

[54] MEANS FOR HYDRAULICALLY
CONTROLLING THE OPERATION OF
INTAKE AND EXHAUST VALVES OF
INTERNAL COMBUSTION ENGINES

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[58] Field of Search..... 123/90.11, 90.12, 90.13,
123/90.14, 90.15, 90.19

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[57] ABSTRACT

In an internal combustion engine wherein the tappet valves are operated by hydraulic means including a delivery pump, in order to ensure a better combustion of fuel during starting and warm-up operation, there is provided an engine temperature-dependent control means for decreasing the output of said delivery pump in the aforementioned two operational ranges to effect a shortening of the length of the tappet valve stroke.

2 Claims, 3 Drawing Figures

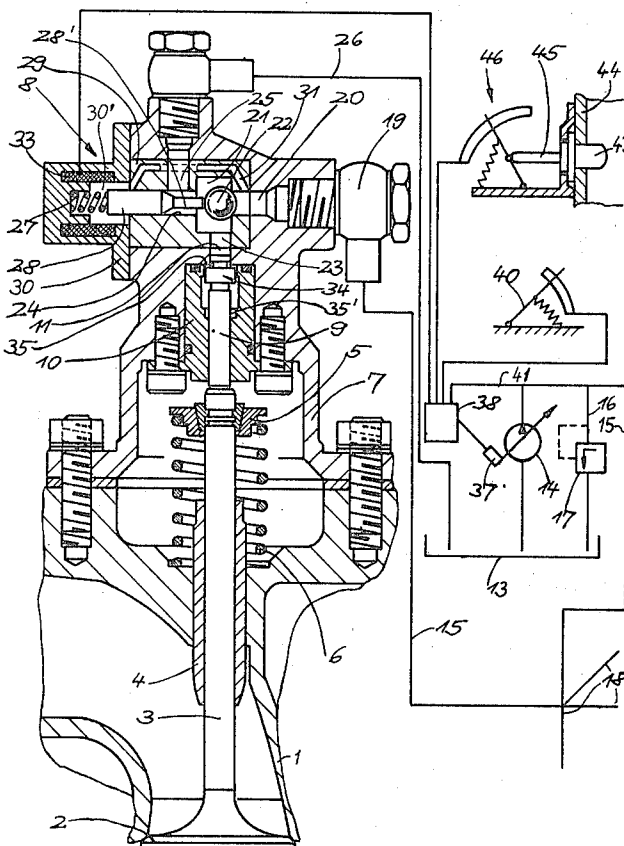


Fig. 1

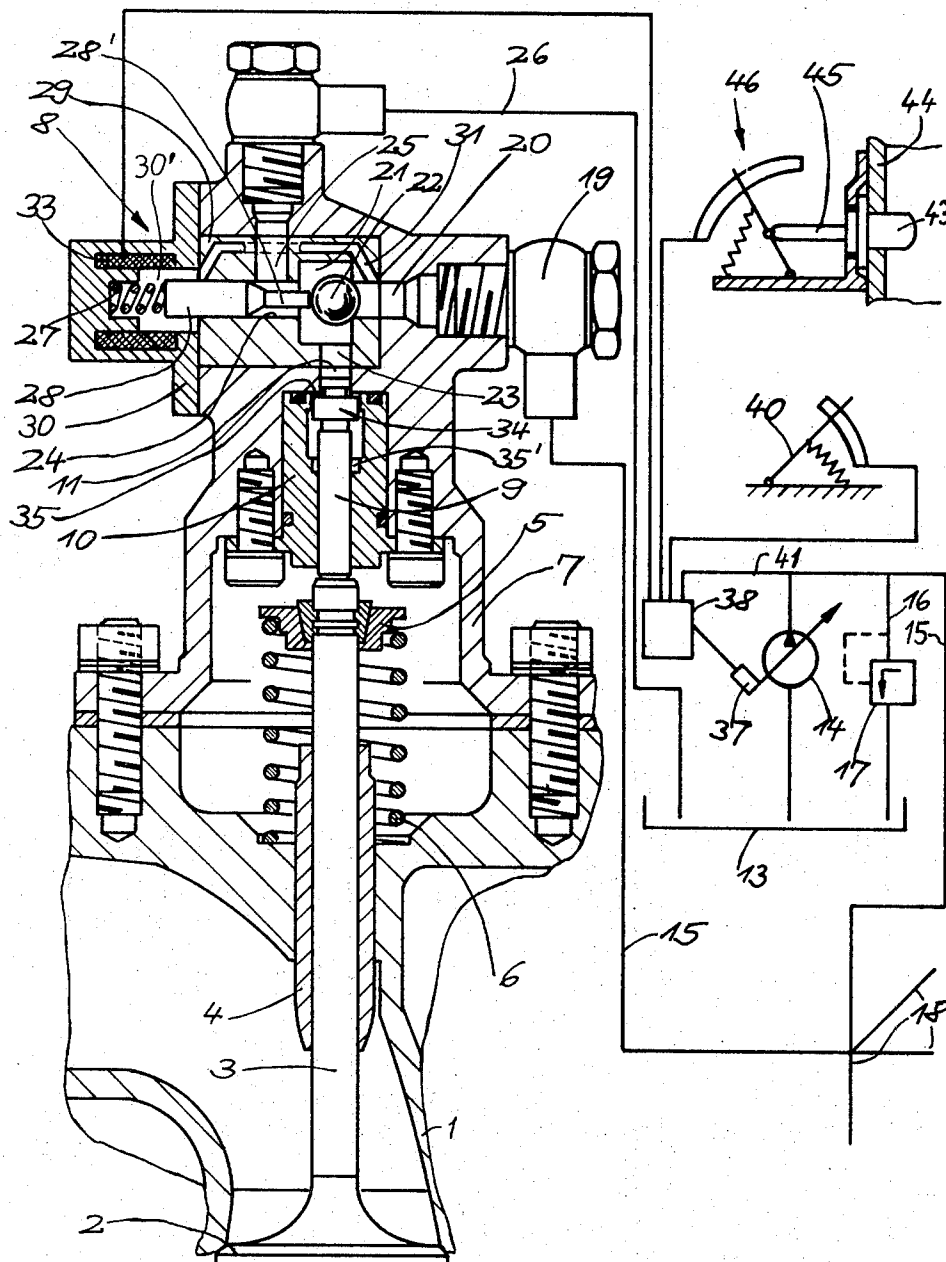


Fig. 2

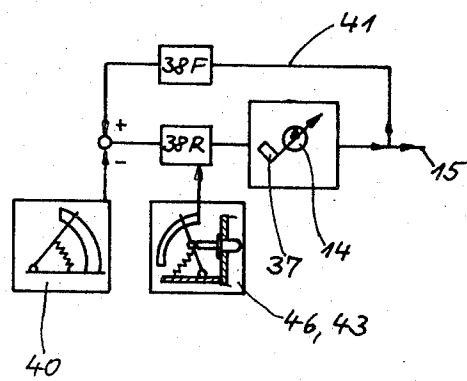
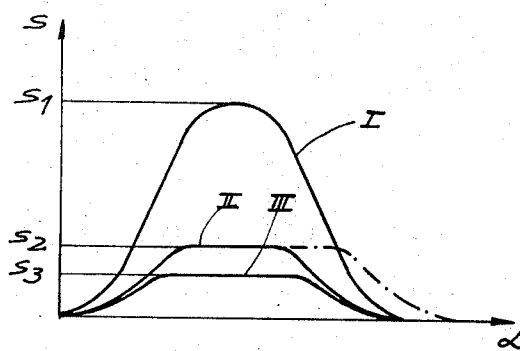


Fig. 3



MEANS FOR HYDRAULICALLY CONTROLLING THE OPERATION OF INTAKE AND EXHAUST VALVES OF INTERNAL COMBUSTION ENGINES

BACKGROUND OF THE INVENTION

This invention relates to a control of the intake and exhaust valves of internal combustion engines by means of a liquid which in the valve opening direction intermittently exerts a force on the frontal face of a work piston, one associated with each valve to cooperate at least indirectly with the valve stem against the force of a closing spring. The liquid is supplied in a substantially continuous manner by a delivery pump and the intermittent control is effected by means of a solenoid valve. For varying the length of the valve stroke, the effective work capacity of the liquid is variable as a function of the engine rpm or the engine load as described in German Pat. Application P 19 62 916.4

For the starting and the warm-up run of the engine it is known to supply a so-called excess fuel quantity, while the intake air is enriched with fuel far beyond the stoichiometrically necessary extent. This is required first, because in a cold engine, on the one hand, the engine friction is greater and, on the other hand, the intake air enters the engine cylinder with a relatively low speed, resulting in a deficient mixture formation and deficient preparation of fuel and, secondly, because upon warm starting of the engine the fuel is vaporized in the intake tube during the preceding period of standstill. As a result, in these operational ranges the exhaust gases contain unpermissibly high proportions of pollutants such as CO and CH.

OBJECT, SUMMARY AND ADVANTAGES OF THE INVENTION

It is an object of the invention to provide an improved hydraulic control means wherein even in the aforementioned operational conditions a good preparation of the fuel is achieved, that is, the imperfect fuel vaporization caused by the low air temperatures is compensated by a finer splitting of the fuel droplets while simultaneously providing a thorough mixing with air, and the fuel vapor deposited during hot engine operation is drawn faster and in a more turbulent manner into the engine cylinder.

Briefly stated, according to the invention, the effective work capacity of the hydraulic liquid is variable by means of a thermostatic control element as a function of the engine temperature, so that in case of a cold engine the amplitude of the opening stroke of particularly the intake valve is smaller than in the case of a warm and particularly a hot engine.

In the aforeoutlined manner it is ensured that during cold engine operation up to a predetermined temperature which is higher than the warm starting temperature, the intake speed of the air is high, resulting in a finer division of the fuel droplets. Further, by virtue of the high air velocity, there is achieved at the valve inlet, a turbulent air flow which ensures a very satisfactory air-fuel mixture. This turbulence is maintained in the engine cylinder even during the compression stroke, so that the formation of condensates and thus the thinning of the lubricating oil occurs in a reduced extent.

In view of the afore-outlined advantages, an excess fuel quantity for starting of the cold engine may be dispensed with. In order to obtain the necessary additional air quantities for the warm-up run, the period of open

condition of the intake valves is accordingly lengthened.

The invention will be better understood as well as further objects and advantages become more apparent from the ensuing detailed specification of a preferred, although exemplary embodiment taken in conjunction with the drawing.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a longitudinal sectional view of the invention including schematic representation of the pump and solenoid control means;

FIG. 2 is a block circuit diagram of an electronic regulator for the control of the intake and exhaust valves according to the invention and

FIG. 3 is a diagram showing different lifting curves of the tappet valves.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Turning now to FIG. 1, in an only partially shown cylinder head 1 of an internal combustion engine there operates an intake or exhaust tappet 2 which has a valve stem 3 axially displaceably guided in a sleeve 4. To one end of the valve stem 3 remote from the tappet 2 there is secured a spring seat disc 5. Between the latter and the cylinder head 1 there is inserted a closing spring 6. To the head 1 there is affixed a housing block 7 in which there is contained a solenoid valve S and a hydraulically operated work piston 9. The latter is axially slideably guided in a fluid-tight manner in a sleeve 10 securely held in the housing block 7. One radial face of the work piston 9 is in an operative engagement with the valve stem 3 during the operation of the valve. The other end of the work piston 9 projects into a chamber 11 which communicates with the solenoid valve 8. From a liquid tank 13 there is admitted liquid to the solenoid valve 8 through a conduit 15 by means of a delivery pump 14. The liquid, which may be fuel, may have a pressure of, for example, 100 kg/cm². From the conduit 15, downstream of the pump 14, there extends a conduit 16, which contains a pressure control valve 17 (pressure limiting valve) and which is in communication with the liquid tank 13.

Further, from the conduit 15 there extend conduits 18 which lead to the control units (not shown) of the other engine valves and which carry pressurized liquid supplied thereto by the same delivery pump 14.

Through a nipple 19 the liquid is admitted into a bore 20 which merges into a control chamber 21 of the solenoid valve 3. The mouth of the bore 20 is formed as a valve seat and is controlled by a valve ball 22 which is movably situated in the control chamber 21. From the latter there extends a bore 23 to the chamber 11 which receives the upper terminus of the work piston 9. Further, from the control chamber 21 there extends a bore 24 from which there leads a discharge channel 25 and an adjoining return conduit 26 to the liquid tank 13. The opening between the control chamber 21 and the bore 24 is also formed as a valve seat for the valve ball 22. The latter is urged against the mouth of the bore 20 by a spring 27 with the interposition of an armature 28 which has a pin 28' and which forms part of the solenoid valve 8.

The control chamber 21, the bore 24 and the armature 28 are supported in a valve block 29 which is arranged in the housing block 7 and which is locked in

position by the flanged housing 30 of a solenoid 33. The chamber 30' receiving the spring 27 as well as the bore 20 are interconnected by means of a channel 31 to equalize the pressure in chamber 30' and bore 20. Further, the diameter of the armature 28 in the range of its fluid-tight engagement with the wall of the bore 24 is identical to the diameter of both valve seats for the valve ball 22. Thus, as long as the ball 22 assumes its position shown in FIG. 1, the force exerted thereon in the opening direction by virtue of the pressure prevailing in the bore 20 is equal to the force of the liquid pressure exerted on the ball through the armature 28 in the closing direction. The spring 27 exerts an additional force in the closing direction so that the valve ball 22 is in a stable manner urged against its seat at the mouth of bore 20.

As soon as the solenoid 33 of the electromagnet is energized, for example, by means of an electronic control device, the force of the spring 27 is overcome and the armature 28 is shifted to the left. The pressurized liquid in bore 20 displaces the valve ball 22 and as it enters into the control chamber 21, it presses the valve ball 22 against the oppositely disposed valve seat whereby the bore 24 is closed so that fuel admitted under pressure may enter the chamber 11 through the bore 23. The liquid under pressure in the chamber 11 causes the work piston 9 to shift downward which results in an opening of the valve 2.

As soon as the solenoid 33 is de-energized, the armature 28 and the valve ball 22 are shifted back to the right by the spring 27 until the bore 20 is again closed. By thus opening the bore 24 the liquid may enter from the chamber 11 through the bore 24, the channel 25 and the return conduit 26 into the tank 13. As a result, the chamber 11 is depressurized, whereupon the valve 2 closes.

The motion of the work piston 9 may be hydraulically braked towards the end of either of its strokes. For this purpose there is provided on the lateral face of the work piston 9 a collar 34 which, at either end of the reciprocating motion of the piston 9, enters in one of the cavities 35 or 35', which have a slightly larger diameter than the collar 34. Upon this occurrence, the collar 34 displaces the liquid present in the respective cavity and causes it to escape between the narrow clearance formed by the collar 34 and the wall of the cavity 35 or 35'. In this manner a braking (dashpot) effect is achieved.

The output of the delivery pump 14 may be varied by means of a servo device 37. The latter may be hydraulic servo piston, or an electric servo motor or any other similar device. The servo device 37 receives its command signals from an electronic control apparatus 38. The latter evaluates the signals representing actual values, particularly those characterizing engine variables (for example, the position of the gas pedal 40 or that of the brake pedal, the pressure of the servo liquid admitted through a conduit 41 as well as the engine temperature) to form a desired value of pump output. Signals representing this desired value are applied by the electronic control apparatus 38 to the servo device 37.

The magnitude of the opening stroke of the valve 2 is dependent upon the pressure of the liquid admitted to the chamber 11 and delivered by the pump 14. It is apparent that the valve 2 is displaced only to such an extent until the closing force which is exerted by the spring 6 and which continuously increases due to the

progressive compression of the spring 6 during the opening stroke of valve 2, is in equilibrium with the opening force exerted by the pressurized liquid on the radial work face of the piston 9. Thus, the magnitude or amplitude of the opening stroke of the valve 2 depends upon the output of the delivery pump 14. Independently from the foregoing effect, the length of the open or closed valve period as well as the beginning thereof may be determined by appropriate signals applied to the solenoid valve 8 by the same electronic control apparatus 38.

The engine temperature is sensed by a thermostat member 43 which may comprise a heat-expandable component and which is installed in the housing 44 of the engine. A setting pin 45 of the thermostat member 43 controls a potentiometer 46 or the like, the electric output signals of which thus are a function of the engine temperature and are applied to the electronic control apparatus 38.

FIG. 2 illustrates in a block circuit diagram an example of the structure of the electronic control apparatus 38. Thus, the latter includes a regulator 38R and in a feed-back circuit 41 to the regulator 38R, a converter 38F. In the simplest case the regulator 38R is formed as an amplifier, the output signal of which is applied to the electromagnetically operating servo device 37 associated with the pump 14 for changing the output thereof. In order to ensure that the output pressure of the pump 14 maintains exactly the value predetermined by the accelerator pedal 40, the feedback circuit 41 forms a closed pressure regulating circuit with the converter 38F. The latter generates, from the pressure values of the output side of pump 14, an electric signal which is compared with the output signals (functioning as control magnitude) of the accelerator pedal 40. The electric signal which depends on the temperature and which is generated by the signal transmitting device 46 affects the regulator 38R in such a manner that in case of an increasing engine temperature, the output of the pump 14 also increases, whereby, accordingly, a greater opening stroke of the valve 2 is obtained.

Turning now to FIG. 3, the diagram shown therein well illustrates the advantage of the aforescribed thermostatic control. In this diagram the stroke S of the engine valve, measured along the ordinate, is shown as a function of the rotational angle α (abscissa) of the engine cam shaft. The stroke S₁ indicates the structurally possible maximum stroke of the engine valve 2 during normal operation. The associated lifting curve is indicated at I. The curves II and III, on the other hand, correspond to the opening stroke of the engine valve during warm-up operation and starting operation, respectively. By virtue of the small stroke S₂ during the warm-up operation and warm starting and the stroke S₃ during cold starting there is obtained a smaller flow passage section at the valve 2 so that at the inlet of the engine cylinder a high air speed will prevail, resulting in a turbulent flow. By virtue of this turbulent flow, on the one hand, the fuel droplets are torn into smaller particles and, on the other hand, the fuel and air are thoroughly intermixed so that the disadvantage of a defective fuel vaporization during cold engine run is compensated.

What is claimed is:

1. In means for hydraulically controlling the operation of a tappet valve in a cylinder associated with an

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internal combustion engine, the improvement comprising

- A. a slidably held work piston in engagement with said tappet valve, said work piston, during its reciprocation, causing similar displacement of said tappet valve,
- B. means for delivering liquid under pressure to said work piston for causing reciprocation thereof, said last-named means including a liquid delivery pump,
- C. intermittently actuated solenoid valve means disposed in the path of said liquid between said pump and said work piston for intermittently admitting liquid under pressure to said work piston,
- D. means for controlling the output of said pump to

- control the pressure of said liquid for controlling the stroke amplitude of said tappet valve and
- E. engine temperature-responsive thermostatic control means applying signals to said means defined in (D) for varying the stroke amplitude of said tappet valve as a function of the engine temperature in the same sense as the change of the latter.
- 2. An improvement as defined in claim 1, wherein said means for controlling the output of said pump includes an electronic control apparatus having an input connected to said engine temperature-responsive thermostatic control means and an output connected to said pump for varying the output thereof.

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