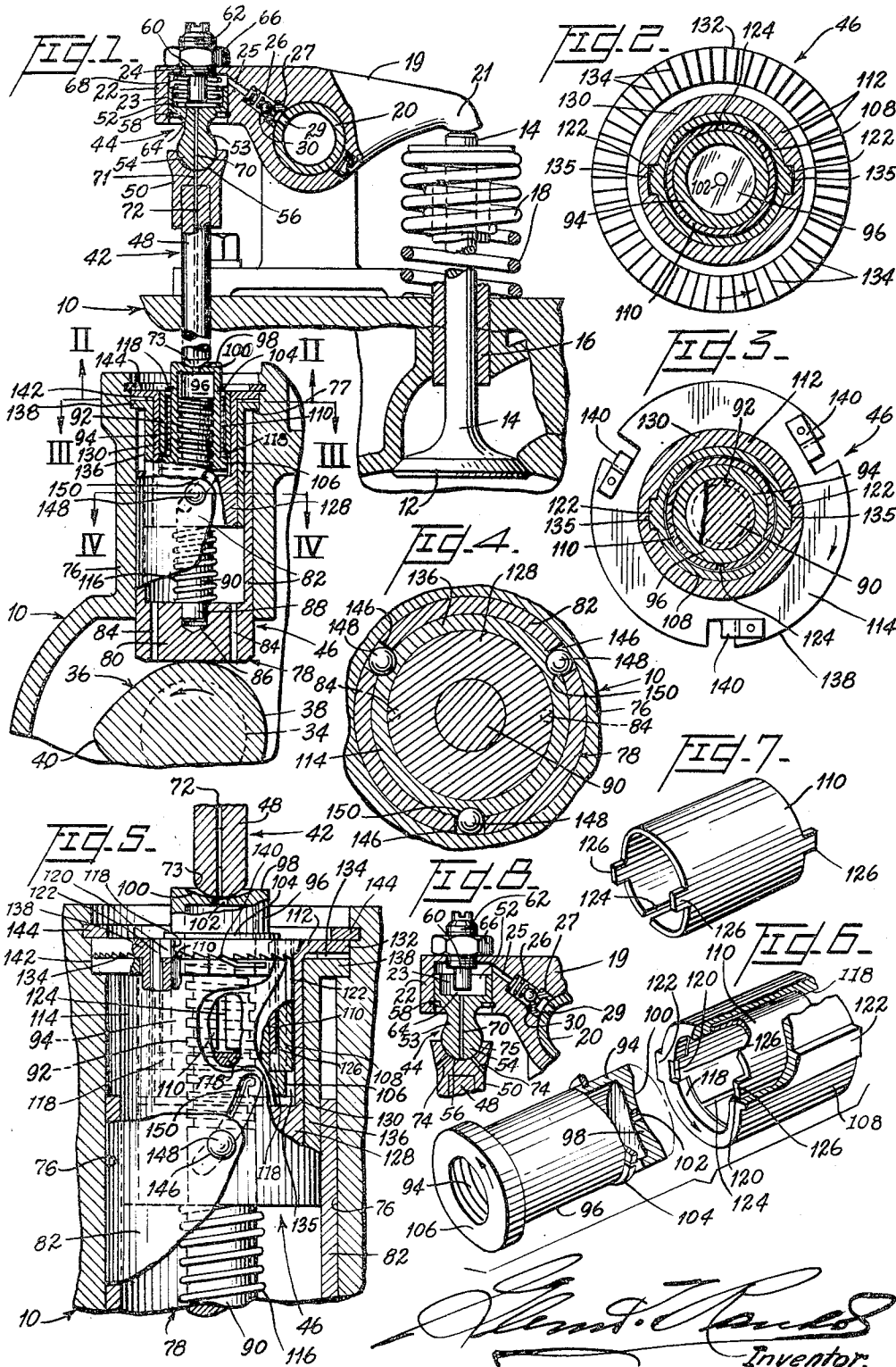


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COMPENSATING VALVE LIFTER MECHANISM
FOR INTERNAL-COMBUSTION ENGINES
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COMPENSATING VALVE LIFTER MECHANISM FOR INTERNAL-COMBUSTION ENGINES

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The present invention relates generally to valve tappets or lifters for internal-combustion engines and which are automatically operable to compensate for expansile or contractile clearance between the parts of the valve drive train, and more especially relates to novel and improved mechanism for incorporation in said valve drive train to maintain a desirable clearance therein under all engine operating conditions.

Broadly, this invention concerns novel mechanism, interposed in the valve drive train of an internal-combustion engine between the valves and the camshaft, which mechanism is adapted to automatically maintain zero-clearance in said valve train during engine operations while accommodating a clearance gap operably adapted to control said mechanism to adjust in one direction only and simultaneously compensate for thermal expansion and contraction of the parts.

To this end, my invention provides novel means for initial manual adjustment (in accordance with known thermal-responsive characteristics of the particular engine) of an element included in the mechanism whereby a predetermined clearance gap is establishable in the linkage of the valve drive train; means for locking the said adjustable element in the selected position of adjustment; novel means for utilizing the fluid pressure of the engine lubrication system in cooperation with a preloaded compression spring, or either per se, for resisting closure of said gap from its normal cold-engine status, said gap being subject to slight variations therein resulting from thermal changes only in said valve train drive when the engine is in operation; novel valve lifter means of the mechanical extensible type adapted to automatically absorb or eliminate in cooperation with said hydraulic-spring means, any backlash due to wear etc. that may develop in said valve train external to the aforesaid clearance gap; and other inter-related and combined arrangements of elements to produce the novel operativeness of the mechanism, as will appear.

A further important object of my invention is the provision in mechanically-operated valve tappets of the type under consideration of novel combined ratchet and friction clutch mechanism operatively disposed between the principal body parts of such tappets, and operable responsive to engine camshaft rotation to separate the body parts automatically to take up all backlash or play between the camshaft and valve without interfering with the proper seating of the valve under all engine operating conditions, and which, moreover, will automatically compensate for expansion and contraction caused by thermal changes in the engine.

Preferably, the clearance gap in the valve drive train is manually established with the engine cold, allowance for the expansion factor only being thus requisite.

Heretofore, numerous attempts have been made to eliminate backlash in the mechanism interposed between the camshaft and the stem of the engine valve which is to be intermittently opened and closed. As a general rule, the resultant prior disclosures have been directed to improve-

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ments in the valve lifter mechanism per se, whereas my invention contemplates not only a novel and improved lifter construction, but additionally contemplates the incorporation of a novel assembly of elements adapted to provide a force-transmitting connection of limited yieldable character between the upper end of the push rod and one end of the rocker arm element included in such valve train. As a result, the present invention provides an arrangement which insures optimum valve operation irrespective of the thermal conditions of the engine, and which automatically obviates the heretofore periodically required adjustment of tappets to eliminate noise and so on.

The primary object of my invention therefore is to provide, for incorporation in the valve drive train of a conventional internal-combustion engine, an improved one-way adjustable valve lifter mechanism to automatically compensate for backlash in combination with a two-way adjustable clearance mechanism responsive to expansion and contraction of the engine due to thermal changes, whereby to maintain substantially zero-clearance in said valve train during engine operations by the inter-cooperative relation obtaining between said mechanisms.

It is a further object of the invention to facilitate the attainment of the recited primary objective by the provision of novel mechanical lifter mechanism of the compensating type which is automatically adjustable in one direction only, and when actuated by a cam on the camshaft, is adapted to intermittently effect the cyclic opening and closing of a conventional engine valve in silent fashion, and efficiently.

Further, it is an object of this invention to facilitate the attainment of said primary objective by the provision of means for manually setting the backlash absorbing mechanism when the engine is cold, whereby to establish a selected clearance gap in the valve train which may vary in extent responsive to thermal conditions only of the engine, but cannot be varied in consequence of the operative action of said lifter mechanism after lost-motion in the valve drive linkage has been taken up by the latter.

An important object is to provide in an automatically adjustable valve lifter mechanism, manually-adjustable means whereby a given clearance of backlash condition within the valve drive train may be established and so maintained by operation of the aforesaid lifter mechanism.

A further object related to that next above is to associate with said manually-adjustable means, novel mechanical lifter mechanism of the automatically compensating type which is adapted to adjust in one direction only, and further adapted to maintain zero-clearance substantially in the mechanism of said valve drive train at all points except at the established clearance point aforesaid.

A novel provision related to the two immediately preceding objects resides in the said manually-adjustable means once it has been adjusted to provide a predetermined cold-engine clearance, in that such clearance cannot be altered in consequence of the aforesaid lifter mechanism operation after lost-motion external to the clearance setting has been taken up, the latter mechanism being unidirectionally operative only to take up backlash outside of said predetermined clearance.

A further important object of the present invention is to associate with automatically adjustable tappets, novel manually-adjustable means whereby a given clearance in the valve drive mechanism may be established at will and which is subject to automatic modulation within the range of said clearance to compensate for expansion and contraction due to variable thermal conditions of the engine, and wherein operations of said tappet are only effective to maintain substantially zero-clearance between the parts of the valve drive mechanism external to those parts controlling the aforesaid given clearance.

An object importantly related to the object next above is to associate with said manually-adjustable means, novel mechanical valve lifter mechanism of the automatically compensating type and which is adapted to adjust in a lengthening direction only to maintain substantially said zero-clearance in said valve drive mechanism, said manual means being effective at all times to prevent said lifter mechanism from over-adjusting said drive mechanism.

A further salient object relating to the aforesaid novel manually-adjustable means is the inclusion of novel combined spring and hydraulic pressure means operable jointly or separately to accommodate expansion and contraction of the valve drive linkage due to thermal changes only, said valve drive system including means for lubricating the parts thereof from the engine pressure lubricating system which also supplies the hydraulic pressure for the manual clearance means.

There are other objects, features and advantages which the instant invention provides. These will be set forth in the course of the detailed description of the construction and operation of the invention to follow with reference to the accompanying drawing, which exhibits two exemplary embodiments of the invention in the valve drive train of an otherwise conventional internal-combustion engine.

In the drawings:

Figure 1 is a vertical sectional view with some elements portrayed in elevation, of an internal-combustion engine incorporating the valve lifter mechanism of the present invention;

Figure 2 is an enlarged horizontal sectional view taken substantially along the line II—II of Figure 1, this view exhibiting especially the toothed ratchet under surface of a rotatable sleeve included in the mechanism;

Figure 3 is an enlarged horizontal sectional view taken substantially along the line III—III of Figure 1, this view exhibiting particularly a circular series of leaf-spring pawls adapted to cooperate with the ratched under surface of said rotatable sleeve whereby to actuate the latter;

Figure 4 is an enlarged horizontal sectional view taken on the line IV—IV of Figure 1, this view portraying especially a series of circumferentially spaced spherical elements and the raceways wherein they are operatively retained for purposes to appear;

Figure 5 is an enlarged view, partly in section and partly in elevation, of a fragmentary portion of Figure 1, this view more clearly presenting features of my valve lifter mechanism which may not appear clearly in said Figure 1;

Figure 6 is an exploded perspective view, partly in elevation and partly in section, of the internally threaded adjusting head, the torque sleeve associated therewith, and the torque spring associated with said sleeve;

Figure 7 is a perspective view of the said torque sleeve spring per se; and

Figure 8 illustrates a slight modification, being a reproduction of a fragmentary portion of Figure 1, a compression spring appearing in the latter view being omitted.

Referring to the drawings, and particularly to Figure 1, a preferred embodiment of my invention is shown incorporated in a conventional internal-combustion engine, fragmentarily shown and designated by the numeral 10. Such an engine includes a plurality of valves, one of which is exemplarily shown at 12, and includes an integral stem portion 14 slidably mounted in a bushing 16. A relatively heavy compression spring 18 about said stem normally biases the valve to closed position, as is well understood.

Numeral 19 designates a rocker arm pivotally mounted intermediate its ends on an oil distribution pipe 20. One end 21 of said rocker arm is normally in contact with the upper end surface of the valve stem 14, whereby clockwise movement of the rocker arm about its pivotal mounting will open the valve 12 against the force of spring 18.

In accordance with the concepts of my invention, the other end 22 of the rocker arm is bored as at 23, and

counterbored as at 24 for reasons to appear. The counter-bore 24 is internally threaded, and the bore 23 is in fluid communication with the oil distribution pipe 20 via successively enlarged diagonal passageways 25, 26, and 27 formed in said arm, and an aligned discharge port 29 provided in the pipe 20, as shown.

A check valve assembly 30, which includes a valve seat member press-fitted into the passageway 27 and a ball normally maintained seated against said member by a conical compression spring in the passageway 26, is provided as clearly shown, and for a reason to be explained.

Illustrated in broken lines and designated 34, is the camshaft of the engine. Numeral 36 indicates a cam on said shaft, said cam including a concentric portion 38 and a lobe portion 40. Interposed between cam 36 and rocker arm 19 is a valve train, generally designated 42, adapted to oscillate said rocker arm responsive to the rotation of the camshaft 34, whereby to intermittently effect the opening and closing of the valve 12, as is well understood.

The valve train 42 herein disclosed incorporates the novel mechanism comprising the instant invention. That is to say, my invention contemplates the attainment of its objectives by the provision of what will be termed a predetermined clearance mechanism, generally designated 44, positioned for illustrative purposes only in the rocker arm end 22; the provision of a novel valve lifter mechanism, designated generally by the numeral 46, which is in contact at its lower end with the cam 36 and is actuated by the latter; and the provision of axially aligned restrictive passageways in the conventional push rod 48, in its connector socket member 50, and in a component of the aforesaid clearance mechanism 44, as will be explained in more detail hereinafter.

With particular reference now to the predetermined clearance mechanism 44, it is seen to include a cup-shaped hollow piston member 52 having a depending shank 53 which terminates in a substantially spherical portion 54 seated in a semi-circular socket 56 of the connector member 50. The piston member 52 is slidably operable in bore 23 between a split lock or retainer ring 58 and the reduced depending shank 60 of a screw member 62 adjustably engaging the threaded counterbore 24 as shown in Figure 1, and also in the Figure 8 modification.

Numeral 64 indicates a gap or clearance which obtains between shank 60 and the upper surface of the bottom wall of cup-shaped piston member 52. The extent of the gap 64 is obviously variable dependent upon engine requirements as is understood. Any specific cold-engine gap may be normally maintained by the lock nut 66, as should be obvious.

Interposed between said bottom wall of the cup-shaped piston member 52, and the under surface of the top wall defining the bore 23, is a compression spring 68 which normally biases said piston downwardly against the retainer ring 58, as demonstrated in Figure 1. However as will be clarified hereinafter, and as demonstrated in Figure 8, inclusion of the spring 68 is not requisite for the operativeness of the aforesaid novel clearance mechanism 44.

From the foregoing it should be apparent that the gap 64 may be selectively increased or decreased by means of the screw member 62; that the resultant disposition of the shank 60 relative to the bottom of the piston cup may be maintained by means of lock nut 66; and that when the engine is in operation, oil under pressure from the distribution pipe 20 will enter the bore 23 above the piston member 52 via port 29, check-valve valve assembly 30, and connecting passageways 26 and 25. In the embodiment presented in Figure 1, this oil pressure aids and abets the spring 68 in normally biasing the piston 52 downwardly against the retainer ring 58 to bring the shank 53 into engagement with the push rod socket 56 in which disposition of the parts, the valve drive gear is free of backlash. In the embodiment presented in Figure 8, the

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oil pressure per se normally seats the piston against said retainer ring.

It should consequently be manifest that the mechanism 44 exhibits a hydraulic dashpot arrangement (spring-loaded in the Figure 1 embodiment but not in the Figure 8 embodiment), including a manually dispositionable element 69 for establishing a predetermined limit stroke of the dashpot movable piston member 52.

Obviously the oil introduced as described above, also serves to lubricate the piston 52, but impounding of the oil is obviated to enable modulation of the clearance gap 64 by the provision of a restricted passageway 70 in the depending shank 53 of said piston, a similar passageway 71 in the socket member 59, and a similar passageway 72 extending the full length of push rod 48. As clearly seen in Figure 1, these three passageways are in alignment and are all of a much smaller diameter than that of the diagonal passageway 25. With this arrangement, the clearance mechanism 44 is operative to compensate for thermal changes in the engine and the movable elements of the valve gear train are also lubricated, the oil flowing through the said aligned passageways 70, 71, and 72 and onto the lifter mechanism 46, and therefrom into the engine sump for recirculation, as will appear.

In the modified Figure 8 embodiment, the passageways 71 and 72 are omitted, a plurality of angularly disposed passageways 74 provided in the socket member 50 being substituted therefor. With this arrangement, the spherical portion 54 combines with the semi-circular socket 56 to form what will be termed a valve which is designated 75, and the function of which will be explained hereinafter.

The lifter mechanism 46 of the valve train 42 is located directly above the camshaft 34 in a conventional cylinder portion 76 of the engine 10, said cylinder being modified with a counterbore at the outer end to form an annular internal shoulder 77. Said mechanism includes a cup-shaped piston member 78 having a bottom wall 80 which, as will appear, is at all times maintained in contact with the cam 36, and a cylindrical side wall 82. Said bottom wall 80 has formed therein one or more oil drainage ports 84, and centrally, a socket 86 for the reception of the lower reduced extremity 88 of a stem 90 which projects above the top marginal edge of the piston side wall 82. The thus projecting section 92 of said stem is provided in the present embodiment with left-hand threads to cooperate with the right to left one-way adjusting movements of the mechanism 46 as viewed from Figures 1 and 5, preferably of square contour as illustrated, for engagement with mating internal threads provided in the circular side wall 94 of what will be termed an adjustable head member 96.

As seen most clearly in Figure 5, the horizontal top wall 98 of head member 96 is dished out to provide an arcuate seat 100 for the rounded lower extremity 73 of push rod 48, and has formed therein an opening 102 for constantly lubricating the threaded connection referred to in the preceding paragraph. With reference also to Figure 6, the side wall 94 of the head member has an external annular groove therein for the reception of a split retainer ring 104, and terminates at its lower end in an outwardly projecting annular flange 106.

The mechanism 46 further comprises a combined ratchet and friction clutch mechanism which includes a torque sleeve 108; a torque sleeve spring 110; the internally threaded thimble 96; a substantially cup-shaped ratchet member 112; a tubular pawl carrier 114; and a compression spring 116.

The torque sleeve 108 is exhibited perspectively in Figure 6, and terminates at each end in an internal circular flange 118 having a pair of diametrically opposed slots 120 therein. Externally, the sleeve 108 is seen to have a pair of diametrically opposed splines 122, each extending longitudinally from end to end of said sleeve.

The torque sleeve spring 110 per se is perspectively

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depicted in Figure 7; its disposition within the torque sleeve 108 is exhibited in Figure 6. Split longitudinally as at 124, said spring has at each end thereof a pair of diametrically opposed projections or lugs 126 formed integrally with its body portion. Preferably but not necessarily, these lugs are each approximately 90° distant from the split 124. In assembled disposition, the lugs 126 engage in the slots 120 of the torque sleeve, whereby manifestly and as will appear, said sleeve 108 and the spring 110 provide an assembly which may be actuated rotationally and axially as a unit.

The ratchet member 112 includes a bottom wall 128 through which the stem 90 extends, and a cylindrical side wall 130 which terminates at its upper end in a lateral flange 132 the under surface of which has formed therein a circular series of closely generated ratchet teeth 134, as portrayed especially in Figures 2 and 5. The internal periphery of the member 112 is provided with a pair of longitudinally extending diametrically opposite grooves 135 which embrace the splines 122 of the torque sleeve in the assembled mechanism, as shown particularly in Figures 2 and 3.

The pawl carrier 114, which is designed to cooperate with the said ratchet member 112 in a manner and for a purpose to be explained, includes a tubular body portion 136 which terminates at its upper end in a laterally projecting annular flange portion 138, and a plurality of flexible pawl elements 140 each rigidly attached at one end to the upper surface of the flange 138 as illustrated in Figure 3.

The diameter of the flange 132 coincides with that of the flange 138, so that both of said flanges extend in superimposed relation into an annular recess 142 provided therefor between the circular shoulder 77 and a split retainer ring engaging an annular groove in the cylinder portion 76, as best seen in Figure 5. The latter view also discloses that the free segments of the pawls 140 project angularly upwardly above the top surface of flange 138 into engagement with the ratchet teeth 134, and that said split retainer ring 144 overlies the marginal edge portion of the flange 132.

The compression spring 116 is interposed about the stem member 90 between the bottom wall 80 of the piston 78 and the bottom wall 128 of the ratchet member 112. With this arrangement, the flange 132 is maintained in frictional engagement with the retainer ring 144, whereby to obviate fortuitous rotary movements of the ratchet member and to operatively stabilize the parts of the lifter unit 46 in their relatively adjusted positions. In other words, all rotational movements of the ratchet member 112 are positive movements imparted thereto by the pawl carrier 114. This compression spring possesses the additional function of restoring the normal relative positioning of the piston member 78 and pawl carrier 114 after the piston member 78 has been positively moved by the engine cam 36 to actuate the ratchet mechanism 140, 134 during which latter operation, the aforesaid relative relation normally obtaining between the piston and pawl carrier is taken up. Restoration of said relative movement between the piston member 78 and pawl carrier 114 by the spring 116 being effective with the bottom wall 80 in contact with the closing ramp and base circle of the cam 36.

Reverting again to the piston member 78 and with reference particularly to Figures 4 and 5, it is noted that the cylindrical wall 82 of said member incorporates a plurality of circular openings 146, three being illustrated. Each opening 146 surrounds a spherical element or ball 148, and each ball is maintained by the wall of cylinder 76 in engagement with one of three circumferentially equally spaced angularly disposed arcuate raceways 150, the latter being formed in the tubular body portion 136 of the pawl carrier 114. It is important to further consider here that the positive upward thrusts and the opposing action of the spring 116 reciprocally act on the piston member

78 to produce slight axial movements thereof relatively to the pawl carrier member 114, which movements are instantly and simultaneously converted into oscillatory movement of said carrier member relatively to the member 78. These oscillating movements adjust the pawls 140 relatively to the toothed member 112 and into engagement with the teeth thereof, thereby rotatably raising the head member 96 to adjust the parts of the valve drive train in accordance with the backlash present therein.

With reference to the split ring 104 associated with the head member 96, it is noted that the projecting portion of said ring overlies the upper end of the torque sleeve 103, so that it should be manifest that said head member, said sleeve and the torque sleeve spring 110 are limited to axial movements as a unit in an upward direction only.

Operation

It will be assumed that with the engine 10 cold, the clearance mechanism 44 has been manually adjusted in accordance with known characteristics of said engine so as to insure that a gap 64 will obtain between the shank 60 and the piston member 52 when in operation, heat generated by the engine results in the expansion of these elements. It is further to be observed that the mechanism 44 has been designated hereinbefore as "the predetermined clearance mechanism," because the pre-selected clearance gap 64 is variable only in consequence of the heat responsive expansion of the engine block and the parts which define the gap.

With the engine 10 cold, the spring 18 maintains the valve 12 closed and simultaneously biases the rocker arm in a counterclockwise direction, the compression spring 68 being inherently strong enough to preserve the selected gap clearance 64. One of the features which the mechanism 44 of this invention provides should however be noted in this connection. That is to say as an inspection of Figure 1 visibly indicates, the conventional spring 18 is obviously capable of overcoming the compressive resistance of the spring 63, so that positive seating of the valve 12 is automatically assured at all times despite the established clearance gap 64.

Assuming now that the engine 10 is energized so that the camshaft 34 rotates in a counterclockwise direction as viewed in Figure 1, the valve drive train 42 remains in the status thereof illustrated so long as the concentric portion 38 of cam 36 rides along the undersurface of lifter piston bottom wall 80. Simultaneously with the energization of the engine, the oil pump associated therewith initiates the flow of oil under pressure through the distribution pipe 22. Passing through discharge port 29, the oil unseats the ball of the check-valve assembly 30, and thus flows onwardly via passageways 26 and 25 into the chamber formed by the bore 23. However, as hereinbefore noted, although oil pressure is thus built up in said chamber because of the restricted size of the conduit formed by passageways 70, 71 and 72 relative to that of the passageway 25, the oil is not impounded but proceeds downwardly via said conduit, lubricating those elements of the valve drive train including the lifter mechanism 46, which are interposed between the rocker arm end 22 and the camshaft 34.

In the Figure 1 embodiment of the mechanism 44, built up oil pressure serves to aid and abet the compression spring 68 in the maintenance of the established or automatic variations in clearance gap 64. In the Figure 8 embodiment however, said spring is dispensed with, the pressure of oil serving as the sole clearance gap maintenance medium.

Now as the shaft 34 approaches the completion of a revolution, the non-concentric lobe portion 40 of cam 36 incrementally elevates the lifter mechanism piston member 78 whereby, in a manner which will now be explained, to unseat the valve 12 until when the apex of said lobe is perpendicularly disposed, the said valve momentarily occupies its fully opened disposition, and

instantly thereafter begins to approach its fully closed disposition for a repetition of its actuating cycle. Therefore, during each revolution of the camshaft, as the cam lobe 40 engages against the undersurface of the piston member 78, the latter is elevated or lifted axially. As a result, the ball elements 148 move upwardly also, thereby rotating the pawl carrier 114 to the left as viewed in Figure 5. In so doing, the pawls 140 effect the incremental leftward rotation of the ratchet member 112 by the interengagement of said pawls and the teeth 134. The ratchet member 112 in turn rotates the torque sleeve 108 because of the splined connections 122—135, and obviously also, the sleeve spring 110 since the lugs 126 of the latter extend into the slots 120 of the torque sleeve. Further, in consequence of the longitudinal split 124 in the torque-sleeve spring, the latter frictionally engages the internally threaded head member 96 so that this member 96 is also rotated to the left.

Therefore, from the immediately foregoing and earlier description, it should be apparent that upward axial movement of the piston member 78 automatically effects the simultaneous rotation as a unit of pawl carrier 114, ratchet member 112, torque sleeve 108, and torque-sleeve spring 110, and axial extension of head member 96. This entire unit rotates leftwardly as viewed in Figures 1 and 5, and therefore clockwise as viewed in Figure 3.

In consequence of this rotary movement, the adjustable head member 96 simultaneously also moves vertically upwardly by reason of its engagement with the threaded upper end 92 of the stem member 90. Bearing in mind that the oil pressure built up in chamber 23 above the piston member 52, augmented by the expansive force of compression spring 68, is sufficient to yieldably maintain the established clearance gap or any reduction thereof, the said upward movement of the member 96 elevates the push rod 48 thereby actuating the rocker arm 19 in a clockwise direction to open the valve 12, as should be manifest. Obviously, the effectiveness of the hydraulic dashpot arrangement of the mechanism 44 is insured by the ball check-valve assembly 30 while the engine is running.

Further elaborating on the operational behavior of the clearance mechanism aforesaid, it is desired to point out that the valve closure springs 18 when the engine is stopped with certain of the valves in unseated open condition, are effective on the spring 63 supplemented by the pressure charged hydraulic dashpot comprising the mechanism 44 tending to close the existent gap 64 aided by possible leakdown via push rod end seats 56 and 100. This condition is brought about when pressure on the oil in the engine pressure lubricating system ceases, thus enabling the biasing effect of the springs 18 to continuously act on the static fluid confined in the dashpot chamber and thereby cause a gradual discharge therefrom via passageways 70, 72 or 74 and seats aforesaid back to the engine sump. The aforesaid condition obtains where the ball-socket connections 54, 56 and 73, 100 at opposite end respectively of the push rod 48 fail to be fitted in fluid-tight sealing relationship. In the event, the clearance mechanism gap 64 becomes closed in the manner abovedescribed, upon restarting the engine, before maximum idling speed is reached, the pressure from the engine lubricating pump supplemented by the pretensioned spring 63 is sufficient to quickly restore the clearance gap 64 to its normal status as determined by the existent thermal condition of the engine. This novel restorative feature of the present invention is effected in the manner just described while the lifter assembly 46 is being reciprocated slowly. Such slow reciprocating velocities of the lifter unit impart a relative rotational movement to the piston member 78 with respect to the sleeve member 136 influenced by inertia differences of the members and the ball-raceway connections 148, 150 therebetween; but upon increase of the reciprocating velocities of said members

responsive to maximum idling or higher engine speeds, the piston member 78 tends to move in a straight line causing the connected sleeve member 136 to rotate relatively with respect thereto with consequent operation of the ratchet mechanism 134, 140 to adjust the valve drive gear lengthwise if backlash is existent therein, otherwise, due to resistance of the mechanism 44, such operation of the lifter unit would be ineffective to adjust despite its uninterrupted operation aforesaid.

From the foregoing, it will be appreciated that one of the primary purposes of the clearance gap 64 is to prevent interference with proper seating of the engine valves due to expansion and contraction. Without this gap, which resists lengthening of the valve gear train after backlash has been eliminated, the lifter mechanism 46 with a cold engine would remove all play in the valve actuating linkage causing frequent unseating of the valves, or at least interfere with the tight closing thereof when the engine becomes hot with consequent burning of the valves and inefficient operation of the engine. However, with the gap, whenever expansion of the engine and valve drive parts occur due to heat tending to unseat the valves, the gap modulates to compensate for such expansion but at all times is effective to prevent the one-way backlash adjusting operation of the valve lifter assembly 46 from modifying said gap in excess of its predetermined status or any variations within its limits. Thus, the lifter mechanism 46 is operatively prevented from changing the status of the gap 64 responsive to the compression spring 68 and/or fluid-pressure charged dashpot 64 comprising the clearance mechanism 44, whereby all play in excess of the gap is taken up by operation of the lifter unit 46, whereupon said unit is rendered ineffective to adjust despite its continued operation responsive to rotation of the engine driven camshaft.

Therefore, the clearance mechanism 44 and lifter unit 46 are operatively inter-dependent to cooperatively produce and maintain substantially zero-lash adjustment between the engine-driven cam and valves for improved engine operating efficiency and quietness with minimum wear on the parts. In addition, the clearance mechanism 46 possesses the distinctive novel feature of providing a predetermined cold engine gap which is capable of modulating within its limits only, such modulation occurring according to thermal changes in the engine with the extent of gap change being determined solely by such thermal conditions, and in no case is the predetermined gap fully taken up with each valve operating cycle except where thermal conditions of the engine induce, nor is a wider gap condition required to render operation of the lifter unit 46 effective to take up existent backlash.

It is important to note here that, as pointed out above, the cold-engine clearance gap 64, or any modulation therein responsive to thermal changes in the engine, is never fully closed during valve opening operations where the hydraulic dashpot 44 is utilized since slight emission of oil through passageway 72, or 74 enables only slight contraction of the gap against the oil-pressure charged chamber 23. However, should the thermal change in the engine be of such magnitude that the operating gap 64 is substantially reduced to almost closed status, then opening of the engine valve 14 causes the end wall of the piston element 52 to engage the confronting face of the manually-adjustable pin 60, thus a solid one-way connection is formed for transmitting the thrust action of the engine cam through the linkage to open the selected valve 14. Upon the valve becoming fully seated in closed condition the oil pressure combined with the force exerted by the compression spring 68 quickly restores the operating gap 64 according to the thermal operating condition of the engine in readiness for another valve opening cycle. In the case where the compression spring 68 alone is employed to reset the clearance gap after completion of each valve opening cycle, since this spring is of less strength than the valve closure spring 18, it fol-

lows that with inauguration of a valve opening cycle it is additionally compressed to the extent of the existent gap 64 so that the end wall of the piston 52 engages the confronting end of the stem 60 to form such solid one-way connection through which the action of the engine cam is transmitted to the push rod 48 and rocker arm 19 to open the valve 14 as is understood. Upon closure of the engine valve, spring 68 instantly restores the clearance gap 64 according to the thermal condition of the engine structure in readiness for another valve opening cycle. It is thus seen that the novel relationship between the one-way adjusting tappet 46 and the two-way adjusting gap means 44 provides for one-way adjustments to remove wear-lash only from the valve drive gear while the clearance gap accommodates two-way adjustments within a predetermined maximum obtaining with cold-engine, to compensate for expansion and contraction resulting from engine operation. After wear has been taken up by the tappet mechanism 46 its operation to further adjust in a lengthening direction is prohibited by the hydraulic and/or spring forces within the mechanism 44.

From the foregoing statement of the mode of operation of the present tappet mechanism, it should be manifest that the tappet 46 and clearance mechanism 44 are operably inter-dependent to bring about the novel advantages described. If the tappet 46 were eliminated compensation for thermal changes only would, in such case, be provided by the clearance mechanism 44, thus in time, a noisy valve operation would result from the wear factor alone. On the other hand, were the gap mechanism 44 eliminated the tappet 46 would over-adjust, since at cold-engine the normal backlash set in the valve drive linkage would be taken up and upon expansion due to engine operation, the engine valves would not effectively close resulting in reduced engine efficiency and possible power failure. The tappet 46 operation is in a lengthening direction only making the clearance mechanism 44 requisite to obtain substantially zero-lash in the valve drive gear with the present compensating mechanism.

Installation of the lifter unit of the present invention should be made during cold engine, with the adjustable head member 96 threaded down on its complemental threaded section on the stem member 90 to provide, for example, .025 to .030 inch lost-motion in the valve drive parts with the engine valves freely seated; whereupon, starting the engine will enable the dashpot clearance mechanism 44 to become fully charged with oil from the engine pressure lubricating system prior to the backlash aforesaid being eliminated by operation of the lifter unit 46 responsive to separation of the head member 96 relatively to the stem member into operative contact with the push rod 48 and rocker arm 19 to effectively and quietly open and close the selected engine valve as is understood.

Reverting now to that portion of the operational description wherein the incremental elevation of the piston member 78 was treated, it is deemed advisable to elaborate somewhat on the axial-rotational cooperation of the balls 148 and their raceways 150 in actuating the pawl carrier 114. Thus, as the piston 78 ascends with the balls 148 confined in the openings 146 between said raceways 150 and the cylinder wall 76, the pawl carrier 114 is caused to rotate clockwise in consequence of impact thrust upwardly of said piston, and to rotate counterclockwise when the force of the spring 116 is effective to lower said piston 78 relatively to the pawl carrier 114 upon the bottom wall 80 engaging the closing ramp and base circle of the cam 36. The piston 78 moves bodily axially, but does not rotate with respect to the pawl carrier, because of the frictional contact obtaining between the cylindrical piston wall 82 and the wall of engine cylinder 76. Therefore, a sudden thrust applied to the piston 78 will cause the balls and associated raceways to rotate the pawl carrier alternately in one, then in the other direction as the thrust is applied and released.

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The camshaft 34 revolves continuously as is understood, so that subsequent to the complete opening of the engine valve 12, cam lobe 40 passes from beneath the central region of the under-surface of piston 78 which then returns to the position thereof illustrated in Figure 1. In so doing, the pawl carrier 114 is caused to rotate counterclockwise as a result of the previously explained coaction of the balls 148 and their associated raceways 150. Consequently the adjustable head member 96 is lowered as the compression spring 18 effects the counterclockwise rotation of the rocker arm and returns the valve 12 to closed position.

It should be observed that during valve closing operations as well as during valve opening operations, the clearance gap 64 remains adjusted in accordance with thermal conditions of the engine, as previously emphasized. In other words, considering the Figure 1 embodiment of the mechanism 44, the body of oil under pressure in chamber 23 in conjunction with the spring 63 functions to maintain substantially the established clearance gap which is subject to slight variations only within said gap and solely in consequence of expansion and contraction of the parts responsive to and as a result of temperature changes in the engine. Considering the Figure 8 embodiment of said mechanism, the body of oil under pressure in chamber 23 performs the said function per se, that is without the aid of a spring such as the spring 68 aforesaid.

From the foregoing it should be apparent that this invention provides novel means incorporated in an otherwise conventional valve drive train for alternately opening and closing the valve 12 in a quiet and efficient manner.

Assuming now that a backlash condition should develop at any point in the valve train linkage except at the aforesaid clearance gap, then the novel valve lifter mechanism 46 will automatically effect a compensating adjustment to immediately eliminate the backlash, and restore the linkage to substantially zero-clearance condition.

To this end, it is noted that the length of the angularly disposed raceways 150 is greater than necessary for normal lifter operations when backlash is absent in the linkage. It should be apparent therefore, that a backlash or lost-motion condition in the linkage will be automatically eliminated so to speak, immediately subsequent to its inception, because the instantaneously obtaining diminished resistance to the upward movement of the member 96 accommodates the requisite additional elevation of the latter to immediately restore substantially zero-clearance.

Further discussing the operation of the clearance mechanism 44, and particularly with respect to the functional behavior of the spring 63 and associated dashpot in relation to the split tubular torque-transmitting spring 110, it is desired to stress that the pre-energized tension of the spring 63 is sufficiently strong to prohibit the spring 110 from adjustably turning the head 96 notwithstanding operations of the lifter unit 46 are effective to rotate the torque-sleeve 108 intermittently. As the torque-sleeve rotates step-by-step with each engine valve opening operation of the lifter unit 46, the head 96 is urged in a corresponding direction of rotation by the clamping action of the spring 110 on the outer cylindrical surface of the said head. Under such conditions, however, the torque-sleeve 108 can only rotate intermittently relatively to the head 96 as accommodated by the frictional engagement between the spring 110 and said head with backlash conditions in the valve drive parts substantially at zero since the head 96 can adjustably rotate only when backlash is present. Moreover, the hydraulic dashpot when charged with fluid under pressure from the engine pressure lubricating system, likewise prevents the torque-sleeve 108 from adjustably turning the head 96 in response to valve lifter operations following backlash elimination thereby.

Thus it may be said that the forces of the pre-tensioned spring 63 and the pressure charged hydraulic dashpot acting separately or together offer sufficient axial thrust re-

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sistance on the head 96 to prevent rotation thereof by the tappet 46 while backlash conditions in the valve drive train are substantially at zero with the clearance mechanism 44 at any adjusted positions within the predetermined clearance gap 64 to compensate for engine expansion and contraction as the case may be. Stated differently, operations of the lifter unit 46 are incapable of overruling the adjusted condition of the clearance mechanism 44, said lifter unit being operable to maintain substantially zero-clearance between the valve drive parts without affecting the settings within the clearance gap 64 which latter are automatically effected once the correct predetermined clearance gap is manually established in the manner already described.

Reference is again made to the structure and operation of the novel ratchet mechanism comprising the flexible pawls 140 and cooperating teeth 134 carried by the annular flanges 138 and 132, respectively. These pawls are constructed of rectangular strips of springy material and are secured at one end as by rivets to the upper end face of member 114 in circumferentially equally spaced relation, while their free flexible ends are angularly disposed upwardly for coaction with the aforesaid confronting ratchet teeth 134 which latter are closely generated radially in the underface of the annular flange 132. The oscillatory action of the member 114 previously referred to causes the pawls 140 to progressively flex into and out of engagement with the teeth 134 to thus produce a ratcheting effect therebetween on said driven member 112 with consequent intermittent rotation of said driven member in one direction only. Thus minute initial portions of the axial reciprocations of the lifter unit 46 in effecting a cyclical opening and closing of a selected engine valve 12 are converted into said oscillatory movements of the pawl carrier member 114 by the ball and raceway connections 148, 150 operably incorporated between said pawl member and the drive member 78. During engine valve actuations, the ball and raceway connections 148, 150 are rendered inoperative to produce the above described rotary motion but with the engine valve closed, said connections are capable of axially adjusting to effect relative oscillation of member 114 thereby correspondingly rotating the member 112 in the one direction which movement is transmitted through the torque-sleeve 108 to frictionally adjust the head 96 as required to eliminate substantially such backlash as may be present in the valve drive linkage, otherwise the sleeve 108 will rotate relatively to the head without imparting any movement thereto.

In view of the drawings and reference to the preceding description, it is believed prolix to further elaborate on the disclosed means for lubricating the valve drive train. Briefly however, it is noted that in the Figure 1 disclosure, the oil flow is direct from source 20 through the linkage and ultimately via opening 102 to the lifter mechanism 46, thence to the engine sump via drainage ports 84.

In the modified Figure 8 disclosure the oil flow is controlled by the previously referred to valve arrangement designated by the numeral 75. Said valve arrangement so to speak, hermetically seals or impounds substantially the body of oil within chamber 23, except, however, for losses due to pressure leakby at the lower end of the passage 70 which forms the valve 75 to close off said passage by engagement with the central surface portion of the concavity in the upper end of the push rod 43. This valve is closed on the upstroke of said push rod in opening an engine valve, thus compensating for the absence of the spring 68 included in the Figure 1 disclosure. On each downstroke however, the valvular hermetic seal is relaxed sufficiently to enable a quantity of oil to escape via the passage 70 and diagonal passageways 74, and to flow downwardly on the outer surface of the push rods onto the lifter units 46, thence to the engine sump for recirculation via drainage ports 84.

If the engine equipped with this modified structure is stopped, the fluid pressure charge in the chamber formed

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between the bore 23 and piston 52, created during operation of the engine will not maintain the clearance status of the mechanism 44 due to loss of pressure via passageways 70, 72 or 74 and ball-socket connections 54, 56 and 73, 100 influenced by the action of spring 18 when a valve is open but upon restarting the engine the oil pressure from the engine lubricating system would be effective to instantly restore the gap of the clearance mechanism 44 to its status as determined by the existent thermal condition of the engine. This re-establishing of the clearance gap precedes operation of the lifter assembly 46 to adjust in one direction to lengthen the valve drive gear which latter, if effective coincidentally, would prevent restoration of the clearance gap as aforesaid, notwithstanding the engine may have reached substantially maximum idle speed at which the lifter unit is reciprocated at what may be termed "slow velocities," such action of the lifter unit causing the piston member 78 to rotate relatively with respect to the ball and race connected sleeve member 136 due to differences in the inertia status of these two members and as a consequence the ratchet mechanism shown in Figures 2 and 3 cannot operate to modify the length of the valve drive gear while the gap status aforesaid of the clearance mechanism is temporarily closed. However, upon speeding up the engine with consequent increase in the velocity of the relatively movable parts of the lifter assembly, the piston member 78 which carries the balls 148 tends to move in a straight line with respect to the connected sleeve member 136 having the angular raceways 150 on its outer surface. This action of the piston member 78 is converted automatically by the ball and raceway connections into rotary motion of the sleeve member 136 in a counterclockwise direction as viewed from the bottom of Figure 5 to thus operate the pawls 140 into engagement with the shoulders of the ratchet teeth 134 causing a lengthening adjustment in the lifter assembly to be effected should a backlash condition be existent. From the foregoing, it is seen, that upon starting the engine the operational sequence of the clearance mechanism 44 and lifter assembly 46 to adjust is such that the clearance mechanism is charged with pressure fluid to restore the gap prior to any lengthening operation of the lifter unit as may be required to take-up play in excess of the existent thermal status of the engine, and that such operation of the lifter unit to lengthen should a backlash condition exist, should be effective immediately prior to the engine reaching maximum idling speed. Considering the terminology used in the foregoing description and in the appended claims, the identifying expressions and/or terms employed are intended to convey meanings which include the range of reasonable equivalents in the patent sense. For example, the expressions "wear," "backlash," and "lost-motion" are intended to convey similar meanings with respect to exclusive idle travel within the valve drive gear, while such expressions as "operating clearance," "clearance gap," "relative movement," etc. relate to the operational characteristic of the lifter mechanism as is understood. The terms "upper," "vertically," "top," "bottom," "lower," "upwardly," and other directional words or characters are intended to have only relative connotation for convenience in describing the structure as illustrated in the drawings, and are not intended to be interpreted as requiring any special orientation with regard to associated structure external to the present disclosure.

Further considering the operational characteristics of the ball and raceway connections 148, 150 which operatively interconnect the piston member 78 and the pawl carrier 114, in relation to the reciprocable movements imparted to the piston 78 by the engine cam 36 and the compression spring 116, it is desired to emphasize that when the spring 116 is employed to actuate the head 96 through its backlash adjusting phase, then reversal of the operating connections for the ratchet mechanism would be required; such as, disposition of the raceways 150 in

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the opposite direction in relation to their engaging balls 148, opposite positioning of the pawls 140 and cooperating teeth 134 and the threaded connection of the head member 96. Thus the present invention provides utilization of either the upward cam thrust of the piston 78 to adjust for backlash elimination and subsequently open the engine valve; or the final portion of the lowering movement of the member 78 under influence of the compression spring 116 to effect backlash adjustment upon the engine valve becoming closed in readiness for another cyclical opening and closing thereof. Moreover, the presence of limited controlled-backlash in the valve drive train is desirable for efficient engine operation free of valve malfunctions which novel advantage is provided by utilizing only the initial portions of lifter unit reciprocations to define the degree of backlash present in the valve drive parts essential to enable operation of the backlash adjusting mechanism 140, 134 whereby the valve drive parts outside of those comprising the clearance mechanism 44, are operatively maintained in their relative zero-clearance relation substantially.

From the preceding description augmented by an inspection of the drawings, it is believed that a comprehensive understanding of the instant invention may be had. It is to be understood however, that my invention shall not be limited to the precise details of structure illustrated and described. That is to say, my invention contemplates any and all modifications, substitutions of parts and equivalents in structure which may fall within the scope of the claims hereunto appended.

I claim as my invention:

1. Mechanism interposed between the camshaft and the valve included in the valve drive train of an internal-combustion engine, comprising: a valve lifter unit having a frictional element mechanically operable in response to the rotations of said camshaft to automatically adjust in a lengthening direction only to eliminate backlash only in said valve drive train; adjustable clearance mechanism including a yieldable force-transmitting connection operably controlling the operation of the lifter unit to adjust as aforesaid and for establishing a predetermined clearance gap in the valve drive train, whereby said mechanism is operable automatically to vary said clearance within the limits of said gap according to contraction and expansion due to thermal conditions resulting from engine operations; and manually-adjustable means incorporated in said clearance mechanism for selectively establishing said predetermined clearance gap therein.

2. In an internal-combustion engine including a camshaft, a valve, and an actuatable drive train for controlling the action of said valve: a lifter mechanism having a frictional element mechanically operable by a cam on said camshaft to actuate said drive train for intermittently opening said valve; self-adjustable means in the lifter mechanism for eliminating a backlash condition only should one exist or develop in the drive train; clearance establishing mechanism operably interposed between the cam and valve included in said drive train to provide a gap normally predetermined relative movement therebetween; manually-adjustable means in the last-named mechanism for establishing said predetermined relative movement therein; and a preloaded compression spring combining with oil under pressure from the pressure lubricating system for automatically opposing decrease in said relative movement during engine operation.

3. In a reciprocable push rod and a valve drive train of the character described including a rocker arm pivotally mounted on an oil distribution pipe of an internal-combustion engine, clearance establishing mechanism interposed in said drive train between one end of the push rod and the adjacent end of said rocker arm, said mechanism including: a cup-shaped hollow piston member slidably positioned in a bore formed in the designated end of the rocker arm and retained therein by means of a split retainer ring; an integral shank depending from the piston

and terminating in a substantially spherical portion seated in a semi-circular socket provided on said one end of the push rod; a manually-adjustable screw member engaging a reduced counterbore in the rocker arm and having an integral depending shank extending into the cup-shaped piston to establish a selected gap between said shank and the inner surface of the end wall of the piston; a lock nut for maintaining the screw member in the selected position of adjustment; means for introducing oil under pressure from said oil distribution pipe into the chamber formed by said bore above the piston aforesaid to yieldably oppose closure of said selected gap during opening of the corresponding engine valve; and one-way adjustable mechanism actuated by said engine, and operably controllable to adjust by said clearance mechanism, to compensate for any backlash in the valve drive train external to the selected gap aforesaid.

4. In a reciprocable push rod and a valve drive train of the character described including a rocker arm pivotally mounted on an oil distribution pipe of an internal-combustion engine, clearance establishing mechanism interposed in said drive train between one end of the push rod and the adjacent end of said rocker arm, said mechanism including: a cup-shaped hollow piston member slidably positioned in a bore formed in the designated end of the rocker arm and retained therein by means of a split retainer ring; an integral shank depending from the piston and terminating in a substantially spherical portion seated in a semi-circular socket provided on said one end of the push rod; a manually-adjustable screw member engaging a reduced counterbore in the rocker arm and having an integral depending shank extending into the cup-shaped piston to establish a selected gap between said shank and the inner surface of the end wall of the piston; a lock nut for maintaining the screw member in the selected position of adjustment; a preloaded compression spring interposed about said depending shank between the end wall of said piston and the wall surrounding said reduced counterbore in the rocker arm to yieldably oppose closure of said selected gap; means for introducing oil under pressure from said oil distribution pipe into the chamber formed by said bore above the piston aforesaid to supplement the action of said compression spring; and one-way adjustable mechanism actuated by said engine, and operably controllable to adjust by said clearance mechanism, to compensate for any backlash in the valve drive train external to the selected gap aforesaid.

5. The valve drive train set forth in claim 3 wherein the means for introducing oil under pressure from the recited distribution pipe includes a plurality of successively enlarged diagonal passageways formed in the rocker arm, a discharge port formed in said pipe and disposed in alignment with said passageways, and a check-valve assembly interposed in said passageways between the discharge port and the passageway of smallest diameter, the latter having direct fluid communication with the recited pressure chamber in a non-interfering plane with the movement of the cup-shaped piston.

6. The valve drive train set forth in claim 4 wherein the means for introducing oil under pressure from the recited distribution pipe includes a plurality of successively enlarged diagonal passageways formed in the rocker arm, a discharge port formed in said pipe and disposed in alignment with said passageways, and a check-valve assembly interposed in said passageways between the discharge port and the passageway of smallest diameter, the latter having direct fluid communication with the recited pressure chamber in a non-interfering plane with the movement of the cup-shaped piston.

7. A valve lifter mechanism including: a cup-shaped piston axially reciprocable in a cylinder of an internal-combustion engine in response to the rotation of the engine camshaft, said piston having an end wall and a cylindrical side wall; a socket formed centrally in the

inner side of said end wall; a stem member fixed at one end in said socket with a section of its other end provided with threads; an adjustable head member having an internally threaded cylindrical side wall in threaded engagement with said threaded section of the stem member; means rotatable in response to the axial reciprocations of said piston for rotating said head member in one direction only to effect simultaneous unthreading movements thereof; means for preventing axial displacement of said last-mentioned means with respect to said cylinder; and preloaded spring means for cooperating with said piston to rotate said rotatable means and stabilize the latter in its adjusted positions.

8. A valve lifter mechanism including: a cup-shaped piston axially reciprocable in a cylinder of an internal-combustion engine in response to the rotation of the engine camshaft, said piston having an end wall and a cylindrical side wall; a socket formed centrally in the inner side of the end wall; a stem member fixed at one end in said socket with a section of its other end provided with threads; an axially extendible head member having an internally threaded cylindrical side wall in threaded engagement with said threaded section of the stem member; an external annular flange adjacent one end of said head member; an external annular groove provided with a split retainer ring on the head member spaced from said annular flange for retaining the head member and torque sleeve in assembled relationship therebetween a torque sleeve disposed about the head member; a longitudinally split torque sleeve spring mounted in said torque sleeve for endwise movement therewith in frictionally engaging relation with said head member to extend the same; a rotatable cup-shaped ratchet toothed member having a splined connection with said torque sleeve; a tubular pawl carrier including a movable pawl operatively engaging the teeth of the ratchet member aforesaid; means responsive to the axial reciprocations of said piston for effecting incremental rotations of the pawl carrier; an internal annular groove in the wall of said cylinder adapted to receive a split retainer ring for presenting axial displacement of the ratchet member and pawl carrier with respect to said cylinder; and a preloaded compression spring interposed about a portion of said stem member between the end walls respectively of the piston and the pawl carrier for stabilizing the latter during rotary movements thereof against the last mentioned retainer ring.

9. The valve lifter mechanism of claim 8 wherein the recited means for effecting incremental rotations of the pawl carrier comprises a plurality of circumferentially equally spaced angularly disposed arcuate raceways formed in the tubular body portion of the pawl carrier, an identical plurality of circumferentially spaced circular openings formed in the cylindrical side wall of the piston, and a spherical element disposed in each of said circular openings and maintained in engagement with its raceway by the wall of the cylinder wherein said piston is reciprocable.

10. The valve lifter mechanism of claim 8 wherein the recited adjustable head member terminates at its outer end into a horizontally disposed concave wall to provide an arcuate seat for the rounded extremity of a valve drive train push rod; and wherein said wall has formed centrally therein an opening for the admission of lubricant to the recited threaded section of the stem member.

11. In a valve lifter mechanism interposed between the lower extremity of a valve train push rod and the camshaft of an internal-combustion engine: a rotatable and simultaneously vertically movable head member having a threaded bore closed at its upper end which engages said lower extremity of the push rod; an external annular flange adjacent the lower end of said head member; a cup-shaped ratchet member terminating at its upper end in a laterally projecting annular flange the under surface of which has formed therein a circular series of radially disposed ratchet teeth; an external annular groove in the side wall of said head member spaced from its upper end

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aforesaid, adapted to receive a split retainer ring; a torque sleeve having a split sleeve spring encircling said head member between the annular flange and split retainer ring aforesaid and operably interposed between said ratchet and head members; means for locking said torque sleeve and ratchet member for conjoint rotational movement and for enabling relative axial movement thereof; a tubular pawl carrier terminating at its upper end in a laterally projecting annular flange the diameter of which corresponds to that of the ratchet member flange; a plurality of flexible pawl elements attached at one end to the upper surface of said carrier flange and disposed in operative engagement with the teeth of said ratchet member; means for insuring the cooperative disposition of said ratchet teeth and pawl elements; a cup-shaped piston reciprocable by said camshaft in a bore of said engine, said piston including a bottom wall and a cylindrical side wall, the upper portion of the latter surrounding the lower portion of the tubular pawl carrier; a socket formed centrally in said piston bottom wall; a stem member fixed at its lower end in said socket and which extends upwardly through the bottom wall of the pawl carrier and terminates at its upper end in a square-threaded segment in threaded engagement with said threaded bore of said adjustable head member; a plurality of circumferentially equally spaced angularly disposed arcuate raceways formed in said lower portion of the pawl carrier; a plurality of similarly spaced ball elements each surrounded by a circular opening provided therefor in the piston side wall and maintained in operative engagement with said raceways by the wall of the cylinder wherein the piston is reciprocally disposed; and preloaded spring means for stabilizing the ratchet member and pawl carrier in relative operated disposition.

12. In combination with the valve drive train of internal-combustion engine having a push rod, a valve lifter mechanism including: a camshaft reciprocated cup-shaped piston; a cup-shaped ratchet member; a tubular pawl carrier having a movable pawl element cooperatively disposed relative to said ratchet member; a rotatable and simultaneously movable internally threaded head member operably engaging one end of the push rod of said drive train; a torque sleeve having a splined connection with said ratchet member, to provide relative axial movement therebetween; a longitudinally split torque sleeve spring in frictional engagement with the head member and having at each end thereof a pair of diametrically opposed projections extending into complementary slots provided therefor in said torque sleeve; a stem member projecting from the inside of the piston end wall with a section of its opposite end externally threaded for reception of the threaded head member; an external annular groove formed on the head member in longitudinally spaced relation to an external annular flange formed thereon, for the reception of a split retainer ring for maintaining the torque sleeve and spring in assembled relationship on the head member; means effective for maintaining the piston, ratchet member and pawl carrier aforesaid in cooperative disposition; and a plurality of ball elements carried by the piston, each operably engaging complementary angularly disposed arcuate raceways, whereby oscillatory movement is imparted to the pawl carrier during each reciprocatory stroke of said piston in response to the rotation of said camshaft.

13. A valve actuating device for an internal-combustion engine comprising relatively extensible members mechanically connected for converting limited axial movement of one member into predetermined rotary movement of the other member, mechanism including a frictional element operably associated with one of said members to unidirectionally followup and maintain the extending movements of said members in response to the axial-rotational movements of said members aforesaid, and hydraulic pressure means operably associated with one of said members normally establishing a predetermined cold engine gap

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therebetween and operable to modulate said gap within the limits of its predetermined status according to thermal expansion and contraction of the engine and simultaneously prohibit the extending movements aforesaid without interrupting operation of said mechanism relatively to the extensible members.

14. A valve actuating device according to claim 13 plus manually-adjustable means for establishing said predetermined gap.

15. A valve actuating device according to claim 14 in which the action of said hydraulic pressure means is supplemented by a pre-energized spring.

16. A valve actuating device according to claim 14 including in combination an engine valve and a valve-actuating cam, said device being operably disposed between said valve and cam.

17. A valve actuating device according to claim 15 including in combination an engine valve and a valve-actuating cam, said device being operably disposed between said valve and cam.

18. A valve actuating device according to claim 13 in which the mechanical connection between the extensible members comprise a plurality of circumferentially spaced elements carried by one of said members engaging complementary angularly disposed raceways carried by the other member.

19. The combination according to claim 12, including oil passageway means incorporated in the valve drive train and lifter mechanism for restrictively conducting oil from the distribution pipe aforesaid of the engine pressure lubricating system, to lubricate said drive train and mechanism, and return oversupply thereto back to the engine sump.

20. The combination according to claim 19, including a fluid-pressure charged clearance mechanism operably incorporated in said passageway means, comprising: a predetermined contractible gap in said valve drive train; manually settable means for establishing said gap; and operative connections between said clearance mechanism and lifter mechanism for enabling said gap to control adjusting operations by the latter responsive to the restrictive flow aforesaid of oil through said passageway means.

21. In valve operating mechanism for internal-combustion engines, the combination with a valve and a valve-actuating cam of a cam-operated valve actuating device operably interposed between said cam and valve, comprising: a plurality of relatively movable members; means interconnecting certain of said members for inducing intermittent relative separation only of a pair of said members different from said certain members; friction clutch means operably disposed between said pair of members responsive to limited relative movement of said certain members, and normally effective to separate said pair of members to the extent of existent lost-motion between the cam and said valve; a fluid-pressure charged device operably disposed between the first-mentioned device and said valve, and operable to normally provide a predetermined cold-engine gap between said cam and valve, to accommodate modulation of said gap automatically within the limits thereof according to thermal changes in the engine, and to simultaneously prohibit the operation of the first-mentioned device to be effective to separate said pair of members in excess of said lost-motion; means effective to maintain cooperative relationship between all of said members; manually settable means for establishing the predetermined gap status in said fluid-pressure device; a source of pressure-fluid production; fluid passageway means for controlling fluid under pressure from said source to charge the fluid-pressure device; and check-valve means including a spring-loaded movable element for preventing pressure conditions within said fluid-pressure device from working against said source during valve opening cycles.

22. The invention as defined in claim 21 including modulatory preloaded compression spring means for sup-

plementing the action of said fluid-pressure device for the purpose.

23. In valve operating mechanism for internal-combustion engines, the combination with a valve and a valve-actuating cam of a cam-operated valve actuating device operably disposed between said cam and valve, comprising: a plurality of relatively movable members; means interconnecting certain of said members for inducing intermittent relative separation only of a pair of said members different from said certain members; friction clutch means operably disposed between said pair of members responsive to limited relative movement of said certain members, and normally effective to separate said pair of members to the extent of existent lost-motion between the cam and said valve; a spring-loaded device operably disposed between the first-mentioned device and said valve for prohibiting the operation of the first-mentioned device to be effective to separate said pair of members in excess of said lost-motion; a predetermined cold-engine operating gap defined by two spaced elements operably associated with said spring-loaded device, said gap being adapted to modulate automatically within the limits thereof according to thermal changes in the engine; means effective for maintaining co-operative relationship between said movable members; and manually settable means for adjusting one of said spaced elements to establish the predetermined gap status.

24. In compensating tappet mechanism for the valves of an internal-combustion engine including a cam-actuated tappet body having a longitudinal cylindrical bore closed at one end and opened at the other: a pair of interfitting cylindrical members rotatable relatively and in unison in said bore with each member having an outturned annular end flange in overlying relationship adjacent the open end of said bore; means for operationally stabilizing said members with respect to each other and said bore; one-way rotational drive mechanism operably disposed between said interfitting members; means for imparting oscillatory movement to one of said members responsive to reciprocating movement of said tappet body; a threaded stem centrally disposed within said bore and fixed at one end to the end wall thereof; a cylindrical element having a threaded longitudinal bore closed at one end, for engaging the threaded portion of the stem aforesaid; a friction clutch having a split radially contractible sleeve connected to a carrier sleeve for unison movement therewith, the carrier sleeve being slidably splined to the other interfitting member with the split sleeve disposed in clamping engagement about the cylindrical element whereby one-way rotation of the split sleeve is transmitted to the cylindrical element for elevating the latter relatively to said stem to eliminate backlash only from the valve drive train; clearance mechanism having a predetermined cold-engine operating gap and a co-operating hydraulic pressure charged chamber supplemented by a preloaded compression spring; a pair of elements operatively associated with said last-mentioned mechanism and adapted to be biased in opposite directions by said hydraulic and spring forces into engagement with the cylindrical element and engine valve respectively, when backlash is eliminated, to establish said predetermined gap and accommodate two-way adjustments in the valve drive train as defined by the limits of said gap, to compensate for thermal changes only, the biasing force on said pair of elements being of such magnitude as to resist change within said gap by operation of the one-way rotational mechanism after backlash has been eliminated thereby; and manually-adjustable means operably associated with one of said pair of elements for establishing said cold-engine gap.

25. In compensating tappet mechanism for the valves of an internal-combustion engine including a cam-actuated tappet body having a longitudinal cylindrical bore closed at one end and open at the other: a pair of interfitting cylindrical members rotatable relatively and in unison in said bore with each member having an outturned annular

end flange in overlying relationship adjacent the open end of said bore; means for operationally stabilizing said members with respect to each other and said bore; one-way rotational drive mechanism operably disposed between said interfitting members; means for imparting oscillatory movement to one of said members responsive to reciprocating movement of said tappet body; a threaded stem centrally disposed within said bore and fixed at one end to the end wall thereof; a cylindrical element having a threaded longitudinal bore closed at one end, for engaging the threaded portion of the stem aforesaid; a friction clutch having a split radially contractible sleeve connected to a carrier sleeve for unison movement therewith, the carrier sleeve being slidably splined to the other interfitting member with the split sleeve disposed in clamping engagement about the cylindrical element whereby one-way rotation of the split sleeve is transmitted to the cylindrical element for elevating the latter relatively to said stem to eliminate backlash only from the valve drive train; clearance mechanism having a predetermined cold-engine operating gap and a co-operating hydraulic pressure charged chamber; a pair of elements operatively associated with said last-mentioned mechanism and adapted to be biased in opposite directions by said hydraulic force into engagement with the cylindrical element and engine valve respectively, when backlash is eliminated, to establish said predetermined gap and accommodate two-way adjustments in the valve drive train as defined by the limits of said gap, to compensate for thermal changes only, the biasing force on said pair of elements being of such magnitude as to resist change within said gap by operation of the one-way rotational mechanism after backlash has been eliminated thereby; and manually-adjustable means operably associated with one of said pair of elements for establishing said cold-engine gap.

26. In compensating tappet mechanism for the valves of an internal-combustion engine including a cam-actuated tappet body having a longitudinal cylindrical bore closed at one end and open at the other: a pair of interfitting cylindrical members rotatable relatively and together in said bore with each member having an outturned annular end flange in overlying relationship adjacent the open end of said bore; means for operationally stabilizing said members with respect to each other and said bore; one-way rotational drive mechanism operably disposed between said interfitting members; means for imparting oscillatory movement to one of said members responsive to reciprocating movement of said tappet body; a threaded stem centrally disposed within said bore and fixed at one end to the end wall thereof; a cylindrical element having a threaded longitudinal bore closed at one end, for engaging the threaded portion of the stem aforesaid; a friction clutch having a split sleeve radially contractible connected to a carrier sleeve for unison movement therewith, the carrier sleeve being slidably splined to the other interfitting member with the split sleeve disposed in clamping engagement about the cylindrical element whereby one-way rotation of the split sleeve is transmitted to the cylindrical element for elevating the latter relatively to said stem to eliminate backlash only from the valve drive train; clearance mechanism having a predetermined cold-engine operating gap and co-operating spring-pressure loaded chamber; a pair of elements operatively associated with said last-mentioned mechanism and adapted to be biased in opposite directions by said spring force into engagement with the cylindrical element and engine valve respectively, when backlash is eliminated, to establish said predetermined gap and accommodate two-way adjustments in the valve drive train as defined by the limits of said gap, to compensate for thermal changes only, the biasing force on said pair of elements being of such magnitude as to resist change within said gap by operation of the one-way rotational mechanism after backlash has been eliminated thereby; and manually-adjustable

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means operably associated with one of said pair of elements for establishing said cold-engine gap.

References Cited in the file of this patent

UNITED STATES PATENTS

1,193,913	Manning	Aug. 8, 1916
1,443,940	Anderson	Feb. 6, 1923
1,617,986	Blank	Feb. 15, 1927

5

1,663,345
1,673,478
2,036,936
2,160,257
2,283,536

538,952

22

Lievre	Mar. 20, 1928
Addyman	June 12, 1928
Halford	Apr. 7, 1936
Appel	May 30, 1939
Burkhardt	May 19, 1942

FOREIGN PATENTS

Great Britain	Aug. 22, 1941
---------------	---------------