

Oct. 22, 1968

M. P. DONNELLY ET AL

3,406,668

HYDRAULIC TAPPET ASSEMBLY

Filed Jan. 19, 1968

3 Sheets-Sheet 1

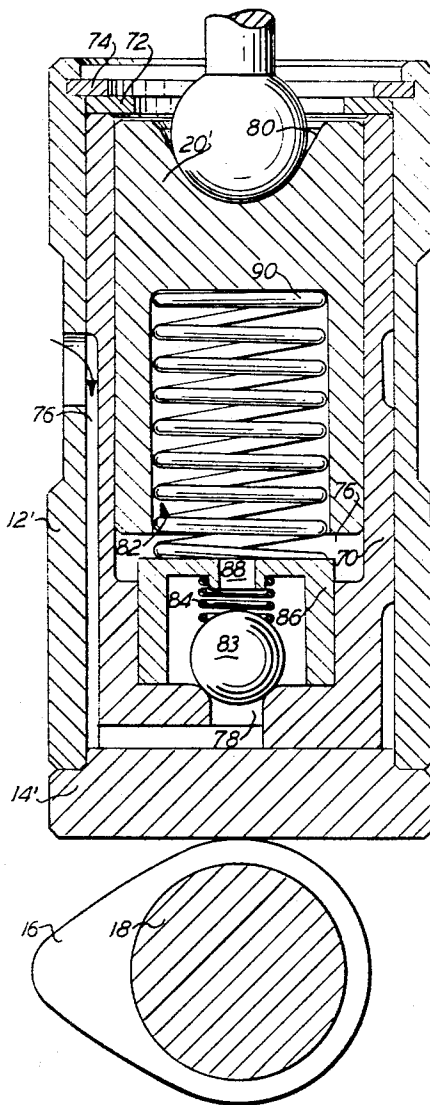


FIG. 1a

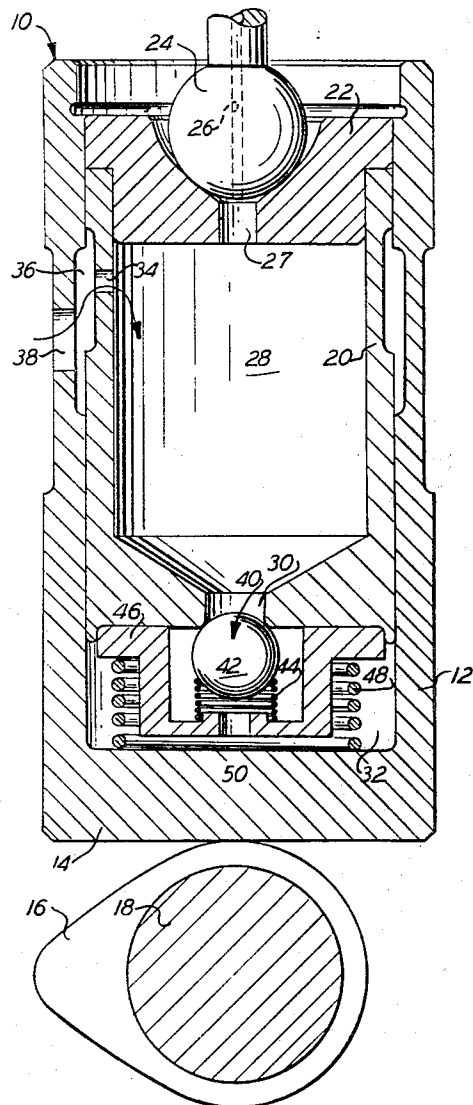


FIG. 1

MURRAY P. DONNELLY
JOHN E. KNOBLOCK
INVENTORS

BY *John R. Faulkner*
Robert E. McCallum

ATTORNEYS

Oct. 22, 1968

M. P. DONNELLY ET AL

3,406,668

HYDRAULIC TAPPET ASSEMBLY

Filed Jan. 19, 1968

3 Sheets-Sheet 2

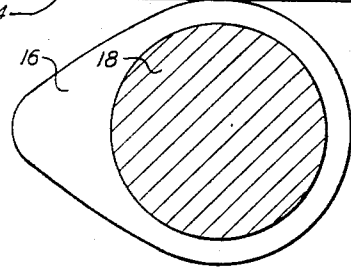
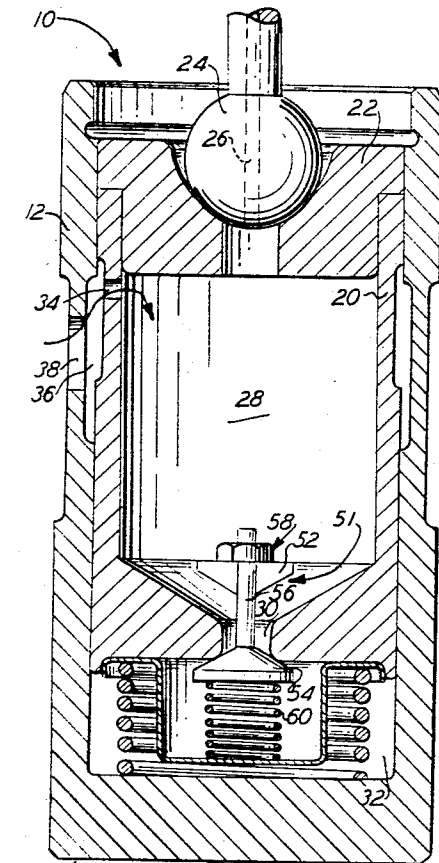


FIG. 2

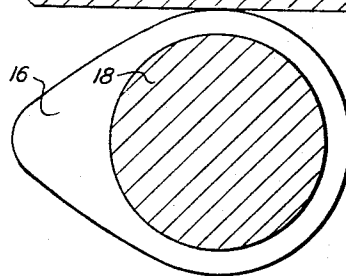
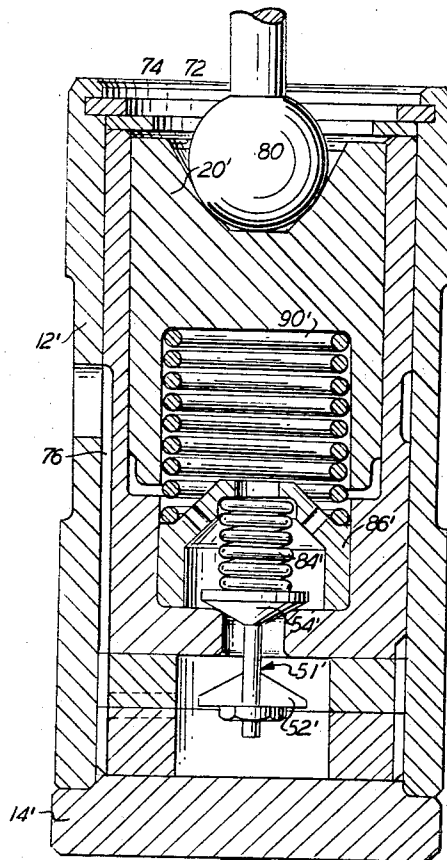


FIG. 2a

MURRAY P. DONNELLY
JOHN E. KNOBLOCK
INVENTORS

BY *John R. Faulkner*
Robert E. McCallum
ATTORNEYS

Oct. 22, 1968

M. P. DONNELLY ET AL

3,406,668

HYDRAULIC TAPPET ASSEMBLY

Filed Jan. 19, 1968

3 Sheets-Sheet 3

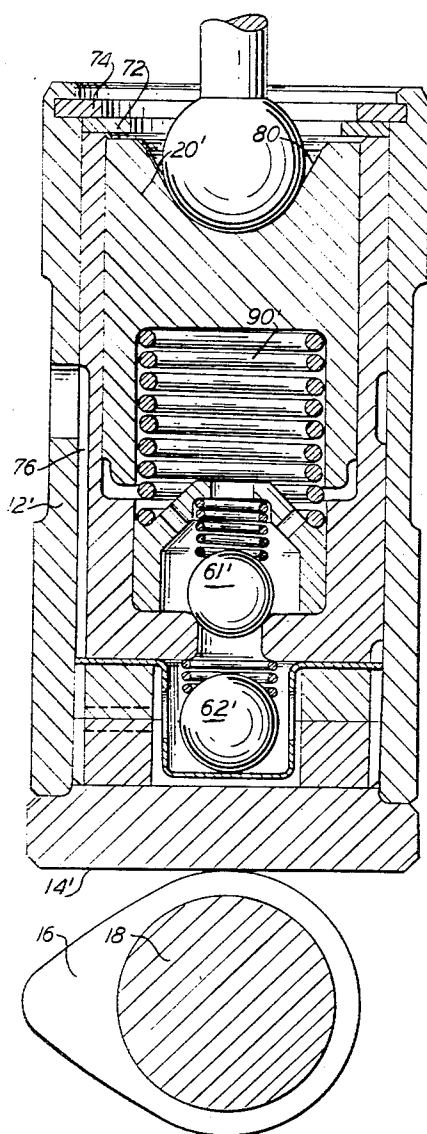


FIG. 3a

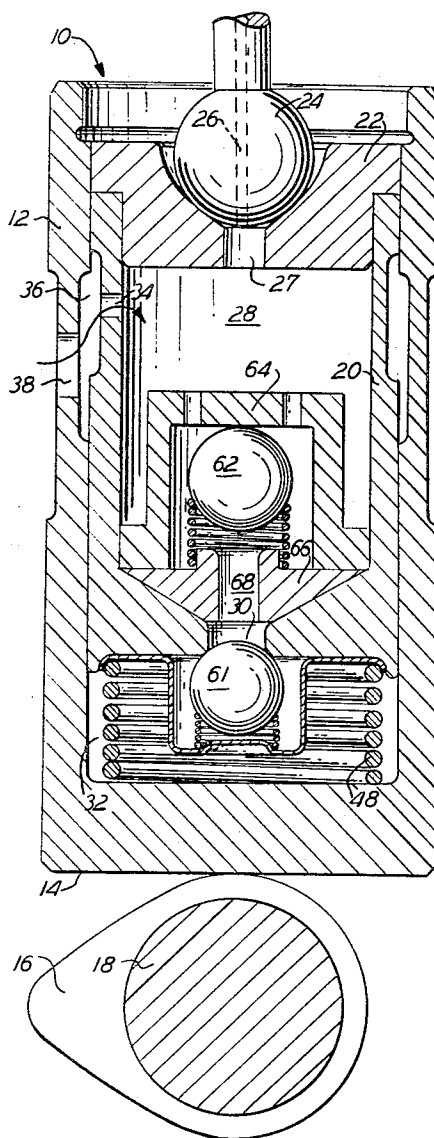


FIG. 3

MURRAY P. DONNELLY
JOHN E. KNOBLOCK
INVENTORS

BY *John R. Faulkner*
Robert E. McCallum
ATTORNEYS

1

2

3,406,668

HYDRAULIC TAPPET ASSEMBLY

Murray P. Donnelly, Inkster, and John E. Knoblock, Detroit, Mich., assignors to Ford Motor Company, Dearborn, Mich., a corporation of Delaware

Filed Jan. 19, 1968, Ser. No. 699,251

11 Claims. (Cl. 123—90)

ABSTRACT OF THE DISCLOSURE

A hydraulic tappet assembly for use in an internal combustion engine valve train, the tappet having bi-directional check valve means inertially responsive during critical engine speeds to prevent pump-up or collapse of the tappet below the toss speed of the valve train by effectively sealing the torque transmitting fluid chamber between the tappet lifter body and plunger.

This invention relates, in general, to a hydraulic tappet assembly. More particularly, it relates to a tappet construction that prevents both pump-up or collapse of the tappet during critical high engine valve train speeds to extend the maximum speed of the engine.

Many of the hydraulic tappets used in internal combustion engine valve trains in present day motor vehicles are of the single ball check type. That is, an outer cylindrical tappet body slidably receives an inner plunger, the two being separated by a fluid that is used to transmit the reciprocating motion of the tappet cylinder to the plunger and therefrom through a pushrod to the intake and exhaust valves. The check valve seats when the engine valve is in motion and the fluid is pressurized, and unseats, after the valve is closed, to add makeup fluid to the chamber. However, in a construction of this type, at some points during high speed operation of the engine, the valve closing spring experiences a phenomenon which might be termed excessive surging, which occurs when the valve train reaches critical speeds where the natural frequency of the spring is a low harmonic of the valve train operating frequency. This condition reduces the effective force of the spring and allows momentary separation of the valve train. This causes an intermittent unseating of the check valve permitting a buildup of fluid in the chamber to a point where the tappet assembly elongates. This ultimately results in a continual unseating of the valves, and a loss of horsepower and an inability on the part of the engine to develop any further horsepower beyond this point.

Attempts have been made to correct the pump-up disadvantage of the conventional hydraulic tappet by redesigning the tappet to collapse at the critical speeds by inverting the ball check valve, as in U.S. 2,902,015. At the high engine r.p.m. at which spring surging occurs, the inertia of the inverted ball valve will unseat it and permit some of the high pressure fluid to flow out of the chamber back into the supply line. Accordingly, the plunger moves towards the tappet base, that is, it collapses, and permits a higher speed operation than the conventional tappet before actual valve train toss speed occurs.

The above collapsing feature type tappet, however, has a disadvantage of being noisy because the tappet collapses causing excessive clearance or lash in the valve train.

The invention relates to a hydraulic tappet construction that incorporates the advantages of both the pump-up and collapsible type. That is, it provides a construction that permits neither a pump-up nor a collapse of the tappet during the critical high speed operation, and, there-

fore, increases the speed range to the point often referred to as the toss speed.

The invention accomplishes the above by using either a single check valve that is alternatively seatable against one or the other of a pair of ports controlling the flow of fluid into and out of the tappet fluid chamber, or a pair of check valves cooperating with a single port in a similar manner. Below the critical valve train speed, the valve and port combinations operate to transmit the upward thrust of the cam to open the intake or exhaust valve, the check valve seating to block flow of fluid out of the chamber except for a predetermined controlled leakage; and unseating during the engine valve closed portion of the valve train cycle to permit makeup flow of fluid into the chamber to restore the liquid to its original level. At the critical valve train speeds during which excessive valve spring surge occurs, the valve mass is chosen such that its inertia is in a direction to block flow into or out of the chamber regardless of whether the tappet assembly is accelerating or decelerating. Thus, no bleed of high pressure fluid from the fluid chamber occurs to cause collapse of the tappet, nor does a bleed of fluid into the chamber occur to effect a pump-up of the tappet.

It is an object of the invention, therefore, to provide a hydraulic tappet assembly that neither pumps-up nor collapses below the toss speed of the valve train.

It is another object of the invention to provide a hydraulic tappet assembly that includes check valve means having a mass responsive action at critical engine speeds to maintain the high pressure fluid chamber closed at all times during valve motion and yet permit an unseating of the check valve in the conventional manner to permit a makeup of leakdown fluid from the chamber when the engine valve is closed.

It is a still further object of the invention to provide a hydraulic tappet assembly consisting of an outer tappet cylinder slidably receiving a thrust transmitting plunger within it in a nesting manner, the nesting arrangement defining a fluid chamber between the two to which the supply of fluid is controlled by either a single check valve alternatively seatable against two ports, or a pair of check valves alternately seatable against a single port, or a pair of check valves operating independently against separate ports, as determined by the fluid pressure differential between the fluid in the chamber and the fluid at the source acting on the valve; the mass inertia of the check valve at the engine valve train critical speeds causing the check valve to seal the fluid chamber at all times during valve motion except for scheduled leakdown, regardless of the accelerating or decelerating condition of operation of the tappet.

Other objects, features, and advantages of the invention will become more apparent upon reference to the succeeding detailed description thereof, and to the drawings illustrating the preferred embodiments thereof, wherein:

FIGURES 1, 2 and 3 are cross-sectional views of three embodiments of a hydraulic tappet assembly embodying the invention; and,

FIGURES 1a, 2a, and 3a are cross-sectional views of three modifications of the FIGURES 1, 2 and 3 embodiments, respectively.

FIGURE 1, which is essentially to scale, shows a hydraulic tappet assembly 10 that preferably is for use in the valve trains of an internal combustion engine. It will be clear, however, that it will have equal use in other applications where the action to be described is desired, without departing from the scope of the invention. Assembly 10 includes an outer cup-shaped lifter body or cylinder 12 having a base portion 14 adapted to abut the cam 16 normally mounted on the camshaft 18 of

an internal combustion engine. An inverted cup-shaped plunger 20 is slidably mounted in cylinder 12 in a nesting manner, as shown, and has fixedly secured to it at its open end a push rod cup 22. The push rod ball has a valve train lubricating passage 26 that is connected by a port 27 to the hollow interior 28 of plunger 20.

The lower closed portion of plunger 20 is provided with an unrestricted port 30 for communication of fluid between chamber 28 and a fluid chamber 32 defined between the base of the plunger and the inner base of the cylinder 12. Chamber 28 is a low pressure fluid chamber, and is connected to a source of fluid under pressure such as engine oil pressure, at say 40 p.s.i., for example, through a plurality of ports 34, and annulus 36 between the plunger and cylinder, and port 38 in cylinder 12.

Communication of the essentially low pressure fluid in chamber 28 to chamber 32 through port 30 is controlled by a ball check valve unit 40. This latter unit consists of a ball valve 42 biased against the port opening by a spring 44 seated at its opposite end against a cup shaped spring retainer 46. The latter is biased upwardly against the bottom of a plunger 20 by a spring 48, and contains a fluid port 50.

Before proceeding to the operation of the tappet assembly, it should be noted that the mass of ball 42 is chosen such that the valve inertia will have essentially no effect on its movement below the critical engine valve train speed. However, at and above this speed, the valve mass inertia will cause it to seat against either port 30 or port 50, as determined by the direction of acceleration of cylinder 20 in combination with the fluid pressure differential between chambers 28 and 32, to maintain high pressure chamber 32 closed at all times during this period except for a predetermined leakdown of fluid from chamber 32 back into the low pressure system through the slight clearance between the walls of cylinder 12 and plunger 20.

In operation, the tappet assembly initially is filled with oil or other suitable liquid, such as engine oil, through port 38, annulus 36, chamber 28 and ports 30 and 50. At low engine operating speed, when cam 16 rotates off its base circle, cylinder 12 is moved upwardly in a known manner. The engine intake or the engine exhaust valve spring immediately exerts a reaction force through push-rod ball 24 and cup 22 onto plunger 20; which pressurizes chamber 32 to a higher value than the supply pressure in chamber 28, maintaining ball 42 sealingly seated against port 30. Since the fluid or oil in chamber 32 is incompressible, cylinder 12 and plunger 20 are moved upwardly essentially as a unit. Some leakdown of fluid from chamber 32 back to the supply source through the clearance between 12 and 20 will occur, as designed for in the normal manner. Therefore, there will be a slight relative movement of plunger 20 in a downward direction relative to the upward movement of cylinder 12.

When the cam has rotated to a point where the cylinder base 14 is contacting the base circle of the cam, the engine valve will have closed and its valve spring no longer will be effective as a force on plunger 20. Leakdown has created valve train separation, and spring 48 will move plunger 20 upwardly relative to cylinder 12 to take up the clearance. This creates a lower pressure in chamber 32 than chamber 28, unseating ball valve 42 and permitting fluid from chamber 28 to flow into chamber 32 to make up for the amount of fluid lost due to the leakdown previously described.

The above action occurs during operation below the critical valve train speed. When this critical speed is reached, at say 5000 r.p.m., for example, the spring coils surge excessively. Accordingly, on plunger 20 reaction is increased and decreased intermittently. Without the construction of the invention the tappet would pump up when the reaction is less than the force of spring 48, which would cause a permanent opening of the engine valve and a resultant limiting of the engine power output.

With the invention, however, the mass inertia of the check valve at this critical speed is now effective to maintain high pressure chamber 32 effectively sealed regardless of the fluttering spring reaction.

More specifically, at the critical speed, a fast upward acceleration of cylinder 12 and plunger 20 causes valve 42 under its own inertia to seat against port 50 and block leak of fluid between chambers 32 and 28. A sudden reversal in acceleration of cylinder 12 and plunger 20, relieving the reaction force on plunger 20, causes ball 42 to unseat from port 50 and seat against port 30, thus again blocking leak of fluid between chambers 32 and 28.

Thus, the seating of ball 42 against port 50 or port 30 will prevent collapse or pump-up by preventing excessive leakage of fluid between chambers 28 and 32.

It will be seen, therefore, that with the prevention of pump-up and collapse of the tappet assembly, the parts will maintain the same relative positions at engine critical speeds as at speeds below this level, and, therefore, the valve train will not be noisy or force the engine valve to remain in an open position below the toss speed. Accordingly, it will permit the engine to continue on to higher speeds until the valve train inertia no longer allows the engine valves to seat.

FIGURE 2 shows a modified construction. The plunger and cylinder constructions are essentially the same as in FIGURE 1, the primary difference being in the use of two valve members controlling a single port instead of a single ball valve controlling two ports. More specifically, FIGURE 2 shows a shuttle valve 51 consisting of a pair of conically shaped valve members 52 and 54 that are located on opposite sides of port 30 and interconnected by a stem 56. In this case, the valve members are shown as being adjustable axially with respect to each other by a nut and screw combination 58 to vary the intervals between the valve members seating. It will be clear, of course, that the shuttle valve could be formed as a single unit without adjustment, without departing from the scope of the invention.

The shuttle valve 51, in this case, is biased against the lower portion of port 30 by a spring 60.

In all other respects, the construction and operation of the tappet assembly of FIGURE 2 is essentially the same as that of FIGURE 1. The valve 51 alternately seats against the opposite faces of port 30 to prevent both pump-up and collapse of the tappet assembly at the engine valve train critical speed level. Again, the mass of the shuttle valve 51 is chosen so that the mass inertia of the valve is ineffective below the critical engine speed.

FIGURE 3 shows a further modification that is essentially the same in operation as the FIGURES 1 and 2 embodiments. In this case, two separate ball valve members are used instead of the shuttle valve 51 of FIGURE 2 and the single ball 42 of FIGURE 1, and one of the ball valves is larger than the other. More specifically, the lower face of port 30 is blocked at times by a ball valve member 61 that is essentially the same as ball valve 42 acting against port 30 in FIGURE 1. The opposite side of port 30 is controlled by a ball valve 62 that is larger and heavier than ball 61, and spring pressed against a hat-shaped retainer 64. The spring is seated at its opposite end against an adapter 66 containing a passage 68 aligned with the port 30.

The operation is essentially the same as that of FIGURES 1 and 2. Lift of cylinder 12 provides a reaction or resistance on plunger 20 to increase the pressure in chamber 32 to a value sufficient to seat ball valve 61 against port 30 and thus permit unitary upward movement of the tappet assembly. When the cylinder 12 returns to the base circle of cam 16, the pressure in chamber 32 is reduced to a value permitting separation of plunger 20 upwardly from cylinder 12 by spring 48, thereby opening the valve 61 and permitting makeup of fluid leakage from chamber 32. The ball valve 62, in this case, does not play any part

in the normal operation of the tappet assembly below the critical speed of operation of the valve train described above.

At the critical valve train speed, there is momentary reduction in reaction force during the valve operating cycle due to the engine valve spring surge. Accordingly, once the critical valve train speed is reached and the reaction force is lower than the force of spring 48, the pressure differential between chambers 28 and 32 is reversed. The load of spring 48 would normally unseat valve 61 allowing pump-up. However, the inertia of ball valve 61 maintains the valve seated during downward acceleration.

During the upward deceleration movement, the heavier mass inertia of valve 62 will have seated it against port 30 before the mass inertia of valve 61 causes it to uncover port 68. This prevents any escape of fluid from or addition of fluid to chamber 32, and thus prevents pump-up or collapse of the tappet assembly.

FIGURES 1a, 2a and 3a show further modifications or other possible constructions for the FIGURES 1, 2 and 3 embodiments, respectively. More specifically, FIGURE 1a again shows the outer cylinder 12', but in this case receiving an inner inverted cup cylinder 70. The inner and outer cylinders are fixed against relative movement by a pair of snap rings 72 and 74. A fluid annulus 76 is provided between the cylinders for communicating the low pressure source fluid to port 78. Inner cylinder 70 slidably receives in a nesting manner the inverted cup-shaped plunger 20' having a push rod cup 80. The ball check valve 83, in this case, is biased by a spring 84 against the port 78, the spring being seated against a retainer 86 having a port 88. The retainer in turn is positioned by a compression spring 90.

Thus, in normal operation, movement of cylinders 12' and 70 by cam 16 immediately increases the pressure in chamber 82 to a high level and maintains the ball valve 83 seated against port 78. Thus, the fluid in chamber 82 will transmit the upward movement of the cylinder to the engine valve. When cam 16 rotates so its base circle contacts the cylinder base 14', the decreases in pressure in chamber 82 to a value below the source pressure permits the ball 83 to be unseated and any fluid makeup needed to flow into chamber 82, in a manner similarly described in connection with FIGURES 1, 2 and 3.

At the engine critical speed, the mass inertia of ball 83 either will seat it against port 78 or port 88, depending upon the direction of acceleration of cylinders 12' and 70, and thus prevent excessive leakage of fluid out of or supply of fluid to high pressure chamber 82. Accordingly, it prevents a pump-up or collapse of the tappet assembly, in the manner already described in connection with the other embodiments.

FIGURES 2a and 3a are similar in construction and operation to the FIGURE 1a showing, and corresponding, so far as the number of and kinds of valve members is concerned, to the FIGURES 2 and 3 embodiments, respectively. That is, in FIGURE 2a, the single, double-acting ball 83 of FIGURE 1a is replaced by the two-piece shuttle valve 51' having members 54' and 52', spring retainer 86' and spring means 84' and 90' remaining essentially the same as already described in connection with FIGURE 1a.

In FIGURE 3a, two separate ball check members or valves 62' and 61' are provided instead of the single ball of FIGURE 2, and corresponding to the two ball check valves 62 and 61 of FIGURE 3. The operation of the FIGURES 2a and 3a constructions are believed to be clear from the above description of operation of the embodiments shown in FIGURES 1, 2, and 3 and 1a, and therefore, are not repeated.

From the foregoing, therefore, it will be seen that the invention provides a hydraulic tappet assembly that not only permits operation in a conventional manner, but

prevents pump-up or collapse of the tappet assembly in the critical range of valve train operation at high engine speeds so that even higher engine speeds can be obtained until the actual valve toss speeds occurs.

While the invention has been described in its preferred embodiments in the drawings, it will be clear to those skilled in the arts to which it pertains that many changes and modifications may be made thereto without departing from the scope of the invention.

We claim:

1. A tappet assembly comprising, an outer cup-shaped cylinder oscillatable at varying frequencies, an inner thrust transmitting plunger slidably mounted in said cylinder in a nesting manner, spring means biasing said cylinder and plunger apart, a fluid chamber between said plunger and cylinder containing fluid for transmitting thrust between said cylinder and plunger, a source of fluid under pressure, passage means connecting said fluid to said chamber, and check valve means of predetermined mass in said passage means movable below a predetermined frequency of oscillation of said cylinder between flow blocking and unblocking positions as a function of the differential fluid pressure force between the chamber pressure and that of said source acting thereon to control the fluid flow to and from said chamber, said valve means being movable above said predetermined frequency to a flow blocking position in response to acceleration of said cylinder in either direction and in response to the inertia of said valve means.

2. A tappet assembly as in claim 1, said passage means having a pair of spaced control ports between said source and said chamber, said valve means comprising a check valve reciprocally movable between and alternately seatable against said ports.

3. A tappet assembly as in claim 1, including means defining a pair of spaced ports connecting said source and said chamber, said valve means comprising a single valve reciprocally movable between and alternately seatable against said ports, said biasing means biasing said valve against one of said ports, the mass inertia of said valve above said predetermined frequency of oscillation of said cylinder alternately seating said valve against one or the other of said ports as a function of the direction of acceleration of said cylinder.

4. A tappet assembly as in claim 1, said chamber having a fluid port, said valve means comprising a pair of spaced check valves alternately seatable against said port.

5. A tappet assembly as in claim 4, said check valves comprising a pair of spaced interconnected valve members straddling said port.

6. A tappet assembly as in claim 4, said valve comprising a shuttle valve having a pair of spaced valve members one on each side of said port and interconnected by means extending through said port.

7. A tappet assembly as in claim 4, said biasing means biasing one of said valves against said port, the mass inertia of each of said valves above the said predetermined frequency of oscillation of said cylinder effecting the seating of one or the other of said valves against said port as a function of the direction of acceleration of said cylinder.

8. A tappet assembly as in claim 4, said valves having different masses, said biasing means comprising spring means biasing one of said valves against said port, the mass inertia of each of said valves above said predetermined frequency of oscillation of said cylinder effecting the movement of both valves in the same direction at different speeds and the seating of one or the other of said valves against said port as a function of the direction of movement of said cylinder.

9. A tappet assembly as in claim 8, including additional spring means biasing the other of said valves away from said port.

10. A tappet assembly as in claim 4, said valves being located on opposite sides of said port, said biasing means

biasing one of said valves against said port, and additional biasing means biasing the other of said valves away from the said port.

11. A tappet assembly comprising, an outer cup shaped cylinder reciprocable at varying frequencies, an inner plunger slidably mounted in said cylinder in a nesting manner, spring means biasing said cylinder and plunger apart, a fluid chamber between said plunger and cylinder containing a fluid for transmitting thrust from said cylinder to said plunger and vice versa, a source of fluid under pressure, passage means connecting said fluid to said chamber, and check valving means in said passage means movable between flow blocking or flow unblocking positions to control the fluid flow to and from said chamber, biasing means biasing said valve means to a first flow blocking position, said valve means being acted upon by the pressure differential between the fluid in said passage means and said chamber and moved thereby alternately between said first position and a second flow unblocking position in response to changes in pressure in said chamber, a reciprocable member engaging said plunger and

acting as a thrust transmitting reaction member upon movement of said cylinder towards said plunger to effect compression of said spring means and pressurization of said chamber and movement of said valve means to said first position while permitting depressurization of said chamber and expansion of said spring means to permit movement of said valve means to said second position upon movement of said cylinder in the opposite direction, the mass inertia of said valve means above a predetermined frequency of oscillation of a portion of said engine valve train effecting movement of said valve means to a flow blocking position regardless of the direction of acceleration of said cylinder.

References Cited

UNITED STATES PATENTS

2,784,707	3/1957	Skinner	123—90
2,790,430	4/1957	Lowther	123—90
2,795,218	6/1957	Heiss	123—90

AL LAWRENCE SMITH, *Primary Examiner*.