(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
13 May 2004 (13.05.2004)

PCT

(10) International Publication Number
WO 2004/040135 A1

(51) International Patent Classification[7]: F04B 17/04, 53/00, F23K 5/14

(21) International Application Number: PCT/DK2003/000732

(22) International Filing Date: 30 October 2003 (30.10.2003)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
PA 2002 01683 1 November 2002 (01.11.2002) DK

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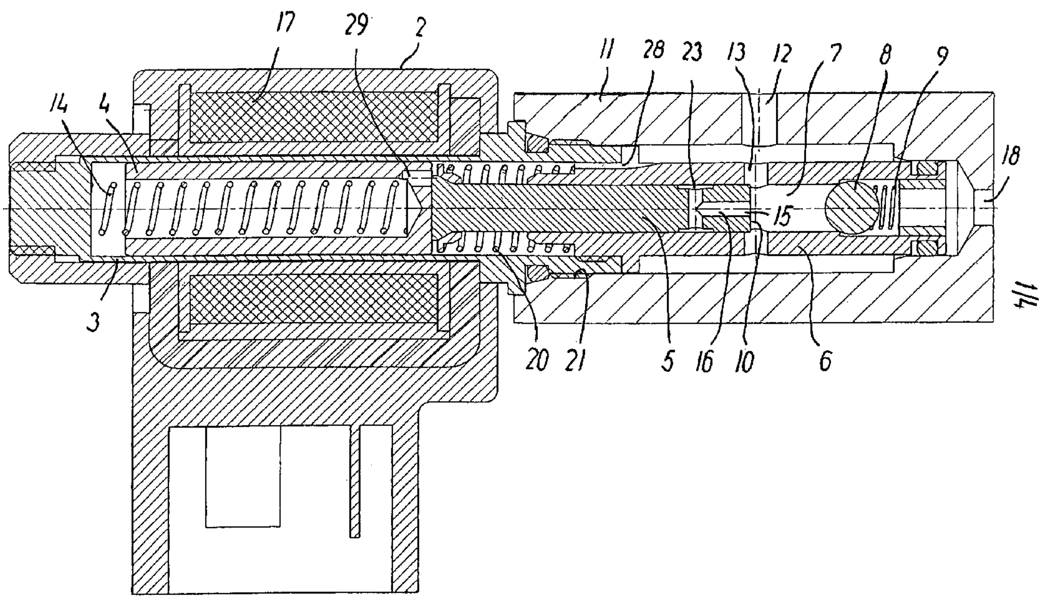
(81) Designated States *(national)*: AM, AT, AU, BA, BG, BR, BY, CA, CH, CN, CZ, DE, DK, EE, ES, FI, GB, GE, HR, HU, IL, IN, IS, JP, KR, KZ, LT, LU, LV, MD, MK, MX, NO, NZ, PL, PT, RO, RU, SE, SG, SK, TR, UA, US, UZ, VN, YU, ZA.

(84) Designated States *(regional)*: Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IT, LU, MC, NL, PT, RO, SE, SI, SK, TR).

Published:
— *with international search report*
— *before the expiration of the time limit for amending the claims and to be republished in the event of receipt of amendments*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) Title: A RECIPROCATING LIQUID PUMP FOR DELIVERY OF LIQUID FUEL TO A DOMESTIC BURNER DEVICE



(57) Abstract: A reciprocating liquid pump (1) for delivery of liquid fuel to a domestic burner device comprises an electromechanical drive means acting on a plunger (5) to reciprocate the plunger in a pump chamber (7) of a pump cylinder (6). Liquid fuel is delivered from the pump during an active strode of the plunger. The active stroke is terminated by pressure relief means (15, 16, 23) which bleed the pump chamber (7) when the plunger is at a distance (d) from the maximum stroke end stop position.

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A reciprocating liquid pump for delivery of liquid fuel to a domestic burner device.

The present invention relates to a reciprocating liquid pump for delivery of liquid fuel to a domestic burner device, said pump comprising an electromechanical drive means which is electronically activated and acts on a plunger to reciprocate the plunger in a pump chamber of a pump cylinder, whereby liquid fuel is delivered from the pump, said plunger having a maximum stroke end stop position.

Liquid fuel burners, such as oil burners, are quite common for domestic use. Ordinarily such oil burners are of the oil pressure atomizing type having high-pressure pumps delivering oil at high pressure. These burners operate on the principle that when oil under pressure is permitted to expand through a small orifice, it atomizes into a spray of very fine droplets, which are suitable for combustion. These burners are designed to operate with oil pressures as high as 40 bar. The principle upon which these burners operate requires that the pressure drop across the orifice be maintained high and as nearly constant as possible in order to achieve the necessary finely atomized droplets and a combustion without significant pulsations. Because it is not possible to maintain the required pressure drop at lower flow rates, turndown, which is defined as the ratio of maximum to minimum input rates, in the operation of such burners has traditionally been very severely limited or has not been used at all, and the burners have been operated in an on-off mode only. This, of course results in inferior temperature control, lower furnace efficiency and increased thermal load of the components, as they will experience a lot of heating and cooling cycles.

Further the minimum output of the burner is controlled by the size of the holes in the orifice. The smallest feasible holes are 0.1 mm in diameter, as smaller holes will clog very quickly due to inevitable particles in the oil or due to soot build-up from the combustion, increasing the need for maintenance to an intolerable level. The ordinary minimum output of oil burners having an orifice with the smallest holes possible, is about 10 kW, which exceeds the static demand of an ordinary household.

The prior art liquid fuel pumps have not been suitable for delivery of the very small quantities of oil needed for an output of less than 10 kW, and existing low volume oil pumps are quite complex and hence expensive to manufacture.

An important parameter for a pump for a burner of a household furnace is that it should be very quiet and preferably noiseless, as most people easily get annoyed and irritated if there is noise in their home.

US patent No. 4,352,645 discloses a prior art fuel oil pump for a burner nozzle of a household furnace. The pump is actuated by an electromagnet and comprises a small diameter pump piston providing a suction force for drawing in fuel oil into an intake chamber. The pump further comprises a main pumping chamber, a series of check valves and a plurality of springs. One advantage of the pump is that noise from armature braking is diminished, but the pump is quite complex and hence expensive to manufacture, and the many moving parts in the pump and also the suction of fuel oil into the intake chamber results in noise. Further the pump is quite voluminous and hence not optimal for installing in a small supply device for a domestic burner.

It is an object of the present invention to provide a liquid pump of simple configuration and a very low noise level.

With a view to this the liquid pump according to the invention is characterized in that the pump comprises pressure relief means to bleed the pump chamber at the end of the active stroke at a distance from the maximum stroke end stop position.

The bleeding of the pump chamber at the end of the active stroke provides a very well-defined and precise termination of the delivery of liquid fuel without using separate moving valve members that can create noise. The bleeding of the pump chamber while the plunger is at a distance from the maximum stroke end stop position provides space for reversing the movement of the plunger without any noisy abutment of the plunger against a stationary part. Simple pressure relief means can be used for bleeding of the pump chamber without necessarily increasing the number of separate moving parts. The pressure relief means allow a noiseless run of the reciprocating liquid pump.

In an embodiment the ratio L/D between an active stroke length L of the plunger and the plunger diameter D is less than 0.4, and preferably less than 0.2. The very small length of the active stroke in relation to the plunger diameter results in small pressure variations in the liquid fuel in the instant where the plunger seals off the pump chamber and initiates the active stroke. The restriction of the size of pressure variations by limiting the plunger movements is an optional measure for assisting the plunger in performing gentle, practically noiseless movements, and the forces on the plunger in order to make it effect the active stroke become so small that the plunger drive means can be also practically noiseless. By keeping the ratio L/D on less than 0.2 the very small active stroke length allows the pump to deliver a certain amount of fuel with a relatively high number of strokes. This promotes a more even flow of fuel delivered from the pump and thus a more stable continuous operation of the burner device despite the fact that the fuel is dosed intermittently by the reciprocating plunger.

In a preferred embodiment the full stroke length of the plunger during the reciprocating cycle is more than two times longer than the active stroke length, and preferably more than three times the active stroke length. This allows the plunger to effect a relatively slow retardation of its speed at termination of the active stroke. In addition to this it is also possible for the plunger to be retracted to an initial position well before the position at which the next active stroke is initiated. This provides a distance for the plunger to accelerate to pumping speed prior to the initiation of the actual active stroke with active pumping of the liquid fuel. The distance for running off its pumping speed and the distance for accelerating to pumping speed enhances the practically noiseless running of the pump because it allows the plunger drive means to act with relatively small forces upon the plunger and yet obtain the desired pumping action.

It is a possibility to let the pressure relief means connect the pump chamber with a fuel inlet to the pump when the plunger is not performing the active stroke. During the return of the plunger to the initial position liquid fuel from the fuel inlet can have access to the pump chamber via a non-return valve opening for inflow of fuel to the pump chamber, and consequently the fuel chamber is maintained filled with fuel at approximately the feeding pressure present in the fuel inlet. However, a more simple design without a non-return valve is preferred. In most cases the plunger does not need to perform any substantial suction of fuel into the pump chamber.

The plunger can be biased towards its initial, neutral position by a first return spring that returns the plunger after termination of the active stroke, and preferably also a second return spring acts on the plunger to advance it in direction of the pump chamber towards said neutral position. Return springs, e.g. of the helical compression spring type are very reliable, easy to mount, and the use of two counter-acting return springs are well suited for obtaining soft, noiseless movements of the plunger.

In an embodiment the plunger is activated by the electromechanical drive means with a frequency of at least 5 Hz. By effecting at least 5 strokes per second and thus delivering at least 5 doses of liquid fuel per second a low pressure burner can be kept continuously running, even though only a small amount of fuel is delivered per dose.

Preferably, the plunger is activated by the electromechanical drive means with a frequency in the range of from 50 to 60 Hz as this allows for use of a plunger drive means driven by alternating current from a public distribution network, which is usually available at all households.

The electromechanical drive means can include an armature comprising ferromagnetic material and mounted in extension of the plunger, and an electromagnet arranged outside the armature. This embodiment is straightforward to manufacture because the armature is a separate member. The armature is of another material than the plunger which can be of hardened steel that has poor magnetic properties.

The pressure relief means include in a preferred embodiment a pressure relief conduit arranged in the plunger and cooperating with a fuel inlet in the pump cylinder. During the entire active stroke the pressure relief conduit in the plunger is kept cut off from the fuel inlet in the cylinder in order to allow the pressure in the pump chamber to exceed the opening pressure of the liquid pump, which opening pressure is determined for instance by a spring force on a ball in a check valve at the outlet from the pump chamber.

Preferably, the pressure relief conduit comprises at least a first bore and an annular recess at the outer surface of the plunger, said first bore extending into the plunger from the pump chamber side thereof towards the annular recess. The annular recess makes it possible for the first bore to communicate with the fuel inlet in the pump cylinder at all rotational positions of the plunger in relation to the pump cylinder, provided the axial positioning of the plunger is outside the range for the active pumping stroke.

The first bore in the plunger can in one embodiment extend in direction of the centre axis of the plunger to join a transverse bore that extends into an annular recess in the outer surface of the plunger, and in another embodiment the first bore extends obliquely to the centre axis of the plunger directly into the annular recess in the outer surface of the plunger. In either case the pressure relief conduit can be manufactured in the plunger in a simple manner. The annular recess in the outer surface of the plunger provides well defined cut off edges. The first bore debouches in the front end (the head) of the plunger and is continuously subjected to the pressure prevailing in the pump chamber. As an alternative the annular recess can be located in the pump cylinder. In another solution, which is more difficult to manufacture, the pressure relief means include a pressure relief passage machined into the pump cylinder. The passage can be a recessed portion in the cylinder bore.

Preferably, the liquid pump is designed with one or more pressure equalizing channels communicating the pressure at the fuel inlet to the drive end side of the plunger. The pressure equalizing channels ensure that the drive end side of the plunger is kept at the pressure prevailing at the fuel inlet so that when the pump chamber is set into communication with the fuel inlet via the pressure relief means, then the plunger obtains a statically fully pressure-balanced state. One advantage of this is that the liquid pump becomes insensitive to pressure variations at the fuel inlet, e.g. pressure variations caused by a primer pump.

It is preferred that the liquid fuel delivery amount per activation of the plunger is in the range of 1 mm^3 to 50 mm^3. If the amount is less than 1 mm^3 it can become difficult to deliver the amount out of the pump with the desired certainty and repeatability. Amounts above 50 mm^3 is of course possible but the size of the pump will be larger. The most preferred amount of fuel per activation of the plunger is in the range of 2 mm^3 to 15 mm^3.

At an activation frequency of 50 Hz and a delivery amount of 2 mm^3 per activation the fuel pump will deliver 0.36 litre of fuel per hour which covers the basic heat consumption of a typical single family house, and the delivery amount of 8 mm^3 per activation at an activation frequency of 50 Hz results in delivery of 1.44 litre of fuel per hour, and that covers the high load heat consumption during consumption of hot water in a typical single family household. These amounts cover the needs of most households using oil fired heating systems.

Prior art high-pressure systems tend to create noise as they run. The low noise level obtained by the present invention can be further enhanced, such as to the point of elimination of noise, by preferably keeping the liquid fuel delivery pressure from the pump in the range of from 0.01 bar to 3 bar above atmospheric pressure. With these low pumping pressures audible sounds will not be created by the pressure changes occurring at initiation and termination of the active stroke of the plunger. A preferred pressure range is from 0.01 to 0.2 bar above atmospheric pressure.

The delivery amount of the liquid pump is preferably adjustable in order to adapt the amount of fuel delivered to the burner device to the current need for heating of for instance the water in a central heating system. The need for heating the water in the central heating system varies considerably during the year from winter to summer. During winter the need for heating can e.g. be proportional to the outdoor temperature, and during summer the need for heating for use in the central heating system can be non-existing. With respect to the need for heating hot water for personal use and for cleaning and use in the kitchen these needs are almost constant during the year, but varies considerably during the day.

In one embodiment the delivery amount is adjustable by turning the pump cylinder in relation to the plunger. The plunger can for instance have an oblique cut-off edge which passes the fuel inlet during the plunger movements, so that turning of the pump cylinder or the plunger results in a adjustment of the active stroke length.

In another embodiment the delivery amount is adjustable by changing the frequency of activation of the plunger. The delivery amount is linearly proportional with the activation frequency and a settable frequency converter can be adjusted to set the frequency of activation at the desired level. The delivery amount can also be adjustable by turning the liquid pump on and off. This can e.g. be relevant during summertime if there is only a need for heating of hot water for consumption.

In the following, examples of embodiments of the invention is described in greater detail with reference to the accompanying schematic drawings, on which

Fig. 1 illustrates a cross-section of a reciprocating liquid pump according to the present invention,

Fig. 2a-2d are enlarged sectional views of a pump chamber and a plunger in the liquid pump of Fig. 1, where the plunger is depicted in a most retracted, initial position in Fig. 2a, in a position of initiation of an active stroke in Fig. 2b, in a position of termination of an active stroke in Fig. 2c, and in a most advanced position in Fig. 2d,

Fig. 3a-3d are similar enlarged sectional views of a second embodiment of a pump chamber and a plunger,

Fig. 4 illustrates a third embodiment of a pump chamber and a plunger with a oblique cut-off edge, and

Fig. 5 illustrates a fourth embodiment of a pump chamber and a plunger.

For purposes of illustration, the present invention is embodied in a liquid pump 1 for metering fuel, such as may be used in dosing small volumes of fuel oil from a tank (not shown) or a primer pump (not shown) to the atomizing unit of a burner in an oil fired heating system, or to a burner device in a car engine heater (not shown). In the context of the present invention the word domestic is to be understood in a broad sense covering both a domestic burner device in a central heating system in a house, and a domestic burner device in peripheral equipment belonging to the household, such as in a car or in a sailboat.

The pump includes a housing 2 for a coil 17 of an electromagnet, which is connectable to a power source providing cycles or pulses of current. The electromagnet is a standard component, such as a Danfoss part order No. 71N0802. An armature cylinder 3 extends through the housing 2, and in cylinder 3 an armature 4 is arranged to be moveable in a reciprocating movement activated by the electromagnet. The armature 4 in turn acts on a plunger 5 reciprocating in a pump cylinder 6 comprising a pump chamber 7 defined by the inner surface of the pump cylinder 6, a check valve, which in this embodiment is a ball 8 biased against an opening of the pump chamber 7 by a compression spring 9, and a plunger head 10 of the plunger 5.

The pump 1 is mounted in a block 11, e.g. by threading an outer thread on the armature cylinder 3 into an inner thread 21 in a bore in blok 11. The block 11 comprises a fuel inlet 12 connectable to a source of liquid fuel, such as oil from a primer pump or a tank (not shown). The fuel inlet 12 is in flow communication with the pump chamber 7 through cylinder openings 13 when the plunger 5 is in a retracted position as shown in Fig. 1. The pump 1 further comprises a first return spring 20 acting on the plunger 5 to bias the plunger in a direction to the left in the figure, i.e. towards the armature in a direction of retracting the plunger 5 from the pump chamber 7 in a return stroke. The pump also comprises a second return spring 14 to bias the armature 4 in a direction to the right in the figure, i.e. in a direction towards plunger 5, and consequently the plunger 5 is biased by the second spring to move in direction of the pump chamber 7. The two counteracting springs 14, 20 are at equilibrium when the plunger 5 is in the neutral position. When the plunger 5 is activated by the drive it will move away from the neutral position, and depending on the direction of movement one or the other of the springs will be compressed, and due to the compression the compressed spring will act on the plunger with a larger spring force trying to restore the equilibrium position. The spring forces thus tend to maintain the plunger at the neutral position.

In operation, when the electromagnet is energized to move the armature 4 forwards in direction of the plunger, the plunger 5 will be subjected to a push accelerating the plunger to the right whereby the edge of the plunger 5 on plunger head 10 seals off the pump chamber 7 and the active stroke is initiated. During the continued travel plunger 5 pressurizes the liquid fuel within the pump chamber 7. When the pressure of the liquid fuel in the pump chamber exceeds the spring force of spring 9, the ball 8 is lifted from its seat, thereby allowing oil to flow from the pump chamber 7 to a pump outlet 18 for delivery to a nozzle of the burner device through a fuel delivery line (not shown) connected to pump outlet 18. A recessed portion in the outer side of the pump cylinder 6 provides a pressure equalizing channel 28 leading into the armature cylinder 3 in the area of the spring 20 on the rear side of the pump cylinder 6. A bore through the end face of armature 4 provides a pressure equalizing channel 29 leading to the rear side of the armature in the area of spring 14. Consequently, the pressure at the fuel inlet is communicated through channels 28, 29 to both sides of armature so that the full the drive end side of the plunger 5 is subjected to the same pressure as the pressure at the fuel inlet.

The function of the pump 1 will be more readily understood by the following description with reference to Figs. 2a-2d, where Fig. 2a is an enlarged view of the pump cylinder 6 and the plunger 5 with the plunger 5 in the most retracted position. In this position pump chamber 7 is in flow communication with the fuel inlet through openings 13 in the cylinder wall, and hence the pump chamber 7 is filled with oil at an oil feeding pressure at the fuel inlet. The openings 13 are manufactured as a single through bore extending through the pump cylinder perpendicular to the longitudinal axis of the pump cylinder. As explained above the pump chamber 7 is closed off at the end by the ball 8 which is seated against the rim of the end opening of pump chamber 7 by the action of spring 9. A ring-shaped end stop 24 has been press-fitted into end area of the central bore of the pump cylinder and acts as a spring guide.

In Fig. 2b the plunger has travelled to a position where the plunger 5 seals off the pump chamber 7 from the fuel inlet because plunger 5 closes the openings 13 in the cylinder wall in the instant the plunger head 10 is moved past the full opening 13, whereby the active stroke of plunger 5 begins. An amount of oil is hence captured in the pump chamber defined by the head 10 of the plunger, the inner surface of pump cylinder 6 and the ball 8, and further travelling of plunger 5 to the right in Fig. 2b cause compression and pressurization of the oil until the opening pressure of the check valve has been obtained and ball 8 is lifted from its valve seat defined by the rim of the end opening of pump chamber 7.

In Fig. 2c the plunger 5 has travelled further on to a position just before flow communication between the pump chamber 7 and the openings 13 is re-established via a pressure relief conduit generally designated by 22. The pressure relief conduit is in the embodiment of Fig. 2 arranged in the plunger and includes a first bore 15 extending coaxially in the longitudinal direction of plunger 5 and a transverse bore 16 debouching into an annular recess 23 in the plunger 5.

During the active stroke of the plunger the oil in the pump chamber is pressurized to a pressure exceeding the feeding pressure at the fuel inlet 12 and exceeding the opening pressure of the check valve because the force generated by the oil pressure on the ball 8 exceeds the closing force of spring 9. During the continued forward movement of the plunger the oil in pump chamber 7 flows out of the outlet 18, until the plunger has been moved through the distance L to the position illustrated in Fig. 2c where a forward cutting off edge 25 on the plunger is abreast of the rim of opening 13. Cutting off edge 25 is delimiting the annular recess 23. When the plunger is moved further forwards the recess 23 and thus the pressure relief conduit 22 is set in communication with opening 13, and fuel oil is caused to escape from the pump chamber 7 via the pressure relief conduit 22 to the opening 13. At this bleeding of the pump chamber 7 the pressure in the pump chamber 7 drops instantly to the pressure at the fuel inlet 12 with the consequence that ball 8 will be seated by the spring force from spring 9 and seal off the opening to outlet 18. At the instant of bleeding the pump chamber 7, the plunger is at the end of the active stroke in a distance d from the maximum stroke end stop position where the end surface of plunger 5 abuts the ball 8.

After bleeding of the pump chamber the plunger continues its forward movement due to inertia in the plunger, but at the same time the spring force from the first return spring causes the plunger to slow down until it obtains the most advanced position illustrated in Fig. 2d where the plunger is momentarily at a stand still and its movement is reversed due to the force from the first return spring 20. The figures show that only a small part of the distance d is utilized.

In the following description of further embodiments the same reference numerals as mentioned above are used for details of the same kind and function.

Fig. 3a to 3d illustrates a second embodiment where the pressure relief conduit 22 is formed partly in the pump cylinder 6 and partly in the plunger. The plunger has a recess 23 but no first bore. The head 10 of the plunger has a diameter adapted to the inner diameter of a shoulder 26 at the bottom of pump chamber 7. The shoulder 26 ends in a cutting off edge 27 where the diameter of the pump chamber is increased to be larger than the outer diameter of the plunger 5. The active stroke is initiated when the head 10 of the plunger cuts off opening 13 as illustrated in Fig. 2b. The active stroke continues until the forward cutting off edge 25 passes the cutting off edge 27 on shoulder 26. At this instant, the pressure relief conduit is established by the recess 23 setting the pump chamber 7 in communication with the opening 13 as illustrated in Fig. 3d.

Fig. 4 illustrates a further embodiment where the head 10 of the plunger 5 is not at right angles to the longitudinal axis of the plunger but extends obliquely at an acute angle α thereto. Another difference with respect to the embodiment of Figs. 2a-2d is that only a single opening 13 is provided in the pump cylinder 6. The angle α has been chosen so that the shortest distance q between the recess 23 and the head 10 corresponds to the diameter of the opening 13 and the longest distance between the recess and the head 10 at the opposite side of the plunger corresponds to the sum of the diameter of the opening 13 and the maximum desired active stroke length Lv. The oblique head 10 makes it possible to vary the active stroke length and thus the delivered amount of oil per stroke by rotating the plunger 7 in relation to the pump cylinder 6. The rotation of the plunger to a new setting of the delivered amount can be effected manually, but is preferably effected by a electro-mechanical setting device (not shown). The above description of the angle α allows a change of the delivery amount between zero (idling) and maximum. The plunger can be manufactured with other inclinations of the head 10, e.g. so that the distance q is smaller than the opening 13 which provides for idling when the plunger/pump cylinder are mutually rotated within a limited range of angles. A continuous change of the setting of the amount is possible with the oblique head 10. If a stepwise change between predefined delivery amounts is desired, the plunger can be manufactured with a stepped head, which brings the advantage that a very coarse rotational setting results in the desired amount.

In yet another embodiment depicted in Fig. 5 the pressure relief conduit 22 consists of the annular recess 23 and the first bore 15 which extends directly between head 10 and recess 23.

In an example of an embodiment the plunger diameter is 3 mm and the active stroke length L of the plunger 5 is 0.4 mm. The volume delivered per stroke of the plunger equals to 2.83 mm^3. In the embodiment the pump is driven by an electromagnet powered by a 50 Hz alternating current, and hence the output of the pump is approximately 0.5 l/h. In the embodiment, fuel oil is delivered to the liquid pump by gravity from a fuel tank located. The liquid pump delivers the fuel oil at a pressure of 0.15 bar. If a primer pump is used, excess oil delivered to the liquid pump can be returned to the oil tank.

In the above embodiment the pump is driven by an electromagnet. However, alternative plunger drive means can be used, such as a piezo-electric actuator.

The output of the pump may be adjusted by adjusting the frequency of activation of the plunger, e.g. by means of a frequency converter. As an alternative to using springs 14, 20 to control return movements of the plunger, an electromagnetic return device can be used, or the plunger can be provided with only the spring 20, and in this case the plunger 5 has to be mechanically connected to the armature 4.

Details from the different embodiments can be combined into further embodiments according to the present invention.

PATENT CLAIMS

1. A reciprocating liquid pump for delivery of liquid fuel to a domestic burner device, said pump comprising an electromechanical drive means which is electronically activated and acts on a plunger to reciprocate the plunger in a pump chamber of a pump cylinder, whereby liquid fuel is delivered from the pump, said plunger having a maximum stroke end stop position, characterized in that the pump comprises pressure relief means to bleed the pump chamber at the end of the active stroke at a distance from the maximum stroke end stop position.

2. A reciprocating liquid pump according to claim 1, characterized in that the plunger has a plunger diameter D and an active stroke length L, that the ratio L/D between the active stroke length L and the plunger diameter D is less than 0.4, and preferably less than 0.2.

3. A reciprocating liquid pump according to claim 1 or 2, characterized in that the full stroke length of the plunger during the reciprocating cycle is more than two times longer than the active stroke length, preferably longer than three times the active stroke length.

4. A reciprocating liquid pump according to claim 3, characterized in that the pressure relief means set the pump chamber in connection with a fuel inlet to the pump prior to and following the active stroke.

5. A reciprocating liquid pump according to any one of claims 1 to 4, characterized in that a first return spring acts on the plunger to retract it from the pump chamber towards a neutral position, and that preferably also a second return spring acts on the plunger to advance it in direction of the pump chamber towards said neutral position.

6. A reciprocating liquid pump according to any one of claims 1-5, c h a r a c t e r i z e d in that the plunger is activated by the electromechanical drive means with a frequency of at least 5 Hz.

7. A reciprocating liquid pump according to claim 6, c h a r a c t e r i z e d in that the plunger is activated by the plunger drive means with a frequency in the range from 50 to 60 Hz.

8. A reciprocating liquid pump according to any one of claims 1-7, c h a r a c t e r i z e d in that the electromechanical drive means can include an armature comprising ferromagnetic material and mounted in extension of the plunger, and an electromagnet arranged outside the armature.

9. A liquid pump according to any one of claims 1-8, c h a r a c t e r i z e d in that the pressure relief means include a pressure relief conduit arranged in the plunger and cooperating with a fuel inlet in the pump cylinder.

10. A reciprocating liquid pump according to any one of claims 1-9, c h a r a c t e r i z e d in that the pressure relief conduit comprises at least a first bore and an annular recess at the outer surface of the plunger, said first bore extending into the plunger from the pump chamber side thereof towards the annular recess.

11. A reciprocating liquid pump according to claim 10, c h a r a c t e r i z e d in that the first bore in the plunger extends in direction of the centre axis of the plunger and joins a transverse bore extending into an annular recess in the outer surface of the plunger, or the first bore extends obliquely to said centre axis directly into the annular recess in the outer surface of the plunger.

12. A reciprocating liquid pump according to any one of claims 1-11, c h a r a c t e r i z e d in that the pressure relief means includes a pressure relief passage arranged in the pump cylinder.

13. A reciprocating liquid pump according to any one of claims 1-12, c h a r a c t e r i z e d in that the liquid pump includes one or more pressure equalizing channels communicating the pressure at the fuel inlet to the drive end side of the plunger.

14. A reciprocating liquid pump according to any one of claims 1-13, c h a r a c t e r i z e d in that the liquid fuel delivery amount per activation of the plunger is in the range of 1 mm^3 to 50 mm^3, preferably in the range of 2 mm^3 to 15 mm^3.

15. A reciprocating liquid pump according to any one of claims 1-14, c h a r a c t e r i z e d in that the liquid fuel delivery pressure from the pump is in the range of from 0.01 bar to 3 bar above atmospheric pressure, preferably from 0.01 bar to 0.2 bar above atmospheric pressure.

16. A reciprocating liquid pump according to any one of claims 1-15, c h a r a c t e r i z e d in that the delivery amount of fuel from the pump is adjustable.

17. A reciprocating liquid pump according to claim 16, c h a r a c t e r i z e d in that the delivery amount is adjustable by turning the pump cylinder in relation to the plunger.

18. A reciprocating liquid pump according to claim 15, c h a r a c t e r i z e d in that the delivery amount is adjustable by changing the frequency of activation of the plunger.

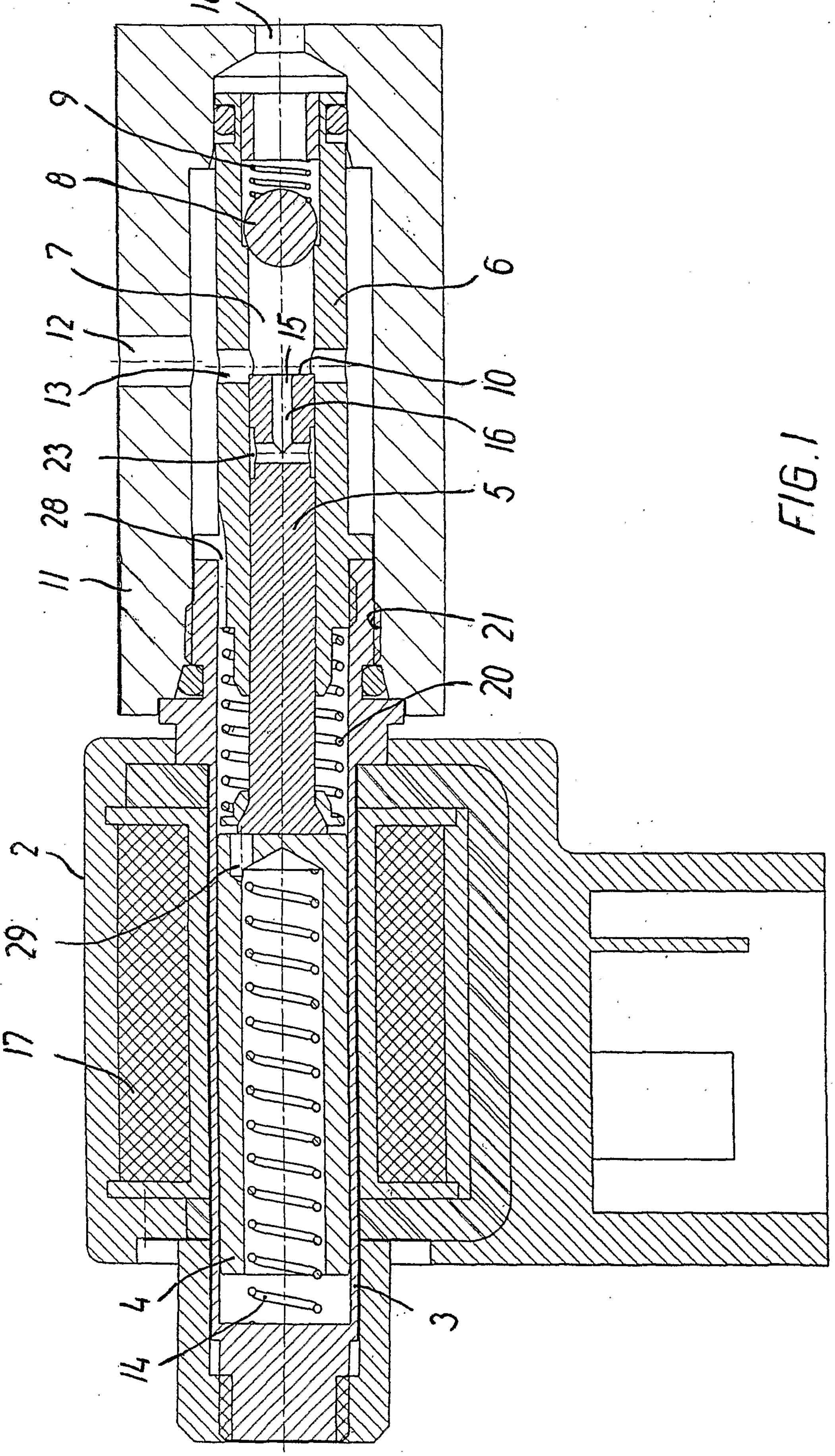

FIG. 1

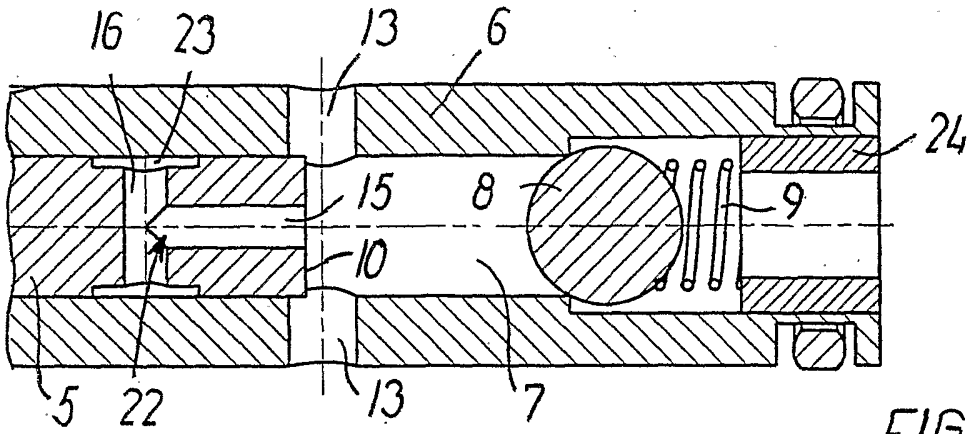

FIG.2a

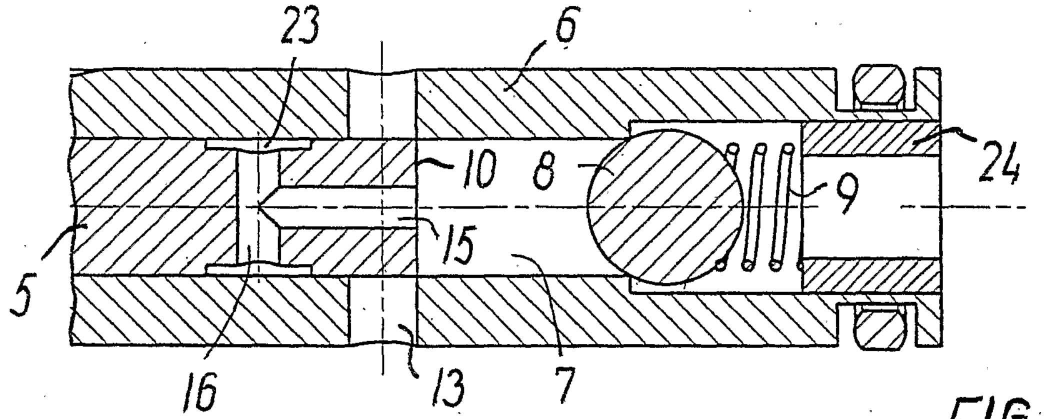

FIG.2b

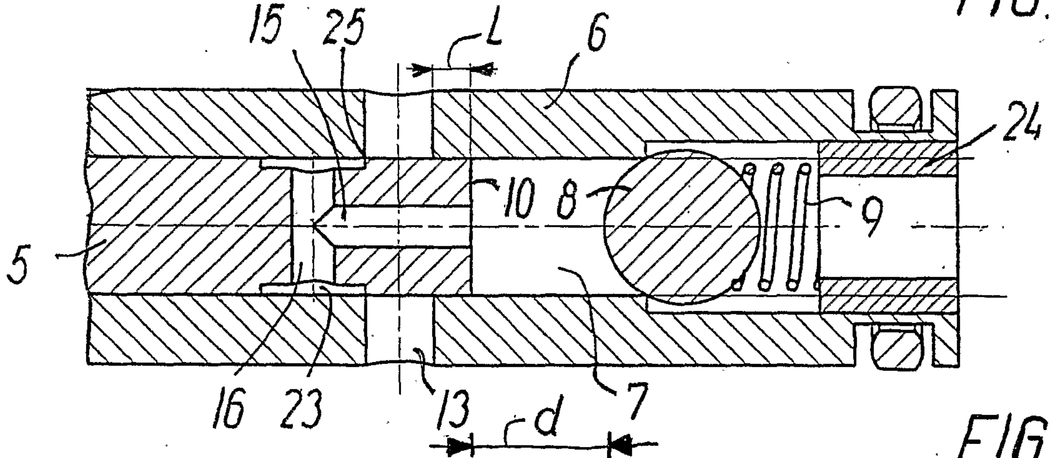

FIG.2c

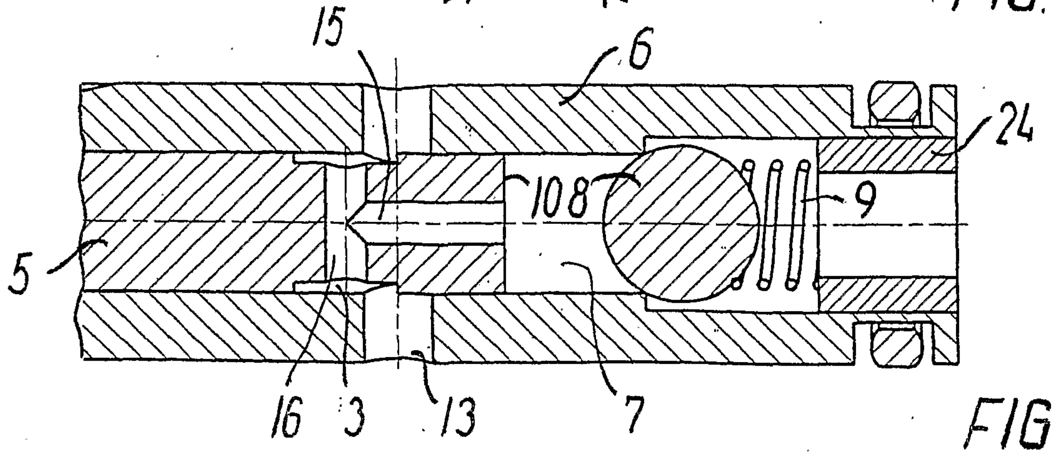

FIG.2d

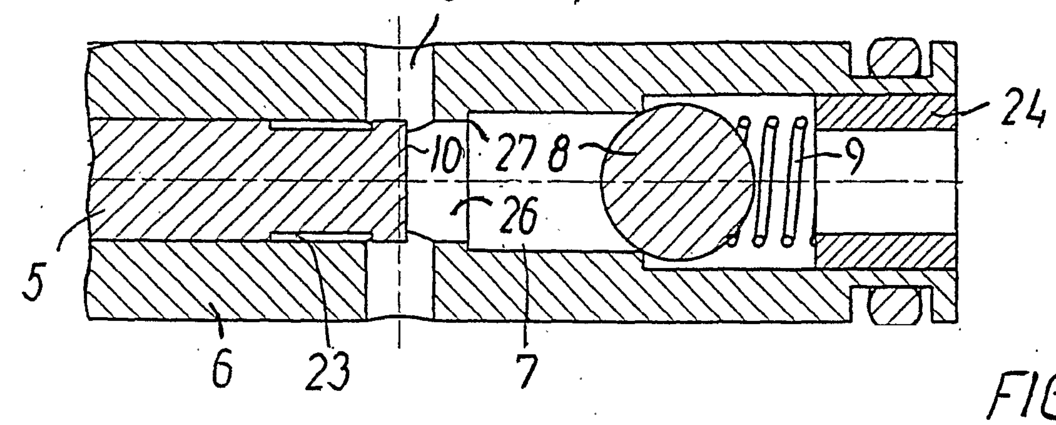

FIG.3a

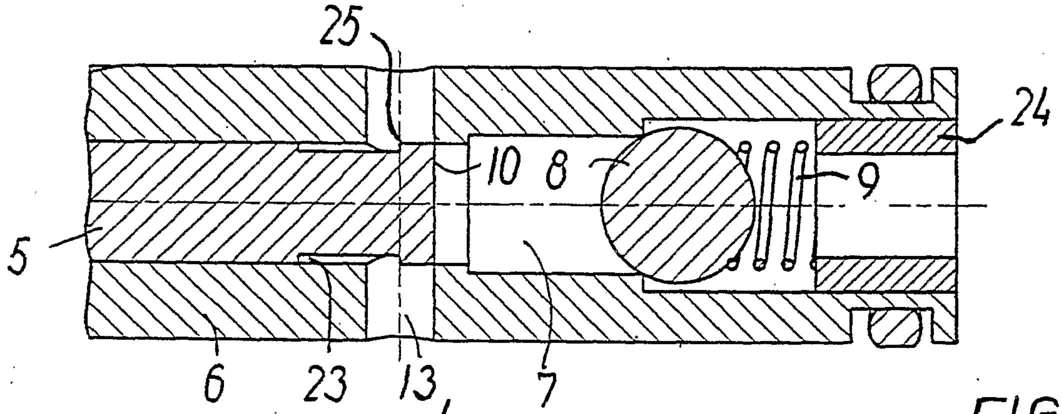

FIG.3b

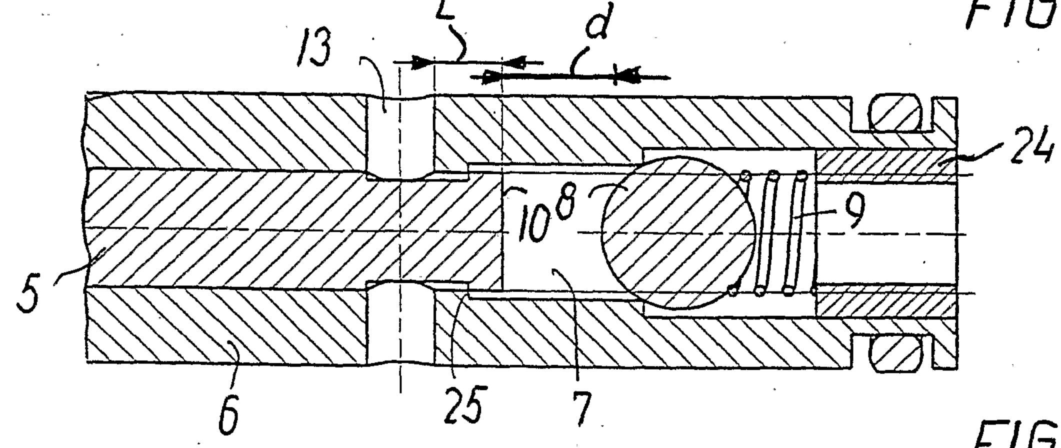

FIG.3c

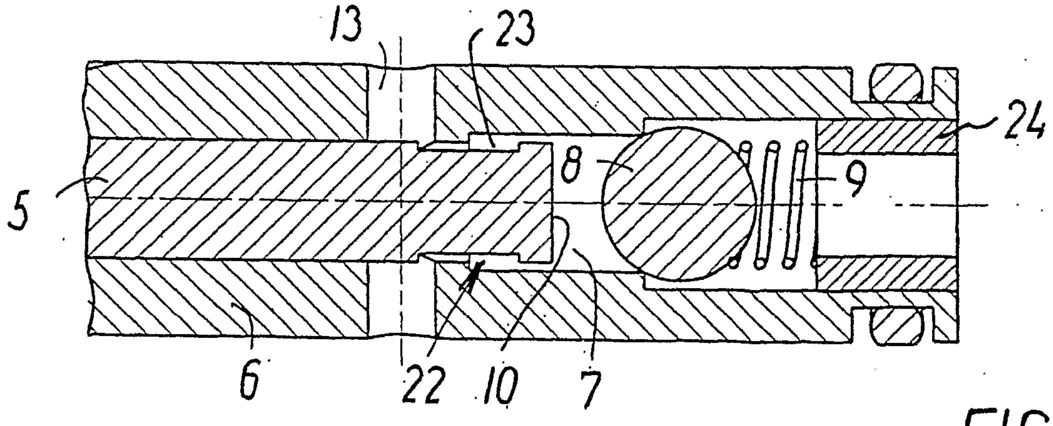

FIG.3d

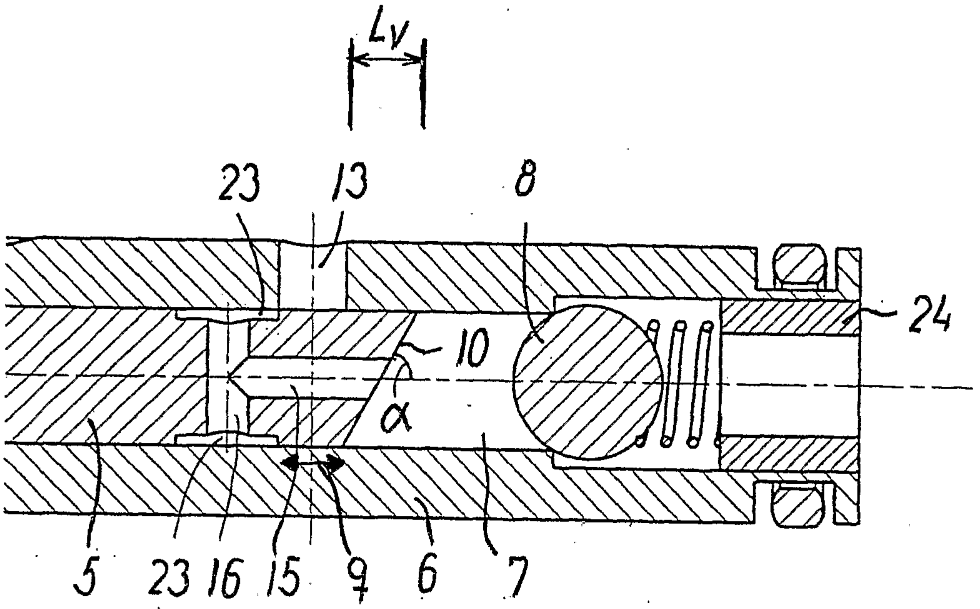

FIG. 4

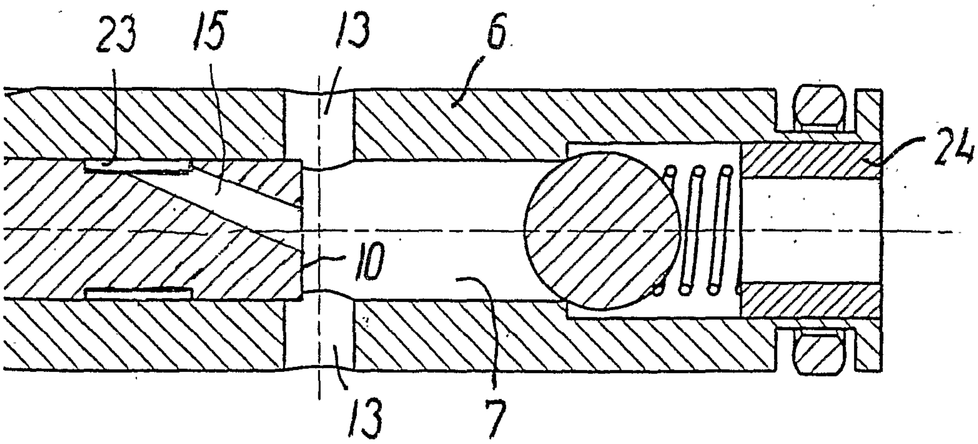

FIG. 5

INTERNATIONAL SEARCH REPORT

International Application No
PCT/DK 03/00732

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F04B17/04 F04B53/00 F23K5/14

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F04B F23K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal, WPI Data, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 4 352 645 A (MEYER JAMES H M) 5 October 1982 (1982-10-05) abstract; figure 1	1-18
A	US 2001/043873 A1 (ENOMOTO KIYOSHIGE ET AL) 22 November 2001 (2001-11-22) abstract; figures 1,3	1-18

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
25 February 2004	16. 03. 2004

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	LENA NILSSON/JAA

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/DK 03/00732

Patent document cited in search report		Publication date	Patent family member(s)			Publication date
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			CA	1142031	A1	01-03-1983
			DE	2929061	A1	31-01-1980
			FR	2431620	A1	15-02-1980
			JP	55017690	A	07-02-1980
			SE	7906157	A	19-01-1980
US 2001043873	A1	22-11-2001	JP	2002048073	A	15-02-2002