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(74) Representative: **Gregory, John David Charles**
Delphi Diesel Systems
Patent Department
Courteney Road
Gillingham
Kent ME8 0RU (GB)

(71) Applicant: **Delphi Technologies Holding S.à.r.l.**
4940 Bascharage (LU)

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(72) Inventor: **Hopley, Daniel**
Chislehurst, Kent BR7 5EG (GB)

(54) **Fluid Pump Assembly**

(57) A fluid pump assembly comprising a driven cam (18) and a reciprocating member (26, 28; 126, 226, 326; 32, 34) reciprocal within a bore provided in a pump housing (12, 30) as the cam (18) is driven, in use, to cause pressurisation of fluid within a pump chamber (33, 35). The pump assembly further comprises interface means (26a, 18c) between the cam (18) and the reciprocating member, preferably in the form of bevelled surfaces (18c, 26a, 28a; 32a, 34a) of the cam and the reciprocating member, which serve to drive the reciprocating member (26, 28) (i) to translate in a first, axial direction within the bore and (ii) to rotate within the bore in a second, rotational direction. A preferred feature of the fluid pump assembly is that the pump housing (12, 30) defines a bearing (22) for the cam (18) which is provided with a region of weakness (50) to allow the bearing (22) to deflect, in use, thereby to provide an increased lubrication volume between the cam and the bearing (22). The reciprocating member may take the form of a tappet (26, 28) which cooperates with a pumping plunger to pressurise fluid within the pump chamber (33, 35).

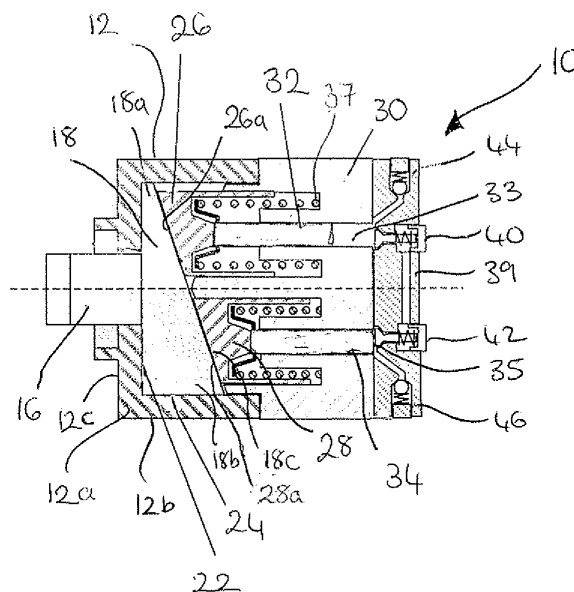


FIGURE 1

Description

TECHNICAL FIELD

[0001] The invention relates to a fluid pump assembly and, in particular, but not exclusively, to a pump assembly for fuel. The pump assembly is suitable for use in a common rail fuel injection system for supplying high pressure fuel to a compression ignition (diesel) internal combustion engine. In particular, the invention has application in a pump assembly of the type in which an engine driven cam imparts reciprocating, pumping motion to a drive member.

BACKGROUND TO THE INVENTION

[0002] One known common rail fuel pump is of radial pump design and includes three pumping plungers arranged at equi-angularly spaced locations around an engine driven cam - such a pump is described in, for example, WO 2004/104409. In this pump, each plunger is mounted within a plunger bore provided in a pump head mounted to a main pump housing. As the cam is driven in use, the plungers are caused to reciprocate within their bores in a phased, cyclical manner. As the plungers reciprocate, each causes pressurisation of fuel within a pump chamber defined at one end of the associated plunger bore. Fuel that is pressurised within the pump chambers is delivered to a common high pressure supply line and, from there, is supplied to a common rail or other accumulator volume, for delivery to the downstream injectors of the common rail fuel system. The fuel pump has an inlet valve for admitting fuel under low pressure and an outlet valve for letting out the pressurised fuel.

[0003] In this pump assembly, the cam carries a cam rider that extends co-axially with the drive shaft for the cam. The cam rider is provided with a plurality of flat surfaces ("flats"), one for each of the plungers. An intermediate drive member in the form of a tappet co-operates with the flat on the cam rider and couples to the plunger so that, as the tappet is driven upon rotation of the cam, drive is imparted to the plunger.

[0004] A fuel pump of radial pump design necessarily occupies a relatively high volume and, for some engine applications, this can be a disadvantage. Furthermore, the tappets are prone to wear due to the side loads experienced as they reciprocate, in use, and there can be significant damage to the tappet face that cooperates with the cam rider due to inadequate lubrication.

[0005] It is an object of the present invention to provide a fluid pump assembly which alleviates these problems when used to pump fuel in a fuel injection system.

SUMMARY OF THE INVENTION

[0006] According to a first aspect of the present invention, there is provided a fluid pump assembly comprising a driven cam and a reciprocating member reciprocal with

in a bore provided in a pump housing as the cam is driven, in use, to cause pressurisation of fluid within a pump chamber. The fluid pump assembly further includes interface means between the cam and the reciprocating member which cause the reciprocating member (i) to translate in a first, axial direction within the bore and (ii) to rotate within the bore in a second, rotational direction, as the cam is driven.

[0007] In one embodiment, the reciprocating member is an intermediate drive member, typically in the form of a tappet, which is cooperable with a pumping plunger to cause pressurisation of fluid within the pump chamber as the pumping plunger is driven by the intermediate drive member.

[0008] In another embodiment, the reciprocating member is a pumping plunger which interfaces directly with the cam.

[0009] The invention is particularly applicable to fuel injection systems for internal combustion engines in which a fuel pump assembly pressurises fuel to a relatively high pressure suitable for injection. Such a fuel pump assembly is particularly suitable for use in a common rail fuel injection system. However, the invention has wider application than fuel pumps for engines, and may be used as a pump for any other type of fluid also.

[0010] In a preferred embodiment, the interface means includes a bevelled face of the reciprocating member and a correspondingly bevelled face of the cam which cooperate so as to cause axial and rotational motion of the reciprocating member as the cam rotates. As the reciprocating member rotates within its bore, the constant relative velocity between the parts aids lubrication so as to reduce the effects of wear due to friction.

[0011] Preferably, the reciprocating member rotates at substantially the same angular velocity as the cam.

[0012] The fluid pump assembly preferably comprises an axial bearing for the cam which is defined by an axially-facing internal surface of the pump housing. The fluid pump assembly may further comprise a radial bearing for the cam which is defined by a radially-facing internal surface of the pump housing.

[0013] The cam may be provided with a low friction coating, for example a soft phosphate or PTFE coating, which deforms, in use, to the profile of the radial bearing. The profile of the coating on the cam being matched with the profile of the radial bearing provides good conditions for promotion of a hydrodynamic film.

[0014] The axial bearing may be provided with at least one recess to provide a volume for receiving lubricating fluid. The recess therefore provides for a supply of lubricating fluid to the axial bearing to aid lubrication between the rotating cam and the axial bearing.

[0015] Furthermore, the axial bearing may include an un-recessed area which defines a load bearing surface for the cam.

[0016] In a particularly preferred embodiment, the axial bearing is provided with a region of weakness to allow the axial bearing to deflect, in use, thereby to create an

increased volume for lubricating fluid between the axial bearing and the facing surface of the cam. Deflection of the axial bearing in this way opens up an enlarged gap between the cam and the axial bearing to encourage lubricating fluid to be drawn between the parts.

[0017] In a further preferred embodiment the axial bearing is further provided with a cut-away section to define a lead-in edge for lubricant drawn between the axial bearing and the facing surface of the cam.

[0018] The fluid pump assembly may comprise at least two intermediate drive members (e.g. tappets) and at least two pumping plungers, each of the intermediate drive members being cooperable with a respective one of the plungers and each of the intermediate drive members being cooperable with a cam common to all intermediate drive members. In one embodiment, for example, the fluid pump assembly includes three intermediate drive members and three pumping plungers, associated pairs of the drive members and the pumping plungers being arranged at equi-angularly spaced locations about a central pump axis. In an embodiment in which the reciprocating members are pumping plungers which interface directly with the cam, the pumping plungers are arranged at equi-angularly spaced locations about the central pump axis.

[0019] In one embodiment, the pump chambers are defined within the pump housing and are closed by a plate mounted to the pump housing. Alternatively, the pump chambers may be defined entirely within the pump housing.

[0020] Depending on the nature of the drive through which the engine is coupled to the drive shaft, an output end of the drive shaft may extend rearward of the cam and act against a bearing defined by the pump housing so as to counter side loads applied to an input end of the drive shaft. Such an arrangement is particularly suitable for belt, chain or gear drive applications where the nature of the input drive causes side loads to be imparted to the drive shaft.

[0021] According to a second aspect of the invention, there is provided a fluid pump assembly comprising a driven cam and a reciprocating member reciprocal within a bore provided in a pump housing as the cam is driven, in use, so as to cause pressurisation of fluid within a pump chamber. The reciprocating member includes a bevelled face which cooperates with a correspondingly bevelled face of the cam so as to impart drive to the reciprocating member as the cam rotates.

[0022] In the second aspect of the invention, the reciprocating member is preferably driven both axially and rotationally within the bore.

[0023] According to a third aspect of the invention, a fluid pump assembly comprises a driven cam and a reciprocating member reciprocal within a bore provided in a pump housing as the cam is driven, in use, so as to cause pressurisation of fluid within a pump chamber. The pump housing defines a bearing for the cam which is provided with a region of weakness to allow the bearing

to deflect, in use, thereby to provide an increased lubrication volume between the cam and the bearing. Preferably it is an axial bearing defined by the pump housing that is provided with the region of weakness. Such an arrangement provides the aforementioned advantages for lubrication between the rotating cam and the axial bearing.

[0024] It will be appreciated that preferred and/or optional features of the first aspect of the invention, as set out above and in the dependent claims, may be included in the second or third aspects of the invention also, alone or in appropriate combination.

BRIEF DESCRIPTION OF THE DRAWINGS

[0025] The invention will now be described, by way of example only, within reference to the following drawings in which:

Figure 1 is a cross sectional view of a fuel pump assembly of a first embodiment of the invention, having two pumping plungers;

Figure 2 is a cross sectional view of a part of the fuel pump assembly in Figure 1 to illustrate a spring seat for a return spring;

Figure 3a is a cross sectional view of a cam and a pump housing of the fuel pump assembly in Figure 1;

Figure 3b is an end view of an axial bearing defined by the pump housing in Figure 3a;

Figure 4a is a cross sectional view of the pump housing in Figure 3a to illustrate an area of weakness on the external surface;

Figure 4b is an end view of the internal surface of the pump housing in Figure 4a;

Figure 5 is cross sectional view of a fuel pump assembly of a second embodiment of the invention having two pumping plungers;

Figure 6 is a cross sectional view of a fuel pump assembly of a third embodiment of the invention having two pumping plungers;

Figure 7 is a cross sectional view of a fuel pump assembly of a fourth embodiment of the invention having a single pumping plunger;

Figure 8 is a perspective view of a part of a fuel pump assembly of a fourth embodiment of the invention having three intermediate drive members for three pumping plungers; and

Figure 9 is a cross sectional view of a fuel pump

assembly of a fifth embodiment of the invention in which the intermediate drive members of previous embodiments are removed.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

[0026] Referring to Figure 1, a first embodiment of the fuel pump assembly 10 of the invention includes a pump housing having a first housing part 12 which is provided with a central bore for receiving a drive shaft 16 (only a part of which is shown). The first housing part includes a front plate 12a of the pump housing and a cylindrical body 12b towards the rear. The rear end of the drive shaft 16 carries a cam 18 which rotates with the drive shaft 16, in use. Typically, the front or input end of the drive shaft is driven by the engine through an Oldham coupling, as would be familiar to a person skilled in the art.

[0027] The cam 18 is wedge-shaped and so has a thin end 18a and a thick end 18b with a bevelled contact surface 18c on its front face. The back face of the cam 18 is planar and acts against an axially-facing internal surface of the pump housing 12, which therefore acts as an axial bearing 22 for the cam 18 as it rotates. The outer surface of the cam, at its thick end 18b, bears against a radially-facing internal surface of the first housing part 12, which therefore acts as a radial bearing 24 for the cam 18 as it rotates.

[0028] The pump assembly includes first and second reciprocating members, in the form of tappets 26, 28, each of which has a bevelled surface 26a, 28a, respectively, for contact with the correspondingly bevelled surface 18c of the cam 18. Each tappet 26, 28 is received within an associated tappet bore provided in a second housing part 30 mounted to the first housing part 12, and is coupled to an associated pumping plunger, 32, 34 respectively, in axial alignment with its tappet 26, 28. The tappets 26, 28 therefore form an intermediate drive member between the cam 18 and the associated plunger 32, 34.

[0029] Each pumping plunger 32, 34 is received within an associated plunger bore provided in the second housing part 30. An end of the pumping plunger 32, 34 remote from the tappet 26, 28 defines an internal surface of a pump chamber 33, 35 which receives fuel to be pressurised during a plunger pumping stroke, in use, as described in further detail below.

[0030] Referring also to Figure 2 (which shows only the first tappet 26), each tappet 26 takes the form of a bucket tappet of generally U-shaped cross section having a base 26b, which defines the bevelled contact surface 26a, and a cylindrical upper body 26c. Within the internal volume of the tappet 26, on the side of the base 26b opposed to the bevelled contact surface 26a, the tappet includes a projection 26d which defines a contact surface for the associated plunger 32. A spring seat assembly for a plunger return spring 37 is received within the internal volume of the tappet 26 defined within the cylindrical

upper body 26c. The return spring 37 serves to provide a return load to the plunger 30 and the tappet 26 to effect a return stroke of the plunger, as described in further detail below.

[0031] The spring seat assembly has two parts. A first part 36 is of top-hat construction and is located at the base of the plunger 32, the plunger 32 extending through a central bore of the first part 36. The first part 36 defines an abutment surface for one end of the return spring 37, the other end of the return spring 37 remote from the spring seat assembly 36, 38 abutting an internal surface 41 of the second housing part 30. A second part 38 of the spring seat assembly is an annular piece forms a push-fit on the base end of the plunger 32 and serves to retain the first part 36 of the assembly in place.

[0032] In an alternative embodiment (not shown), the return spring 37 may be a smaller component located within the pump chamber 33, 35, rather than surrounding the plunger 32, 34.

[0033] Referring again to Figure 1, the pump chambers 33, 35 are closed by a plate 39 at the rear end of the pump assembly 10. The closure plate 39 is provided with a plurality of drillings to allow relatively low pressure fuel to be conveyed into the pump chambers 33, 35 and to allow pressurised fuel to be conveyed from the pump chambers 33, 35 to a pump outlet (not shown). An inlet drilling is provided for each of the pump chambers 33, 35, each inlet drilling having a respective spring-biased inlet valve 40, 42 through which relatively low pressure fuel passes to enter the associated pump chamber 33, 35, prior to pressurisation. An outlet drilling is provided for each of the pump chambers 33, 35, each outlet drilling having a respective spring-biased outlet valve 44, 46 through which pressurised fuel is delivered to the common outlet of the pump assembly when the pressure level in the pump chambers 33, 35 reaches a predetermined amount. The common outlet is connected to a common rail or accumulator volume of the fuel injection system, from where fuel is delivered to the fuel injectors of the engine.

[0034] Operation of the fuel pump assembly will now be described in further detail.

[0035] Considering the first tappet 26 and its associated plunger 32, as the drive shaft 16 rotates, in use, cooperation between the rotating bevelled surface 18c of the cam 18 and the bevelled surface 26a of the tappet 26 results in the tappet 26 reciprocating axially within its tappet bore and, thus, the plunger 32 is caused to reciprocate within its plunger bore also. As the plunger 32 is driven it performs the pumping stroke, in which fuel within the associated pump chamber 33, 35 is pressurised to a high level suitable for injection, followed by the return stroke which is effected by means of the associated return spring 37.

[0036] At the start of the return stroke, the outlet valve 44 is closed under its spring force. As the plunger 32 moves outwardly from its bore to expand the volume of the pump chamber 33, the pump chamber 33 fills with

fuel at relatively low pressure from a supply pump (e.g. transfer pump) through the inlet valve 40 which is open. As the cam 18 continues to rotate and the plunger 32 completes its return stroke, cooperation between the bevelled surfaces 18c, 26a of the cam and the tappet causes the tappet, and hence the plunger, to move inwardly within their bores to reduce the volume of the pump chamber 33. Soon after the volume of the pump chamber 33 starts to decrease, fuel pressure in the pump chamber 33 starts to increase and the force due to fuel pressure acting on the inlet valve 40 causes it to close. The pressure within the pump chamber 33 continues to rise as the plunger 32 continues through its pumping stroke, until such time as the pressure in the pump chamber 33 is sufficient to overcome the closing force of the outlet valve 44, which is then urged open to allow pressurised fuel to be delivered through the pump outlet.

[0037] As a result of the rotating bevelled surface 18c of the cam 18 interacting with the correspondingly bevelled surface 26a of the tappet 26, the tappet is driven to move axially within its bore, hence driving axial motion of the plunger. Importantly, cooperation between the rotating bevelled surface 18c of the cam 18 and the correspondingly bevelled surface 26a of tappet 26 also means that the tappet is driven to rotate within its bore at the same angular velocity at which the cam 18 is driven by the drive shaft 16. The interface between the cam and the tappet therefore results in a deliberately driven, continuous rotation of the tappet about its axis.

[0038] Due to the nature of the two-part spring seat assembly, rotation of the tappet 26 also causes the plunger 32 to rotate within the plunger bore as it reciprocates. The spring seat assembly is configured such that the frictional force between the return spring 37 and the first part 36 of the spring seat assembly is greater than the frictional force between the second and first parts 38, 36 of the spring seat assembly. Hence, as the tappet 26 rotates, the plunger 32 may also rotate, whereas the first part 36 of the spring seat assembly and the return spring 37 remain static. In this way relative movement between the end of the return spring 37 and the internal surface 41 of the second housing part 30 is prevented, to avoid unwanted wear, whilst the plunger 32 is allowed to rotate. Unwanted relative movement between the first part 36 of the spring seat assembly and the return spring 37 is also avoided.

[0039] The second tappet 28 and the second plunger 34 are driven in a similar manner to operate in phased, cyclical motion with the first tappet/plunger 26/32, with both pump chambers 33, 35 filling a common rail with pressurised fuel through the respective outlet valves 44, 46.

[0040] A clearance between each tappet and its tappet bore provides a volume for lubricating fluid and so, due to the relative motion between the rotating tappet and its bore, lubrication of parts is promoted to reduce wear.

[0041] As illustrated in Figure 3(a), the return load on the cam 18 due to pressurised fuel within the pump cham-

ber 33 is exerted on the cam 18 in a direction perpendicular to the bevel angle of the cam surface 18c. The thick end 18b of the cam 18 bears against the radially-facing internal surface of the first housing part 12, which therefore acts as a radial bearing 24 for the cam 18 as it rotates. The rear face 18d of the cam (i.e. the face opposed to the bevelled surface 18c) bears against the axially-facing internal surface of the pump housing 12, which therefore acts as an axial bearing 22 for the cam 18 as it rotates.

[0042] Figure 3(b) illustrates a coating that is applied to the radially-facing surface of the cam 18. It is preferable for the surfaces of the cam 18 which bear against the axial and radial bearings 22, 24 to be provided with a soft lubricating coating, for example phosphate or PTFE. The dashed line illustrates the profile of the coating 25 on the cam 18, in use. As the coating 25 is soft, the coating deforms as the cam 18 rotates so as to conform to the profile of the bearing surface 24, hence providing good conditions for promotion of a hydrodynamic film. The soft phosphate coating is also applied to the bevelled face 18c of the cam 18 which cooperates with the bevelled surface 26a of the tappet.

[0043] Referring to Figures 4(a) and 4(b), the axial bearing 22 is modified, on its front and rear faces, so as to aid lubrication between the parts 12, 18. Firstly, as shown in Figure 4(b), the axial bearing 22 includes first and second raised segments 46a, 46b, or pads, separated by first and second recessed segments 48a, 48b. The raised segments 46a, 46b are positioned so as to be axially aligned with a respective one of the tappets 26, 28 so as to absorb the tappet return load. The recesses 48a, 48b define an enlarged volume for lubricating fluid to aid lubrication between the cam 18 and the bearing 22 as the cam rotates.

[0044] In addition, and as can be seen in Figure 4(a), the opposite face 12c of the first housing part 12 to the axial bearing 22 is provided with a further recess 50 to define a weakened region of the first housing part 12. As the pump is driven and the cam 18 is loaded by the tappet and bears on the axial bearing 22, the weakened region of the pump housing 12 allows the housing to deflect causing a wedge-shaped gap (not shown) to open between the axial bearing 22 and the facing surface 18d of the cam 18. This provides a lead-in edge for lubricating fluid and allows fluid to be drawn between the parts 12, 18, allowing a hydrodynamic bearing to be generated between them as the cam 18 rotates.

[0045] In addition to the lead-in edge provided by deflection of the pump housing 12, the axial bearing 22 may also be provided with a chamfer, radius or bevel (not shown) at the lead-in edge to further encourage lubricating fluid to be drawn between the parts 12, 18 as the cam rotates.

[0046] In Figure 1, where an Oldham coupling is provided between the engine and the drive shaft 16, there are only an insignificant side loads on the drive shaft 16. However, where a belt, chain or gear drive is used between the engine and the drive shaft 16, a significant side

load is exerted on the drive shaft which causes unwanted tilt and translation forces to act on the cam 18. For belt, chain or gear drive applications it is therefore necessary to counter these side loads to prevent unwanted translation and/or tilt of the cam 18 by providing a different bearing arrangement to that shown in Figure 1.

[0047] Figure 5 shows an embodiment of the invention which is appropriate for a belt, chain or gear drive coupling (not shown) between the engine and the drive shaft 16. Similar parts to those shown in Figures 1 to 4 are denoted with like reference numerals. In this embodiment a rear or output end of the drive shaft 16 extends further rearward into the pump assembly 10, and beyond the bevelled contact face 18c of the cam 18, to be received within a central bore provided in the second housing part 30. At the rearmost end of the drive shaft 16 the internal surface of this central bore defines a radial bearing 52 which counters the tilting force acting on the front end of the drive shaft 16 and, hence, prevents unwanted tilt of the cam 18 as it rotates. In addition, the thin end 18a of the cam 18 is provided with an axially-extending flange 54, the outer surface of which bears on the radially-facing internal surface 56 of the pump housing 12. The axially-extending flange 54 bearing against the radially-facing internal surface 56 of the pump housing 12 counters the translation force acting on the front end of the drive shaft 16 and, hence, prevents unwanted translation of the cam 18 as it rotates. The bearing arrangement 52, 54, 56 of Figure 5 therefore prevents unwanted tilting and translation of the cam 18 due to side loading of the drive shaft 16 at its front end.

[0048] It will be appreciated that the pump assembly in Figure 5 is of greater width than that in Figure 1 due to the need for the drive shaft 16 to extend further rearward into the pump housing 12, 30, and beyond the cam 18, to define the rear bearing 52, and hence the need for separation between the tappets 26, 28 to be greater. The width is also increased due to the provision of the flange 54 on the cam 18.

[0049] In the Figure 5 embodiment, the radially-facing internal surface of the pump housing provides a bearing surface 24 for the wide end 18b of the cam 18 and the axially-facing internal surface 22 of the pump housing 12 defines a bearing surface for the front face of the cam 18, as in the Figure 1 embodiment.

[0050] Another alternative bearing arrangement suitable for use with a belt, chain or gear drive is shown in Figure 6. As in Figure 5, the drive shaft 16 extends further into the second housing part 30 and beyond the cam 18 so as to define a radial bearing 52 at the rear end of the drive shaft 16 which counters the tilting force applied to the cam 18 due to the side loads at the front end of the drive shaft 16. In this case, however, the flange on the thin end 18a of the cam 18 is removed, and instead the cam 18 is made of increased thickness in this region 18a' (i.e. the length of the cam along the axis of the drive shaft is increased at its thinnest end). The thin end 18a' of the cam 18 is therefore of greater thickness than in Figure 5

and bears against the radially-facing internal surface 56 of the first housing part 12 to counter the translation force acting on the cam 18 due to the side loading at the front end of the drive shaft 16. This arrangement results in a pump assembly of longer axial length than the Figure 5 embodiment due to the increased thickness of the cam 18 at region 18a', but one of reduced width due to the removal of the flange 54 in the Figure 5 embodiment.

[0051] A further alternative embodiment is shown in Figure 7 which, again, is appropriate for use with a belt, chain or gear drive. Here, the drive shaft 16 does not extend rearward beyond the cam 18, but instead the wide end 18b of the cam is provided with a radially-extending flange 58 which engages with a radially-extending, axially-facing surface 60 of the second housing part 30. The bearing provided by the radially-extending surface 60 of the second housing part 30 counters both the tilting and translation forces exerted on the cam 18 due to the side loads at the input end of the drive shaft 16.

[0052] Although the pump housing in Figure 7 is still of two-part construction, the first housing part 12 which defines the axial bearing 22 is a much smaller component than in previous embodiments, with the second housing part 30 extending further towards the front end of the pump assembly 10 to define the bearing surface 60. The radially-extending flange 58 bears against the bearing surface 60 which counters the side loads on the input end of the drive shaft. The bearing surface 60 therefore takes the place of the bearing surfaces 52, 56 in Figures 1, 5 and 6.

[0053] Another difference between the embodiment in Figure 7 and those described previously is that in Figure 7 there is only a single pumping plunger 32 having a single associated tappet 26 cooperating with the bevelled cam 18. In practice, the pump assembly may include any number of plungers/tappets, depending on delivery requirements. As shown in Figure 8, for example, the pump assembly may include three plungers (not shown), each having an associated tappet 126, 226, 326 which cooperates with a common bevelled cam 18. In a tri-tappet assembly the tappets 126, 226, 326, and their associated plungers, are arranged at equi-angularly spaced locations around a central axis of the pump assembly which is aligned with the drive shaft axis.

[0054] Another arrangement of the bearings (not shown) involves removing the flange 58 in the Figure 7 embodiment, and creating a bearing between a flat portion 18e of the front face of the cam (i.e. a portion that isn't bevelled) and the facing surface of the second housing part 30. In this embodiment, the pump assembly has two axial bearings (at 22 and 18e), facing in opposite directions, to counter the side loads on the input end of the drive shaft 16.

[0055] In another example, as shown in Figure 9, the tappets 26, 26, 126, 226, 326 of previous embodiments may be removed altogether and the bevelled surface 18c of the cam 18 may act directly on a correspondingly bevelled surface 32a, 34a of the reciprocating plungers 32,

34. In other arrangements a single plunger, or more than two plungers, may be provided to interface directly with the bevelled cam 18, again avoiding the need for an intermediate drive member.

[0056] Other embodiments of the invention are also envisaged without departing from the scope of the invention as set out in the claims. For example, the rear closure plate 39 in Figures 1, 5, 6, 7 and 9 may be replaced by a housing part (not shown) which includes regions extending into the second housing part 30 so as to define the plunger sealing lengths and the pump chambers 33, 35. In this way the main pump housing 30 does not have to have the required material strength to accommodate the high pressures of fuel within the pump chambers 33, 35, and only the closure plate 39 needs to be made from high-strength, expensive material.

Claims

1. A fluid pump assembly comprising:

a driven cam (18);
a reciprocating member (26, 28; 126, 226, 326; 32, 34) reciprocal within a bore provided in a pump housing (12, 30) as the cam (18) is driven, in use, to cause pressurisation of fluid within a pump chamber (33, 35); and
interface means (26a, 28a; 32a, 34a; 18c) between the cam (18) and the reciprocating member which cause the reciprocating member (i) to translate in a first, axial direction within the bore and (ii) to rotate within the bore in a second, rotational direction, as the cam is driven.

2. A fluid pump assembly as claimed in claim 1, wherein the reciprocating member is an intermediate drive member (26, 28; 126, 226, 326) which is cooperable with a pumping plunger (32, 34) to cause pressurisation of fluid within the pump chamber (33, 35) as the pumping plunger (32, 34) is driven by the intermediate drive member (26, 28).

3. A fluid pump assembly as claimed in claim 1, wherein the reciprocating member is a pumping plunger (32, 34) which interfaces directly with the cam (18).

4. A fluid pump assembly as claimed in any of claims 1 to 3, wherein the reciprocating member (26, 28; 126, 226, 326; 32, 34) rotates at substantially the same angular velocity as the cam (18).

5. A fluid pump assembly as claimed in any of claims 1 to 4, wherein the interface means (26a, 18c) includes a bevelled face of the reciprocating member (26, 28) and a correspondingly bevelled face (18c) of the cam which cooperate so as to impart axial and rotational motion to the reciprocating member (26,

28) as the cam rotates.

6. A fluid pump assembly as claimed in any of claims 1 to 5, further comprising (i) an axial bearing (22) for the cam defined by an axially-facing internal surface of the pump housing (12) and/or (ii) a radial bearing (24) for the cam defined by a radially-facing internal surface of the pump housing (12).

7. A fluid pump assembly as claimed in claim 6, further comprising a coating (25) applied to the surface of the cam which deforms, in use, to the profile of the radial bearing (24).

8. A fluid pump assembly as claimed in claim 6 or claim 7, wherein the axial bearing (22) is provided with at least one recess (48a, 48b) to provide a volume for receiving lubricating fluid.

9. A fluid pump assembly as claimed in any of claims 6 to 8, wherein the axial bearing (22) includes an unrecessed area (46a, 46b) which defines a load bearing surface for the cam (18).

10. A fluid pump assembly as claimed in any of claims 6 to 9, wherein the axial bearing (22) is provided with a region of weakness (50) to allow the axial bearing (22) to deflect, in use, thereby to encourage lubricating fluid to be drawn between the axial bearing (22) and the cam (18) as it rotates.

11. A fluid pump assembly as claimed in any of claims 6 to 10, wherein the axial bearing (22) is provided with a cut-away section to define a lead-in edge for lubricant drawn between the axial bearing (22) and the cam (18) as it rotates.

12. A fluid pump assembly as claimed in any of claims 1 to 11, wherein either the pump chambers are defined within the pump housing (30) and are closed by a plate (39) mounted to the pump housing (30), or wherein the pump chambers are defined entirely within the pump housing (30).

13. A fluid pump assembly as claimed in any of claims 1 to 12, wherein an output end of the drive shaft (16) extends rearward of the cam (18) and acts against a bearing (52) defined by the pump housing (30) so as to counter side loads applied to an input end of the drive shaft (16).

14. A fluid pump assembly comprising:

a driven cam (18); and
a reciprocating member (26, 28; 126, 226, 326; 32, 34) reciprocal within a bore provided in a pump housing (12, 30) as the cam (18) is driven, in use, to cause pressurisation of fluid within a

pump chamber (33, 35);

wherein the reciprocating member (26, 28) includes a bevelled face (26a) which cooperates with a correspondingly bevelled face (18c) of the cam so as to impart drive to reciprocating member (26, 28) as the cam rotates.

15. A fluid pump assembly comprising:

a driven cam (18); and
a reciprocating member (26, 28; 126, 226, 326; 32, 34) reciprocal within a bore provided in a pump housing (12, 30) as the cam (18) is driven, in use, to cause pressurisation of fluid within a pump chamber (33, 35);

wherein the pump housing (12, 30) defines a bearing (22) for the cam (18) which is provided with a region of weakness (50) to allow the bearing (22) to deflect, in use, thereby to provide an increased lubrication volume between the cam and the bearing (22).

Amended claims in accordance with Rule 137(2) EPC.

1. A fluid pump assembly comprising:

a driven cam (18);
a reciprocating member (26, 28; 126, 226, 326; 32, 34) reciprocal within a bore provided in a pump housing (12, 30) as the cam (18) is driven, in use, to cause pressurisation of fluid within a pump chamber (33, 35); and
interface means (26a, 28a; 32a, 34a; 18c) between the cam (18) and the reciprocating member which cause the reciprocating member to be driven so as to translate in a first, axial direction within the bore and driven to rotate continuously about its own axis within the bore in a second, rotational direction, as the cam is driven.

2. A fluid pump assembly as claimed in claim 1, wherein the reciprocating member is an intermediate drive member (26, 28; 126, 226, 326) which is co-operable with a pumping plunger (32, 34) to cause pressurisation of fluid within the pump chamber (33, 35) as the pumping plunger (32, 34) is driven by the intermediate drive member (26, 28).

3. A fluid pump assembly as claimed in claim 1, wherein the reciprocating member is a pumping plunger (32, 34) which interfaces directly with the cam (18).

4. A fluid pump assembly as claimed in any of claims 1 to 3, wherein the reciprocating member (26, 28;

126, 226, 326; 32, 34) rotates at substantially the same angular velocity as the cam (18).

5. A fluid pump assembly as claimed in any of claims 1 to 4, wherein the interface means (26a, 18c) includes a bevelled face of the reciprocating member (26, 28) and a correspondingly bevelled face (18c) of the cam which cooperate so as to impart axial and rotational motion to the reciprocating member (26, 28) as the cam rotates.

6. A fluid pump assembly as claimed in any of claims 1 to 5, further comprising (i) an axial bearing (22) for the cam defined by an axially-facing internal surface of the pump housing (12) and/or (ii) a radial bearing (24) for the cam defined by a radially-facing internal surface of the pump housing (12).

7. A fluid pump assembly as claimed in claim 6, further comprising a coating (25) applied to the surface of the cam which deforms, in use, to the profile of the radial bearing (24).

8. A fluid pump assembly as claimed in claim 6 or claim 7, wherein the axial bearing (22) is provided with at least one recess (48a, 48b) to provide a volume for receiving lubricating fluid.

9. A fluid pump assembly as claimed in any of claims 6 to 8, wherein the axial bearing (22) includes an unrecessed area (46a, 46b) which defines a load bearing surface for the cam (18).

10. A fluid pump assembly as claimed in any of claims 6 to 9, wherein the axial bearing (22) is provided with a recess to define a region of weakness (50) to allow the axial bearing (22) to deflect, in use, thereby to encourage lubricating fluid to be drawn between the axial bearing (22) and the cam (18) as it rotates.

11. A fluid pump assembly as claimed in any of claims 6 to 10, wherein the axial bearing (22) is provided with a cut-away section to define a lead-in edge for lubricant drawn between the axial bearing (22) and the cam (18) as it rotates.

12. A fluid pump assembly as claimed in any of claims 1 to 11, wherein either the pump chambers are defined within the pump housing (30) and are closed by a plate (39) mounted to the pump housing (30), or wherein the pump chambers are defined entirely within the pump housing (30).

13. A fluid pump assembly as claimed in any of claims 1 to 12, wherein an output end of the drive shaft (16) extends rearward of the cam (18) and acts against a bearing (52) defined by the pump housing

(30) so as to counter side loads applied to an input end of the drive shaft (16).

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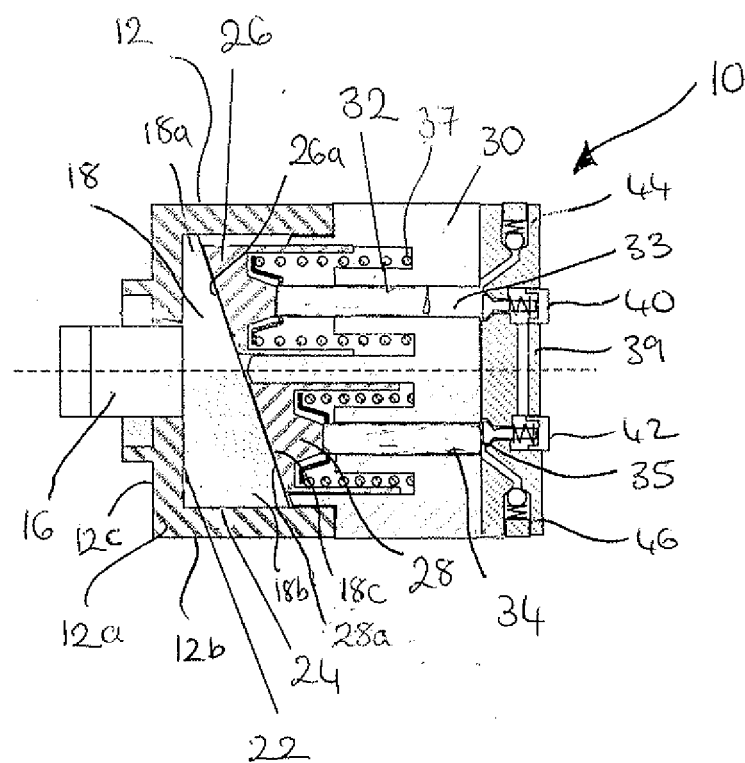


FIGURE 1

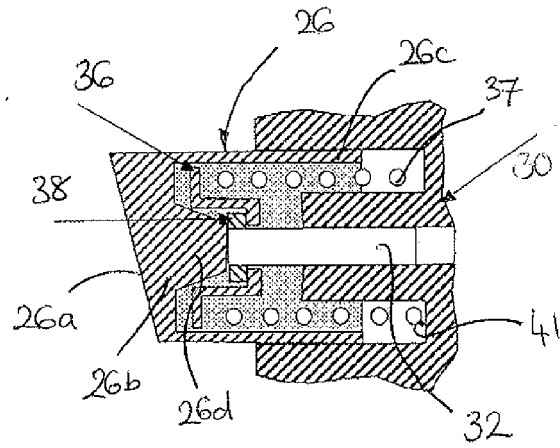


FIGURE 2

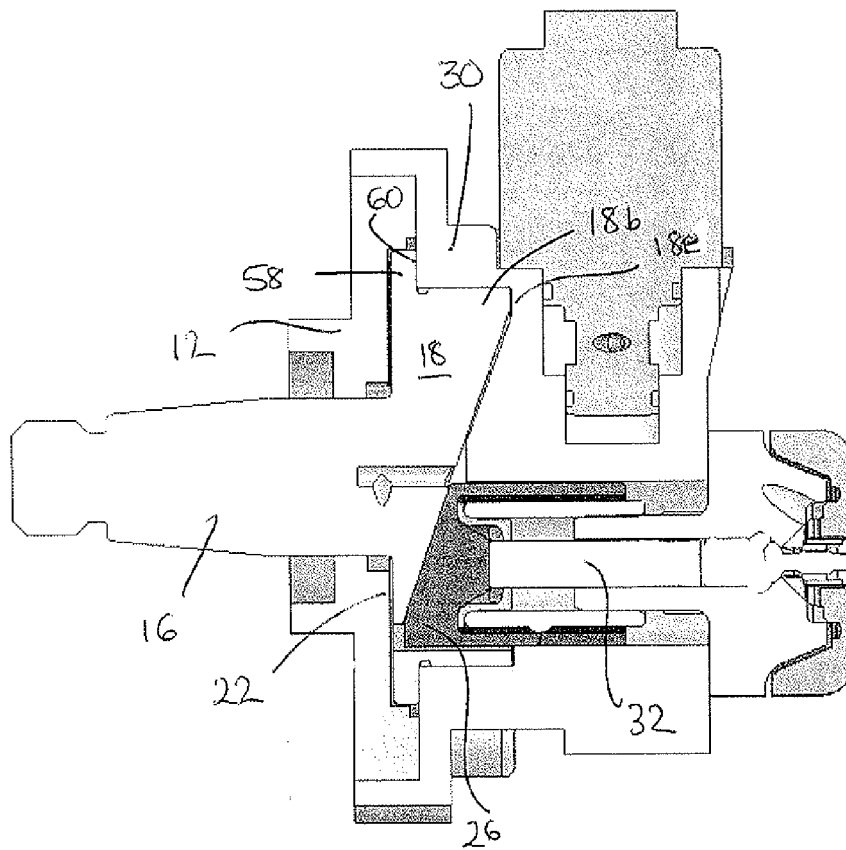
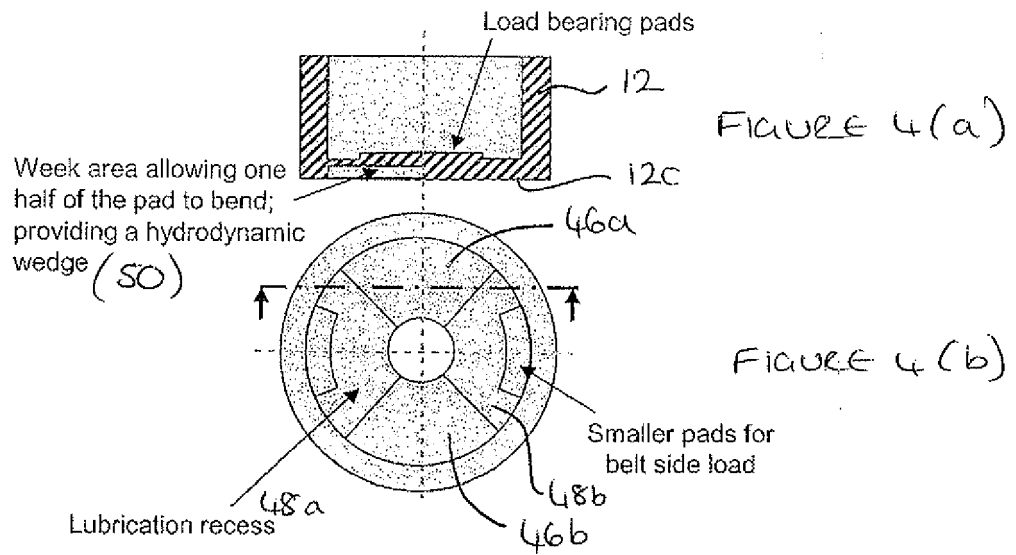
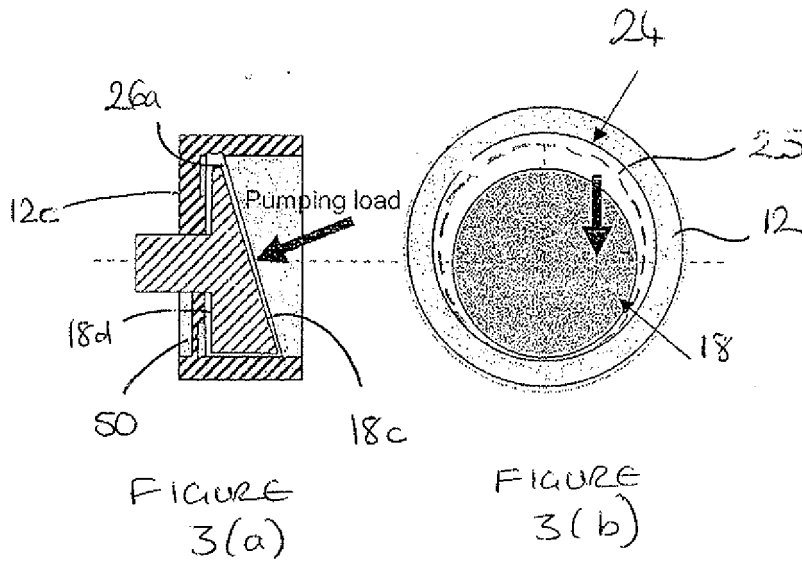


FIGURE 7



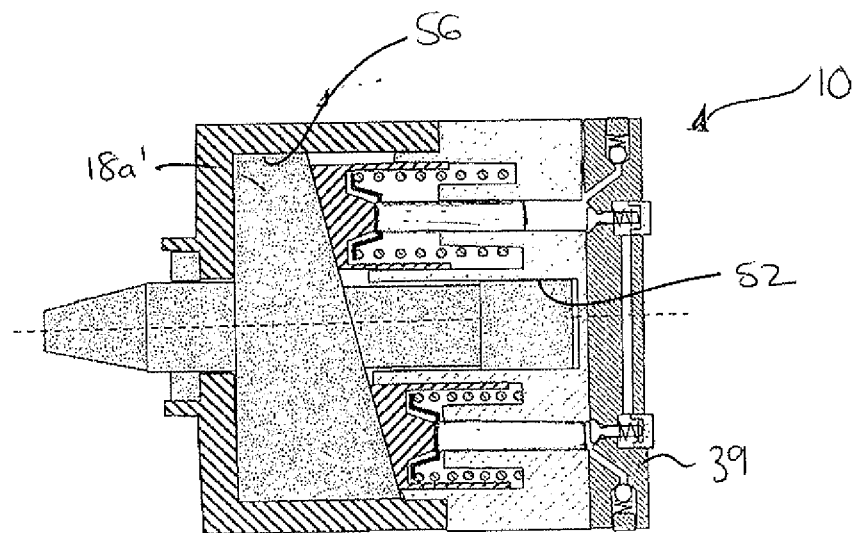
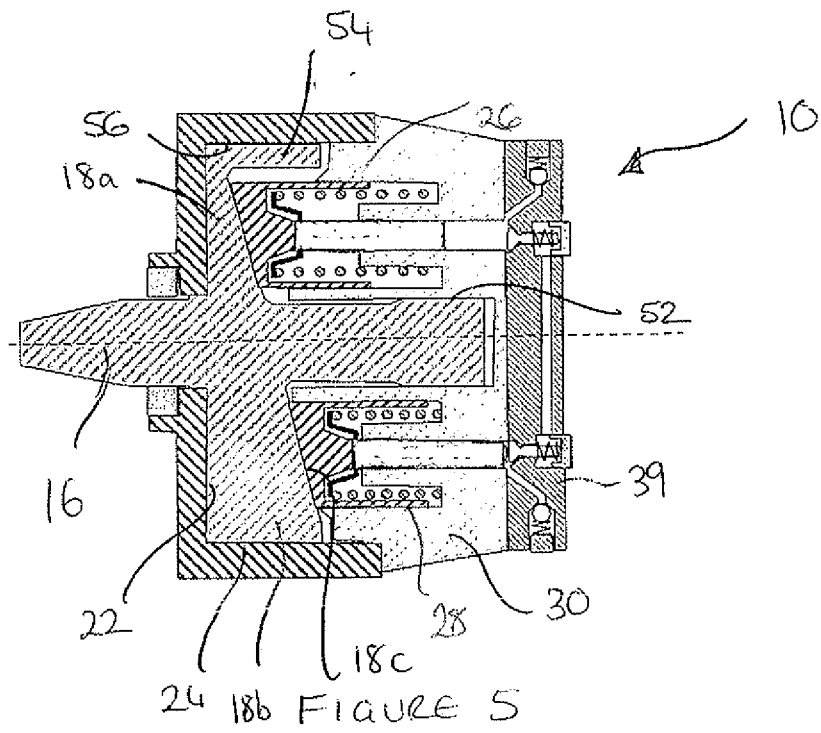


FIGURE 6

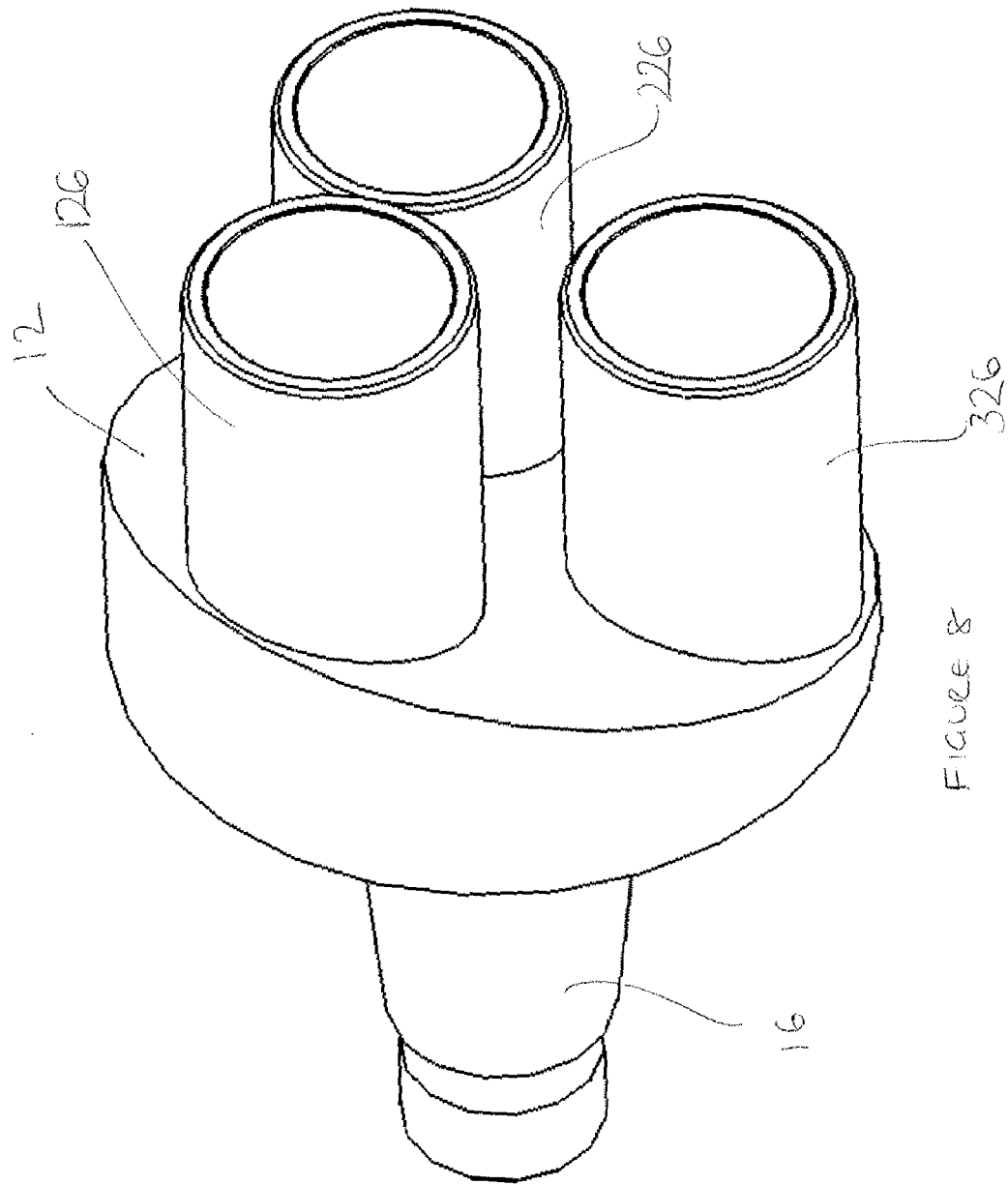


FIGURE 8

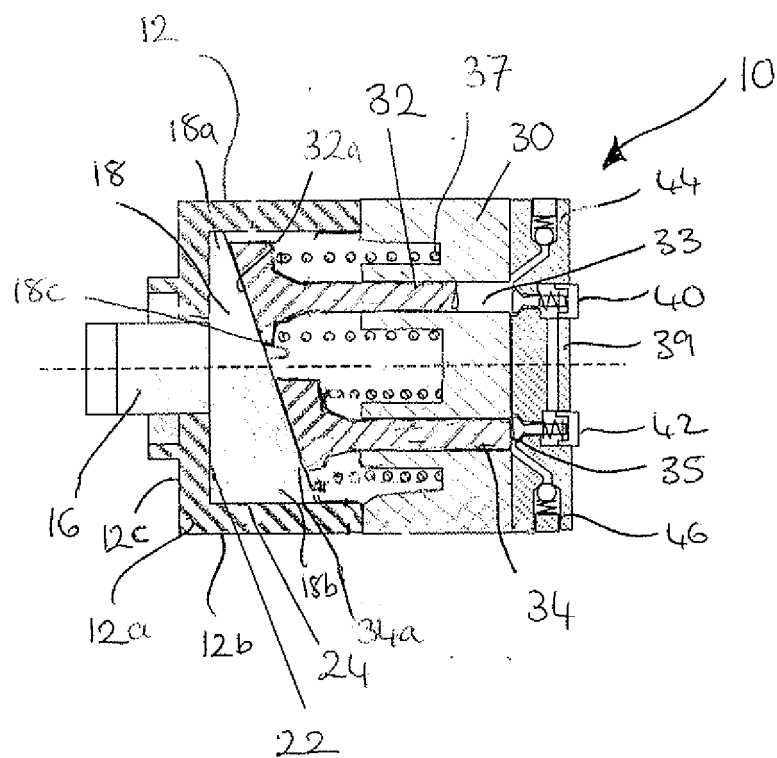


FIGURE 9



EUROPEAN SEARCH REPORT

Application Number
EP 08 16 9804

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	US 3 208 396 A (TADEUSZ BUDZICH ET AL) 28 September 1965 (1965-09-28) * figure 1 * * column 1, line 67 - column 2, line 22 * * column 3, lines 34-40 * -----	1-9,14	INV. F04B1/04 F04B1/12 F04B1/32 F04B9/04 F02M59/10
X	US 4 753 581 A (HISCOCK BENJAMIN [US]) 28 June 1988 (1988-06-28) * figure 3 * * column 6, line 46 - column 7, line 50 * -----	14	
A		2,3,5-9	
X	GB 2 308 623 A (ATSUGI UNISIA CORP [JP]) 2 July 1997 (1997-07-02) * abstract * * page 3, lines 12-33 * * page 17, line 16 - page 18, line 6 * -----	14	
A		2,3,5-9	
X	EP 1 091 170 A (EATON WILLIAMS GROUP LTD [GB]) 11 April 2001 (2001-04-11) * paragraph [0023]; figure 2 * -----	14	
A		2,3,5,6	
A	US 2 910 008 A (WEISENBACH CHARLES O) 27 October 1959 (1959-10-27) * column 2, lines 23-43; figure 2 * -----	1-9,14	TECHNICAL FIELDS SEARCHED (IPC) F04B F02M
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 20 May 2009	Examiner Richmond, Robin
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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EPO FORM 1503 03.82 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
ON EUROPEAN PATENT APPLICATION NO.**

EP 08 16 9804

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
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20-05-2009

Patent document cited in search report		Publication date		Patent family member(s)	Publication date
US 3208396	A	28-09-1965	DE	1291203 B	20-03-1969
			GB	1032516 A	08-06-1966

US 4753581	A	28-06-1988	CA	1275199 C	16-10-1990
			EP	0278739 A1	17-08-1988
			JP	63309783 A	16-12-1988

GB 2308623	A	02-07-1997	DE	19653158 A1	03-07-1997
			US	5782161 A	21-07-1998

EP 1091170	A	11-04-2001	CA	2322302 A1	08-04-2001
			US	6386148 B1	14-05-2002

US 2910008	A	27-10-1959	FR	1143303 A	30-09-1957

EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- WO 2004104409 A [0002]