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(54) Pump assembly

(57) A pump assembly comprises first and second gear pumps (10, 13), each of which has a respective driver gear (11, 14) and a respective driven gear (12,

15), wherein the driven gear (12) of the first gear pump (10) is arranged to drive the driver gear (14) of the second gear pump (13).

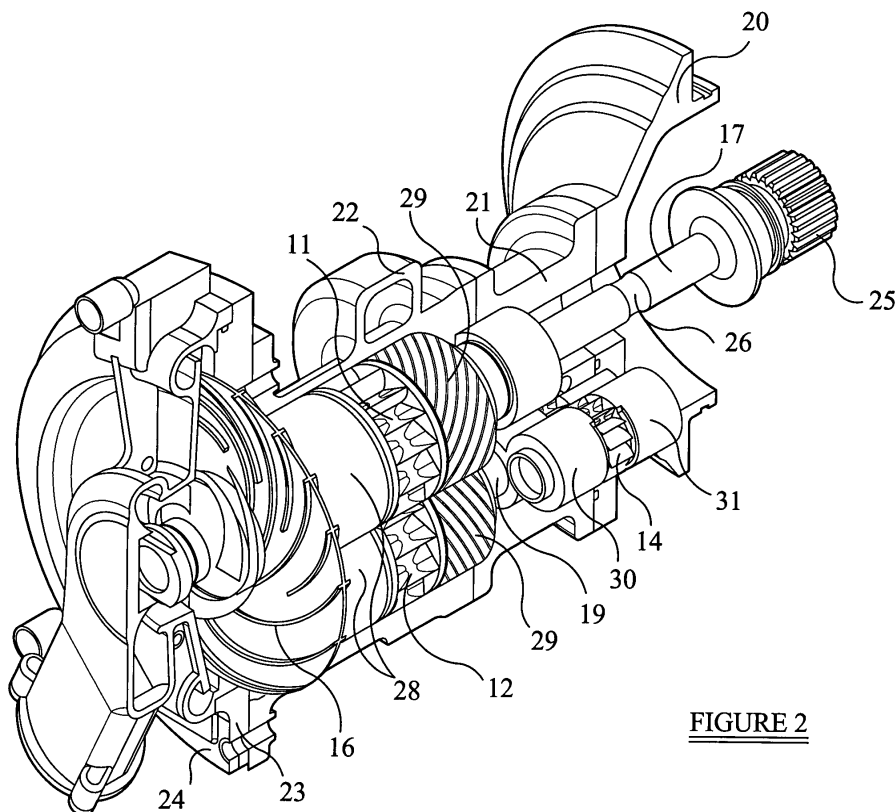


FIGURE 2

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Description

[0001] This invention relates to a pump assembly, particularly for use in a fuel supply system for an aircraft.

[0002] Aircraft fuel supply systems are known in which is provided a low pressure centrifugal pump, and a positive displacement pump, in the form of a gear pump for the high pressure supply stage. With such a conventional gear pump system at relatively low engine power settings; such as cruise or descent, the gear pump delivers considerably more fuel flow than the engine requires. Excess fuel is recirculated around the gear pump and since the pressure rise across the pump is quite high, in order to ensure that there is adequate force available to the variable geometry actuators, considerable 'waste' heat is added to the fuel. This heat energy is dissipated in the burnt fuel and must be managed to ensure fuel temperature does not rise to excessive levels. Hence low flow conditions, such as idle descent, are often the critical conditions with regard to sizing of the engine heat management system. At these conditions the temperature rise across the gear stage can be as high as 50°C. The heat rejection into the fuel from the gear pump also restricts the amount of additional heat which can be dissipated in the fuel for cooling other engine functions, such as the lubricating oil, thus necessitating the use of air/oil heat exchangers which add weight and complexity.

[0003] Fuel pumping arrangements are known in which two separate positive displacement pumps are used to reduce the heat rejection into the fuel by pressurising one of the pump outputs during periods of low fuel demand. Such a fuel system is shown in US Patent No. 4,245,964 which discloses a fuel system using two gear pump stages of different displacements. However, the problem with such prior arrangements is that they do not address the problem of providing a twin positive displacement pumping assembly in which the weight increase over a single pump system is minimised and an increase in the complexity of the drive arrangement from the associated engine accessory gearbox is avoided.

[0004] An object of the invention is to provide a pump assembly in which the above mentioned problem is mitigated.

[0005] According to a first aspect of the invention there is provided a pump assembly comprising a first gear pump, a second gear pump, each gear pump having a respective driver gear and a driven gear, and the driver gear of the second gear pump being arranged to receive drive from the driven gear of the first gear pump.

[0006] Preferably the pump assembly further comprises a rotodynamic pump arranged to receive drive from the driver gear of said first gear pump.

[0007] Preferably the rotodynamic pump is a centrifugal pump.

[0008] The first gear pump is thus used as a step-aside gearbox for the second gear pump.

[0009] Preferably the second gear pump has a small-

er displacement than the first gear pump.

[0010] Preferably the first gear pump is disposed between the second gear pump and the rotodynamic pump. More preferably the centrifugal pump is mounted on an extension of the drive shaft, for example a link drive shaft, from the driver gear of the first gear pump. In a preferred embodiment the centrifugal pump is directly mounted on the extension, for example by screw thread means.

[0011] Conveniently drive is transmitted to the driver gear of the second gear pump from the driven gear of the first gear pump by a second drive shaft. More conveniently opposite ends of the second drive shaft are provided with splines.

[0012] Advantageously internal leakage of fuel from the first gear pump is utilised to lubricate the respective splines of the second drive shaft at the driven gear of the first gear pump and internal leakage from the second gear pump is utilised to lubricate the splines of the second drive shaft at the driver gear of the second gear pump.

[0013] To achieve the fuel lubrication of the splines in the driver gear of the second gear pump it is necessary either to provide a seal at the end face of the gear pump or to ensure that the driven gear of this pump does not have an axial through passage for fuel. Otherwise the internally leaked fuel will take the easier path through the driven gear and not flow through the splines internal to the driver gear.

[0014] According to a second aspect of the invention there is provided a fuel system for a gas turbine engine incorporating a pump assembly in accordance with said first aspect of the invention.

[0015] According to a third aspect of the invention there is provided a pump assembly comprising a first gear pump, and a second gear pump, each gear pump having a respective driver gear and a driven gear, wherein for each of the first and second gear pumps the timing of the meshing of teeth on the driver gear with teeth on the respective driven gear is controlled so that respective peak torques and/or peak pressures of the first and second gear pumps occur out of phase.

[0016] Preferably the second gear pump has a smaller displacement than the first gear pump.

[0017] Preferably this third aspect of the invention, namely control of the timing of the meshing cycles of the two gear pumps, is applied to a pump assembly of said first aspect of the invention and/or to a fuel system of said second aspect of the invention.

[0018] The invention will now be described, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a diagrammatic view of a pump assembly of the invention,

Figure 2 is a cut away interior perspective view of a pump assembly of the invention,

Figure 3 is an exploded perspective view of the components of the pump assembly of Figure 2,

Figure 4 is a perspective view as in Figure 2, but with casing structure of the pump assembly removed, and

Figure 5 is a schematic interior sectional side view of the pump assembly of Figures 1 to 4.

[0019] Although a pump assembly of the present invention can have various applications, it is primarily intended for use in an aircraft fuel system, and will be described hereinafter in relation thereto.

[0020] As described in the introduction, a problem with using conventional positive displacement gear pumps in aircraft fuel systems is that there is excessive heat rejection as a consequence of the pump system delivering considerably more flow than the engine requires at certain operating conditions. Accordingly a pump assembly of the present invention is designed to provide pressurised fuel supply to the aircraft engine and actuators at all speeds and flight conditions with a minimum of excess heat being produced in the pump stages and rejected into the fuel.

[0021] In particular a pump assembly of the invention comprises a low pressure centrifugal pump, and two gear pumps of different displacements for the high pressure stage. At low engine power settings, e.g. idle descent and cruise, the output from the smaller of the two gear pumps is more than adequate to provide for the engine and actuator supplies, and therefore, only this small displacement pump is pressurised. The very much larger volume fuel flow from the large displacement pump is recirculated across a small pressure drop (sufficiently high to keep the fuel in liquid phase), and hence it generates very little waste heat input into the fuel. At take off and low speed start conditions, where respectively the maximum volume of pressurised fuel is required and where the input drive speed to the pumps is very low, both of the gear pumps are pressurised. At each of these conditions however, virtually no flow is recirculated and hence there is virtually no waste heat generated. A flow combining spill valve is used automatically to switch/combine the outputs of the pumps and controls the level of recirculation. The combining spill valve does not form part of the present invention, and will thus not be discussed further.

[0022] The basic elements of a pump assembly of the invention are schematically represented in Figure 1. Shown in this Figure is a first gear pump 10 having a driver gear 11 and a driven gear 12, a second gear pump 13 having a driver gear 14 and a driven gear 15, and a centrifugal or impeller pump 16. As with conventional gear pumps, the respective pinion teeth of the gears are arranged to mesh as the gears rotate, and as will be described hereinafter, appropriate timing of the two gear pumps is possible in order to arrange for the peak torque

and the pressure from the two pumps to occur out of phase. As will be described, this can be effected by ensuring a predetermined angular relationship between the respective sets of pinion teeth of the gears.

[0023] Figure 1 also shows a drive shaft 17 which extends from an accessory gear box (not shown) on the aircraft engine. As will be described, this drive shaft 17 is provided with a shear neck region to disconnect the pump in the event of a serious jam, and thus prevent damage to the engine gear box. As shown in Figure 1, the drive shaft 17 connects directly into the driver gear 11 of the larger displacement gear pump 10 and continues via an extension, in the form of a link drive shaft 18 integral with the driver gear, to an input of the centrifugal pump 16. Extending from the driven gear 12 of the first, larger, gear pump 10 is a further drive shaft 19 which extends into the driver gear 14 of the second, smaller, gear pump 13 to which it is connected. Accordingly the larger displacement gear pump 10 functions as a step-a-side gear box for the smaller gear pump 13.

[0024] The arrangement shown diagrammatically in Figure 1, is shown in detail in Figures 2 to 4, and the pump assembly will now be described in detail by way of those Figures.

[0025] Firstly with regard to the outer casing of the pump assembly, there is shown in these Figures a mounting flange 20 followed by a housing 21 for the second, smaller, gear pump 13, the housing 21 then being followed by a housing 22 for the first, larger, gear pump 10. A centrifugal stage back plate 23 acts as an end cover for the housing 22 and additionally as a back plate for the centrifugal pump 16. Finally at the end of the casing remote from the mounting flange 20 is a low pressure stage housing 24 for the centrifugal pump 16, this housing including the centrifugal stage inlet.

[0026] As shown best in Figure 3, the drive shaft 17 which extends from the engine accessory gear box has male spline couplings 25 at each end. The drive shaft 17 is designed with a weak link, in the form of a shear neck region 26, to disconnect the pump in the event of a serious jam or seizure, and thus prevent damage to the engine gear box. This shear neck region 26 can be designed to provide far higher drive torque to the unit than is required during normal operation, and at the same time ensure that the shear neck region will fail before the gear box is damaged.

[0027] As far as the gear pumps 10 and 13 are concerned, namely the high pressure stage in the pump assembly, these are two twin pinion gear pumps with different displacements mounted for parallel pumping of high pressure fuel. The exact split in the displacements between the two pumps depends on the exact operating conditions of the engine. Something in the region of a 3:1 ratio in the displacements is envisaged. The aim is to ensure that at cruise flight conditions and below only the smaller displacement pump 13 is pressurised.

[0028] The drive shaft 17 connects directly into the driver gear 11 of the gear pump 10, and continues via

the integral link drive shaft 18 to the input of the centrifugal pump 16. There is thus a common, in-line drive for the gear 11 and pump 16. This approach has been chosen as the most desirable configuration since it allows the rotating elements of the centrifugal pump to be mounted on the driver gear of the large displacement pump. The driver gear is the more lightly loaded of the two gears in the larger gear pump 10, and has the required load carrying capacity to cope with the centrifugal pump loads. The bearings, to be described, in the smaller gear pump 13 are not sized to have adequate load capacity to carry these centrifugal pump loads.

[0029] The centrifugal pump impeller assembly is directly mounted on the link drive shaft 18. Preferably the link drive shaft is brazed into the driver gear 11 and forms a seal between the fuel and the spline lubricant. The impeller assembly is screwed on the extension, with the hand of the thread being such as to tighten the assembly under normal rotation. A secondary locking feature can be provided, this being a nut which clamps the impeller to the extension, with the hand of the threads such that should the impeller tend to loosen, the nut will tighten. Preferably a steel insert is bonded into the impeller, so that the threaded joint is steel-on-steel.

[0030] The centrifugal stage takes unfiltered fuel from the aircraft tank feed system (usually fed from the tank pump) often at very low pressure and potentially in the form of a mixed liquid and vapour flow. The centrifugal pump increases the pressure of the fuel sufficiently to ensure that the high pressure stage receives only liquid fuel, and thus allows the gear pumps to operate satisfactorily, taking into account interstage pressure losses associated with heat exchangers, filters, etc. The centrifugal stage has to provide sufficient flow for both of the high pressure stage pumps at the take-off condition (maximum flow). The large displacement high pressure stage pump 10 determines the maximum pressure rise that the centrifugal stage has to generate. It is highly desirable for the rotating elements of the centrifugal pump, i.e. the inducer/impeller, to be directly mounted on an extension of the driver gear 11 in the high pressure stage which, in this embodiment, is provided by the link drive shaft 18. Mounting the inducer/impeller in this way eliminates the need for separate bearings which have to operate on contaminated (unfiltered) fuel. It also means that a splined drive shaft is not required. These lightly loaded drive shafts are often prone to wear due to fretting. Thus this method of mounting the inducer/impeller results in low parts count, excellent reliability, long life, minimum size, weight and cost, and short installation length.

[0031] As shown in Figure 3, the further drive shaft 19 has male splines 27 on its respective opposite ends. Accordingly one splined end of the further drive shaft 19 is engaged internally with the driven gear 12, whilst its opposite splined end is engaged internally with the driver gear 14 of the second, smaller, gear pump 13, so that drive is directly transmitted from the driven gear 12 of

the gear pump 10 to the driver gear 14 of the gear pump 13 without the need for any transfer gears etc. Thus as stated previously, the large displacement pump 10 is used as a step-a-side gear box for the smaller pump 13.

As will be described below, the splines 27 are lubricated with fuel, this significantly simplifying mechanical design of the pump assembly.

[0032] The first, larger, gear pump 10 incorporates, by way of bearings, two fixed bearing blocks 28 and two floating bearing blocks 29. The fixed bearing blocks abut against a rigid surface formed by the back plate 23 whilst the floating bearing blocks are spring loaded against the respective side faces of the gears, this loading being augmented by hydraulic forces generated by gear stage discharge pressure, which is ported to a sealed area outside an eccentric nose at the end of each bearing. As far as the smaller gear pump 13 is concerned, the bearings are of 'figure-of-eight' designs rather than the split bearing block designs used with the gear pump 10.

[0033] The two bearing blocks 30, 31 for the gear pump 13 are shown in Figures 2 to 4. Cardioid shaped elastomeric seals 33 act on the back faces of the bearing blocks 30, 31, and for the bearing block 31 the respective seals are received in seal grooves in the inner end of the mounting flange 20. Similarly the pump housing 22 contains seal grooves for the cardioid seals of the bearing block 30.

[0034] Although the bearing blocks 30, 31 are illustrated as being of figure-of-eight form, they could alternatively be of a split form similar to the bearing blocks of the first gear pump 10, in which case the cardioid-shaped seals 33 can be omitted.

[0035] As far as lubrication of the pump assembly is concerned, it can be seen from Figure 5 that the drive shaft 17 is hollow, and this allows, in the direction of arrow A, one shot oil lubrication from the auxiliary gear box from which the drive shaft 17 extends, this oil lubrication being applied to the splines 25 at the end of the drive shaft 17 where it is connected to the driver gear 11, as shown by the arrows exiting the end of the drive shaft just upstream of the extension 18. Some fuel leaks across the bearing faces of the driven gear 12 of the gear pump 10 (exhaust flow) and a lesser volume leaks past the smaller gear pump bearing faces. As shown by the arrows B in Figure 5, the splines 27 in the driven gear 12 of the gear pump 10 are lubricated by the bearing exhaust flow from both of the solid bearing blocks 28, which are adjacent to the centrifugal stage. This exhaust flow passes down the hollow driven gear, through the splines, and is collected in a cavity adjacent to the pressure loaded bearings 29. The sealing arrangement 33 at the remote end of the small displacement pump 13 ensures that the bearing exhaust flow from the bearing block 30, arrows C, lubricates the splines in the driver gear 14 by passing through the hollow driver gear, and hence through the splines, before being collected in the same cavity as the bearing flow from the gear pump 10. If the sealing arrangement were not present,

the preferred pathway for internally leaked fuel from the small displacement pump would be through the axial hole in the small displacement pump driven gear and not through the splines of the driver gear. If the small displacement pump driven gear is designed without an axial flow pathway the inner cardioid shaped elastomeric seal will not be required. The use of fillet root side fit splines means that there is a significant flow area through the splines which minimises the pressure drop. This flow of fuel provides lubrication and washes out any debris or contamination that may be produced in the pump.

[0036] A drain 32, from said fuel collection cavity, is provided adjacent a centre region of the further drive shaft 19 in order to return the lubrication/leaked fuel to the common input of the two gear pumps. Since the fuel lubricated splines are on the internal drive shaft between the two gear stages, very close control of alignment can be maintained and hence the contact stresses and sliding velocities can be minimised.

[0037] A further aspect of the present invention, which may have application generally to twin stage gear pumps of a configuration different from that shown in Figures 1 to 5, will now be described. It will however be appreciated that this third aspect has particular application to the pump assembly of the first aspect of the present invention.

[0038] In the arrangement so far described, additional loads are imposed on the gears in the first gear pump, since in addition to having to transmit the 'pumping' torque to the driven gear 12, it also has to transmit the total torque required by the smaller gear pump 13. However for most of the operating cycle (cruise) the loading on the large displacement pump gears/bearings will be less than they would normally experience in a single gear pump system where the full output of the gear pump will be pressurised at all times with the excess fuel being spilled back to low pressure.

[0039] The input torque to a gear pump varies with the position of the gears during the meshing cycle. By appropriate timing of the meshing cycles in the two gear pumps, it is possible to minimise the additional peak torque which has to be transmitted through the gears in the larger displacement pump, which is essential to maximising their life. Accordingly it is possible to minimise the stresses on the gears in the gear pump 10 which have to transmit the pumping torque between the drive and driven gears, in addition to the total torque required by the small displacement pump 13. This timing can be achieved by controlling the angular position of the internal spline drives relative to the gear teeth, and by including a datum feature or a register in the drive, such as by missing out one of the teeth on the spline. Male splines on the intermediate driveshaft will have mating registers, and the angular position of these registers will also be closely controlled relative to each other. As both gear pumps will have the same number of teeth (12 in the current design), the intention is to arrange for the peak

torque of the two pumps to occur out of phase by ensuring a predetermined angular relationship between the respective sets of teeth.

[0040] In addition to minimising the loading of the gears of the gear pump 10 by timing the two gear pumps, it is also possible to minimise the combined flow ripple generated by the two gear pump high pressure stage. The flow delivered by a gear pump varies with the position of the gears in the meshing cycle. This flow ripple generates pressure ripple, the impedance of the pump and the hydraulic circuit into which it discharges determining the magnitude and phase shift of the resultant pressure ripple. Minimising the magnitude of the pressure ripple is desirable since it can adversely affect downstream equipment, such as the fuel-metering unit. Once again timing of the two gear pumps is required to minimise the peak pressure that results from the superposition of the flows from the two pumps. There will likely be a compromise between the minimising of peak input, torque, and output flow ripple due to the flow characteristics of the outlets and components downstream of the pump assembly, the exact relationships only being capable of calculation when the housings and connections are fully developed. Accordingly in this third aspect of the invention, for one or both of the first and second gear pumps, the timing of the meshing of teeth on the driver gear with teeth on the driven gear is controlled so that respective peak torques and/or peak pressures of the first and second gear pumps occur out of phase. Whilst this could be at 180° out of phase, this is not necessarily the case.

Claims

1. A pump assembly comprising a first gear pump (10), and a second gear pump (13, each gear pump (10, 13) having a respective driver gear (11, 14) and a driven gear (12, 15) and the driver gear (14) of the second gear pump (13) being arranged to receive drive from the driven gear (12) of the first gear pump (10).
2. A pump assembly according to Claim 1, further comprising a rotodynamic pump (16) arranged to receive drive from the driver gear (11) of said first gear pump (10).
3. A pump assembly according to Claim 2, wherein the rotodynamic pump (16) is a centrifugal pump.
4. A pump assembly according to Claim 2 or Claim 3, wherein the first gear pump (10) is disposed between the second gear pump (13) and the rotodynamic pump (16).
5. A pump assembly according to Claim 4, wherein the rotodynamic pump (16) is mounted on an extension

(13) of the drive shaft (17) from the driver gear (11) of the first gear pump (10).

6. A pump assembly according to Claim 5, wherein the centrifugal pump (16) is directly mounted on the extension (18). 5
7. A pump assembly according to any of the preceding claims, wherein the second gear pump (13) has a smaller displacement than the first gear pump (10). 10
8. A pump assembly according to any of the preceding claims, wherein drive is transmitted to the driver gear (14) of the second gear pump (13) from the driven gear (11) of the first gear pump (10) by a second drive shaft (19). 15
9. A pump assembly according to Claim 8, wherein opposite ends of the second drive shaft (19) are provided with splines (27). 20
10. A pump assembly according to Claim 9, wherein internal leakage of fuel from the first gear pump (10) is utilised to lubricate the respective splines (27) of the second drive shaft (19) at the driven gear (11) of the first gear pump (10) and internal leakage from the second gear pump (13) is utilised to lubricate the splines (27) of the second drive shaft (19) at the driver gear (14) of the second gear pump (13). 25
30
11. A pump assembly according to any of the preceding claims, wherein for each of the first and second gear pumps (10, 13) the timing of the meshing of teeth on the driver gear (11, 14) with teeth on the respective driven gear (12, 15) is controlled so that respective peak torques and/or peak pressures of the first and second gear pumps (10, 13) occur out of phase. 35
12. A gas turbine engine incorporating a pump assembly as claimed in any of the preceding claims. 40
13. A pump assembly comprising a first gear pump (10), and a second gear pump (13), each gear pump (10, 13) having a respective driver gear (11, 14) and a driven gear (12, 15) wherein for each of the first and second gear pumps (10, 13) the timing of the meshing of teeth on the driver gear (11, 14) with teeth on the respective driven gear (12, 15) is controlled so that respective peak torques and/or peak pressures of the first and second gear pumps (10, 13) occur out of phase. 45
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14. A pump assembly according to Claim 13, wherein the second gear pump (13) has a smaller displacement than the first gear pump (10). 55

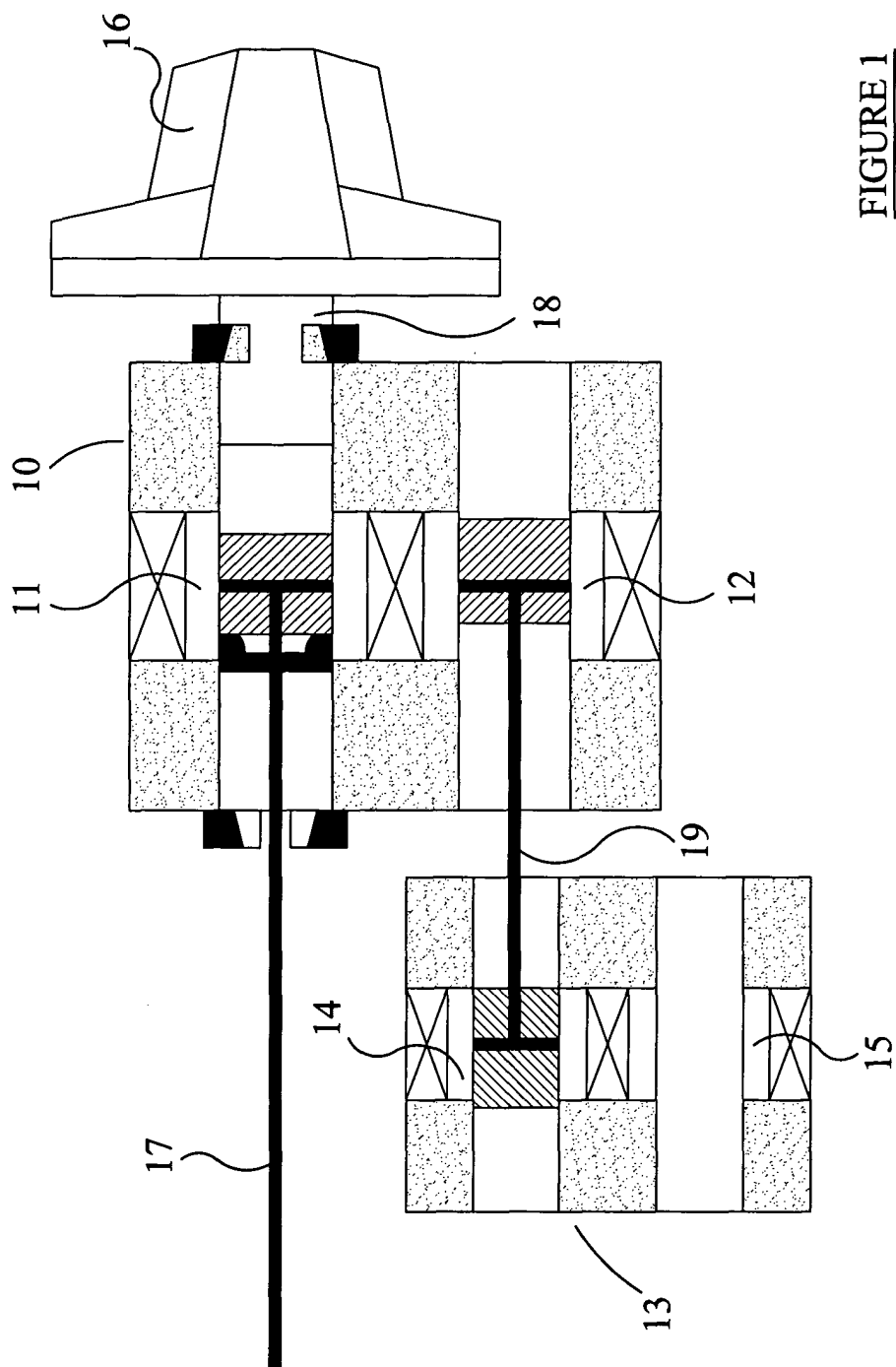
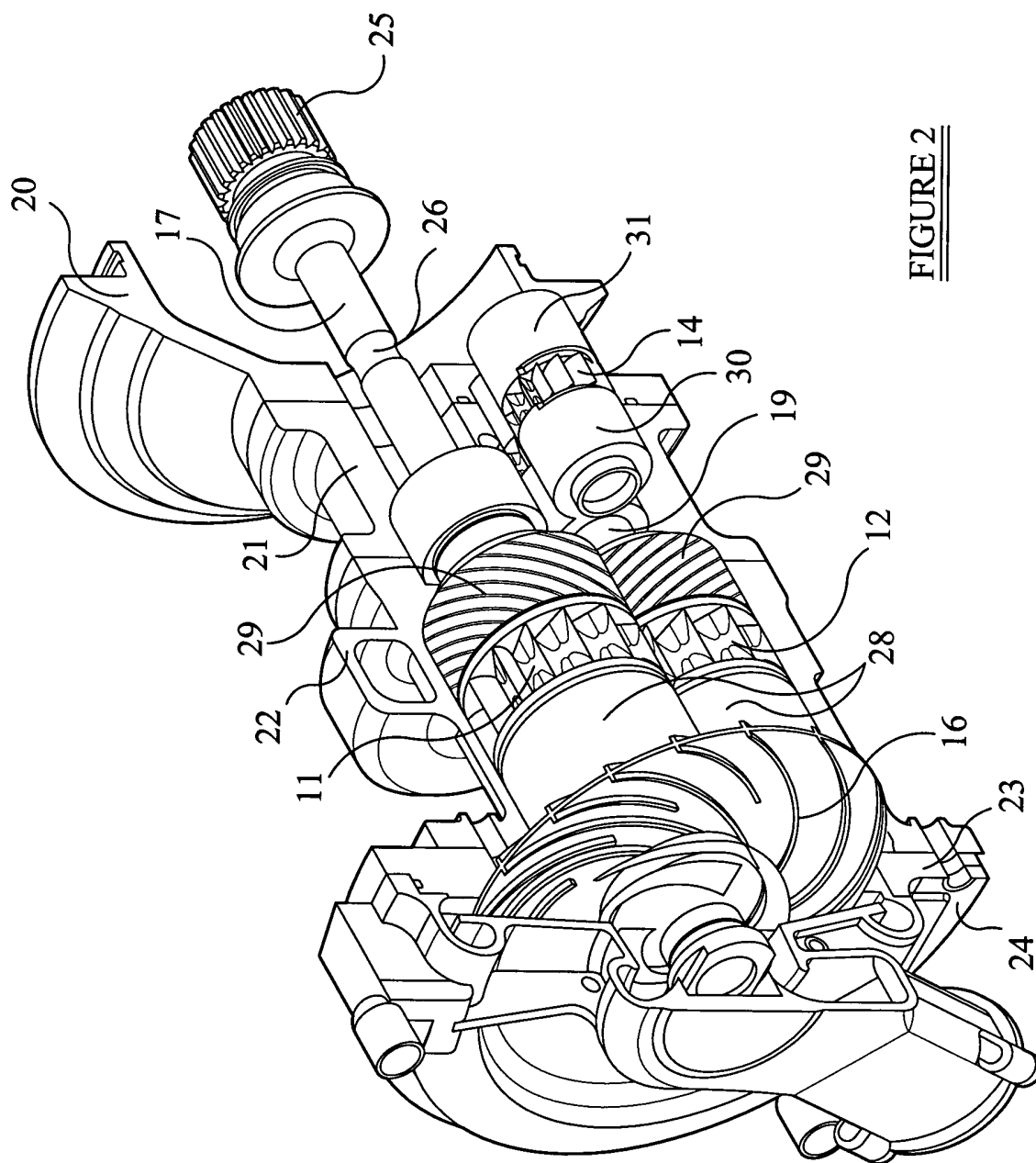


FIGURE 1



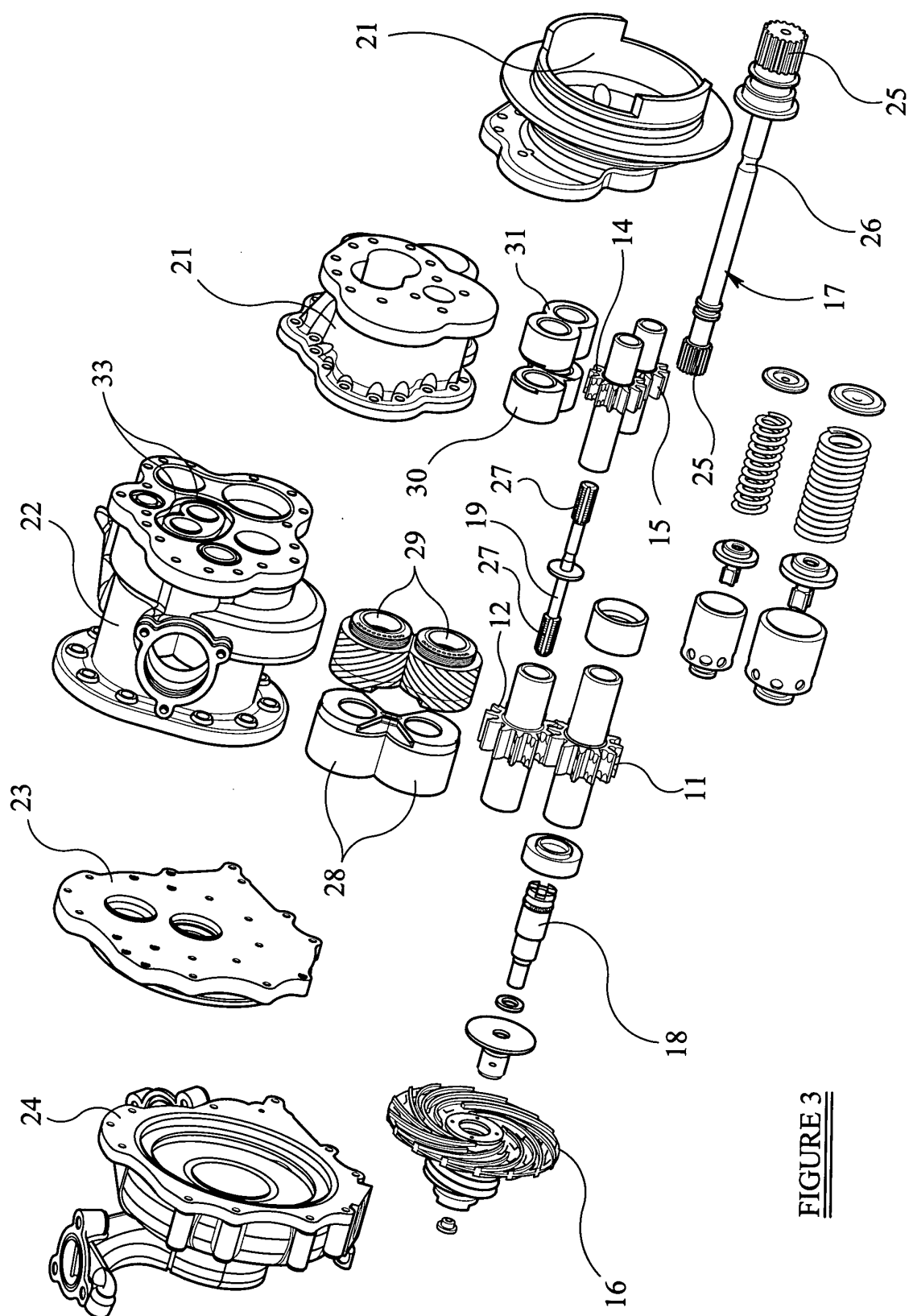


FIGURE 3

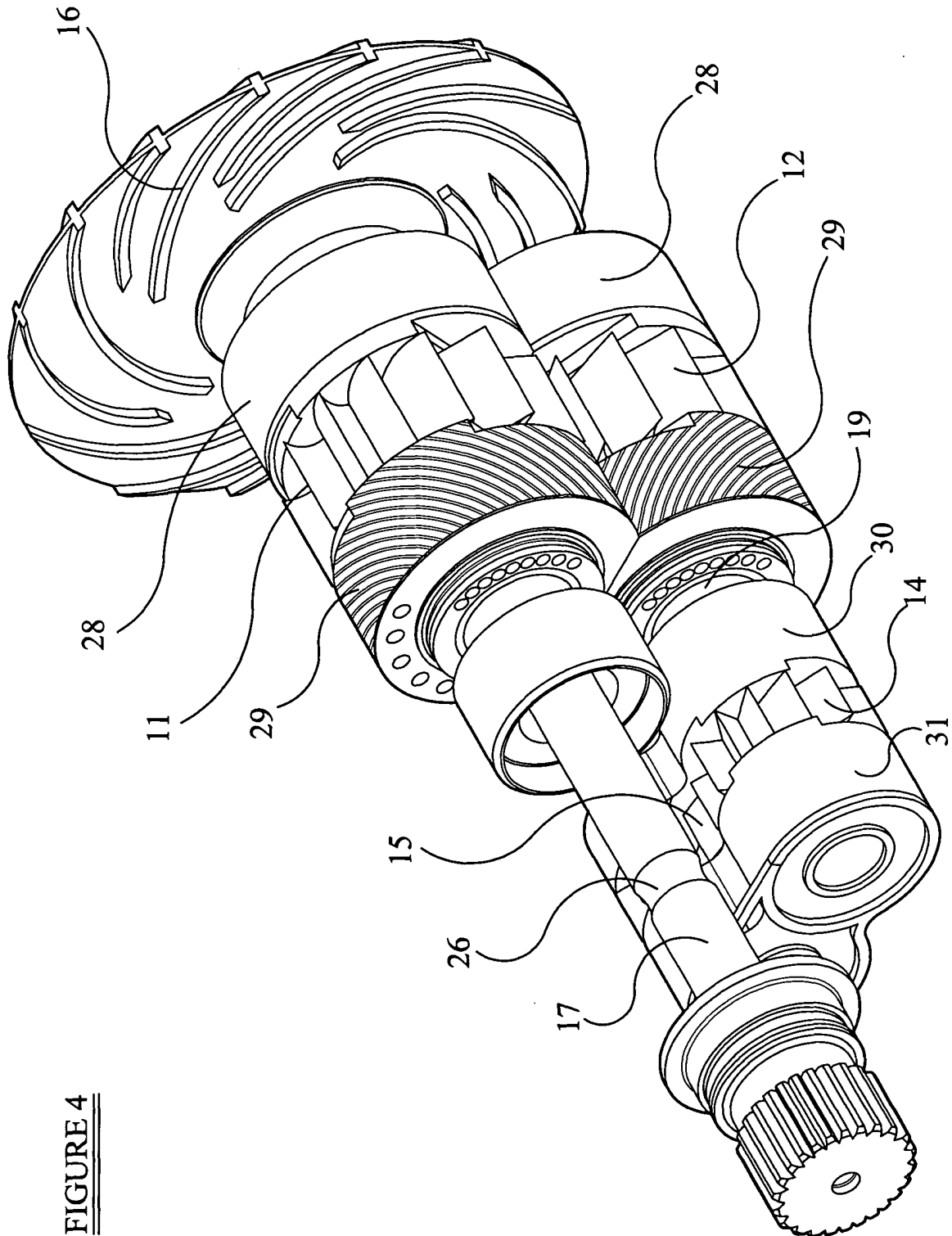
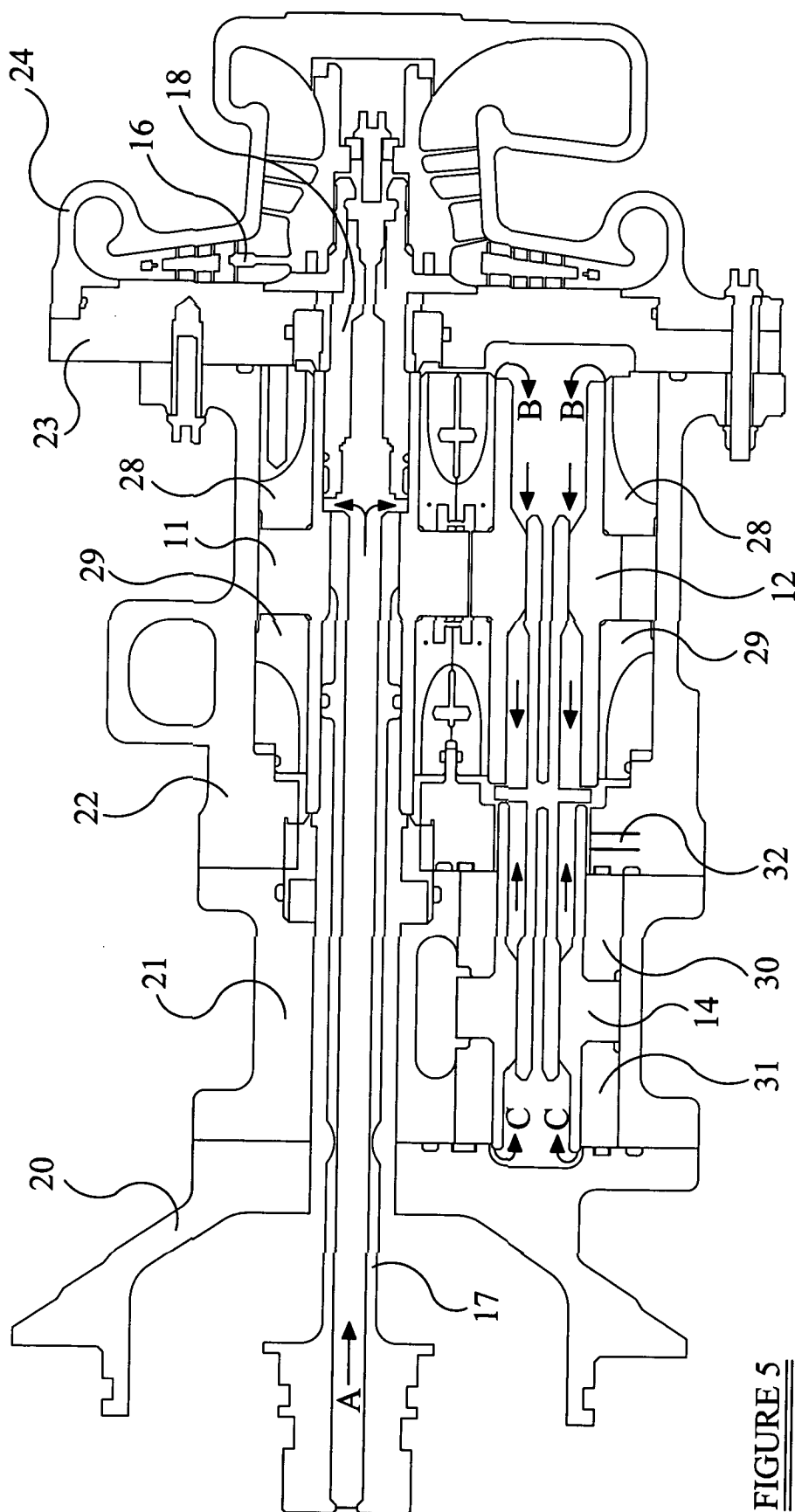


FIGURE 4





European Patent
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EUROPEAN SEARCH REPORT

Application Number
EP 05 25 2373

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
X	PATENT ABSTRACTS OF JAPAN vol. 1997, no. 09, 30 September 1997 (1997-09-30) -& JP 09 133084 A (KUBOTA CORP), 20 May 1997 (1997-05-20)	1,7,8	F04C15/04 F04C11/00 F04C15/00 F02C7/236
Y	* abstract *	2-6,9, 11,12	
A	* figure 5 *	10	
X	----- PATENT ABSTRACTS OF JAPAN vol. 1998, no. 03, 27 February 1998 (1998-02-27) -& JP 09 287578 A (KOYO SEIKO CO LTD), 4 November 1997 (1997-11-04)	1,8	
A	* abstract *	2-7,9, 11,12	
	* figures 1-3 *		
X	----- PATENT ABSTRACTS OF JAPAN vol. 017, no. 392 (M-1450), 22 July 1993 (1993-07-22) -& JP 05 071475 A (ANLET CO LTD), 23 March 1993 (1993-03-23)	13,14	TECHNICAL FIELDS SEARCHED (Int.Cl.7)
Y	* abstract *	11	F04C
A	* figures 1-7 *	7	F02C
X	----- EP 0 797 002 A (ULTRA HYDRAULICS LIMITED) 24 September 1997 (1997-09-24)	13,14	
A	* figure 5 *	7,11	
	* column 4, line 23 - line 28 *		
	* claim 1 *		
Y	----- US 3 026 929 A (BURNS LOUIS G) 27 March 1962 (1962-03-27) * figures 1,3 * * column 1, line 10 - line 20 * * column 1, line 60 - line 67 *	2-6,12	
	----- -/--		
The present search report has been drawn up for all claims			
Place of search The Hague		Date of completion of the search 13 July 2005	Examiner Lequeux, F
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 (F04C01)



European Patent
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EUROPEAN SEARCH REPORT

Application Number
EP 05 25 2373

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
Y	US 2 713 244 A (CHAUDLER MILTON E) 19 July 1955 (1955-07-19)	9	
A	* figures 1-3 * * column 1, line 15 - line 52 * * column 3, line 55 - line 65 * * column 5, line 28 - line 33 *	2,3,8,12	
A	----- US 3 041 974 A (JUDKINS EDWIN R) 3 July 1962 (1962-07-03) * figure 1 * * column 1, line 8 - line 40 * * column 3, line 53 - line 66 * -----	8,9	
The present search report has been drawn up for all claims			TECHNICAL FIELDS SEARCHED (Int.Cl.7)
Place of search The Hague		Date of completion of the search 13 July 2005	Examiner Lequeux, F
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)



European Patent
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Application Number

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CLAIMS INCURRING FEES

The present European patent application comprised at the time of filing more than ten claims.

☐ Only part of the claims have been paid within the prescribed time limit. The present European search report has been drawn up for the first ten claims and for those claims for which claims fees have been paid, namely claim(s):

☐ No claims fees have been paid within the prescribed time limit. The present European search report has been drawn up for the first ten claims.

LACK OF UNITY OF INVENTION

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

see sheet B

☐ All further search fees have been paid within the fixed time limit. The present European search report has been drawn up for all claims.

☒ As all searchable claims could be searched without effort justifying an additional fee, the Search Division did not invite payment of any additional fee.

☐ Only part of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the inventions in respect of which search fees have been paid, namely claims:

☐ None of the further search fees have been paid within the fixed time limit. The present European search report has been drawn up for those parts of the European patent application which relate to the invention first mentioned in the claims, namely claims:



European Patent
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LACK OF UNITY OF INVENTION
SHEET B

Application Number
EP 05 25 2373

The Search Division considers that the present European patent application does not comply with the requirements of unity of invention and relates to several inventions or groups of inventions, namely:

1. claims: 1-6

Pump assembly with centrifugal pump

2. claims: 1,7,14

Pump assembly with different displacements

3. claims: 1,8-10

Pump assembly with splined shaft

4. claims: 1,11,13

Pump assembly with shifted gears

5. claims: 1,12

Gas turbine with pump assembly

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

EP 05 25 2373

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
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