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O. E. SEAY

3,301,410

HYDRAULIC SHOCK ABSORBING APPARATUS

Original Filed Feb. 13, 1964

3 Sheets-Sheet 1

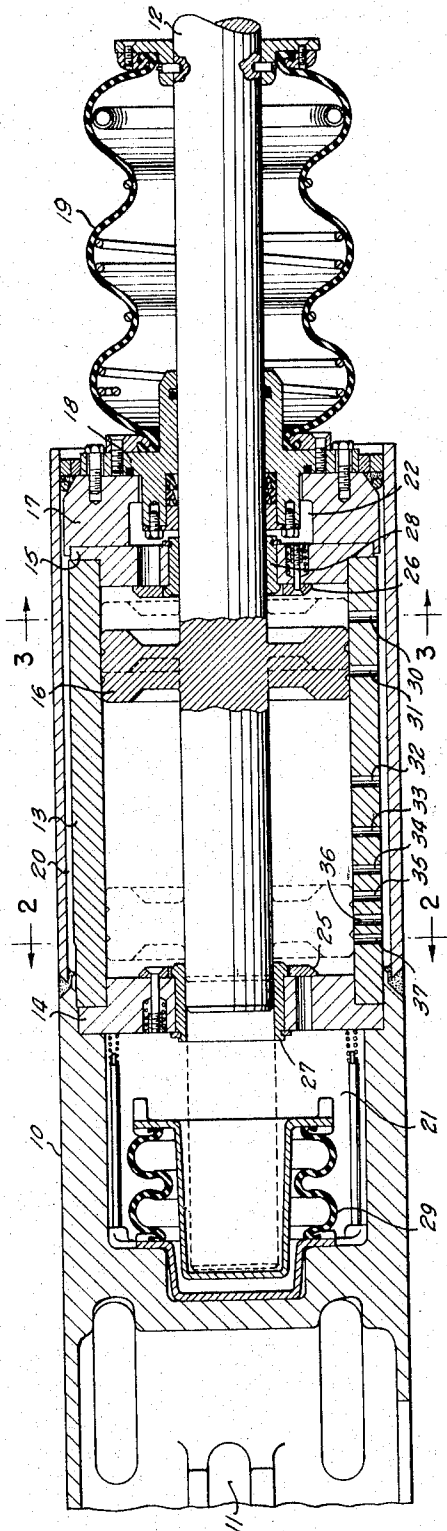


Fig. 1

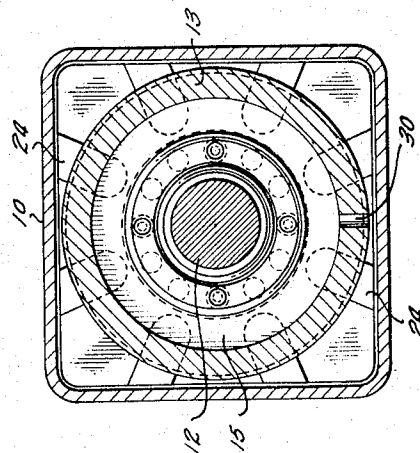


Fig. 3

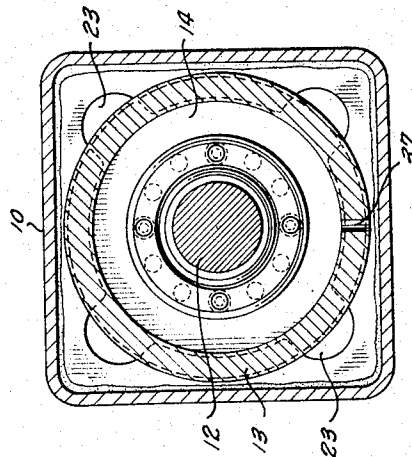


Fig. 2

INVENTOR
Orum Elwyn Seay

BY

Hyon Hyon

ATTORNEYS

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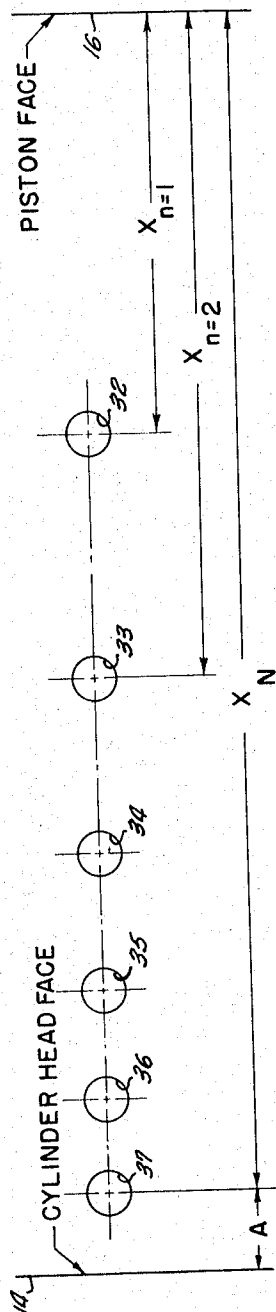


Fig. 4

INVENTOR

Orum Elwyn Seay

BY

Lyon Lyon

ATTORNEYS

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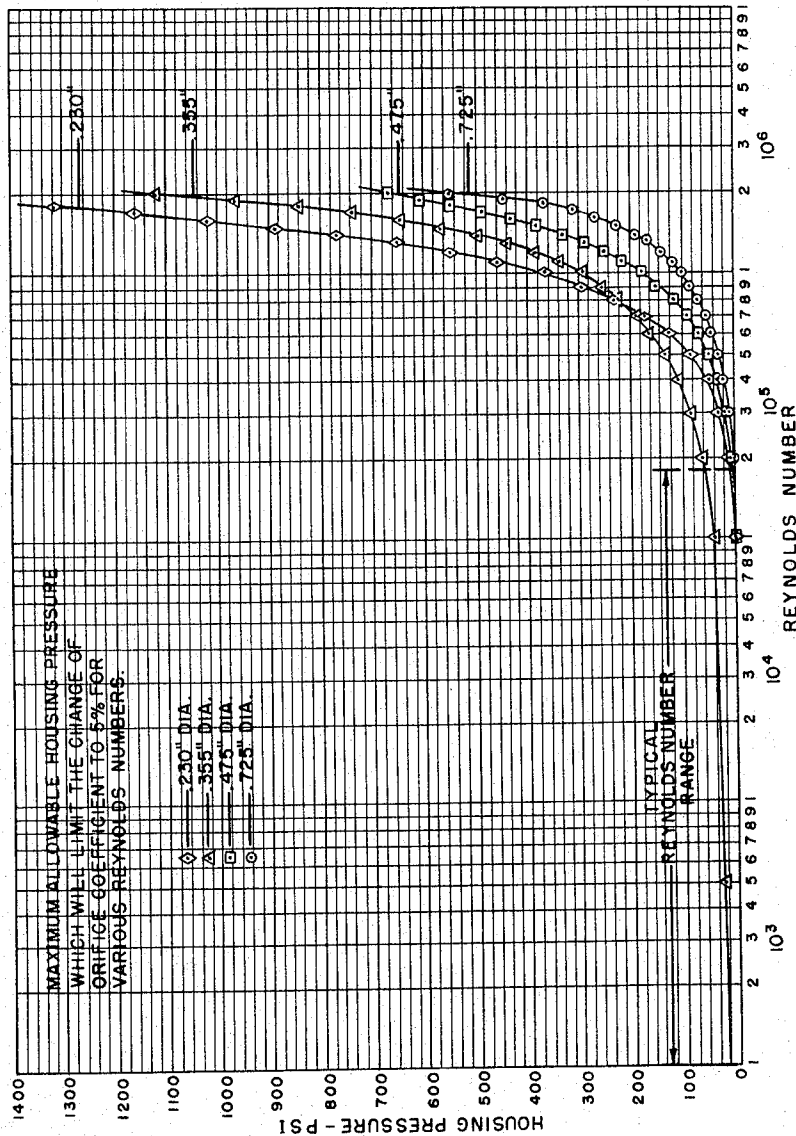


Fig.5

INVENTOR
Orum Elwyn Seay

BY

Lyon & Lyon

ATTORNEYS

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3,301,410

HYDRAULIC SHOCK ABSORBING APPARATUS
Orum Elwyn Seay, Duncan, Okla., assignor to Halliburton Company, Duncan, Okla., a corporation of Delaware
Continuation of application Ser. No. 344,579, Feb. 13, 1964. This application Feb. 14, 1966, Ser. No. 527,347
7 Claims. (Cl. 213-43)

This application is a continuation of my copending application Serial No. 344,579, filed February 13, 1964.

This invention relates to hydraulic shock absorbing or cushioning apparatus, and more particularly to double-acting hydraulic apparatus for use in cushioning railway cars.

The cushioning apparatus of the present invention is an improvement in the apparatus shown and described in U.S. Patent No. 3,047,162, issued July 31, 1962 to W. T. Blake, for "Hydraulic and Resilient Cushioned Railway Car Draft Assembly," and in the apparatus shown and described in U.S. Patent No. 2,944,681, issued July 12, 1960 to W. T. Blake, for "Railway Draft Appliance." Although the apparatus of the present invention is particularly adapted for use in cushioning railway cars, the hydraulic mechanism thereof has general application in the shock absorbing art.

The railway industry is constantly seeking to improve freight car cushioning devices so that damage to the cars and to the car lading is minimized. Railway cars are subjected to severe shocks in normal movement. Railway freight cars, and their contents in particular, are subject to severe impact forces when a train is made up in the freight yards by coupling free-rolling, individual freight cars into a stationary car or group of stationary cars. Because of the slack that exists between the couplers of railway cars, additional shocks are imparted to the cars and their contents when a train is in motion. When a train is started, each car is jerked by the car ahead of it and, upon braking a train to a stop, each car is bumped by the car behind it, thus imparting additional damaging shocks to the cars and their contents.

In hydraulic cushioning apparatus, it is desirable that the operating characteristics of the unit, insofar as possible, be independent of ambient temperature. Hydraulic draft gears for railway cars are exposed to the extremes of temperature encountered in winter and in summer, and in northern and southern climates. As the viscosity of the hydraulic fluid varies with changes in temperature, the functions of the hydraulic units of the known apparatus have been found to change; such units operate more sluggishly as the viscosity of the oil or other hydraulic fluid increases at lower temperatures. Moreover, in such hydraulic cushioning apparatus, the absorption of kinetic energy should take place over the full stroke of the apparatus regardless of the magnitude of the impact, within the range of operation.

Therefore, it is an object of the invention to provide hydraulic cushioning apparatus that operates in a substantially uniform manner over a wide range of temperatures.

Another object of the invention is to provide hydraulic cushioning apparatus in which kinetic energy is absorbed smoothly.

Another object is to provide hydraulic cushioning apparatus which reacts to applied impacts with a uniform force throughout the full stroke of the apparatus.

A further object is to provide complete operational travel in such apparatus regardless of impact speed, within the range of operation.

The foregoing and other aims, objects and advantages are achieved in this invention by means of a hydraulic cushioning mechanism having a piston and cylinder op-

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erable to force hydraulic fluid through metering orifices in the cylinder wall and into a housing chamber, the metering orifices each having a diameter so chosen in relation to Reynolds number and housing backpressure as to minimize variations in flow characteristics occasioned by changes in ambient temperature, namely, a diameter of from 0.28 inch to about 0.38 inch, the metering orifices comprising a plurality of separate ports spaced exponentially along the length of the cylinder.

In the drawings:

FIGURE 1 is an axial, sectional view of a hydraulic cushioning apparatus embodying the invention.

FIGURE 2 is a transverse, sectional view taken along the line 2-2 of FIGURE 1.

FIGURE 3 is a transverse, sectional view taken along the line 3-3 of FIGURE 1.

FIGURE 4 is a diagrammatic view illustrating the principle of exponential spacing of the metering orifices.

FIGURE 5 is a graph on semi-log paper showing maximum allowable housing pressures which will limit the change of orifice coefficient to 5% for various Reynolds numbers.

Referring to the drawings, particularly to FIGURES 1 to 3, the hydraulic cushioning apparatus shown has a housing 10 that may be adapted to slide in one end of the center sill of a railway car, as disclosed in the foregoing Blake patents. A car coupler (not shown) may be carried by the key 11. The piston rod 12 may be fixed to the center sill, also, if desired, in the manner disclosed in said Blake patents.

A high pressure working cylinder 13 is mounted in the housing and spaced from the walls thereof. The front of the cylinder is closed by a head 14 and the rear is closed by another head 15. A piston 16, integral with the piston rod 12, is mounted in the cylinder, the latter sliding over the piston as the housing is reciprocated.

The rear end of the housing 10 is closed by a rear cover plate 17 welded to the housing. A shaft-sealing bushing 18 is carried by the rear cover plate and seals the piston rod 12 against leakage therearound. A bellows 19 protects the exterior sliding surfaces of the piston rod from dust and moisture.

A chamber 20 within the housing 10 surrounds the working cylinder 13 and this chamber 20 includes spaces 21 and 22 which communicate through channels 23 (see FIGURE 2) and channels 24 (see FIGURE 3) formed in the housing 10. A front check valve 25 opening into the cylinder establishes communication with the front space 21 of the housing chamber 20. Similarly, an inwardly opening rear check valve 26 establishes communication between the rear end of the cylinder with the rear space 22 of the housing chamber 20. Forward and rear bushings 27 and 28 are carried in the front and rear cylinder heads and accommodate the piston rod.

A compensating device 29 is mounted in the front space. This device is filled with gas, preferably at sub-atmospheric pressure, and expands and contracts as necessary to compensate for temperature and for displacement of hydraulic fluid occasioned by movement of the piston rod 12 into and out of the closed housing 10.

A number of metering ports or orifices 30 to 37 are provided in the wall of the cylinder 13. The orifices are longitudinally spaced in an exponential fashion as described below, and they serve to control the cushioning action of the apparatus as forces are applied to it.

As seen in FIGURE 1, the apparatus is shown in the neutral or rest position. As explained at length in the aforesaid Blake Patent No. 3,047,162, when an impact in buff is applied to the housing 10 through the key 11, the housing slides to the right as seen in FIGURE 1. In the fully closed position, the piston is adjacent to the front cylinder head 14, as shown in dotted lines in FIG-

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FIGURE 1. When draft forces are applied to the housing, it moves to the left, as seen in FIGURE 1. In the fully extended condition, the piston is adjacent to the rear cylinder head 15, as shown in dotted lines in FIGURE 1.

An inspection of FIGURE 1 shows that metering orifices 32 to 37 inclusive are operative during the buff stroke of the apparatus from the neutral position. As the cylinder moves to the rear over the piston, the orifices are successively closed by the piston, thereby progressively diminishing the number of orifices discharging fluid from the cylinder. As the cylinder returns from the fully closed position to the neutral position, orifices 30 to 34 are initially operative; orifices 31 to 34 are closed in reverse sequence by the piston 12 as the cylinder 13 moves forwardly to neutral position.

When the cylinder moves forwardly from the neutral position to the fully extended position, orifice 30 controls the movement. Because there is a slight clearance between the piston and the cylinder, hydraulic fluid may flow between the piston and the cylinder to permit the cylinder to reach the extreme positions in buff and draft shown in FIGURE 1, even though in each of such extreme positions the terminal orifice 37 or 30 is closed by the piston. It will be understood that, in operation, the compensating device 29 expands or contracts to compensate for temperature changes and piston displacement. Moreover, the check valves 25 and 26 open to admit hydraulic fluid to that end of the cylinder which is moving away from the piston. The conjoint action of the compensating device and the check valves limits pressures in the housing chamber 20 to low values. The low pressures in the housing chamber not only simplifies the sealing of the piston rod in the rear cover plate but also contributes to the achievement of substantially uniform operation of the hydraulic unit over a wide range of temperatures, as described below.

If the hydraulic cushioning apparatus of the invention is to perform in substantially the same way at all temperatures, the same internal cylinder pressures and flow rates of hydraulic fluid through the metering orifices must exist for the various conditions encountered. If the orifice coefficient of the metering orifices of the working cylinder can be kept constant over the range of Reynolds numbers at which the apparatus is normally operated, the desired internal cylinder pressures and flow rates will be achieved. The terms "orifice coefficient" and "Reynolds number" have their ordinary meaning as used in hydraulic design. Through careful scientific study and experiments, it has been found that for low values of housing chamber backpressure, the orifice coefficient will remain substantially constant in the normally encountered range of Reynolds numbers if the diameter of the metering orifices is made about 0.355 inch. Although this diameter gives the best practicable uniformity of operation, the diameter of the metering orifices may be between 0.28 inch and 0.38 inch and still give excellent uniformity of operation of the apparatus over the temperature range normally found in practice. The data leading to this discovery are pictorially represented in FIGURE 5.

FIGURE 5 is a semi-log graph showing the relationship between housing pressure and Reynolds number for metering orifices having diameters from 0.230 inch to 0.725 inch. The curves show the maximum allowable housing pressure which will limit the change of orifice coefficient to 5% for various Reynolds numbers. The Reynolds number range for operation of hydraulic cushioning apparatus of the type disclosed is indicated in FIGURE 5 by the horizontal arrow labeled "Typical Reynolds Number Range." From the graph it is seen that if the orifice coefficient is to remain constant within acceptable limits, for example plus or minus 5%, for Reynolds numbers varying from about 10^3 to about 2×10^5 , the orifice diameter must be about 0.355 inch and the housing pressure (i.e. the pressure in the housing

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chamber 20) should not be allowed to rise above about 20 p.s.i. at a Reynolds number of 10^3 or above about 60 p.s.i. at a Reynolds number of 2×10^5 . It is also seen from FIGURE 5 that if the metering orifice has a diameter of 0.230 inch, which is substantially below the range of orifice diameters in accordance with the invention, it is not possible to hold the orifice coefficient within 5% of a constant value when operating at Reynolds numbers less than about 10^5 , where the 0.230 inch diameter curve intersects the horizontal axis of the graph corresponding to zero housing pressure. The same may be said with regard to the metering orifice having a diameter of 0.475 inch which is above the range of metering orifice diameters of the invention. For an orifice having a diameter of 0.725 inch, operation at even higher Reynolds numbers must occur, if the orifice coefficient is to be held within 5% of a constant value.

Stated in different terms, the chart of FIGURE 5 shows housing backpressure plotted against Reynolds numbers for certain orifice sizes. It is desired to obtain repeatable performance which does not vary more than plus or minus 5% and the problem is to determine which orifice size best gives this result. From experimental data for any one orifice size, correlation between (a) Reynolds number and (b) orifice coefficient is obtained for various values of housing pressure.

First example.—Assume:

$$\text{Reynolds number } R_e = 1 \times 10^6$$

$$\text{Housing pressure} = 100 \text{ p.s.i.}$$

Question: What hole sizes could be used to get repeatable forces at various temperature extremes?

Answer: From FIGURE 5 find the Reynolds number on horizontal axis (1×10^6). Find Housing Pressure on vertical axis, 100 p.s.i. This defines a point on the chart. Any curves above this point indicate orifice sizes which could be utilized and still provide an orifice coefficient which does not change more than 5%, and in actual practice this means that the force of impact, accelerations of impact, etc., would likewise not vary more than 5%. In this particular case, all orifice sizes below 0.725" would produce variations of less than 5% (0.725" is shown as exactly 5%).

Second example.—Assume:

$$\text{Reynolds number } R_e = 6 \times 10^5$$

$$\text{Housing pressure} = 100 \text{ p.s.i.}$$

The result from a consideration of the chart of FIGURE 5 shows that orifice sizes of 0.230" and 0.355" produce the desired result of performance within 5%, whereas orifice sizes of 0.750" and 0.725" will give greater variations than 5%.

Third example.—Assume:

$$\text{Reynolds number } R_e = 1.9 \times 10^5$$

$$\text{Housing pressure} = 60 \text{ p.s.i.}$$

(This is the high end of the Reynolds number range shown on the chart.)

The result from a consideration of the chart of FIGURE 5 shows that an orifice size of 0.355" is satisfactory. All other orifice sizes shown on chart give more than 5% variation. (Hole sizes of 0.233" and 0.475" are next best but are somewhat over 5%.)

Fourth example.—Assume:

$$\text{Reynolds number } R_e = 1.0 \times 10^3$$

$$\text{Housing pressure} = 15 \text{ p.s.i.}$$

(This is the low end of Reynolds number range shown on the chart.) From chart, only orifice size of 0.355" will give results within less than 5% variation.

It is thus seen that the use of metering orifices having a diameter of from 0.28 to 0.38 inch permits operation of hydraulic cushioning apparatus at Reynolds numbers of from 10^3 to 2×10^5 , with the orifice coefficient remaining within about 5% of a constant value over this

Reynolds number range, if the backpressure on the metering orifice is kept below about 60 p.s.i. The practical result is substantially uniform operation over a wide range of ambient temperature.

I have also found that the spacing of the metering orifices in accordance with an exponential equation, particularly where the metering orifices have a diameter within the range of the invention, provides a hydraulic cushioning apparatus of greatly improved kind. The energy absorbed by the apparatus is represented by the area under a force-travel curve. For the absorption of a given quantity of energy, minimum force exists only when this area is rectangular; under those conditions, the force is uniform throughout the stroke. With the metering orifices spaced exponentially, kinetic energy is absorbed uniformly throughout the stroke of the apparatus; thus pressure within the working cylinder and the force applied to the apparatus approach uniform and minimum values throughout the stroke. As a result, minimum accelerations are imparted to the system cushioned by the apparatus.

Moreover, with exponential spacing of the metering orifices, full travel of the apparatus is obtained for all impact forces above a threshold value insufficient to cause damage to the car or contents.

Referring to FIGURE 4, the diagram shown represents a portion of the interior of the working cylinder of the hydraulic apparatus. Metering orifices 32 to 37 are shown. At the left of FIGURE 4, the interior face of the front cylinder head 14 is shown. At the right of the view, the left hand face of the piston 16 is indicated, the piston being in the neutral position. Let it be assumed that N metering orifices are desired to be spaced in accordance with the invention along a distance of X_N . Then n is the ordinal number of any orifice with reference to the piston face, and X_n is the distance from the piston face to the center of orifice n . Then:

$$X_n = C \log_b (n+1)$$

where:

$$C = \frac{X_N}{\log_b (N+1)}$$

b =any positive number except 1; and

A =distance from last port to cylinder head face.

Therefore:

$$X_n = \frac{X_N \log_b (n+1)}{\log_b (N+1)} = C \log_b (n+1)$$

Applying this equation, for example, to the particular case in which it is desired to use 6 metering orifices in a 9 inch stroke device:

$$b=10$$

$$N=6$$

$$X_N=9 \text{ inches}$$

$$C = \frac{X_N}{\log_b (N+1)} = \frac{9}{\log_{10} 7} = \frac{9}{0.846} = 1.064$$

$$X_n = C \log_b (n+1) = 9.52 \log_{10} (n+1)$$

TABLE I

n	$\log (n+1)$	X_n (inches)
1	0.301	3.20
2	0.477	5.08
3	0.602	6.41
4	0.699	7.44
5	0.778	8.28
6	0.846	9.00

The column headed X_n in Table I gives the distance of each of the 6 metering orifices from the piston in neutral position. These distances provide the exponentially spaced arrangement of orifices in accordance with the invention.

It will be understood that the metering orifices do not have to be arranged in a straight line longitudinally of the cylinder; the orifices may be offset circumferentially of the cylinder, if desired, provided that their distances from the piston at its starting position are substantially in accordance with the foregoing exponential equation. Although optimum results are obtained when the metering orifices are positioned in accordance with the foregoing equation, reasonable deviations from such positioning are permissible without departing from the invention; thus, so long as the orifices are positioned within about 15% of the distances between them as given by the foregoing equation, the advantages of the invention are substantially realized.

The operation of the apparatus described hereinbefore has been set forth in connection with its description or is apparent therefrom.

Although a preferred embodiment of the invention has been shown and described herein, the invention is not limited to the disclosed embodiment but is defined by the following claims.

I claim:

1. A hydraulic cushioning apparatus of the type described, comprising, in combination: a cylinder adapted to contain hydraulic fluid, a piston adapted to move axially relative to said cylinder, means forming a chamber, a plurality of metering ports establishing the sole communication between said cylinder and said chamber, all of said metering ports being exponentially spaced axially, successive metering ports being spaced closer together in the direction of the working stroke of the piston, each metering port having a diameter of from about 0.28 inch to about 0.38 inch, relative axial movement of said piston and cylinder acting to force hydraulic fluid from said cylinder through said metering ports and into said chamber.

2. The combination of claim 1 in which the metering ports are provided in the wall of the cylinder.

3. The combination of claim 2 in which all of the metering ports are the same size.

4. A hydraulic cushioning apparatus of the type described, comprising, in combination: a cylinder adapted to contain hydraulic fluid, a piston adapted to move axially relative to said cylinder, means forming a chamber, a plurality of metering ports establishing the sole communication between said cylinder and said chamber, all of said metering ports being exponentially spaced axially, successive metering ports being spaced closer together in the direction of the working stroke of the piston, so that substantially the same extent of relative axial movement occurs between the piston and cylinder over a wide range of applied force, each metering port having a diameter of from about 0.28 inch to about 0.38 inch, relative axial movement of said piston and cylinder acting to force hydraulic fluid from said cylinder through said metering ports and into said chamber.

5. A hydraulic cushioning apparatus of the type described, comprising, in combination: a cylinder adapted to contain hydraulic fluid, a piston adapted to move axially relative to said cylinder, a closed housing enclosing the cylinder and having a chamber therein, a rod fixed to the piston and projecting exteriorly of the housing, seal means between the piston rod and the housing, a plurality of metering ports establishing the sole communication between said cylinder and said chamber, all of said metering ports being exponentially spaced axially, successive metering ports being spaced closer together in the direction of the working stroke of the piston, each metering port having a diameter of from about 0.28 inch to about 0.38 inch, passage means through which hydraulic fluid may pass from the chamber back into the cylinder, relative axial movement of said piston and cylinder acting to force hydraulic fluid from said cylinder on one side of said piston through said metering ports and into said

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chamber, and back into the cylinder on the other side of said piston through said passage means, and means in the housing limiting pressure in the chamber.

6. The combination of claim 5 in which the passage means includes at least one check valve.

7. The combination of claim 5 in which the pressure limiting means comprises a gas filled compensating device.

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ARTHUR L. LA POINT, *Primary Examiner*.
DRAYTON E. HOFFMAN, *Examiner*.