

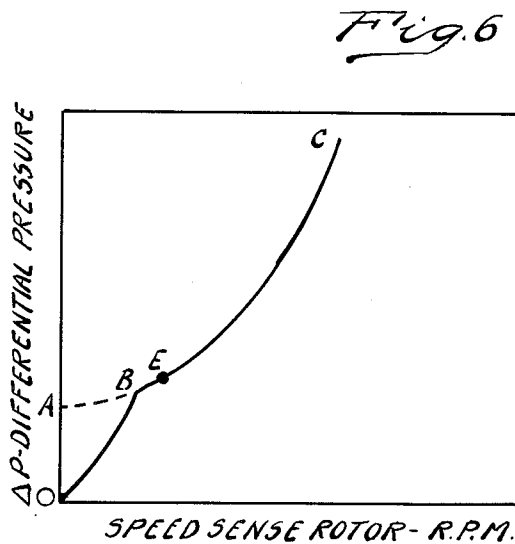
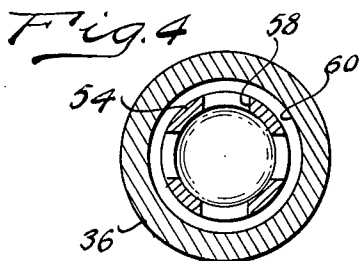
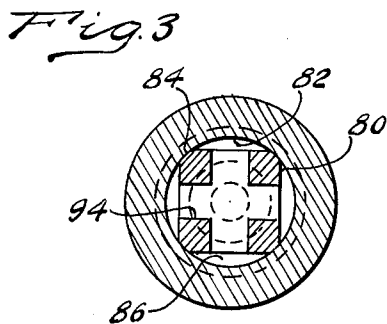
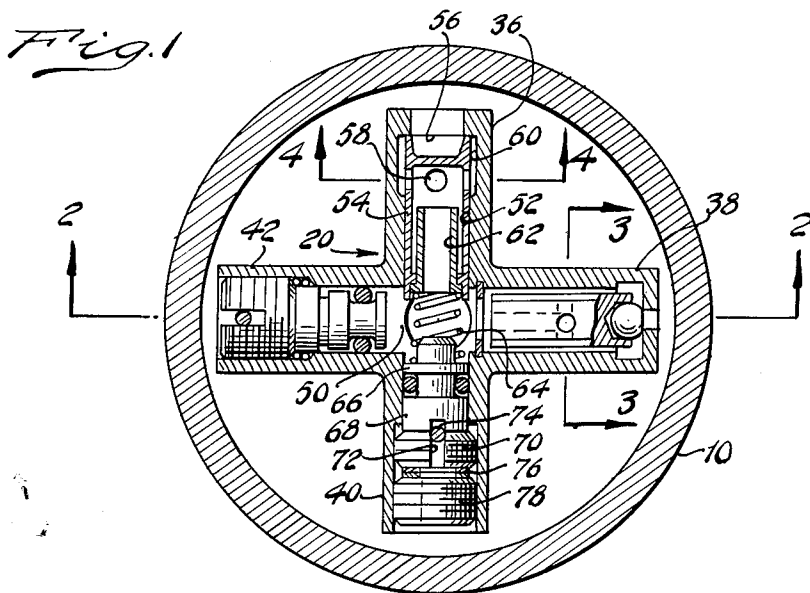
Dec. 28, 1965

R. K. LONDAL
HYDRAULIC RESISTORS

3,225,778

Filed Nov. 6, 1961

2 Sheets-Sheet 1



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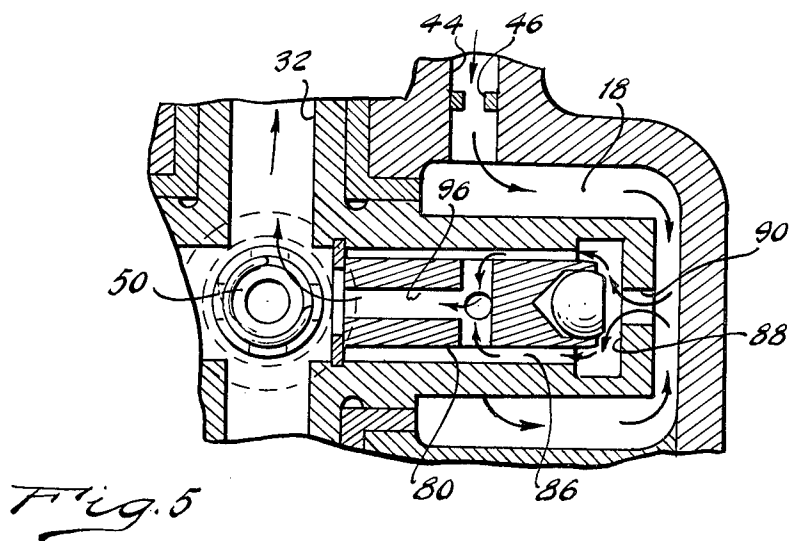
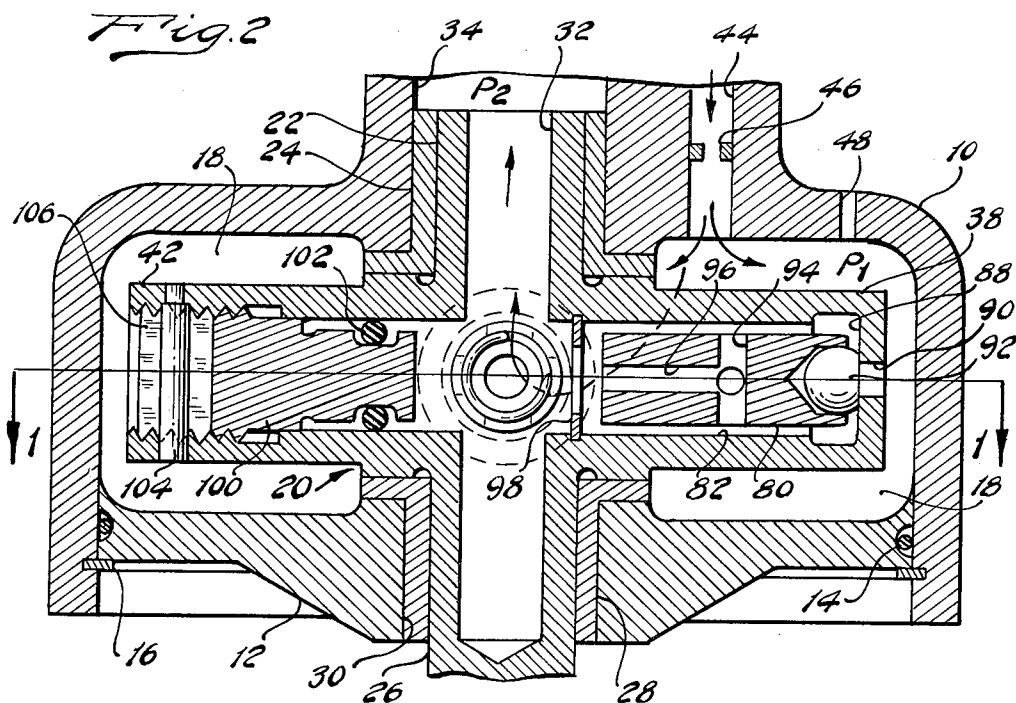
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HYDRAULIC RESISTORS

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5 Claims. (Cl. 137-54)

The present invention relates to hydraulic resistors, and more specifically, to variable hydraulic resistors which are partially controlled by centrifugal force.

In the past, most variable centrifugally actuated resistors produced a hydraulic pressure differential which was used as an indication or signal of some operating condition of the engine. One such resistor is shown in Begian et al. application Serial No. 806,881, now United States Patent No. 3,026,890. In most cases this type of resistor has proven very satisfactory; however, in some applications, such as in certain guided missiles, this type of unit is not satisfactory due to the nature of the differential pressure used as a signal for the accelerating and governing systems.

This pressure differential resulted from centrifugal means, which produced at the low range of operation, a relatively small change in differential pressure for a given change in engine speed.

In many present hydraulic circuits, a device which sets up a reference pressure SS as a function of speed is used as a speed sense. This pressure is used as a signal to the acceleration and governing systems.

The speed sense pressure SS is controlled by a pressure drop from a relatively constant reference pressure CR. The speed sense pressure differential (SS-CR) has been determined by a piston valve in a rotor which is turning at a speed proportional to engine speed. The centrifugal force on the piston valve maintains the SS-CR differential at a value determined by engine speed. The piston valve has been spring-loaded so that SS-CR has a positive value at zero speed.

Because centrifugal force is proportional to the square of speed, the change in pressure differential SS-CR in the low speed region is relatively small for a given change of speed as compared to the changes which occur for a given change of speed in a higher speed region. Considerable difficulty has been encountered in making the acceleration system follow the relatively small changes in pressure differential occurring between zero speed and the first engine operating speed.

In accordance with the present invention, it is now proposed to provide a double piston valve speed sense to provide a larger change in the pressure differential SS-CR between zero speed and the lowest operating speed. This is accomplished by adding a second piston valve to the sensing unit. This second piston valve is not spring-loaded and accordingly, it will open at zero speed and the pressure differential SS-CR will drop to near zero. This secondary piston valve is much heavier than the primary piston valve so that as the rotor speed increases the secondary piston valve will develop relatively large centrifugal forces tending to move it toward closed position, and the pressure differential SS-CR will increase rapidly. By the time the lowest operating speed is reached, centrifugal force will have completely closed the secondary piston valve and thereafter, the pressure difference SS-CR will be determined solely by the primary piston valve over the remainder of the speed range.

It is accordingly an object of the present invention to provide a centrifugal hydraulic resistor adapted to maintain a negligible pressure differential at zero speeds of rotation, to provide such pressure differential which increases rapidly throughout a low speed range, and which

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increases more gradually in the intermediate and high speed ranges.

More specifically, it is an object of the present invention to provide a double centrifugal speed sense valve including centrifugal valves connected in parallel, one of which is relatively heavy and is not spring-biased, whereas the other is relatively light and has a positive spring-bias.

Other objects and features of the invention will become apparent as the description proceeds, especially when taken in conjunction with the accompanying drawings, illustrating a preferred embodiment of the invention, wherein:

FIGURE 1 is a plan view of the invention in partial section, taken along the line 1-1, FIGURE 2.

FIGURE 2 is an enlarged elevational view in partial section, taken along the line 2-2, FIGURE 1.

FIGURE 3 is an enlarged sectional view on the line 3-3, FIGURE 1.

FIGURE 4 is an enlarged sectional view on the line 4-4, FIGURE 1.

FIGURE 5 is a fragmentary sectional view corresponding to that of FIGURE 2, showing the parts in a different operating position.

FIGURE 6 is a graphic curve showing pressure differential against engine speed of the previous methods and that of the present invention.

Referring now to the drawings, there is provided a stationary housing 10 having a removable cover plate 12, the cover plate including an annular seal 14 and a retaining snap ring 16. The housing 10 is shaped so that in cooperation with the cover plate 12, it defines a chamber 18.

Located in the chamber 18 is a rotary hydraulic resistor assembly 20 including an upper tubular extension 22 rotatably received in a bearing 24, and a drive shaft 26 rotatably received in a bearing 28 mounted in an opening 30 extending through the cover plate 12. The drive shaft 26 will be connected to a rotating part of an engine with which the rotary hydraulic resistor is associated for rotation of the resistor 20 at a speed dependent on engine speed. The tubular extension 22 includes a passage 32 therein in open communication with a chamber 34.

As best seen in FIGURE 1, the rotary hydraulic resistor assembly 20 comprises radially extending tubular arms 36, 38, 40 and 42. In two of these arms, as will subsequently be described, are provided a pair of centrifugal piston valves adapted to control the flow of fluid from the chamber 18 into the interior of the rotary resistor 20 and thence through the passage 32 into the chamber 34. The housing 10 is provided with an inlet passage 44 having a restriction 46 thereon adapted to provide for a flow of high pressure hydraulic fluid from a substantially constant pressure source into the chamber 18. The rotary hydraulic resistor 20 determines the rate of flow of fluid out of the chamber 18 and hence, will determine the actual pressure existing within the chamber 18. An outlet passage 48 is provided to which a suitable conduit may be connected to transmit the signal pressure existing within the chamber 18 to a signal responsive device. The pressure existing within the chamber 18 is here designated P_1 and the pressure existing in the passage 32 and chamber 34 is designated P_2 .

Tubular arms 36, 38, 40 and 42 are all in open communication with a central chamber 50, as best seen in FIGURE 5. The tubular arm 36 has an internal cylindrical portion 52 slidably receiving a piston valve 54, the valve at its outer end engaging a valve seat 56 when the piston valve 54 is in closed position. The piston valve 54 has ports 58 in communication with an annular chamber 60 so that when the end of the piston valve 54

moves away from the valve seat 56, fluid can flow inwardly between the end of the valve and the valve seat into the chamber 60 and thence through the ports 58 and through the hollow interior of the valve to the central chamber 50. The valve 54 includes a separate centrifugal weight element 62.

Adjustable spring means are provided for urging the piston valve 54 toward closed position and this means comprises the spring 64 seated at one end in a recess provided at the inner end of the valve 54, and seated at its other end on a flange 66 on adjustable spring seat 68. The spring seat 68 includes a threaded portion 70 adjustably received in the threaded outer end of the tubular arm 40. The spring seat 68 is slotted as indicated at 72 for the reception of a pin 74 by means of which the spring seat is prevented from accidental displacement after having been adjusted into the position required to produce the necessary spring force.

Inasmuch as the centrifugal hydraulic resistor operates at high speed, it is essential to insure centrifugal balance. Accordingly, means are provided for balancing the rotary resistor with the spring seat in different positions of adjustment, and this means comprises removable and replaceable shims 76 which are positioned between the outer end of the spring seat 68 and a lock screw 78.

The valve 54 in association with the spring 64, may operate to maintain a hydraulic pressure within the chamber 18 as indicated by the line ABEC in FIGURE 6. It will be observed that at zero speed a substantial pressure differential is maintained across the valve 54 because of the force exerted by the compression spring 64.

In order to provide a hydraulic speed sense operating along the curve OBEC which is characterized by a substantial pressure difference developed by a change in speed, from zero to the point E, corresponding to the engine starting speed, there is provided in parallel with the valve 54 a second piston valve which will now be described.

Referring now more particularly to FIGURES 2, 3 and 5, there is shown a piston valve 80 movable longitudinally in a cylinder 82 formed in the radially extending arm 38. As best seen in FIGURE 3, the piston valve 80 is of generally square cross-section, its corner portions as indicated at 84 being rounded to provide a sliding fit within the cylinder 82. At the same time, the square cross-sectional shape of the piston valve 80 provides passages 86 for the flow of hydraulic fluid.

At its outer end, the arm 38 includes an returned valve seat portion 88 provided with a passage 90 adapted to be closed by a partially spherical ball valve element 92 having a flattened end portion adapted to fit over the valve seat. The piston valve 80 is provided with transverse intersecting passages 94 and a longitudinally extending passage 96 communicating with the transverse passages 94. The valve 80 is maintained in the cylinder 82 of the arm 38 by a stop 98.

As best seen in FIGURE 5, under conditions when the valve 80 is in open position, hydraulic fluid flows through the passage 44 past the restrictions 46, into the chamber 18. Hydraulic pressure within the chamber 18 accordingly depends upon the pressure drop across the restriction 46. This in turn depends upon the flow of fluid past the restriction 46 which is regulated by the position of the valve 80 during low speed operation. It will be observed that the valve 80 is not provided with a spring and further, that it is relatively heavy as compared to the structure making up the valve 54. At low speed operation, pressure existing within the chamber 18 will maintain the valve 80 off the valve seat 88 and fluid will flow through the passage 90 along the passages 86 through the passages 94 and 96 to the chamber 50 located centrally of the hydraulic resistor and in communication with the outlet passage 32 therein. In normal operation, the valve 80 is not always in engagement with

the stop 98, in which event the fluid may also pass around valve 80 through chamber 50.

Again, in order to provide for centrifugal balance, the arm 42 is provided with a threaded weight 100 adjustable within the threaded interior of the tubular arm 42. The weight 100 is provided with an O-ring seal 102 and is retained in adjusted position by a pin 104 adapted to extend across a slot 106 formed in the outer end of the weight 100.

Considering the operation of the relatively heavy unbiased piston valve 80 alone, it will be apparent that since there is no spring bias, the piston valve 80 under zero speed condition, is adapted to maintain a substantially zero pressure difference at opposite sides thereof. Thus, as seen in FIGURE 6, and at the zero speed point, the pressure differential is substantially zero as designated by the point O. On the other hand, the piston valve 80 is relatively heavy as compared to the piston valve 54, and accordingly, upon initial rotation of the hydraulic resistor centrifugal force builds up rapidly and the pressure differential across the valve will be maintained along the relatively steep curve OB. It will further be noted that the point B intersects the curve AEC at a point representing a somewhat slower speed than the starting speed indicated by the point E.

Operation

While the description as previously given indicates the overall operation, it is believed desirable to review the complete operation at this time.

Fluid at some elevated and presumably substantially constant pressure is applied to the passage 44 and flows through the restriction 46 into the chamber 18. At relatively low speeds, corresponding to speeds below the point B in the curve of FIGURE 6, the centrifugal force developed by the piston valve 80 will maintain a pressure drop across the valve 80 in accordance with the curve OB. At a speed corresponding to the point B on the curve, the curve OB intersects the curve AEC at which time the relatively light spring-biased centrifugal valve 54 starts to open and thereafter, as speed increases, the pressure drop between the chamber 18 and the passage 32 is in accordance with the curve BEC.

During low speed operation, as the fluid passes through the secondary piston valve 80 it drops in pressure to the lower "sink" pressure P_2 because of communication afforded by passages 86 and the ports or orifices 94 and 96. At a speed below point B in FIGURE 6, the primary piston valve 54 is closed by the force of the spring 64, the secondary piston valve 80 being open.

The secondary piston valve 80 is much heavier than primary piston valve 54 so that as the rotary hydraulic resistor speed increases the secondary piston valve 80 will begin to close and the pressure differential P_1-P_2 will increase rapidly and will follow the curve OB. After the speed reaches a speed corresponding to the point B the pressure differential P_1-P_2 will thereafter increase at a substantially slower rate along the curve BEC.

It is evident that the flow past the primary piston valve 54 will be some function of the centrifugal force of valve 54 and the spring load on the same valve. By the time the lowest operating speed is reached, centrifugal force will have completely closed the secondary piston valve 80 and thereafter, upon further increase in speed, the pressure differential P_1-P_2 will be determined by the primary piston valve 54 over the remainder of the speed range.

The drawings and the foregoing specification constitute a description of the improved hydraulic resistors in such full, clear, concise and exact terms as to enable any person skilled in the art to practice the invention, the scope of which is indicated by the appended claims.

What I claim as my invention is:

1. A hydraulic resistor comprising a pressure chamber, a restricted inlet passage for connecting said chamber

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to a source of fluid under pressure, said chamber having an outlet port for connection of a signal pressure transmission passage thereto, a hollow rotor, an outlet passage connected to the interior of said rotor, a pair of centrifugally biased valves in said rotor connected in parallel to control fluid flow from said chamber through said rotor, said valves each comprising a radially inwardly facing valve seat, a valve element movable radially toward and away from said seat, means for subjecting said valve element to a radially acting resultant force dependent on the pressure differential between said chamber and the interior of said rotor, one of said valve elements being substantially heavier than the other valve elements so as to have a substantially greater rate of change of centrifugal force than the other valve element during change in the speed of rotation of said rotor.

2. A resistor as defined in claim 1 comprising a bias spring operatively connected to the lighter of said valve elements to urge it toward its seat.

3. A hydraulic resistor comprising a stationary pressure chamber, a hollow rotor in said chamber, a passage for connecting the interior of said rotor to exhaust, a restricted supply passage for connecting the interior of said chamber to an external source of pressure, a signal passage for connecting the interior of said chamber to signal responsive means, said hollow rotor having a pair of radially disposed hollow arms having valves therein, said valves comprising ports in said arms affording parallel paths of communication between the interior of said chamber and the hollow interior of said rotor, valve elements movable radially in said arms toward and away from outwardly limited positions in which said valve elements respectively close said ports, said valves including means for subjecting said valve elements to radially inwardly acting forces variable in accordance with fluid pressure in said chamber, one of said valve elements being substantially lighter than the other of said valve elements and having spring means connected thereto to bias said one valve element toward its outer position.

4. A hydraulic resistor comprising a stationary pressure chamber, a hollow rotor in said chamber, a passage for connecting the interior of said rotor to exhaust, a restricted supply passage for connecting the interior of said chamber to an external source of pressure, a signal passage for connecting the interior of said chamber to signal responsive means, said hollow rotor having a pair

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of radially disposed hollow arms having valves therein, said valves comprising ports in said arms affording parallel paths of communication between the interior of said chamber and the hollow interior of said rotor, said ports being provided in the outer ends of said arms and forming radially inwardly facing valve seats, valve elements movable radially in said arms toward and away from said seats, said valves including means for subjecting said valve elements to radially inwardly acting forces variable in accordance with fluid pressure in said chamber, one of said valve elements being substantially lighter than the other of said valve elements and having spring means connected thereto to bias said one valve element toward its outer position.

5. A hydraulic resistor comprising a stationary pressure chamber, a hollow rotor in said chamber, a passage for connecting the interior of said rotor to exhaust, a restricted supply passage for connecting the interior of said chamber to an external source of pressure, a signal passage for connecting the interior of said chamber to signal responsive means, said hollow rotor having four hollow arms extending radially therefrom, said arms being provided in pairs, the arms of each pair being diametrically disposed with respect to each other, one arm of each pair having a valve therein, said valve comprising a port at its outer end defining a radially inwardly facing valve seat, valve elements in said arms movable radially toward and away from said seats, the other arm of each pair having a radially adjustable balance weight therein, the arms containing said valves being in communication with the hollow interior of said rotor and providing parallel flow passages between said hollow rotor and said chamber, one of said valve elements being of substantially less weight than the other of said valve elements, and having a bias spring connected thereto urging said one valve element radially outwardly toward its seat.

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