

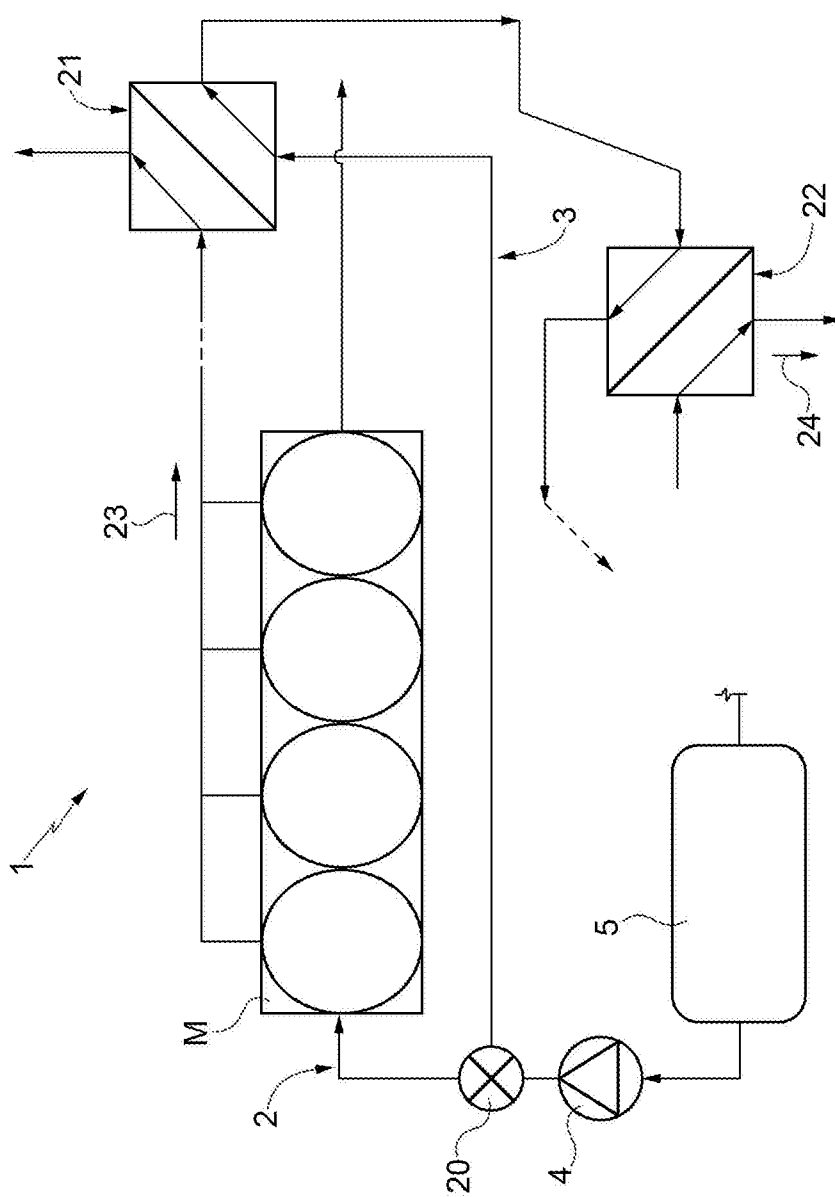


FIG. 1

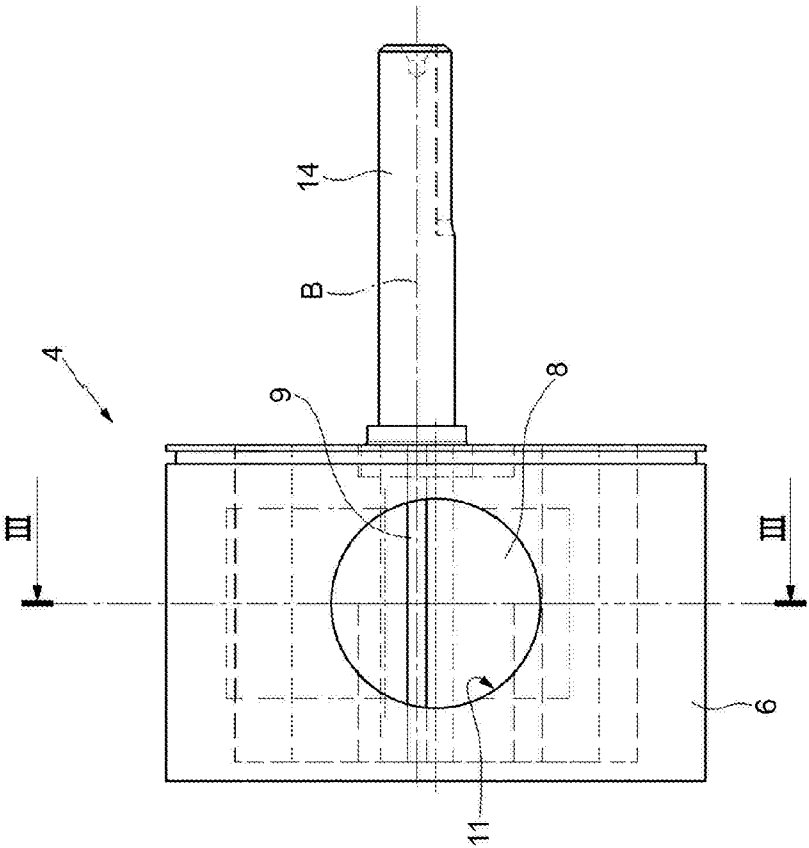


FIG. 2

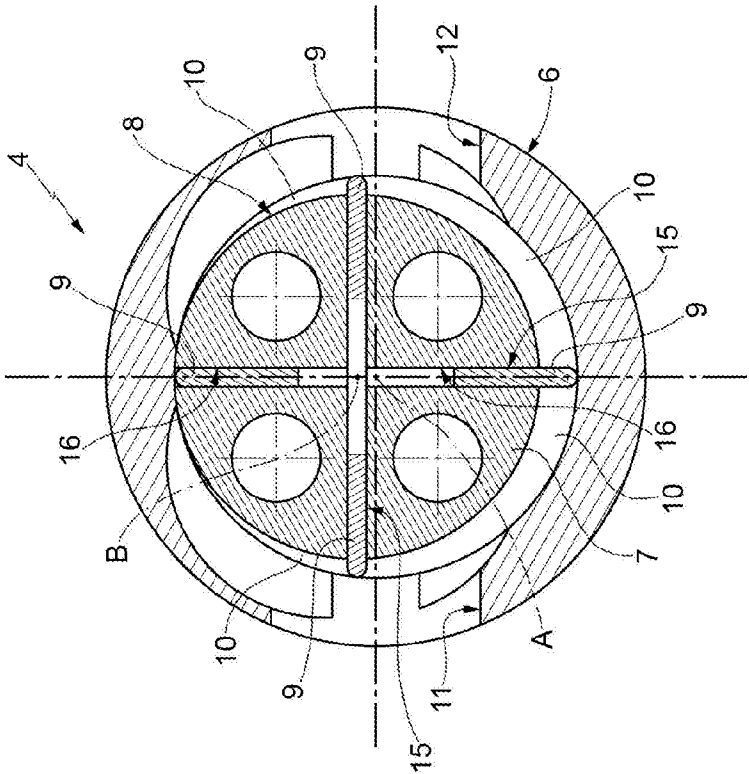


FIG. 3

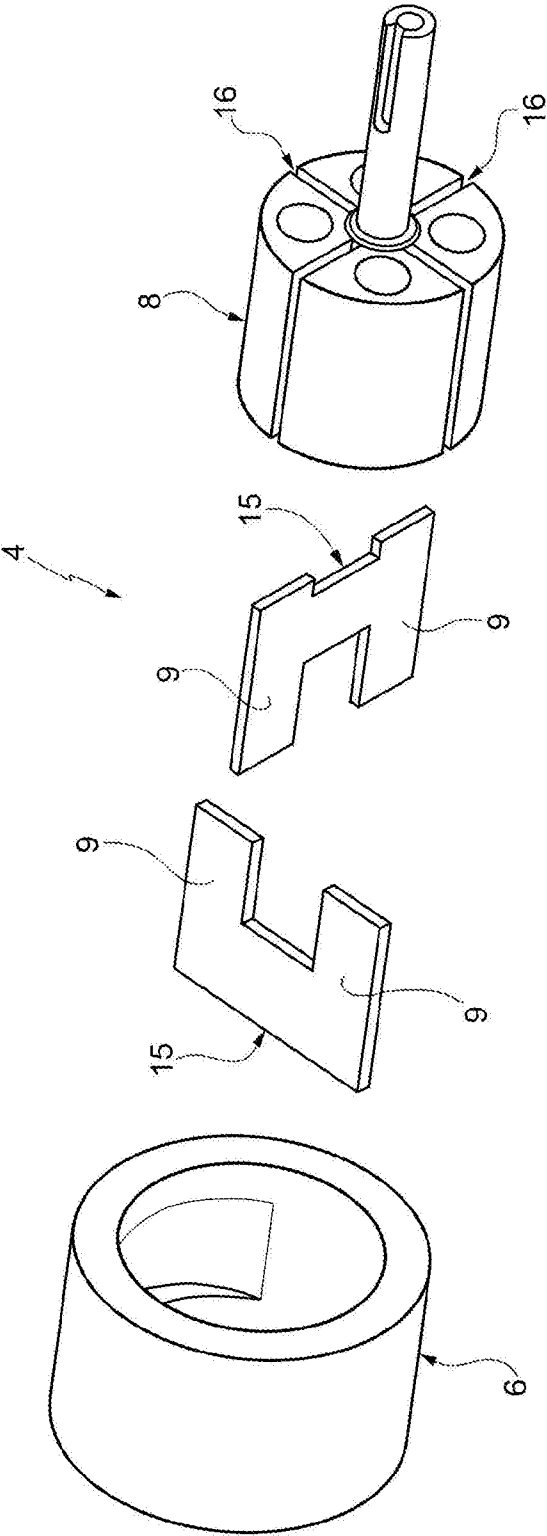


FIG. 4

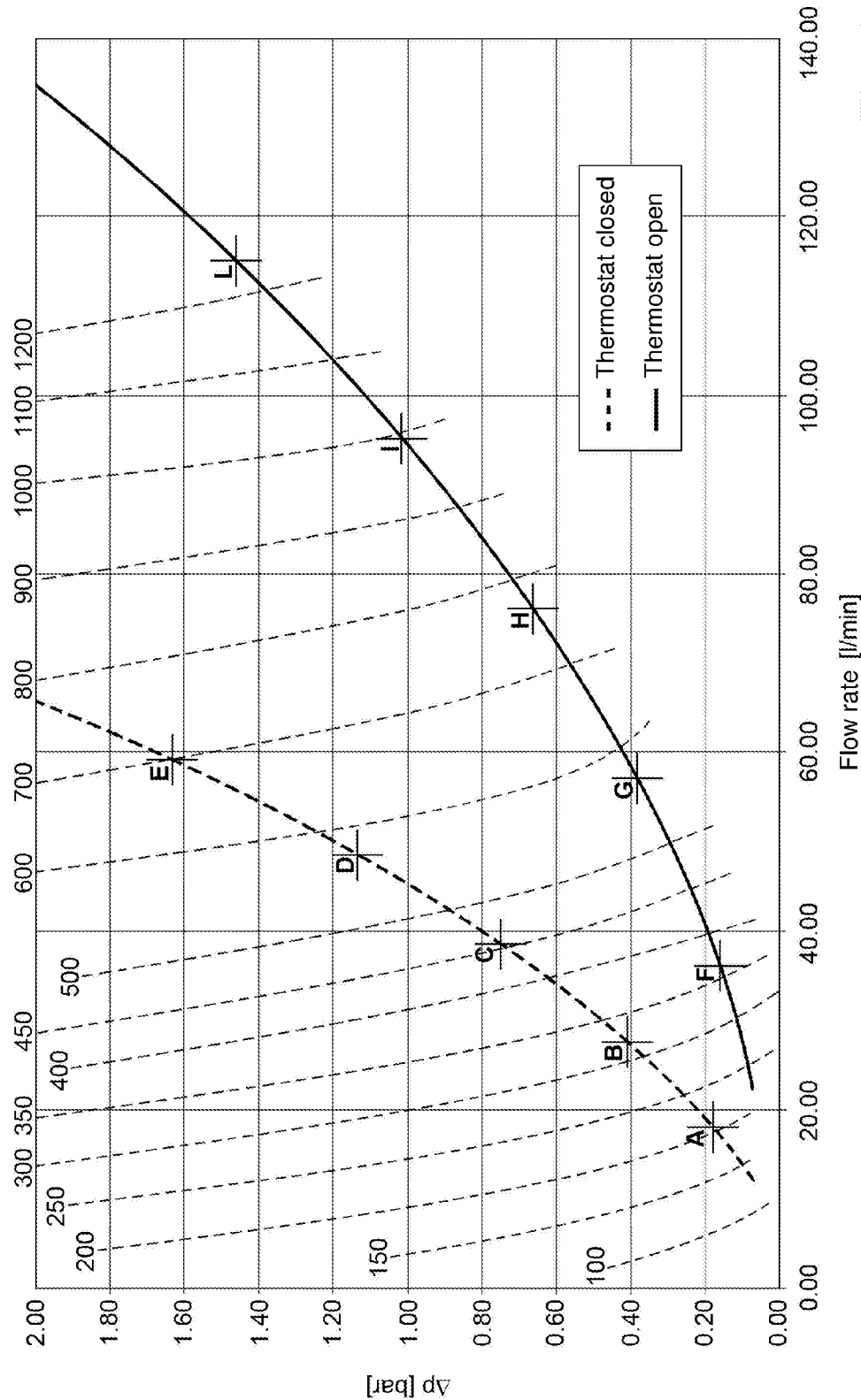


FIG. 5

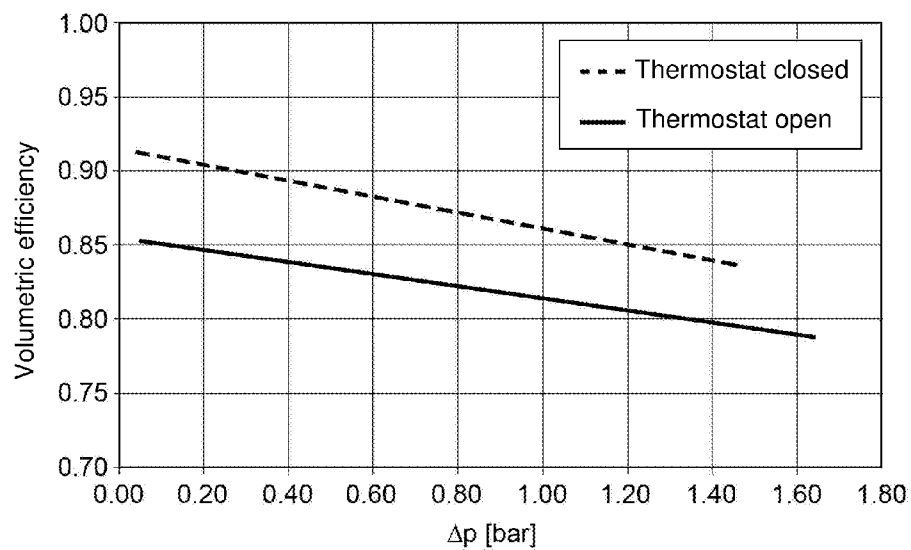


FIG. 6

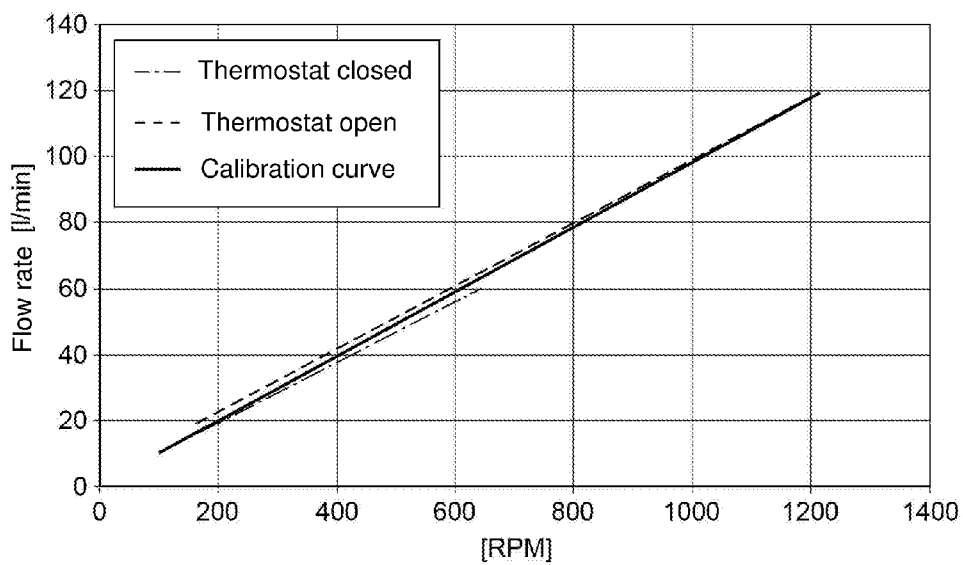


FIG. 7

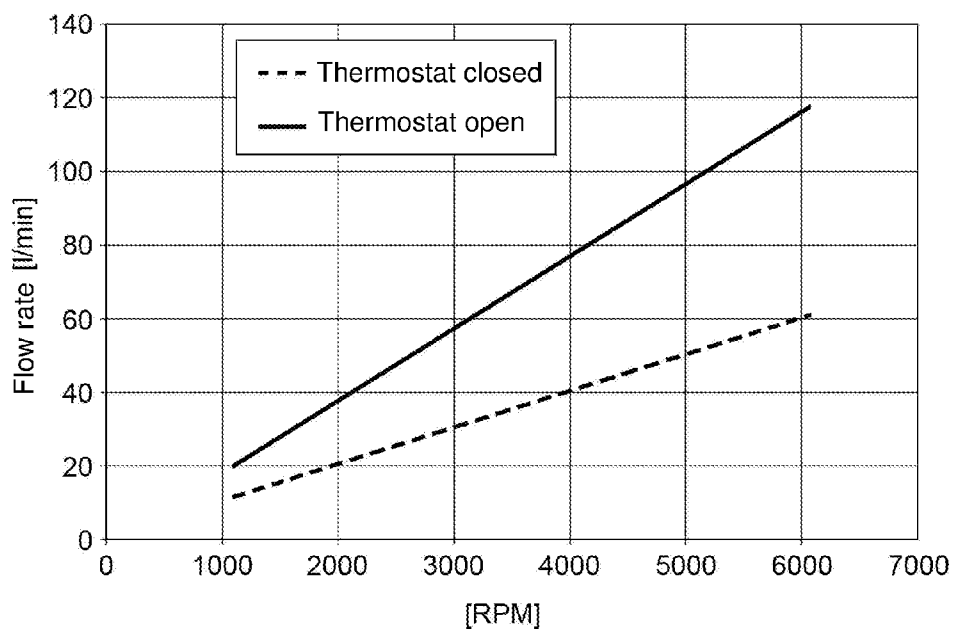


FIG. 8

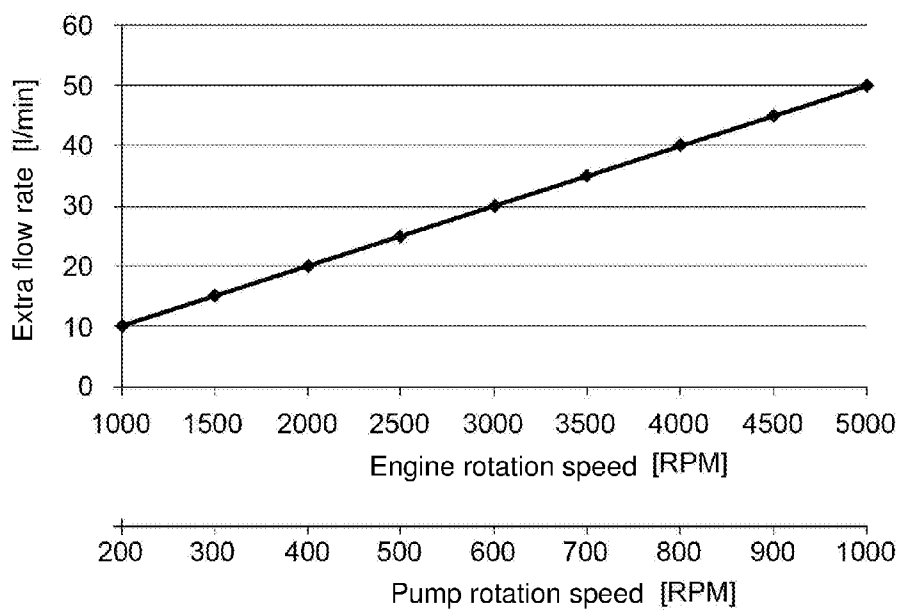


FIG. 9

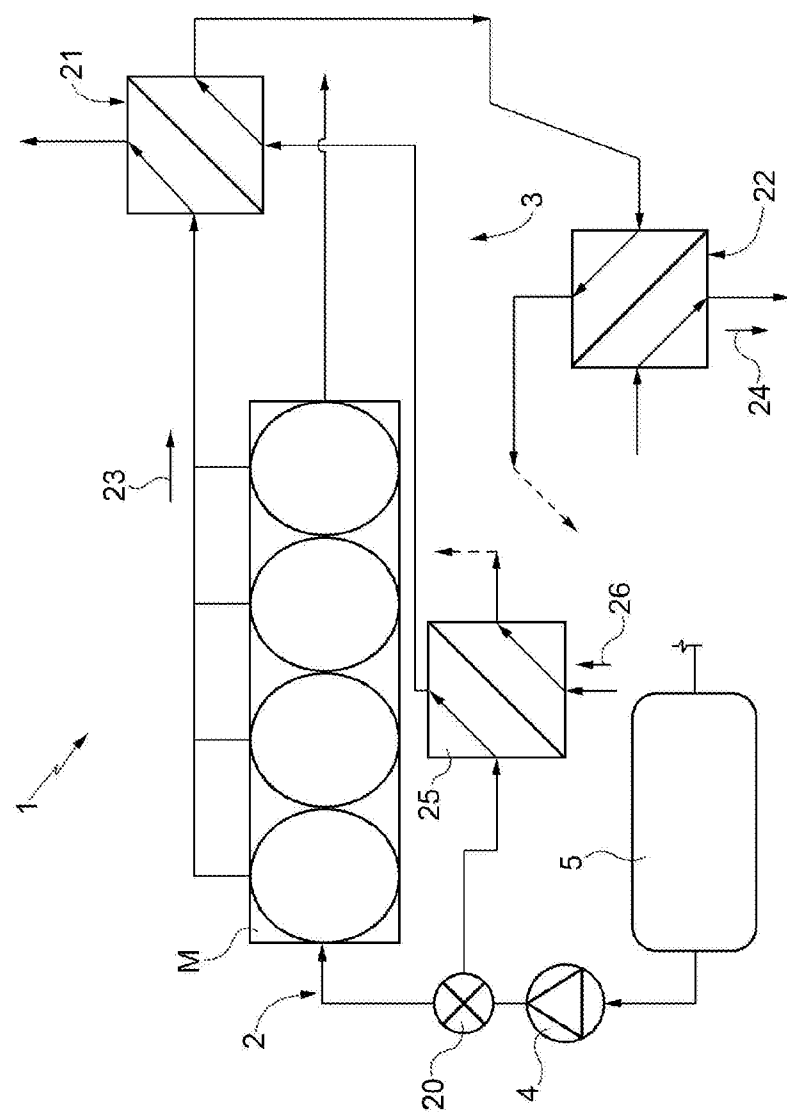


FIG. 10

A HYDRAULIC CIRCUIT ASSOCIATED WITH AN INTERNAL COMBUSTION ENGINE

TECHNICAL FIELD

[0001] The present invention relates to a hydraulic circuit associated with an internal combustion engine.

BACKGROUND ART

[0002] Engine cooling is the subject of major attention by manufacturers because it can significantly contribute to reducing primary pollutant levels.

[0003] Emission limits are defined on European level by the sequence of Euro 1-2-3-4-5-6 standards for passenger cars and Euro I-II-III-IV-V for heavy vehicles.

[0004] The evaluation of the aforesaid limits for light vehicles includes performing a mission with a predetermined speed profile as a function of time, starting from cold engine condition (NEDC cycle).

[0005] When performing the aforesaid cycle, the engine warms up in an interval of time which is equal to about $\frac{2}{3}$ of the total time test (1200 s). Therefore, most of the test is carried out before the engine has warmed up, and thus under disadvantageous conditions for emission levels.

[0006] A quick warm-up of the engine after cranking allows a considerable reduction of emissions; such a reduction is particularly significant, as mentioned, because it favors conditions emission determination according to the standards in force.

[0007] The most recent limitations introduced in the engine sectors concern CO₂ emissions, which are closely correlated to fuel consumption.

[0008] A faster engine warm-up also promotes consumption reduction for various reasons, including the reduction of the power lost by friction due to a faster reaching of the optimal viscosity conditions of the lubricant oil.

[0009] A further reduction of fuel consumption may be obtained by improving the organic efficiency of the engine, which implies reducing the power drawn by the auxiliary members of the engine itself. Of these members, the cooling fluid pump (also named "water pump") plays a significant role.

[0010] A centrifuge pump, which is dimensioned to obtain maximum efficiency under maximum engine power conditions, which corresponds to the maximum thermal power to be removed, is normally utilized to circulate the cooling fluid. When the pump is driven at slower speeds, such as those typical of the type-approval cycle, but also of most of the real operating conditions of the vehicle, especially in cities, efficiency is lower and the power drawn by the pump becomes significant for consumptions.

DISCLOSURE OF INVENTION

[0011] It is the object of the present invention to provide a hydraulic circuit associated with an internal combustion engine which allows to achieve energy advantages at least during the step of warming up the engine.

[0012] The aforesaid object is achieved by means of a circuit according to claim 1.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] For a better understanding of the present invention, a preferred embodiment will now be described by way of non-limitative example, with reference to the accompanying drawings, in which:

[0014] FIG. 1 is a diagram of a first embodiment of a hydraulic circuit according to the present invention;

[0015] FIG. 2 is a side elevation view of a pump of the circuit in FIG. 1;

[0016] FIG. 3 is a section taken along line III-III in FIG. 2;

[0017] FIG. 4 is an exploded perspective view of a rotor of the pump in FIG. 2;

[0018] FIG. 5 is a chart showing the characteristic curves of the circuit in FIG. 1;

[0019] FIG. 6 is a chart showing the efficiency trend of the pump in FIG. 2, as a function of the working pressure, in two operating conditions of the circuit in FIG. 1;

[0020] FIG. 7 is a chart showing the trend of the flow rate of the pump in FIG. 2 according to rotation speed variations;

[0021] FIG. 8 is a chart showing the trend of the flow rate of a conventional centrifuge pump according to rotation speed variations;

[0022] FIG. 9 is a chart showing the extra flow rate made available by the pump in FIG. 2 as compared to a conventional pump according to engine speed variations; and

[0023] FIG. 10 is a diagram of a further embodiment of the circuit of the invention.

BEST MODE FOR CARRYING OUT THE INVENTION

[0024] With reference to FIG. 1, reference numeral 1 indicates as a whole a hydraulic circuit associated with an internal combustion engine M, in particular for a motor vehicle.

[0025] Circuit 1 essentially comprises a main cooling circuit 2 (partially shown) and an auxiliary circuit 3 connected to and branched from the main cooling circuit 2 (also partially shown).

[0026] The main circuit 2 essentially comprises a circulation pump 4 of the cooling fluid (hereinafter referred to as "fluid" for conciseness) and a radiator 5. The main circuit 2 further comprises a thermostatic valve or thermostat (of conventional type and not shown), which is configured so as to assume two positions as a function of the fluid temperature: a closed position under a threshold temperature (i.e. with the "engine cold"), with which the fluid is recirculated between pump 4 and engine M without sending it to the radiator 5 to promote rapidly reaching a warmed up condition, and an open position, when the temperature of the fluid exceeds the aforesaid threshold value, which allows the circulation of the fluid through the radiator.

[0027] The main circuit, of which only one portion is shown, may be of any type and comprise, in addition to radiator 5, other heat exchangers, such as for example a heater for the air entered into the passenger compartment, a heat exchanger for cooling the EGR gases, etc.

[0028] According to a feature of the present invention, pump 4 is a rotary volumetric pump and preferably, but not necessarily, a vane pump.

[0029] According to a preferred embodiment of the invention, the vane pump 4 is made as shown in figures from 2 to 4, and comprises, in particular, a casing 6 defining a cylindrical cavity 7 of axis A, and a cylindrical rotor 8 mounted eccentrically within cavity 7 and integrally rotational with a shaft

14 about an axis B thereof. Rotor 8 has four radial vanes 9 arranged at 90°, adapted to substantially cooperate in a fluid-tight manner with the walls of cavity 7 to delimit four compartments 10 therewith having a volume varying with the rotation of rotor 8.

[0030] Casing 7 is further provided with an intake port 11 and with a delivery port 12, diametrically opposite to each other, with which the compartments 10 cyclically communicate.

[0031] As clearly shown in FIG. 4, the opposite vanes 9 are opposite in pairs and integrally defined by a single element 15 slidably housed in a respective diametral slot 16 of rotor 8.

[0032] The auxiliary circuit 3 (FIG. 1) branches off from the main circuit 2 by means of a three-way, two-position solenoid valve 20, arranged in the illustrated example immediately downstream of pump 4 (FIG. 3).

[0033] The auxiliary circuit 3 comprises a first heat exchanger 21, in which the fluid exchanges heat with (and absorbs heat from) a first fluid 23 at higher temperature, already available when the engine is cold, and a second heat exchanger 22, in which the fluid exchanges heat with (gives heat to) a second fluid 24 at lower temperature which it is intended to warm up as rapidly as possible.

[0034] According to a preferred embodiment, the first fluid 23 consists of the exhaust gases of engine M, and the second fluid 24 consists of the engine oil; alternatively, instead of the engine oil, the second fluid 24 may consist of air entered into the passenger compartment.

[0035] In order to understand the invention and favor comparison with the prior art, with reference to a rotary volumetric vane pump dimensioned to produce a desired volumetric flow rate $Q_{des}=100$ l/min at a rotation speed of $\omega_{des}=1000$ RPM with design head of $\Delta p_{des}=1$ bar, FIG. 5 shows:

[0036] a) the characteristic curves of pump 4 in a rotation speed range from 100 RPM to 1200 RPM (dashed line);

[0037] b) the characteristic curves of the main circuit 3 with the thermostat open (solid line) and closed (dash-dotted line);

[0038] c) the characteristic working points A-M of a type-approval cycle with a conventional type centrifuge pump. The following table shows the pressure difference Δp and flow rate Q values for each of the aforesaid working points, in addition to comparison data of a conventional centrifuge pump to a vane pump according to the present invention, and in particular:

[0039] a. rotation speed ω ;

[0040] b. power P ;

[0041] c. efficiency η

(the values related to the conventional centrifuge pump are tagged by subscript C and those of the volumetric vane pump V are tagged by subscript V)

Point	Δp [bar]	Q [l/min]	ω_c [RPM]	ω_v [RPM]	P_c [W]	P_v [W]	η_c	η_v
A	0.18	18	2000	200	35.6	20.9	0.15	0.26
B	0.41	27.5	3000	315	87	56.4	0.22	0.33
C	0.73	38.5	4000	400	189	123	0.25	0.38
D	1.14	48	5000	521	341	230	0.27	0.40
E	1.64	59.1	6000	639	593	384	0.27	0.42
F	0.16	36	2000	347	37	40	0.26	0.24
G	0.36	57	3000	557	89	131	0.39	0.26
H	0.65	76	4000	744	210	268	0.40	0.31
I	1.1	96	5000	935	379	505	0.42	0.32
L	1.47	116.5	6000	1148	653	836	0.44	0.34

[0042] With the thermostat closed, the higher efficiency of the rotary volumetric pump (of the vane type, in the example) as compared to a conventional centrifuge pump is immediately apparent. With the thermostat open and at high flow rates, the efficiency of the centrifuge pump is higher, but the higher power drawn by the vane pump is negligible under those conditions as the engine power is very high.

[0043] It is worth noting the following aspects for the purposes of the further advantage generated by a rotary volumetric pump.

[0044] The rotation speed of the rotary volumetric pump is determined in relation to the mass flow rate needed to cool the engine; such a flow rate value must be considered as necessary to ensure the cooling of the engine, and is the same which should be supplied by a centrifuge circulation pump of the conventional type.

[0045] Given V_{int} the geometric intake volume of the rotary volumetric pump (solely defined by the machine geometry), the geometric volumetric flow rate will be:

$$V_{geom} = \frac{V_{int} \cdot \omega \cdot n}{60} \quad (1)$$

[0046] ω indicating the rotation speed of the pump in RPM and n the number of compartments characteristic of the machine (equal to the number of vanes of a rotary vane machine).

[0047] As known, the fluid pressure is fixed by the volumetric features of the downstream circuit which, being characterized by a characteristic curve which defines the load losses as a function of the flow rate, will pressurize the fluid delivered by the pump: for such a reason, the delivery head will always be ensured by the engine circuit, the flow rate being equal.

[0048] Actually, the fluid recirculations between the adjacent compartments of the rotary volumetric pump increase as the delivery head increases, so that the fluid-dynamic flow rate (V_{flow}) tends to be different from the geometric flow rate expressed by equation 1. Such an effect is quantified as the volumetric efficiency of the rotary volumetric pump, which is maintained always sufficiently high in all cases (see FIG. 6, for example).

[0049] A control law of the rotation speed of the rotary volumetric pump can be defined by taking volumetric efficiency data into account. A law which corresponds to the real situation being discussed is shown in FIG. 7, in which the upper curve relates to the open thermostat condition and the lower curve to the closed thermostat condition.

[0050] The “intermediate” line represents an average value which may be considered as a good approximation of the two curves.

[0051] It is thus possible to observe that the flow rate delivered by the rotary volumetric pump is only one (regardless of the position of the thermostat, and thus of the load losses in the circuit) and thus linearly variable (as inferred from Equation 1) with the rotation speed of the pump itself. Instead, in the case of the centrifuge pump according to the prior art, the flow rate depends on the position of the thermostat because the working point is defined by the balance between the characteristic curve of the circuit (which is modified) and those of the pump. FIG. 8 shows the flow rates which correspond to a hydraulic circuit with the thermostat open and closed as a function of the rotation speed of the pump

mechanically connected to the thermal engine: the higher hydraulic permeability of the circuit with the thermostat open makes the circulating flow rates higher, the rotation speed being equal.

[0052] The comparison of FIG. 7 to FIG. 8 shows that with the thermostat closed (i.e. during the step of warming up the engine), the speed controlled vane pump (as shown in FIG. 7: a single rotation speed of the pump is identified for a defined cooling fluid flow rate) produces, with the thermostat closed, an “extra flow rate” with respect to the cooling needs of the engine. This extra flow rate is represented, in the discussed case, but with general validity, by the values shown in FIG. 9.

[0053] The same figure shows the possibility of binding the rotation speed of the rotary volumetric pump to that of the engine by means of a fixed rotation ratio (5:1).

[0054] Referring back to the diagram in FIG. 1, it is apparent from the above description that using a rotary volumetric pump, and particularly a vane pump, instead of a conventional centrifuge pump, makes available an extra flow rate of cooling fluid which, by means of the solenoid valve 20, may be utilized in the auxiliary circuit 3, during the step of warming up the engine to remove heat from the exhaust gases of the engine (which warm up very rapidly) and utilize such a heat to preheat the engine oil or the air entered into the passenger compartment.

[0055] FIG. 10 shows a variant of the auxiliary circuit 3 in which the extra flow rate, before reaching the heat exchanger 21, is circulated through a third heat exchanger 25, in which it exchanges heat with the supercharging compressed air 26 of engine M.

1. A hydraulic circuit (1) associated with an internal combustion engine (M) and comprising a main cooling circuit (2) of the engine (M) provided with a circulation pump (4) for a cooling fluid, characterized in that the pump (4) is a rotary volumetric pump driven by the engine (M) with a fixed transmission ratio and generating, at least during a warm-up transient of the engine (M), a flow rate greater than that required for the cooling of the engine (M) and by comprising an auxiliary circuit (3) branched from the main circuit and selectively connectable thereto by valve means (20), said valve means (20) being configured for sending to said auxiliary circuit (3), at least during said warm-up transient of the engine (M), at least a fraction of the flow rate of the cooling fluid pumped by the pump (4), said auxiliary circuit (3) comprising at least a first heat exchanger (21) wherein the cooling fluid absorbs thermal energy from a first fluid (23; 26) at a higher temperature, and at least a second heat exchanger (22)

wherein the cooling fluid transfers thermal energy to a second fluid (24) at lower temperature.

2. The circuit according to claim 1, characterized in that the first fluid (23) is constituted by engine exhaust gases.

3. The circuit according to claim 1, characterized in that the first fluid (26) is constituted by engine supercharging compressed air.

4. The circuit according to claim 1, characterized in that the second fluid (24) is constituted by engine oil.

5. The circuit according to claim 1, characterized in that the second fluid (24) is constituted by air entered into the passenger compartment.

6. The circuit according to claim 1, characterized in that the rotary volumetric pump (4) is a vane pump.

7. The circuit according to claim 6, characterized in that the vane pump (4) has a rotor (8) provided with an even number of vanes (9) and that vanes (9) diametrically opposite to each other are made from a single element (15) housed in a sliding manner in a diametral seat of the rotor (8).

8. The circuit according to claim 7, characterized in that the pump (4) comprises four vanes (8) obtained, two by two, by respective elements (15) housed in a sliding manner in a respective diametral slot (16) of the rotor.

9. The circuit according to claim 1, characterized in that said transmission ratio is a reduction ratio.

10. A method of controlling a hydraulic circuit (1) associated with an internal combustion engine (M), characterized in that it comprises, during the step of warming up the engine, the steps of:

generating a flow rate of cooling fluid by way of a rotary volumetric pump (4) in excess with respect to the flow rate required for cooling the engine (M);

utilizing at least a fraction of the flow rate pumped by the pump (M) to withdraw heat from a first fluid (23; 26) at a higher temperature and transfer heat to a second fluid (24) at the lower temperature.

11. The method according to claim 10, characterized in that the first fluid (23) is constituted by the engine exhaust gases.

12. The method according to claim 10, characterized in that the first fluid (26) is constituted by engine supercharging compressed air.

13. The method according to claim 10, characterized in that the second fluid (24) is constituted by the engine oil.

14. The method according to claim 10, characterized in that the second fluid (24) is constituted by air entered into the passenger compartment.

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