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(54) AN INJECTION SYSTEM WITH VARIABLE ADJUSTMENT OF THE INJECTION TIMING FOR FUEL IN A DIESEL ENGINE WITH HIGH-PRESSURE INJECTION

EINSPRITZSYSTEM MIT REGELBARER VERSTELLUNG DES KRAFTSTOFFEINSPRITZPUNKTES FÜR EINEN DIESELMOTOR MIT HOCHDRUCKEINSPRITZUNG

SYSTEME D’INJECTION A REGLAGE VARIABLE DE L’INJECTION DANS LE TEMPS POUR UN MOTEUR DIESEL A INJECTION HAUTE PRESSION

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GB-A- 2 273 319
GB-A- 2 279 706
US-A- 4 763 873

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The invention relates to a fuel injection system with variable adjustment of the injection timing in a diesel engine with high-pressure injection in order to change the maximum pressure in a cylinder during combustion, which engine has several cylinders, each having at least one fuel injector connected with a fuel pump via a high-pressure conduit, a drain valve arranged on the high-pressure side of the fuel pump being capable of being closed for initiation of an injection sequence, and fuel from the pump being capable of influencing the drain valve in the closing direction.

In a diesel engine the liquid fuel is injected at a high pressure, such as from 500 to 800 bar, and is ignited by the compression heat in the cylinder, whereupon the cylinder pressure increases to the maximum pressure as the combustion progresses. The efficiency of the engine increases with increasing maximum pressure, but so do the mechanical influences on the engine members. To achieve the best possible efficiency without overloading the engine, the combustion process in modern engines is usually controlled in such a manner that the largest possible maximum pressure is achieved in the lower load area of the engine, while the maximum pressure in the upper load area, such as from 80 to 105 per cent of the nominal full load point of the engine, is limited to a substantially constant value determined with due regard to the mechanical strength of the structural members.

The maximum pressure is affected by the compression pressure which, in turbocharged engines, depends on the engine load, and is affected by the injection timing, i.e. the moment of initiation of the fuel injection seen in relation to the point in the engine cycle where the piston is in its top dead centre position (TDC) and the working stroke is to be initiated. The injection timing is normally stated as the relative angular position of the crank at which the injection starts.

In the upper load area the maximum pressure can be restricted by delaying the moment of injection so that the maximum cylinder pressure is achieved later in the cycle, where the gas in the combustion chamber has expanded to some extent because the piston has moved some distance downwards. The injection timing can be changed by changing the moment for initiation of the pressure increase in the high-pressure conduit, to be specific, by changing the closing time of a controlled drain valve arranged in the high-pressure conduit.

The initially mentioned example of an injection system with a controlled drain valve is known from Danish patent No. 1551290, where the drain valve leads the high-pressure fuel back to the suction side of the pump, until an electrically actuated valve is switched to remove a fluid pressure which keeps the slider of the drain valve in an open position, whereupon the slider is moved to a closed position by the fuel flowing out through the drain port. The opening pressure comes from a hydraulic pressure source which is independent of the fuel system and therefore involves rather a lot of extra equipment. The electrically actuated valve is spring-biased to the position where the opening pressure is removed from the drain valve, which means that the drain valve closes as fast as possible in case of failure in the electric system. This known system has certain disadvantages in connection with large diesel engines of high outputs where it is desired to delay the injection moment at high loads. If the electric system fails, the ignition timing is advanced and the maximum pressure rises, thus causing a risk of mechanical failure in the heavily loaded engine components.

GB-A-2 273 319 describes an injection system where the high-pressure side of the fuel pump is connected with a drain valve for controlling the start and stop of the fuel supply to the fuel injector in an injection sequence. The fuel from the fuel pump is used for closing the drain valve. Some of the fuel flows through a restricted flow passage in the drain valve slider and into a pressure chamber in which the pressure build-up results in closure of the drain valve. This injection system seems to relate to injection rate control.

The object of the invention is to provide an injection system of a more simple design for the delivery of large amounts of fuel per injection and having a drain valve which can be actuated electronically by means of suitably small adjustment forces in such a manner that the risk of overloading the engine is minimized.

In view of this, the injection system according to the invention is characterised in that the fuel from the fuel pump used for closure of the drain valve passes through a restricted flow passage, that the injection timing is advanced by actuation of an electrically actuated control valve for increasing the flow area in the restricted flow passage, and that at failure in the electric control system the control valve is adapted to be in the position without increased flow area.

The first amount of fuel delivered from the fuel pump is drained away through the drain valve, and at the same time a small amount of fuel flows through the passage and builds up a pressure which begins closing the drain valve when its actuating pressure is exceeded. When the drain valve is completely closed, the pressure in the high-pressure conduit rises to a level where the fuel injector opens. It is a substantial feature of the invention that the control valve must be actuated to advance the injection timing to achieve a larger maximum pressure at combustion. At failure in the electronic actuation of the control valve the restricted flow passage does not achieve an increased area, and the drain valve closes slowly so that the maximum pressure at combustion is not increased. Mechanical overloading of the engine owing to power failure in the control of the injection system is thus avoided.

It is furthermore an advantage that the flow volume in the restricted flow passage required for closure of the drain valve is substantially smaller than the deliv-
ery volume of the pump per injection, which renders it possible to manufacture the control valve with small dimensions and associated small mass so that the actuating force can be supplied, for example, by a solenoid. The use of the fuel pressure for closure of the drain valve combined with the small control valve provides an injection system of a very simple design, requiring only electrical actuation of the control valves. Control of the injection timing for the whole engine can thus be provided with a simple controller, like an ignition distributor, instead of the heavy and slow-acting mechanical control systems formerly used in large engines.

In a preferred embodiment it is possible, in addition to control of the injection timing, to achieve control of the fuel amount injected, as a secondary control valve can open the drain valve by passing the pressure from the high-pressure side of the fuel pump on to a piston surface acting in the opening direction of the drain valve, which interrupts the injection.

The invention will now be further described in detail below with reference to the very schematic drawing, in which

Fig. 1 shows a diagram of the pressure sequences in the high-pressure conduit during an injection period for two different injection timings,

Fig. 2 shows the corresponding pressure sequences in the cylinder during combustion,

Fig. 3 is an outline of an injection system according to the invention, and

Figs. 4 and 5 are outlines of two different embodiments of the drain valve with associated control valve.

In a large two-stroke diesel engine for propulsion of a ship or for power generation in a stationary power plant, cam-actuated piston pumps of the Bosch type are conventionally used to generate the required injection pressure in the fuel oil. Depending on cylinder size and number, the engines can yield outputs of between 4,000 and 70,000 kW, and a typical cylinder output may be in the interval from 1,000 to 5,000 kW, which means that a considerable volume of fuel must be injected during each engine cycle. In the largest engines, the injection into each cylinder may be in the order of about 200 g of fuel per engine cycle, which may, for example be distributed on three injectors per cylinder. Engines of this type typically have maximum speeds in the interval of 60-190 r.p.m.

Fig. 3 shows an example of a well-known fuel pump 1 of the make MAN B&W Diesel. A cam 2 on a camshaft 3 can, via a cam roller 4, move a pump piston 5 upwards so that the pump performs a delivery stroke, whereby the fuel in a pump chamber 6 delimited by the piston and a surrounding pump cylinder 7 is pressed out of the discharge opening of the pump and over into a high-pressure conduit 9. A compression spring 10 presses the cam roller 4 against the cam 2, so that the pump piston is returned to its starting position after the delivery stroke with simultaneous suction of a new portion of fuel into the pump chamber via an inlet conduit 11 and an annular chamber 12 around the pump cylinder.

The high-pressure conduit 9 leads to a fuel injector 13 having a central through-going fuel passage with a spring-biassed valve, the spring force determining the opening pressure of the injector. The fuel passage opens out downwards in an atomizer 14 with atomizer nozzles from where the fuel can be atomized into the combustion chamber of the cylinder.

Fig. 1 shows in a solid line the pressure sequence on the high-pressure side of the pump, measured in the pump chamber 6, during an injection period with advanced moment of injection. The pressure is indicated in bar as a function of the crank angle, where 180° indicates the crank position with the piston in the top dead centre position.

It is seen that during a crank turn of about 5°, the pressure rises from the pre-pump pressure of about 8 bar to a pressure of about 520 bar stated by point O where the injector opens, and during the next approximately 10° the fuel is injected into the cylinder at a steadily increasing pressure up to about 700 bar marked by the point M, where a drain hole, not shown, in the pump cylinder 7 is uncovered so that the pressure in the pump chamber drops to about 200 bar, and the valve in the injector closes and interrupts the injection.

This injection sequence produces a pressure sequence in the cylinder shown in Fig. 2 drawn up in full lines. During the first 180° crank turn, the air in the cylinder is compressed to the compression pressure P_c of about 124 bar, whereupon the combustion of the fuel makes the pressure rise further up to the maximum pressure P_max of about 165 bar at a crank angle of about 193°.

Figs. 1 and 2 indicate in broken lines the pressure sequences appearing when the moment of injection of the fuel is delayed. In Fig. 1, the time difference between the two injection sequences corresponds to a crank turn of 3.5° so that the points O' and M' are located this angle later in the cycle. It is seen from Fig. 2 that the delayed injection of the fuel has the effect that the compression pressure has time to fall by about 2 bar from the point P_e to the point P'_e, before the combustion makes the cylinder pressure rise, and that the maximum pressure P'_max of about 140 bar appears at a crank angle of 196°. The pressures and angles of rotation are merely one concrete example out of many possibilities, but it shows that a delay of the start of the injection sequence reduces the maximum pressure in the cylinder.

The fuel injection timing can be varied by means of a drain valve 15 which can be set in the first extreme position shown in Fig. 3, where the fuel discharge of the pump is connected with a low-pressure source or a drain 16 preventing the pressure in the conduit 9 from reaching the opening pressure of the injector.
In its other extreme position, the drain valve 15 connects the fuel discharge of the pump with the injector, and the connection to the drain is interrupted. The drain valve is spring-biased in the opening direction, i.e., for movement towards the first extreme position shown in Fig. 3, where the pump pressure is drained away. Instead of a spring bias, the drain valve may be moved to its first extreme position by a hydraulic or pneumatic influence or by some other returning force which is temporarily active in the period between two pump strokes.

At the same time as part of the fuel is drained away through the valve 15, a small amount of fuel flows into a restricted flow passage 17 branching off from the high-pressure conduit on the upstream side of the valve 15. The branched-off amount of fuel is via the flow passage 17 made to act on the drain valve in the closing direction. When a sufficient amount of fuel has flown through the flow passage 17, the drain valve has been moved to a completely closed position, whereupon the pressure in the conduit 9 can rise and the injection can begin.

As a predetermined, fairly constant fuel volume is required to close the drain valve, the time required for the closure can be shortened by increasing the flow area in the restricted flow passage 17. This may be done by using a single flow passage with a restriction adjustable in size by the control valve or by using a continuously open passage and at least one additional passage coupled in parallel, which can be opened or closed by the control valve. Fig. 3 illustrates both options by means of a control valve 18 in a parallel flow passage.

The control valve 18 can be activated electronically and may, for example, be a solenoid valve. Preferably, the control valve is spring-biased for movement in the direction of a starting position without increased flow area, a spring influence from a mechanical spring being completely independent of failures in the electronic system. The spring bias may also be provided in any other manner, for example by a permanent magnet.

The simplest and most reliable system can be obtained with a control valve with only two fixed extreme positions, but the control valve can also be designed so that the activated extreme position with the increased area is adjustable, and that the size of the increased area depends on this adjustment. Thus, the control valve becomes adjustable in a number of intermediate positions with a gradual increase of the flow area in the restricted flow passage. As an example of such a control valve, not shown, may be mentioned a solenoid valve which creates a magnetic field by the electrical actuation, which field pulls the valve body to an extreme position determined by the valve body encountering a mechanical stop, which is adjustable in the direction of movement of the valve body. The valve body may comprise a rod extending transversely in through the flow passage and constituting a restriction of the flow area. The rod may, for example, have a number of wedge-shaped depressions of increasing depth towards one end of the rod so that a longitudinal displacement away from the starting position moves still larger depressions into the flow passage, which renders possible a step-wise increase of the flow area without the fuel pressure in the passage influencing the rod with a resulting force.

A first embodiment of the control valve 18 is shown in Fig. 4 where the flow passage 17 extends from the high-pressure conduit 9 to a pressure chamber 19. The movable body 20 of the drain valve comprises a seat 21 which, through a connecting portion 22 in the high-pressure conduit, is supported by a cylindrical guide section 23 passing in a pressure-sealing manner through a guiding bore 24 into the pressure chamber where the guide section is joined with a cylindrical piston 25 of a larger diameter than the guide section and constituting a movable end wall in the pressure chamber.

When delivery of fuel from the pump stops, the pressure in the conduit 9 drops, and a compressed compressed spring 28 presses the valve body 20 into the open extreme position and at the same time the pressure chamber 19 is emptied of fuel.

The flow area in the restricted flow passage is adjustable by means of the control valve 18, which in a first embodiment comprises a slider 30, which is longitudinally displaceable in a transverse direction in relation to the passage 17 between the starting position shown in Fig. 4, where a first, relatively small flow opening 31 is arranged in the passage and restricts the fuel flow, and an activated extreme position, where a second, relatively larger flow opening 32 is arranged in the passage. In the example shown, the flow openings in the slider 30 are designed as sections with a smaller diameter, where the section with the opening 32 is larger than the section with the opening 31.

The slider 30 is moved between the extreme positions by means of an electrically/magnetically acting drive comprising two coils 33, 34, each with associated magnetisable core material, and a circular disc 35 acting as a fitting, fastened at the end of the slider and movable from one to the other extreme position by a magnetic field created by passing current through one of the coils 33, 34 through associated wires 36, 37. As mentioned, the magnetising current pulls the disc 35 over to the magnet and leaves a certain residual magnetism in the core material. The residual magnetism is sufficient to retain the disc after interruption of the current. When the control valve is to be switched, it may be advantageous
to pass a small surge of current through the coil to be left by the disc to remove the residual magnetism in the core material, as it is thus possible to reduce the magnetising current which it is necessary to apply to the opposite coil to attract the disc 35. To ensure that the control valve is positioned in the starting position shown in Fig. 4 if the electrical system fails, the coil 33 may be connected to a capacitor which is discharged through the coil at power failures, whereby the disc is fixed in the starting position until the electric control system is operating again. The control valve is seen to be a digital valve.

[0029] Instead of the drive described, a usual solenoid can be used, i.e., a single coil which, when magnetising current is applied, can pull the disc from the starting position to the actuated position with simultaneous compression of a compression spring which returns the slider to the starting position when the magnetising current of the solenoid is interrupted.

[0030] In the embodiment shown in Fig. 5, the drain valve is in principle designed in the same manner as described above, and the same reference numerals are therefore used. The flow passage 17 is divided into a primary passage 40 permanently interconnecting the high-pressure conduit 9 and the pressure chamber 19. The flow area of the primary passage has been chosen to achieve the same fuel flow through the passage as through the passage in Fig. 4 when the slider 30 is in its starting position. The pressure chamber 19 can further be connected with the conduit 9 through a secondary passage 41 which, like the passage 40, is arranged on the upstream side of the drain port 27.

[0031] The control valve 18 is designed as a solenoid valve having a valve needle 42 which is pressed by a compression spring 43 towards a starting position, in which the needle is in contact with a stationary seat at the inlet opening to the secondary passage and bars access to it. When a solenoid is magnetised by current passing through the wires 45, the valve needle is retracted from the stationary seat so that fuel can also flow through the secondary passage into the pressure chamber 19.

[0032] It is also possible to design a second pressure chamber 50 on the side of the piston 25 of the drain valve which is biased by the compression spring 28. The chamber 50 may have a drain passage, not shown, with an orifice to prevent sustained substantial pressure build-up on the spring side of the piston 25. Alternatively the second pressure chamber could have a large volume and the lifting height of the drain valve 15 could be small.

[0033] A passage 51 may interconnect the second pressure chamber and the high-pressure conduit 9 on the downstream side of the drain port 27, when a secondary control valve 52 is actuated to open for access to the passage 51. The secondary control valve 52 may be constructed in the same manner as the control valve 18. In the embodiment shown in Fig. 5, the secondary control valve comprises a valve needle 53, which can be moved away from the inlet opening of the passage by means of a solenoid.

[0034] When the passage 51 is open and both pressure chambers and the conduit 9 are filled with fuel at a high pressure, the valve body 20 is influenced by a resulting force in the opening direction, because the piston 25 has an effective area acting in the opening direction and corresponding to the cross-sectional area of the connecting portion 22. Actuation of the secondary control valve 52 results in opening of the drain valve, which interrupts injection. The secondary control valve 52 can be used to adjust or meter the amount of fuel injected.

[0035] In the drawing, the drain valve with associated control valve is shown in connection with the high-pressure conduit 9. It should be understood that the valves may be designed in a valve unit which can be connected to the high-pressure conduit 9. The valve unit may, for example, be mounted on top of the housing of the fuel pump. It is possible to retrofit the valve unit on engines already existing to enable adjustment of the injection timing in a simple manner. It is also possible to build in the valve unit as an integral part of the fuel pump.

Claims

1. A fuel injection system with variable adjustment of the injection timing in a diesel engine with high-pressure injection in order to change the maximum pressure \( P_{\text{max}} \) in a cylinder during combustion, which engine has several cylinders, each having at least one fuel injector connected with a fuel pump via a high-pressure conduit, a drain valve arranged on the high-pressure side of the fuel pump being capable of being closed for initiation of an injection sequence, and fuel from the pump being capable of influencing the drain valve in the closing direction, characterized in that the fuel from the fuel pump used for closure of the drain valve passes through a restricted flow passage, that the moment of injection is advanced by actuation of an electrically actuated control valve for increasing the flow area in the restricted flow passage, and that at failure in the electric control system the control valve is adapted to be in the position without increased flow area.

2. A fuel injection system according to claim 1, characterized in that the control valve (18) is spring-biased for movement in the direction of a starting position without increased flow area.

3. A fuel injection system according to claim 2, characterized in that the control valve (18) has a slider (30) with two flow openings (31, 32) of different dimensions, that the smaller flow opening (31) is located in the restricted flow passage (17) when the
control valve is in a starting position, and that at the electrical actuation the slider is displaced against a spring bias away from the starting position to a second position where the larger flow opening (32) is located in the restricted flow passage.

4. A fuel injection system according to claim 1 or 2, characterized in that the restricted flow passage (17) comprises a permanently open primary passage (40) and a secondary passage (41) which can be barred by the electrically actuated control valve.

5. A fuel injection system according to any one of claims 1-4, characterized in that the control valve is adjustable in a number of intermediate positions with gradual increase of the flow area in the restricted flow passage.

6. A fuel injection system according to claim 1, characterized in that the electrically actuated control valve (18) is a digital valve which can be retained in at least one of the two positions by residual magnetism between actuations.

7. A fuel injection system according to any one of claims 1-5, characterized in that the electrically actuated control valve (18) is a solenoid valve.

8. A fuel injection system according to any one of claims 1-7, characterized in that a secondary control valve (52) can open the drain valve (15) by passing the pressure from the high-pressure side of the fuel pump on to a piston surface acting in the opening direction of the drain valve.

Patentansprüche


2. Kraftstoff-Einspritzsystem nach Anspruch 1, dadurch gekennzeichnet, dass das Steuerventil (18) unter Federspannung steht zur Bewegung in der Richtung einer Startposition ohne vergrößerten Strömungsbereich.

3. Kraftstoff-Einspritzsystem nach Anspruch 2, dadurch gekennzeichnet, dass das Steuerventil (18) einen Schieber (30) mit zwei Strömungsoffnungen (31, 32) mit verschiedenen Abmessungen hat, dass die kleinere Strömungsoffnung (31) im eingeschränkten Strömungsweg (17) angeordnet ist, wenn sich das Steuerventil in einer Startposition befindet, und dass bei der elektrischen Auslösung der Schieber gegen eine Federspannung von der Startposition weg in eine zweite Position verschoben wird, wo die größere Strömungsoffnung (32) im eingeschränkten Strömungsweg angeordnet ist.

4. Kraftstoff-Einspritzsystem nach Anspruch 1 oder 2, dadurch gekennzeichnet, dass der eingeschränkte Strömungsweg (17) einen permanenten offenen Hauptdurchgang (40) aufweist und einen Nebendurchgang (41), welcher durch das elektrisch ausgelöste Steuerventil versperrt werden kann.


7. Kraftstoff-Einspritzsystem nach einem der Ansprüche 1 bis 5, dadurch gekennzeichnet, dass das elektrisch ausgelöste Steuerventil (18) ein Solenoidventil ist.

8. Kraftstoff-Einspritzsystem nach einem der Ansprüche 1 bis 7, dadurch gekennzeichnet, dass ein Nebensteuerventil (52) das Ablässventil (15) öffnen kann durch Weiterleiten des Druckes von der Hochdruckseite der Kraftstoffpumpe auf eine Kolbenfläche, welche in der Öffnungsrichtung des Ablässventils wirkt.
Revendications

1. Système d'injection de carburant à réglage variable du temps d'injection pour un moteur diesel à injection haute pression dans le but de modifier la pression maximale ($P_{\text{max}}$) dans un cylindre durant la combustion, ladit moteur comportant plusieurs cylindres dont chacun comprend au moins un injecteur de carburant (13) relié à une pompe de carburant (1) par l'intermédiaire d'une conduite haute pression (9), une soupape de drainage (15) disposée du côté haute pression de la pompe de carburant étant capable d'être fermée pour engager une séquence d'injection, le carburant provenant de la pompe étant à même d'influer sur la soupape de drainage dans le sens de la fermeture, caractérisé en ce que le carburant provenant de la pompe de carburant et utilisé pour la fermeture de la soupape de drainage s'écoule à travers un passage d'écoulement restreint (17), en ce que l'instant d'injection est avancé par actionnement d'une soupape de commande (18) actionnée électriquement pour accroître la superficie d'écoulement dans l'écoulement restreint, et en ce qu'en cas de défaut dans le système de commande électrique, la soupape de commande (18) est conçue pour se retrouver dans la position où il n'y a pas de superficie d'écoulement accrue.

2. Système d'injection de carburant selon la revendication 1, caractérisé en ce que la soupape de commande (18) est précontrainte par ressort pour mouvement en direction d'une position de départ sans superficie d'écoulement accrue.

3. Système d'injection de carburant selon la revendication 2, caractérisé en ce que la soupape de commande (15) comporte un tiroir (30) à deux orifices d'écoulement (30, 31) de dimensions différentes, en ce que l'orifice d'écoulement le plus petit (31) est situé dans le passage d'écoulement restreint (17) lorsque la soupape de commande se trouve dans une position de départ, et en ce que lors de l'actionnement électrique, le tiroir est déplacé, contre une précontrainte par ressort, pour s'éloigner de la position de départ jusqu'à une seconde position dans laquelle l'orifice plus large d'écoulement (32) est situé dans le passage d'écoulement restreint.

4. Système d'injection de carburant selon les revendications 1 ou 2, caractérisé en ce que le passage d'écoulement restreint (17) comprend un passage primaire (40) ouvert en permanence et un passage secondaire (41) qui peut être refermé par la soupape de commande actionnée électriquement.

5. Système d'injection de carburant selon l'une quelconque des revendications 1 à 4, caractérisé en ce que la soupape de commande est réglable dans un nombre de position intermédiaires avec augmentation graduelle de la superficie d'écoulement dans le passage d'écoulement restreint.

6. Système d'injection de carburant selon la revendication 1, caractérisé en ce que la soupape de commande à actionnement électrique (18) est une soupape numérique qui peut être retenue dans au moins l'une des deux position par magnétisme résiduel entre actionnements.

7. Système d'injection de carburant selon l'une quelconque des revendications 1 à 5, caractérisé en ce que la soupape de commande à actionnement électrique (18) est une soupape à solénoïde.

8. Système d'injection de carburant selon l'une quelconque des revendications 1 à 7, caractérisé par le fait qu'une soupape secondaire de commande (52) peut ouvrir la soupape de drainage (15) en transférant la pression provenant du côté haute pression de la pompe de carburant à une surface de piston agissant dans le sens de l'ouverture de la soupape de drainage.