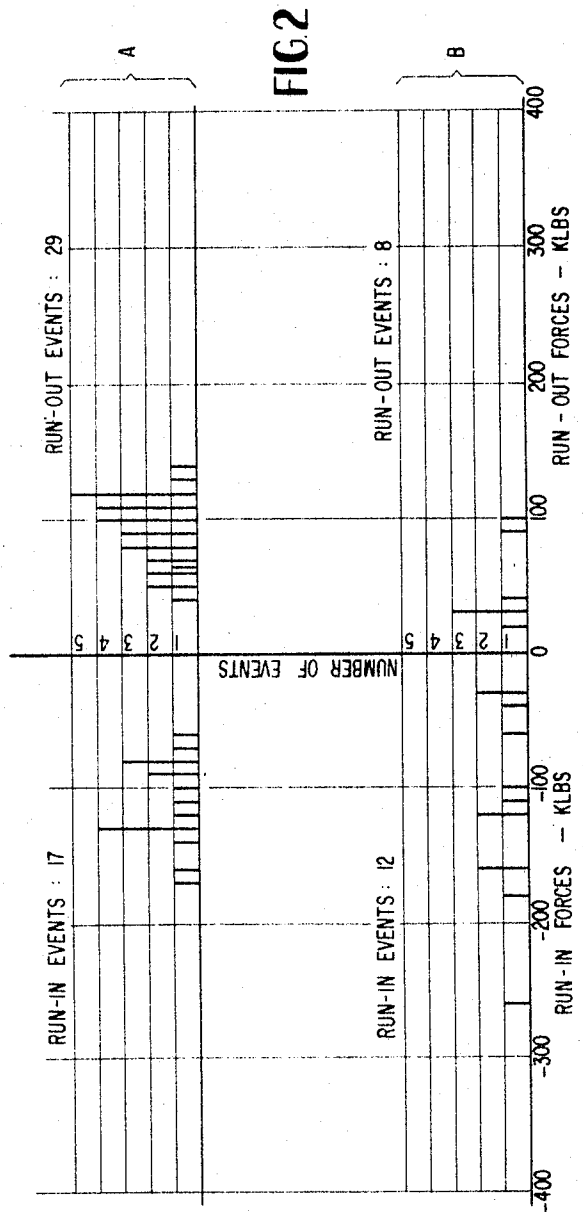


FIG. 1

FIG. 2



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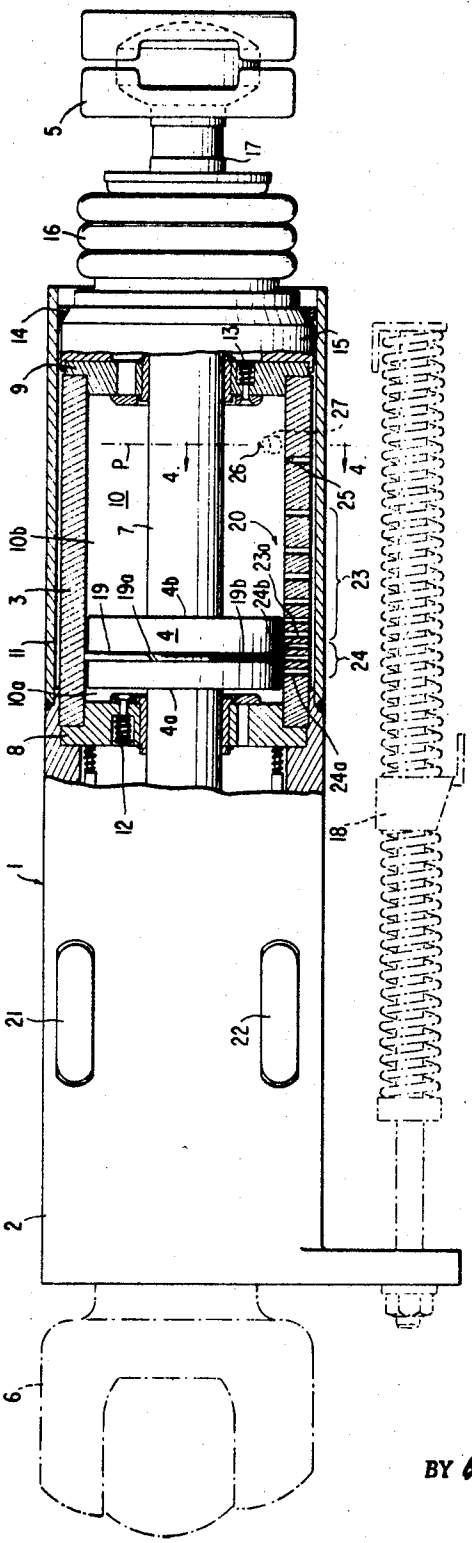


FIG 3

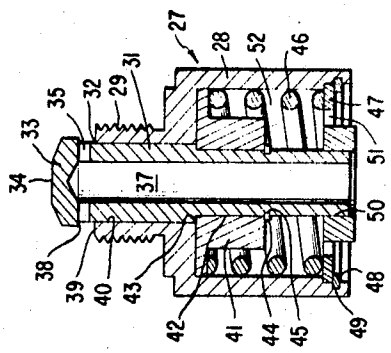


FIG 5

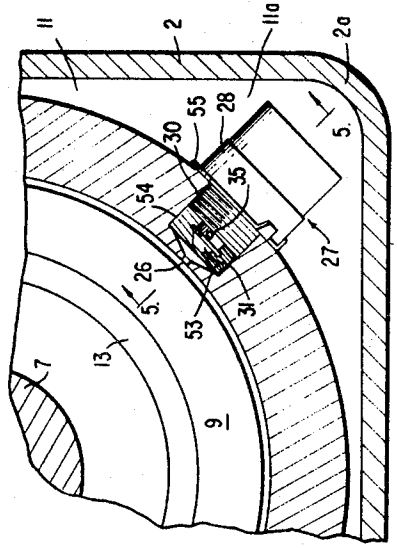


FIG 4

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FIG. 6

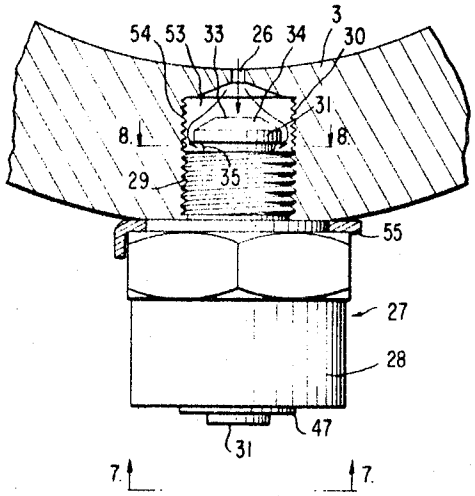
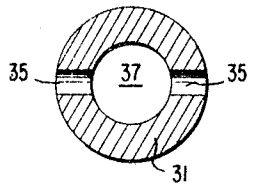


FIG. 8



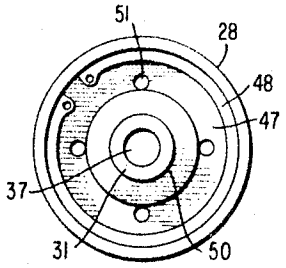


FIG. 7

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United States Patent Office 3,463,328
Patented Aug. 26, 1969

3,463,328

DRAFT GEAR TRAIN ACTION CONTROL VALVE

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Filed Jan. 26, 1967, Ser. No. 611,931
Int. Cl. B61g 9/06, 9/04, 11/02
U.S. Cl. 213—43 2 Claims

ABSTRACT OF THE DISCLOSURE

A valve apparatus for inclusion in railway draft gear of the type which includes a fluid shock absorbing assembly comprising cylinder means and piston means. The valve means is disposed to regulate an outflow of fluid from a piston biasing fluid zone of the assembly. In the context of this draft gear, the valve is characterized by port means carried by a movable valve member. This port means causes the movable valve member to be closable in response to fluid flow therethrough emanating from the piston biasing zone of the assembly. Resilient means tend to bias the movable valve member to an open port position.

BACKGROUND OF INVENTION

In recent years, considerable attention has been paid to the development of hydraulic shock absorbing mechanisms to cushion forces imposed on railway car couplings.

A primary function of these shock absorbing mechanisms is to absorb the exceedingly high buff forces which are often imposed on couplings during the intercoupling of railway cars in railway yards.

Particularly effective railway shock absorbing mechanisms are described in Blake Patent 2,944,639 and in the Seay Patent 3,301,410.

In certain railway systems, such as some in Canada, extremely adverse track conditions are encountered which tend to induce an extraordinarily high degree of train action. Train action is a term applied to phenomena which occur as a consequence of slack in the couplings between railway cars, which slack enables cars to undergo relative movement while a train is in motion. Specifically, "train action" denotes the equalizing of speed of adjacent cars which have undergone relative movement. Where this relative movement is characterized by the adjacent cars moving apart, this action is termed "run-out." Where adjacent cars are tending to converge, this train action is referred to as a "run-in."

Such run-out and run-in phenomena produces a variety of undesirable effects. For example, during the period while couplings are extending or contracting, train crewmen have reported that they experience a "floating" sensation. Once couplings have ceased their extension or contraction, train action shocks of a high magnitude are imposed on cars which are noticeably severe and occasionally severe enough to jolt and injure crewmen. Such train action shocks also obviously are injurious to train components and may produce derailment tendencies.

The magnitude of the train action problem is discussed in connection with experimental and evaluation efforts pertaining to Canadian railway systems in a paper authored by F. E. King and R. W. Radford, entitled "The Effect of Freight Car Cushioning Characteristics on Train Action," and presented at the winter annual meeting of the American Society of Mechanical Engineers held in New York in November and December of 1966. Some appreciation of the track profile conditions which produce excessive train action may be gathered by reference to this article.

Coping with the train action phenomena presents requirements which are not aligned with those involved in maintaining the desired characteristics of coupling bars while they are not being subjected to heavy, extraneous forces. For example, in order to restore at a satisfactory fast rate from a full buff condition, hydraulic shock absorbing mechanisms are necessarily characterized by relatively high capacity restricted passage means for venting fluid from coupler bar, movement resisting, fluid bodies. However, in order to effectively limit coupling movement while trains are in motion where coupling forces of a considerable lower magnitude are generally involved, it is necessary to provide a more restricted venting of fluid from such fluid bodies.

SUMMARY OF INVENTION

An overall apparatus and technique for effectively controlling and minimizing train action events without adversely interfering with the shock absorbing capability of railway draft gear is described in an application of Jack G. Stephenson and Robert E. Abbott entitled Draft Gear Train Action Control System, Ser. No. 610,553, filed Jan. 20, 1967. In the context of this overall apparatus and technique, it is a principal object of the present invention to provide a valve mechanism which affords effective control over train action run-out events, i.e., tendencies of a coupling bar to extend while a train is in motion.

It is another principal object of the invention to provide a valve apparatus for enabling railway car shock absorbing devices to minimize train action run-out events while at the same time permitting such devices to restore at a satisfactory rate from a full buff position to a neutral position when not subjected to train action forces.

A further object of the invention is to provide a valve apparatus for effectively controlling train action run-out phenomena without requiring drastic alteration of existing shock absorbing mechanisms such as those described in the aforesaid Blake patent and Seay application.

It is a related object of the invention to provide such an improved valve structure which is compact, structurally rugged, and readily incorporated in existing railway draft gear.

Yet another object of the invention is to provide such an improved valve structure which is positive in its valve closing and opening action and which is positively protected against undesired valve chattering tendencies.

A still further object of the invention is to provide such an improved valve apparatus which is operable in response to a flow of fluid from a piston biasing zone of hydraulic draft gear, with the valve being biased to a normally open position and closable in response to such a flow generated by train action forces acting upon a coupling bar.

In accomplishing these objects, there is provided an improved valve mechanism which is integrally incorporated in railway draft gear of the type which includes a fluid shock absorbing assembly comprising cylinder means and piston means. In this apparatus setting for the improved valve mechanism, the piston means is connected with one of a coupling portion and body portion of a railway car and the cylinder means is connected with the other of these coupling and body portions. The valve means, in this context, serves to regulate an outflow of fluid from a piston biasing zone of the assembly.

The improved valve means includes a valve body, a movable valve member, and a valve seat carried by the valve body. Port means carried by the movable valve member serves, in response to a fluid flow therethrough emanating from the piston biasing zone of the assembly, to create a pressure differential across the movable member tending to move the movable valve member into port means closing engagement with the valve seat. A resilient

means biases the valve member away from the valve seat so as to yieldably position the valve member with its ports open.

In the context of this improved valve arrangement, an additionally and independently significant combination involves this improved valve structure in combination with a fluid reaction surface on the movable valve member upon which fluid outflow from the assembly is adapted to impinge and apply a valve closing reaction force.

A further independently significant combination involves the improved valve arrangement as above described in further combination with dashpot means, i.e., movement retarding means, which tend to retard movement of the valve member toward the valve seat.

DESCRIPTION OF DRAWINGS

In describing the invention, reference will be made to the preferred embodiment of the apparatus illustrated in the appended drawings.

In the drawings:

FIGURE 1 provides an elevational profile of a portion of a test track over which trains were run in order to evaluate the improved ability of the cushioning or shock absorbing mechanism of the present invention to minimize train action events and reduce their severity;

FIGURE 2 provides a graphical comparison of the performance of railway cushioning gear constructed in general accordance with the present invention as opposed to the performance of conventional, friction-type, draft gear;

FIGURE 3 provides a partially sectioned, elevational view of a shock absorbing mechanism of the present invention in the format of an improvement in the mechanism disclosed in the aforesaid Seay application, Ser. No. 527,347;

FIGURE 4 provides an enlarged transverse sectional view of a portion of the FIGURE 3 mechanism illustrating structural details of a valve controlled vent, as viewed along the section line 4—4 of FIGURE 3;

FIGURE 5 provides a still further enlarged transverse sectional view of the valve mechanism shown in FIGURE 4 as viewed along the section line 5—5 of FIGURE 4;

FIGURE 6 provides an enlarged elevational view of the valve shown in FIGURE 5;

FIGURE 7 provides an end view of the FIGURE 6 valve as viewed along the view direction 7—7; and

FIGURE 8 provides a transverse sectional view of the flow restricting ports included in a valve member of the FIGURE 6 valve assembly.

OVERALL APPARATUS

FIGURES 3 through 8 illustrate structural details of a railway cushioning device fabricated in accordance with the present invention.

FIGURE 3 illustrates an improved form of the cushioning device illustrated and described in detail in the aforesaid Seay Patent No. 3,301,410. Briefly reviewing the structural characteristics of this cushioning device **1**, without unnecessarily redescribing structure which is known in the art, it will be appreciated that the basic components of this device **1** comprise a housing **2** having a generally rectangular cross section, a high pressure cylinder **3** contained within housing **2** and having a cylindrical cross section, a piston **4** mounted for telescoping movement within the cylinder **3**, and an anchor assembly **5**. A conventinal coupling bar **6**, shown in phantom lines in FIGURE 3, is adapted to be connected by conventional mounting keys or a pivot pin to housing **2**.

Housing **2** is adapted to be movably mounted, i.e., slidably mounted in a sill beneath a railway car body, with anchor assembly **5** being fixedly anchored to a sill portion. In this manner, anchor assembly **5**, which is connected to a piston rod **7** extending from and connected with piston **4**, serves to fixedly position the piston **4** within a railway car sill.

Cylinder end walls **8** and **9** are secured to opposite ends of cylinder wall **3** so as to define a high pressure cavity **10** within cylinder **3** and a relatively low pressure zone **11** encircling cylinder **3** and disposed between the housing **2** and the cylinder **3**. This zone **11** communicates through openings in end walls **8** and **9**, not shown, with check valve means **12** and **13** contained in walls **8** and **9** respectively. This general check valve and passage arrangement is fully described in the aforesaid Blake Patent 2,944,681 and Seay Patent No. 3,301,410, and for that reason, need not be reillustrated.

As described in this Blake patent, and Seay application, the low pressure zone surrounding the cylinder wall **3** and end walls **8** and **9** is sealed at opposite ends of the housing **2**. Thus, as shown in FIGURE 3, the left end of this low pressure zone is sealed by annular seal means **14** interposed between housing **2** and a housing end plate **15**.

Piston rod **7** telescopingly passes through the end plate **15** and is in sealed but slidable engagement with this plate. An axially extendable and contractable seal **16**, connected to the end plate **15** and a piston rod portion **17**, serves to protect the portion of the piston rod **7** which passes slidably through the housing end plate **15**.

A coil spring restoring mechanism **18**, preferably of the types disclosed in the U.S. Blake Patent 3,047,162 or in U.S. Abbott et al. Patent 3,233,747 filed Jan. 22, 1964, is incorporated in device **1**. This restoring mechanism **18** tends to resiliently and yieldably maintain the housing **2** within a railway car sill in a predetermined neutral position so as to obtain a desired position of the piston **4** within the cylinder **3** when the coupling bar **6** is separated from the coupling of an adjacent railway car. In this connection, it will be understood that in FIGURE 3, the piston **4** is illustrated in a position where it has been displaced from its usual neutral position in response to the application of buff force to the coupling bar **6**. In the FIGURE 3 arrangement, it is contemplated that the restoring mechanism **18** would ordinarily tend to position the piston **4** so that a split piston ring **19** having a ring split or gap **19*a*** is carried by the piston **4** will have its draft or right edge **19*b*** substantially aligned with the plane P. This use of this ring provides an improved seal between the piston **4** and cylinder **3** and thus represents an improvement over the devices of the aforesaid Seay application and Blake patents.

Port means **20**, comprising a plurality of radial ports spaced longitudinally of the cylinder wall **3** and extending through the cylinder wall **3**, provide fluid communication between the zones **10** and **11**.

Key slots **21** and **22** may be provided in housing **2** to receive conventional cushioning device mounting keys of the type described, for example, in Blake Patent 3,207,324 and Blake Patent 3,047,162. These mounting keys extend fairly snugly through the slots **21** and **22** and into relatively long, longitudinally extending sill slots. The length of the sill slots may serve to determine the stroke of the cylinder **3** relative to the fixed piston **4** and thus limit the stroke in both buff and draft directions. That is, when the mounting keys engage either end of a sill slot, further movement of the cylinder **3**, housing **2**, and coupling bar **6** is prevented. In some instances, a housing flange carried by the housing **6**, as shown for example at element **65** in the Blake Patent 3,207,324, may abuttingly engage the sill end in response to imposition of buff force so as to delineate the termination of buff movement of the housing **2**, cylinder **3**, and coupling bar **6**.

The basic mode of operation of the previously described components of the FIGURE 3 assembly is described in considerable detail in the aforesaid Blake patents, Abbott et al. patent and Seay application.

In brief review, when buff forces are imposed on the coupling bar **6** with the piston **4** disposed at its neutral position, fluid will be displaced from an annular zone **10***a* to the left of the piston buff end **4***a* through the port means **20** into the zone **11**. This displaced fluid will travel from the zone **11** through the previously described openings in the plate **9** to pass through the check valve means **13** into the zone **10***b* to the right of the piston **4**. The port means **20**, in restricting flow from the zone **10***a*, will cause the fluid within the zone **10***a* to absorb buff forces and thus cushion the imposition of buff shock on a railway car.

Conversely, when draft force is applied to the coupling bar **6** with the piston **4** disposed as shown in FIGURE 3, fluid will be displaced by the piston draft end **4***b* from the annular zone **10***b* through the port means **20**. This displaced fluid will flow through the previously described openings in the end plate **8** and return to the annular zone **10***a* through the check valve means **12**. As will be appreciated, check valve means **12** and **13** are biased to a closed position by conventional coil spring mounting arrangements and open inwardly into the zones **10***a* and **10***b* in response to a pressure differential across them characterized by a low pressure zone adjacent the check valves within the cylinder cavity **10**.

IMPROVED PORTING AND PISTON ARRANGEMENT

The FIGURE 3 assembly is characterized by an improved arrangement of venting ports and piston components which provide effective control over train action events without impairing the ability of the FIGURE 3 assembly to absorb heavy buff impact under coupling conditions.

This improved porting arrangement is characterized by a first plurality of longitudinally spaced, cylinder wall ports **23**. Ports **23** are exponentially spaced and progressively and exponentially decrease in spacing in a longitudinal direction extending toward the coupling bar **6**. In order to obtain the desired orifice coefficient characteristics of these flow-restricting ports, the dimensional relationships in the aforesaid Seay application should be followed. That is, the port diameters should fall within a range of about .28 inch to about .38 inch. In practice a diameter of 5/16 of an inch for each port **23** has been found to produce satisfactory results, consistent with the teaching of this Seay application.

A second plurality of venting ports **24**, longitudinally displaced from the first plurality of ports **23**, is disposed between the ports **23** and the coupling bar **6**. Ports **24** are preferably also dimensioned in accordance with the teachings of the aforesaid Seay application and in practice, have been found to perform satisfactorily when having a diameter of about 5/16 of an inch. However, the ports **24**, rather than being exponentially spaced, are equally spaced.

In a tested embodiment where the piston **4** had a diameter of about 9 inches and where the relative travel of piston **4** within cylinder **3** is about 10 inches, i.e., when housing **2** and cylinder **3** have a full stroke of 10 inches, satisfactory results have been obtained with six ports in the port means **23** and three port means in the port means **24**, with all of these ports being 5/16 of an inch in diameter.

As will be appreciated, the ports **23** and **24** are disposed to the left of the piston **4** when it has assumed its neutral position, as determined by the spring mechanism **18**. Thus exponentially spaced ports **23** and uniformly spaced ports **24** serve to control fluid flow from the cylinder cavity **10** during the imposition of buff force when the cylinder is moved from its neutral position.

As shown in FIGURE 3, the cylinder **3** has been moved so as to position the piston **4** at the end of its buff stroke. In this position, the leftmost port **24***a* is displaced from the leftmost end **4***a* of the piston **4** by a distance of .75 inch. With these dimensional relationships, the longitudinal width of the piston **4** has been gauged at about 2½ inches with the ring **19** having a width of about .375 inch and being spaced from the left or buff end **4***a* of the piston **4** a distance of 1.22 inches.

Although the split piston ring **19** provides slidable and sealing engagement radially between the piston **4** and cylinder **3**, some limited and throttled fluid flow between the cylinder and piston may take place through the ring split zone **19***a*. Thus, when the piston **4** has been positioned so as to cover the leftmost port **24***a*, continued, but highly resisted, piston movement relative to cylinder **3** may continue, with fluid venting from zone **10***a* through ring split **19***a* into zone **10***b*. As will be further noted, when coupling **6** has moved to its full buff position so as to position piston **4** as shown in FIGURE 3, the port means **24** will have been covered.

It will also be appreciated that after the left edge **4***a* of the piston **4** has covered the port **24***a*, some limited and throttled leakage may take place between the piston **4** and cylinder **3** to the left of ring **19** as shown in FIGURE 3, thereby enabling fluid to flow from the zone **10***a* into one or more of the ports **24**. However, it is believed that the combined flow capacity of this leakage, as well as the leakage through ring split zone **19***a*, is not as great as the flow capacity of individual port **24**. Thus, when the cylinder **3** has moved sufficiently to cause the piston **4** to cover the ports **24**, further buff movement of the cylinder **23** will take place with intensified or increased resistance.

With the buff controlling port means **23** and **24** having been described, it now becomes appropriate to describe the draft controlling port means of the FIGURE 3 apparatus.

It should be here observed, that in the FIGURE 3 embodiment, the cylinder **3** may travel nine inches from its neutral position in response to the imposition of buff force and one inch from its neutral position in response to the imposition of draft force.

This draft controlling port means comprises, in the right or draft end of the cylinder wall **3**, a relatively restricted single port **25** and a valved port **26**. As illustrated, restricted port **25** is longitudinally interposed between valve port **26** and first port means **23**. Thus, restricted port **25** may be viewed as third vent means with valve port **26** being viewed as fourth vent means.

At this point, it should be observed that in indicating that various ports of the port or vent means are longitudinally displaced, what is meant is that the ports are aligned with planes perpendicular to the longitudinal axis of the assembly **1**, with these planes being longitudinally displaced.

Longitudinal spacing, as previously used in this discussion, and as used subsequently, is intended to encompass not only longitudinal spacing with the maintenance of axial port alignment, but also longitudinal displacement where ports are circumferentially displaced from each other. Indeed, in practice, it has been found that it is undesirable to longitudinally align all of the venting ports of the apparatus. Because of the extreme fluid pressures encountered within the cavity **10**, metal deformation in the vicinity of the venting ports may tend to result. Where all of the ports are aligned, this axial alignment of deformed areas may tend to produce an undesired interference action between the piston and cylinder.

As shown in FIGURE 3, valved port **26** is preferably aligned with plane P, i.e., the location of the draft or right edge **19***b* of the piston ring **19** when the piston **4** is in its full draft position. Thus, with the valved port **26** open, even though a portion of the piston **4** covers the port **26**, fluid may flow from the zone **10***b*, to the left as shown in FIGURE 3 and between the piston **4** and cylinder wall **3**, and then through the port **26** into the low pressure zone **11**. This zone outflow, coupled with the by-pass outflows from the zone **10***b* through the ring split **19***b* to the zone **10***a*, will enable a desired rapidity

of cylinder movement near the end of a draft stroke under conditions where the valve port **26** remains open. This restoration rapidity is desirable under "free air" conditions—i.e., when coupling bar **6** is disconnected from an adjacent car.

In practice, it has been found desirable for the longitudinal distance between the center line of the port **26** and the right or draft end **4***b* of the piston **4** at its full draft position to be on the order of 1.77 inches. Similarly, it has been found acceptable for the distance from this draft edge **4***b* of the piston **4** in its full draft position to the center line of the port **25** to be on the order of about 2.37 inches, with port **25** having a diameter of about $\frac{1}{10}$ of an inch.

For a more detailed treatment of the location of port means **23**, **24**, **25** and **26** in cylinder **3**, the criteria for locating these parts, and the overall mode of operation of draft gear **1**, reference should be made to the aforesaid Stephenson application.

A valve mechanism **27** is mounted on the exterior of cylinder wall **3** and cooperates with cylinder wall **3** and port **26** to define a valved, radial flow path extending from cylinder cavity **10**. This flow path is unique in that the valve mechanism **27** is resiliently biased to an open position and tends to remain open until a predetermined pressure is created within cylinder draft zone **10***b* in response to the imposition of a predetermined draft force on coupling bar **6**. This predetermined force is of such a level as to be higher than the force imposed on housing **2** by restoring mechanism **18** but at least equal to the normal level of forces imposed on the coupling **6** as a consequence of run-out train action. As a consequence, port **26** will remain open while the coupling **6** is restoring from a full buff condition as shown in FIGURE 3, with the coupling **6** being disconnected from an adjacent railway car. However, the port **26** will be valved closed in response to the imposition of run-out forces on the coupling **6** while it is connected to the adjacent car of a train in motion.

VALVE STRUCTURE

Structural details of valve mechanism **27** are illustrated in FIGURES 4 through 8.

Mechanism **27** includes a generally cylindrical body portion **28** and a reduced threaded coupling portion **29** which is threadably secured in a threaded socket portion **30** of cylinder wall **3**. With body **28** thus threadably secured to threaded socket **30**, the longitudinal axis of the valve **27** projects generally radially into a corner portion **11***a* of cavity **11**. In other words, valve **27** projects generally radially toward rectangular housing corner **2***a* so as to project into a portion of the reservoir **11***a* large enough to accommodate the valve body without requiring enlargement of the housing **2**.

Valve **27** includes a tubular and movable valve member **31** which is telescopingly received with a valve body opening **32**. Movable valve member **31** is provided at the end facing port **26** with a head **33** providing a fluid reaction surface **34**. This surface **34** extends generally transversely of the radial axis of port **26** and faces this port. A pair of diametrically opposed, restricted, radial ports **35** intersect the tubular wall of valve member **31** adjacent the valve member head **33**.

In practice, it has been found that the diameter of the head **33** should be on the order of .65 inch with the two ports **35** each having a diameter of about $\frac{1}{10}$ of an inch. Each of the ports **35** communicates with a longitudinal passage **37** of tubular member **31** which has been found to have a satisfactory diameter of $\frac{5}{16}$ inch, i.e., the same diameter as port **26**. Thus, passage **37** provides fluid communication, through ports **35**, between port **26** and reservoir **11**.

An annular under lip or abutment **38** on the head **33** of valve member **31** is adapted to abuttingly engage an annular, abutment defining upper end **39** of valve body portion **29** so as to limit radial outward movement of the valve member **31**. With the valve member **31** being telescoped radially outwardly of the valve body **28** so as to bring the abutments **38** and **39** into engagement, the ports **35** will have been brought into port closing engagement with tubular valve seat **40** defined by the cylindrical wall encircling opening **32**.

An annular piston **41** is slidably and telescopingly positioned within valve body portion **28**. Annular piston **40** includes a central aperture **42** through which valve member **31** projects. Annular piston **41** engages an annular shoulder or abutment **43** on valve member **31** and is secured against this abutment by a conventional snap ring **44** which is secured within an annular recess **45** of member **31**.

A coil spring **46** abuttingly engages annular piston **41** and is secured within the housing portion **28** by an annular plate **47**. Annular plate **47**, itself, is secured within the valve body **28** by a snap ring **48** which is anchored within a valve body recess **49**. With annular plate **47** locked in position by snap ring **48**, the spring **46** is maintained so as to exert a desired force transmitted through the piston **41** and acting on the valve member **31** to resist valve closing or port closing movement of this member **31**. Generally, the spring **46** will be preloaded or precompressed by the installed plate **47**. In this manner, the member **31** is yieldably biased to an open position under a load which will be offset by the predetermined loading on the coupling bar **6** necessary to generate sufficient pressure within the zone **10***b* to effect valve opening.

As illustrated in FIGURES 5 and 7, annular plate **47** includes a central aperture **50** through which the open lower end of the member **31** slidably and telescopingly projects. A plurality of ports **51** in the plate **47** provides limited fluid communication between the reservoir **11** and an annular zone **52** within the body **28** communicating with the piston **41**. In this connection, it will be appreciated that the entire valve member **27** is immersed in the hydraulic fluid occupying the reservoir **11** and cylinder cavity **10** and thus will also occupy the space **52**. Thus, piston **41**, in cooperation with ports **51**, will provide a dashpot or movement impedance effect on member **31**, tending to prevent valve "chattering."

As fluid is discharged through the port **26** in response to draft movement of the cylinder **3**, this fluid will be first radially impinged on the fluid reaction surface **34**. This fluid will then be deflected laterally to flow into an annular space **53** defined by wall means **54** of socket **30**. This relatively enlarged annular zone **53** provides communication between the port **26** and the ports **35** of the valve member **31**, when the valve member **31** is in an open port position as shown in FIGURE 4 and FIGURE 5.

Fluid, after having been deflected by the reaction surface **34** into the annular zone **53** defined by wall **54**, will flow through the restricting ports **35** into the valve member passage **37**. These flow restricting orifices or ports **35** will produce a pressure drop between fluid in the space **53** and fluid in the passage **37** so as to tend to produce a valve closing pressure differential force acting on the valve head **33**.

As a result of this pressure differential, probably augmented by the force generated by the impingement of fluid on the reaction surface **34**, a valve closing force will be exerted on the valve member **31**, which of course is normally held open by the biasing spring **46**. When train action draft force is imposed on the coupling bar **6** to produce a pressurized flow through vent **26**, impinging on the surface **34** and passing through the ports **35**, this flow will create sufficient force acting on the valve **31** so as to overcome the biasing effect on the spring **46**. The valve member **31** will then move radially outwardly of the cylinder **3** and close the port **26**. However, when relatively low force is imposed on the cylinder **3** or the coupling bar **6**, such as the force imposed by restoring mechanism **18**, a flow of velocity and pressure insufficient

to effect the closing of the valve **27** will take place through the port **26**. In practice, it has been found that the spring **46** should be adjusted to enable the valve **27** to close in response to a total imposition of draft force on housing **2** on the order of about 21,000 pounds, with about 7,600 pounds of this force being supplied by restoring mechanism **18**. In other words, valve **27** will close in response to a train action draft force of about 13,000 pounds.

As will be appreciated, the valve member **27** is coupled to the cylinder **3** so as to be sealingly engaged with the socket **30**. A conventional lock washer **55** may be interposed between the valve body **28** and the cylinder wall **3** to securely anchor the valve member **27** on the outer periphery of the cylinder **3**.

SUMMARY OF MAJOR ADVANTAGES AND SCOPE OF INVENTION

In discussing the significant advantages attributable to the valve mechanism of this invention, it is appropriate to first review the overall characteristics of the draft gear in which the valve is incorporated. These characteristics may best be depicted by reference to the performance of the gear on adverse rail beds such as that shown in profile in FIGURE 1.

FIGURE 1 illustrates the profile of a portion of a rail bed on a Canadian track extending between Garneau and Riviere a'Piere, Quebec. The maximum grade on this track is about 3.6° which is unusually steep. In addition to the grade problem, the track undulates severely in profile so as to tend to produce conditions uniquely conducive to train action events.

FIGURE 1 illustrates a track profile with vertical guide markers being provided at one mile increments. Track slope in degrees has been noted at varying points along this profile. The lowest point on this profile, i.e., point X, has an elevation above sea level of about 520 feet. The highest point Y on this profile has an elevation of about 632 feet above sea level. In the vicinity of the two-mile marker in the FIGURE 1 profile, a train moving from left to right moves down an incline at about 2.8° and then abruptly up an incline of 2.3° and then subsequently abruptly down another incline of about 1.9°. As will be apparent, by reference to FIGURE 1, these changes in inclination are closely spaced and fairly well typify the grade profile. Thus, in moving down the 2.8° slope and approaching the two-mile marker, the cars of the train will tend to converge so as to produce run-in train action. Then abruptly, after clearing the two mile marker, and while moving up the 2.3° slope, the cars will tend to separate so as to produce train action run-out events.

Where conventional, frictional type, draft gear has been employed, this 40 mile length of track between Garneau and Riviere a'Piere, has produced extraordinary severe train action. The upper Section A of FIGURE 2 provides a graphical indication of a representative number of run-in and run-out train events, and the severity of these train events, which were encountered with standard friction type draft gear traversing this track.

FIGURE 2 dramatically illustrates the extent to which the previously described draft gear reduces over all severity and number of train action events. Section B of the chart shown in FIGURE 2 illustrates run-in and run-out events, both by number and magnitude, for the same track with a similar train, but where train action was governed by cushioning devices generally constructed in accordance with the present invention. The essential similarities in the trains involved in compiling Sections A and B of FIGURE 2, and the inclusion of the draft gear used in compiling Section B of the chart of the exponential port means **23**, the supplemental port means **24**, a valved port **26**, and a restricted port **25**, are believed to provide a valid basis for evaluating the improved train action control afforded by this invention. In this connection, it should be noted that the tests which produced the data for Section B of FIGURE 2 were performed where the vent **25** was less advantageously disposed as an axial orifice in the valve head **34**.

In comparing Section B with Section A of FIGURE 2, it will be noted that run-outs were reduced from 29 to 8 while run-in events were reduced from 17 to 12. The magnitude of run-out forces were substantially reduced, with the number of run-in events at higher run-in force being reduced. In this connection, it should be noted that the single run-in event which occurred with a run-in force between 200,000 and 300,000 pounds is believed to have resulted from train braking and thus probably does not provide valid data for comparison purposes.

A principal advantage of this invention resides in the ability of the improved valve mechanism to effectively control or minimize train action run-out events while allowing for an adequate rate of coupling restoration under free air conditions, i.e., while significant force is not being imposed on a coupling bar by an adjacent railway car.

Another principal advantage of the invention resides in the inherently compact structure of the valve mechanism, resulting from its mutually telescoped components, which yield both structural ruggedness and a unique conservation of space.

A particularly significant advantage of the invention involves the reliable and responsive operating characteristics of the valve due, to a large degree, to the utilization of valve ports which provide a valve closing differential force.

Another important advantage attributable to the improved valve mechanism entails the utilization of the movement damping, dashpot piston which serves to minimize valve chattering tendencies.

The impingement of the outflow of fluid from the draft zone of the cylinder upon the valve fluid reaction surface is also advantageous. This utilizes the outflow of fluid during the extension of a coupling bar to impose a valve closing force on the valve as a fluid velocity phenomenon.

In being submerged in the hydraulic oil of the reservoir **1**, valve **27** is both continuously lubricated and cooled. This factor, coupled with the basic ruggedness of the valve **27**, should tend to virtually eliminate or highly minimize maintenance and replacement problems.

It is also significant to note that this improved valve structure may be easily incorporated in effective shock absorbing mechanisms such as those disclosed in the aforementioned Blake patents and Seay patent without requiring any significant structural alterations of the basic components of these devices.

Those familiar with the disclosure of this invention will at once recognize that its scope is not limited to the port type venting system described, or to the particular piston and cylinder arrangements illustrated. As was noted in the aforesaid Seay patent, limited deviations with respect to optimum port positioning is tolerable within the spirit of the invention.

Thus, those skilled in the art may recognize additions, deletions, substitutions, modifications, or other changes which would fall within the scope of the invention.

I claim:

1. In railway draft gear, including a fluid shock absorbing assembly comprising cylinder means and piston means, with the piston means being connected with one of a coupling portion and body portion of a railway car and the cylinder means being connected with the other of said coupling and body portions, valve means for regulating an outflow of fluid from a piston biasing zone of said assembly, the improvement in valve means comprising:

a valve body;

a movable valve member;

a valve seat carried by said valve body;

a fluid reaction surface on said movable valve member upon which fluid outflow from said assembly is adapted to impinge;

port means carried by said movable valve member and operable in response to a fluid flow therethrough emanating from said piston biasing zone of said assembly to create a pressure differential across said movable valve member tending to move said valve member into valve closing engagement with said seat; and

resilient means biasing said valve member away from said valve seat so as to yieldably position said valve member with the ports thereof open;

dashpot means operable to retard movement of said valve member toward said valve seat so as to close said ports;

first wall means defining an axial flow path leading axially from said piston biasing zone toward said fluid reaction surface and adapted to define an axial flow having a cross-sectional area substantially less than the cross-sectional area of said movable valve; and

second wall means defining an annular chamber providing fluid communication between said axial flow path and the port means of said movable valve member.

2. In railway draft gear, including a fluid shock absorbing assembly comprising cylinder means and piston means, with the piston means being connected with one of a coupling portion and body portion of a railway car and the cylinder means being connected with the other of said coupling and body portions, valve means for regulating an outflow of fluid from a piston biasing zone of said assembly, the improvement in valve means comprising:

a valve body;

a movable valve member;

a valve seat carried by said valve body;

a fluid reaction surface on said movable valve member upon which fluid outflow from said assembly is adapted to impinge;

port means carried by said movable valve member and operable in response to a fluid flow therethrough emanating from said piston biasing zone of said assembly to create a pressure differential across said movable valve member tending to move said valve member into valve closing engagement with said seat;

first wall means defining an axial flow path leading axially from said piston biasing zone toward said fluid reaction surface and adapted to define an axial flow having a cross-sectional area substantially less than the cross-sectional area of said movable valve;

second wall means defining an annular chamber providing fluid communication between said axial flow path and the port means of said movable valve;

first abutment means carried by said valve member;

second abutment means carried by said valve body and operable to engage said first abutment means to limit valve closing movement of said valve member;

annular piston means slidably mounted within said body, with said valve member extending axially through said annular piston means;

third abutment means carried by said valve member;

coil spring means contained within said valve body, engaging said annular piston means and biasing said annular piston means toward the port means of said valve member and into engagement with said third abutment means;

annular plate means fixedly mounted within said valve body and providing fourth abutment means for engaging said coil spring means and holding said coil spring means within said body and in engagement with said annular piston means;

said coil spring means exerting a predetermined biasing force on said annular piston means, with said valve member telescopingly passing through a centrally apertured portion of said annular plate means;

port means in said annular plate means providing restricted fluid communication between the exterior of said valve body and an annular zone within said valve body having longitudinally spaced ends defined by said annular piston means and said annular plate means and having inner and outer side walls defined respectively by the outer peripheral portion of said valve member and an inner peripheral portion of said valve body; and

means defining a fluid reservoir on the exterior of said cylinder means of said shock absorbing assembly substantially surrounding said valve body.

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