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(54) **HYDRAULIC PUMP**

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(57) **ABSTRACT**

A hydraulic pump includes a cylinder block having a cylinder, a rotor rotatable in the cylinder, and a vane movable in a radial direction of the rotor and biased to the rotor. The vane, the cylinder, and the rotor thereamong define an operation chamber. The rotor draws fluid into the operation chamber and sends the fluid outside the operation chamber. The vane is urged to the rotor according to differential pressure between pressure of fluid at high-pressure in the operation chamber and pressure of fluid at low-pressure in the operation chamber. The vane and the rotor define a hardness ratio being calculated by dividing hardness of the vane by hardness of the rotor, and the hardness ratio is greater than or equal to 1.6.

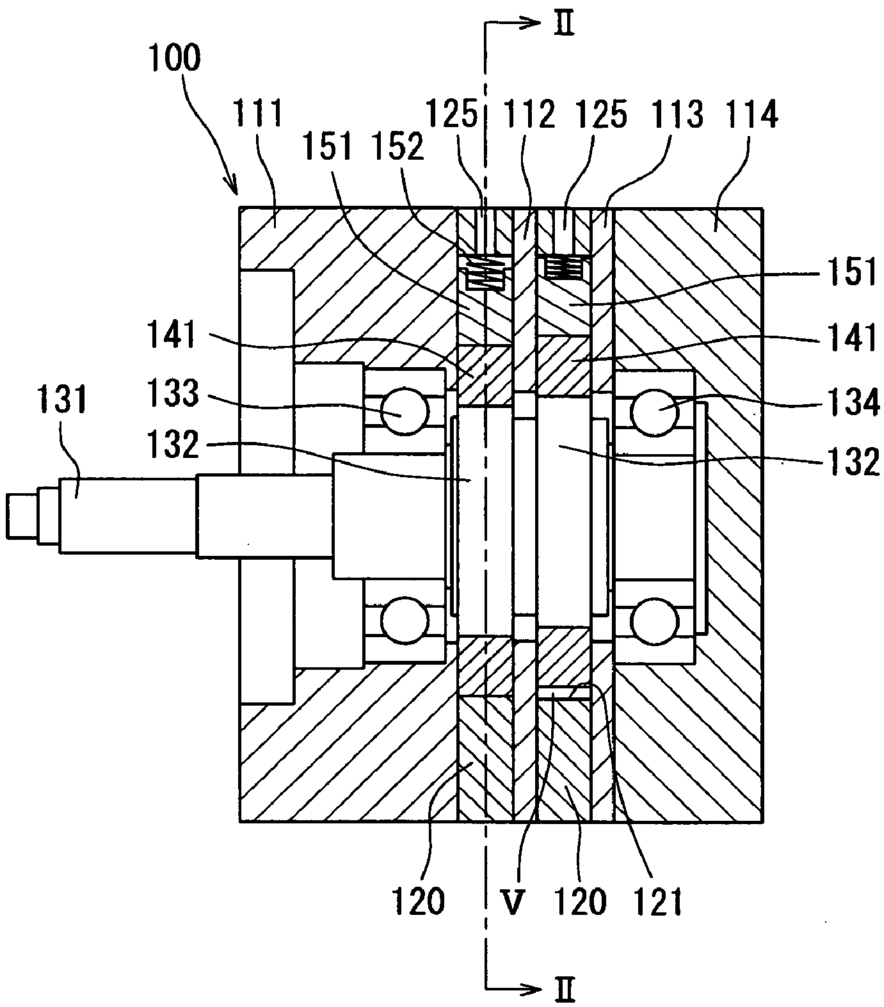


FIG. 1

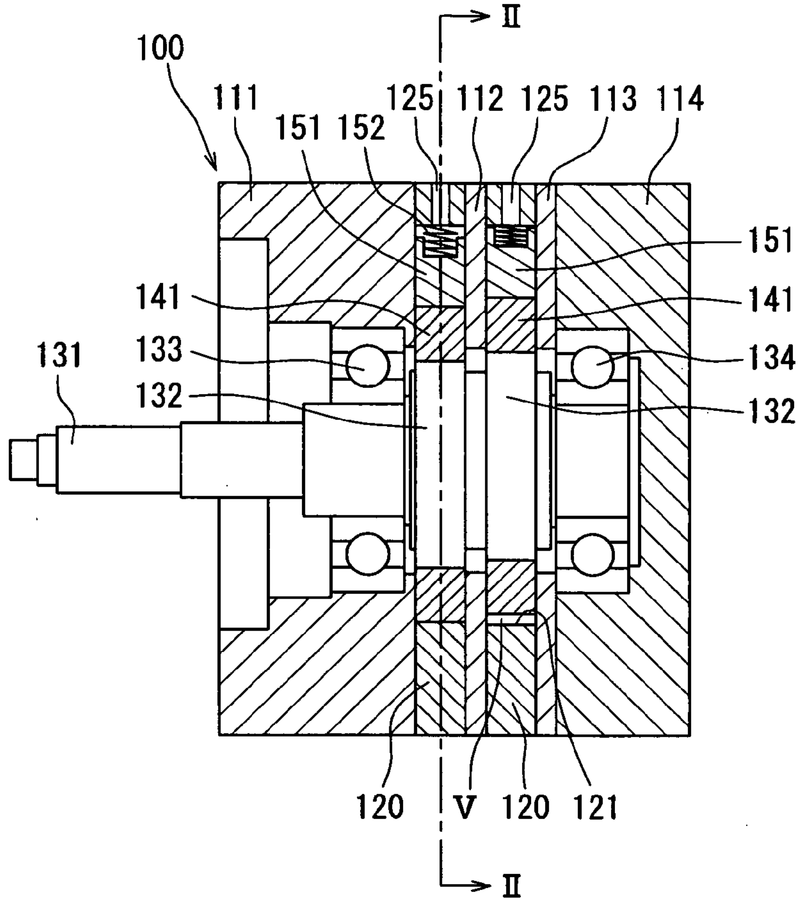

II
100
111
125
112
125
113
114
151
152
151
141
141
133
134
131
132
132
120
V
120
121
II

FIG. 2

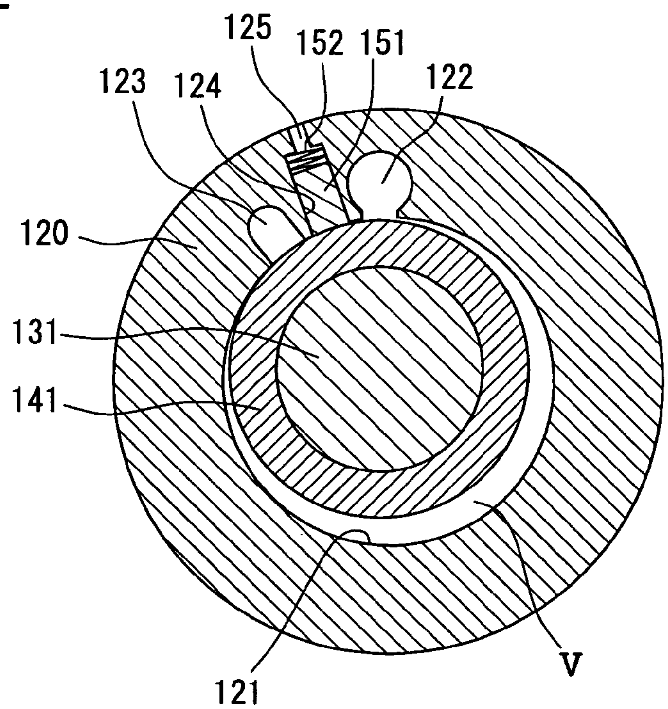

125
152
151
123
124
122
120
131
141
121
V

FIG. 3

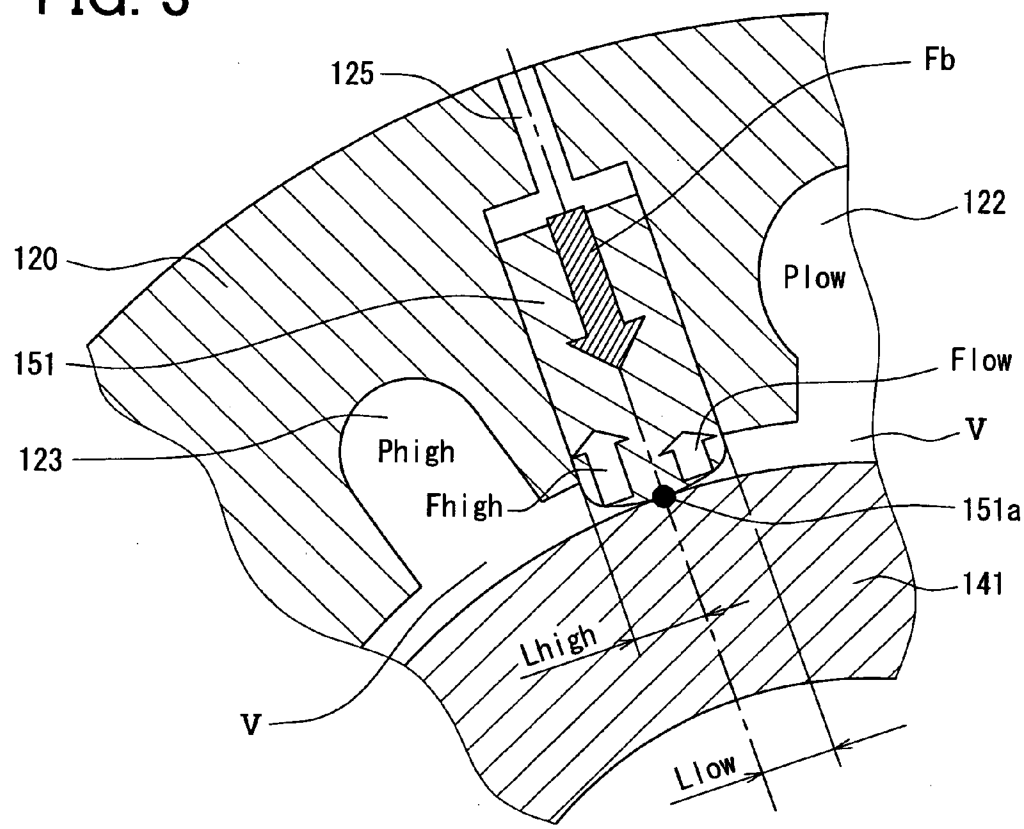

125
Fb
122
Plow
120
151
Flow
V
123
Phigh
Fhigh
151a
141
Lhigh
V
Llow

FIG. 4

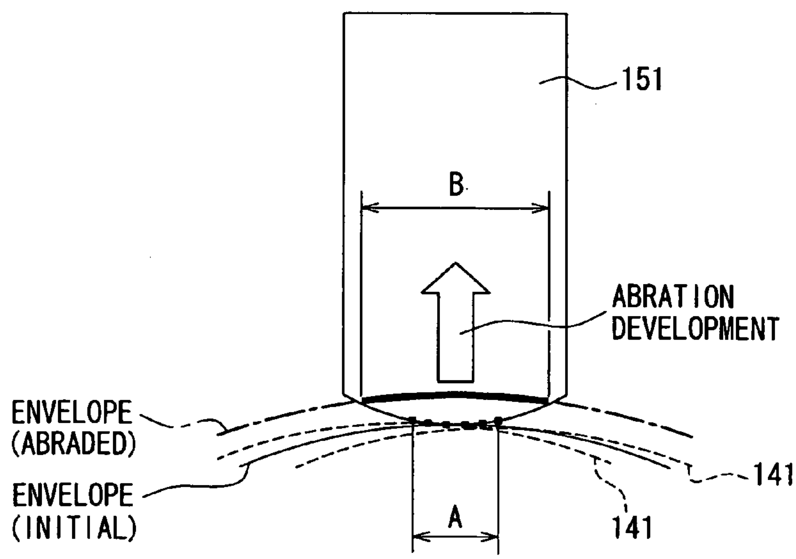

151
B
ABRATION
DEVELOPMENT
ENVELOPE
(ABRADED)
141
ENVELOPE
(INITIAL)
A
141

FIG. 5

LEVEL	VANE				ROTOR				HARDNESS RATIO	SPECIFIC WARE RATE (m^2/N)	
	BASE MATERIAL	SURFACE TREATMENT	HARDNESS Hv	Rz	BASE MATERIAL	SURFACE TREATMENT	HARDNESS Hv	Rz		VANE	ROTOR
1	SCM415	CARBURIZING, QUENCHING, AND TEMPERING	650～700	3.2	SCM415	CARBURIZING, QUENCHING, AND TEMPERING	650～700	3.2	1.0	1.02×10^{-16}	2.40×10^{-18}
2	SKH51	SOFTNITRIDING WITH QUENCHING AND TEMPERING	1100	0.3	↑	↑	↑	↑	1.6	1.40×10^{-17}	2.39×10^{-17}
3	SKD11	CrN	1700	↑	↑	↑	↑	↑	2.5	2.35×10^{-17}	1.10×10^{-16}
4	↑	TiN	2300	↑	↑	↑	↑	↑	3.0	7.64×10^{-18}	4.01×10^{-16}

FIG. 6

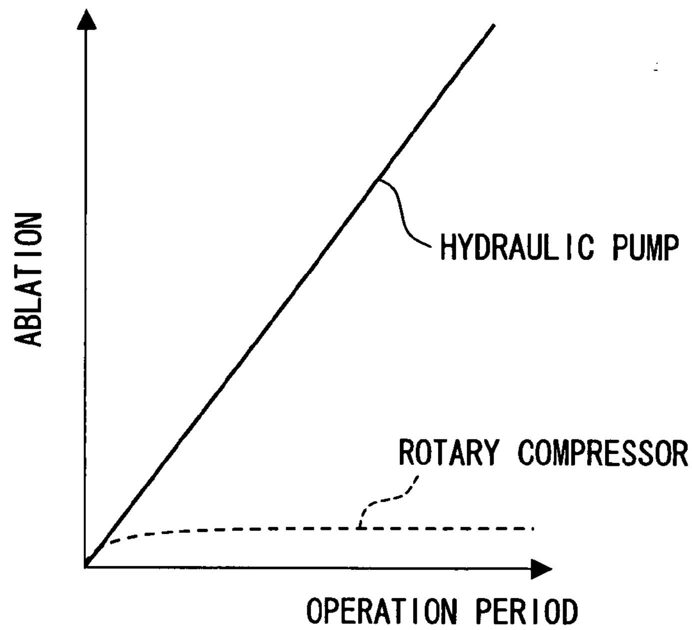
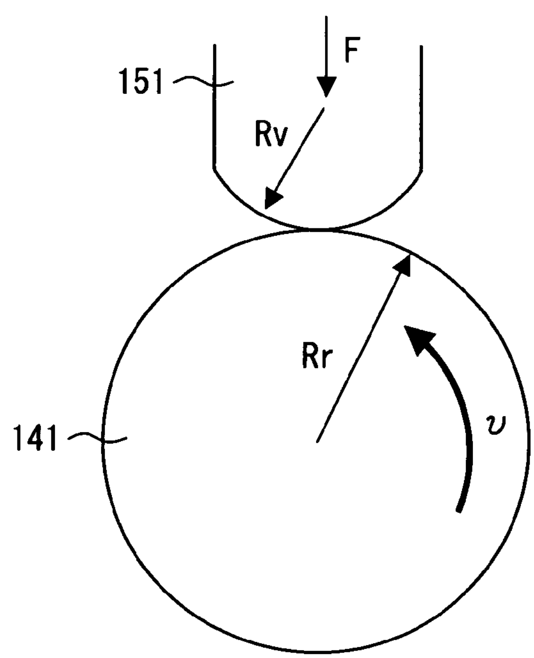

ABLATION
HYDRAULIC PUMP
ROTARY COMPRESSOR
OPERATION PERIOD

FIG. 7


F
151
Rv
Rr
141
v

HYDRAULIC PUMP

CROSS REFERENCE TO RELATED APPLICATIONS

[0001] This application is based on and incorporates herein by reference Japanese Patent Application No. 2007-91582 filed on Mar. 30, 2007.

FIELD OF THE INVENTION

[0002] The present invention relates to a hydraulic pump. In particular, the present invention may relate to a refrigerant pump for pumping refrigerant fluid to a heater in a waste heat recovery system such as a Rankine cycle system.

BACKGROUND OF THE INVENTION

[0003] For example, JP-A-63-277883 discloses a rotary compressor configured to be provided to an airconditioner, a refrigerator, or the like. In the present rotary compressor, a vane is provided to a cylinder accommodating a rotor. The vane is movable in a vane groove provided in the cylinder. The vane has a tip end urged to a rotor and the tip end of the vane is formed from a material being excellent in wear resistance compared with that of a body portion of the vane. In the present structure, the tip end of the vane is enhanced in wear resistance, and abrasion caused in a wall surface, which defines the vane groove, is reduced.

[0004] The rotary compressor is configured to compress vapor-phase refrigerant in a refrigeration cycle of, for example, an airconditioner. In the present structure, the vane and the rotor are capable of therebetween sufficiently forming an oil film in a steady operation, thereby maintaining therebetween a state of fluid lubrication. In the present structure, only in a transitional operation at the time of starting or stopping of the operation of the rotary compressor, the vane and the rotor are in a state of boundary lubrication where the vane and the rotor therebetween do not sufficiently form an oil film. Therefore, as indicated by a dashed line in FIG. **6**, development in abrasion accompanying time progress is not so significant in the rotary compressor. Therefore, in the rotary compressor, wear resistance needs to be considered only in the state of boundary lubrication.

[0005] However, in a hydraulic pump, which pumps low-viscosity fluid, a vane and a rotor are regularly in the state of boundary lubrication at any operations. Therefore, as indicated by a solid line in FIG. **6**, the vane and the rotor are apt to therebetween significantly develop abrasion accompanying time progress. Accordingly, only the above structure of the rotary compressor, in which the tip end of the vane is formed from the wear-resistive material, may not be practical for reducing abreaction in the hydraulic pump configured to pump low-viscosity fluid.

[0006] As follows, an exemplified premise for calculating a minimum oil film thickness t and an oil film parameter Λ is described with reference to FIG. **7**. In the present premise, fluid fed by the hydraulic pump has viscosity of 1 [mPa·s], a vane **151** has a tip radius Rv of 20 [mm], the vane **151** exerts load F of 4000 [N/m] per unit length to a rotor **141**, the vane **151** has surface roughness Rzv of 0.8, the rotor **141** has radius Rr of 20 [mm], the rotor **141** slides at sliding speed v of 1 [mm/s], and the rotor **141** has surface roughness Rzr of 0.32.

[0007] According to the premise and the elastohydrodynamic lubrication theory, the minimum oil film thickness t between the vane **151** and the rotor **141** and the oil film parameter Λ are calculated such that the minimum oil film thickness=0.03 [μm] and the oil film parameter Λ<1. According to the calculation, the vane **151** and the rotor **141** are obviously in the state of boundary lubrication in the hydraulic pump. Therefore, abrasion between the vane **151** and the rotor **141** in the hydraulic pump needs to be steadily suppressed so as to enhance product lives of both the vane and the rotor and maintain a performance of the hydraulic pump for a long period.

SUMMARY OF THE INVENTION

[0008] In view of the foregoing and other problems, it is an object of the present invention to produce a hydraulic pump having a vane and a rotor for pumping liquid-phase fluid, the hydraulic pump being capable of restricting ablation between the vane and the rotor and capable of maintaining a performance thereof.

[0009] According to one aspect of the present invention, a hydraulic pump comprises a cylinder block having a cylinder. The hydraulic pump further comprises a rotor rotatable in the cylinder. The hydraulic pump further comprises a vane movable substantially in a radial direction of the rotor and configured to be biased to the rotor. The vane, the cylinder, and the rotor thereamong define an operation chamber. The rotor is configured to draw fluid into the operation chamber and configured to send the fluid outside the operation chamber. The vane is configured to be urged to the rotor according to differential pressure between pressure of fluid at high-pressure in the operation chamber and pressure of fluid at low-pressure in the operation chamber. The vane and the rotor define a hardness ratio being calculated by dividing hardness of the vane by hardness of the rotor. The hardness ratio is greater than or equal to 1.6.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010] The above and other objects, features and advantages of the present invention will become more apparent from the following detailed description made with reference to the accompanying drawings. In the drawings:

[0011] FIG. **1** is a sectional view showing a hydraulic pump according to a first embodiment;

[0012] FIG. **2** is a sectional view taken along the line II-II in FIG. **1**;

[0013] FIG. **3** is an enlarged sectional view showing a vane, a rotor, and peripheral components of the hydraulic pump;

[0014] FIG. **4** is a schematic view showing an ablation developed in the vane;

[0015] FIG. **5** is a table showing specific wear rates of both the vane and the rotor of the hydraulic pump;

[0016] FIG. **6** is a graph showing a relationship between ablation, which is developed in each of a rotary compressor and a hydraulic pump, and an operation period; and

[0017] FIG. **7** is a schematic view showing a vane and a rotor being in contact with each other.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

First Embodiment

[0018] In the present embodiment, a refrigerant pump **100** is provided to pump liquid-phase refrigerant as fluid in a Rankine cycle for recovery of waste heat. The Rankine cycle as a waste heat recovery cycle is, for example, provided to a vehicle. Next, a structure of the refrigerant pump **100** is described with reference to FIGS. **1**, **2**. The waste-heat recovery Rankine cycle is constructed by combining the refrigerant pump **100**, a heater, an expansion device, and a condenser to define an annual circuit. The condenser condenses therein liquid-phase refrigerant, and the refrigerant pump **100** feeds the condensed refrigerant to the heater. The heater heats the liquid-phase refrigerant using waste heat of an internal combustion engine of the vehicle. The liquid-phase refrigerant in the heater is heated to be superheated steam refrigerant. The superheated steam refrigerant is fed into the expansion device, whereby the expansion device recovers kinetic energy caused by expansion of the superheated steam refrigerant. The refrigerant used in the present waste-heat recovery Rankine cycle is preferably used for a refrigeration cycle of an airconditioner of the vehicle.

[0019] The refrigerant pump **100** is a rotary-vane pump having a cylinder, which accommodates a rotor and a vane. In the present embodiment, the refrigerant pump **100** is a two-cylinder pump having two cylinders **121**, two rotors **141**, and two vanes **151**.

[0020] The refrigerant pump **100** has a body portion, which is constructed by combining a side plate **111**, one cylinder plate **120**, an intermediate plate **112**, another cylinder plate **120**, a side plate **113**, and a bearing holder **114** in order. Each of the components of the body portion of the refrigerant pump **100** is substantially in a flat cylinder shape.

[0021] The side plate **111** has one-end side on the left side in FIG. **1**, and the one-end side has a large opening being configured to be mounted with an additional device such as an electric motor as a driving source of the refrigerant pump **100**. The opening in the one-end side of the side plate **111** is reduced in diameter stepwise toward the other-end side on the right side in FIG. **1**, whereby the opening is communicated with the other-end side of the side plate **111**. The other-end side of the side plate **111** has a communication hole having an inner diameter, which is less than an inner diameter of the cylinder **121** of the cylinder plate **120**. The communication hole of the other-end side of the side plate **111** is inserted with a shaft **131**. The other-end side of the side plate **111** has a center portion fixed with a bearing **133**. The bearing **133** rotatably supports one-end of the shaft **131**.

[0022] The intermediate plate **112** and the side plate **113** are disc-shaped members respectively having center portions defining insertion holes. The insertion holes of the intermediate plate **112** and the side plate **113** are substantially the same as the opening of the side plate **111** on the other-end side in inner diameter. The bearing holder **114** is a bottomed cylindrical member having a center portion defining a recess on the one-end side. The recess of the bearing holder **114** is provided with a bearing **134**, which rotatably supports the other-end side of the shaft **131**.

[0023] Each cylinder plate **120** is a cylinder block member being in a disc shape. The cylinder plate **120** has a center portion defining the cylinder **121** being in a circular shape. One of the cylinder plates **120** is interposed between the side plate **111** and the intermediate plate **112**. The other cylinder plate **120** is interposed between the intermediate plate **112** and the side plate **113**. The side plate **111**, the one cylinder plate **120**, the intermediate plate **112**, the other cylinder plate **120**, the side plate **113**, and the bearing holder **114** are communicated with each other through the insertion hole and the cylinders **121**.

[0024] Each cylinder plate **120** has an inlet port **122**, an outlet port **123**, a vane groove **124**, and a backpressure feed port **125**, in addition to the cylinder **121**. In the present structure, the inlet port **122** as one communicating portion is provided for communicating an exterior of the cylinder plate **120** with an interior of the cylinder **121**. The outlet port **123** as another communicating portion is provided for communicating the interior of the cylinder **121** with the exterior of the cylinder plate **120**. The outlet port **123** is adjacent to the inlet port **122** with respect to a circumferential direction of the cylinder plate **120**. The vane groove **124** extends from the cylinder **121** radially outward in the cylinder plate **120**. The vane groove **124** is located circumferentially between the inlet port **122** and the outlet port **123**. The backpressure feed port **125** as another communicating portion is provided for communicating an interior of the vane groove **124** with a high-pressure chamber in an operation chamber V.

[0025] The shaft **131** has two portions correspondingly to the two cylinders **121**, and each of the two portions of the shaft **131** is provided with a circular cam portion **132**. The circular cam portion **132** is eccentric relative to an axis of the shaft **131**.

[0026] The rotor **141** is a flat cylindrical member and rotatably equipped to an outer circumferential periphery of the cam portion **132** via a bearing (not shown) for feeding liquid-phase refrigerant. An outer diameter of the rotor **141** is smaller than an inner diameter of the cylinder **121**. The rotor **141** is inserted into the cylinder **121** such that the rotor **141** is configured to revolve around the cam portion **132** in the cylinder **121**.

[0027] The vane **151** is a plate-shaped member and accommodated in the vane groove **124** such that the vane **151** is movable in the vane groove **124**. A spring **152** is interposed between a recess in a bottom of the vane groove **124** and the vane **151**. The spring **152** biases the vane **151** toward the rotor **141** such that a tip end of the vane **151** is in contact with an outer circumferential periphery of the rotor **141** mainly in a condition where the refrigerant pump **100** stops. The rotor **141** and the vane **151** define the operation chamber V in the cylinder **121**.

[0028] Here, hardness of a material of each of the vane **151** and the rotor **141** is predetermined by selecting a material of each of the vane **151** and the rotor **141** and selecting heat treatment applied to the material. More specifically, a hardness ratio Hr between hardness Hvv of the vane **151** and hardness Hvr of the rotor **141** is predetermined to be greater than or equal to 1.6. Here, the hardness ratio Hr is defined by dividing the hardness Hvv of the vane **151** by the hardness Hvr of the rotor **141**.

[0029] The refrigerant pump **100** with the structure described above has the inlet port **122** and the outlet port **123**. The inlet port **122** is connected with an outlet of the condenser, and the outlet port **123** is connected with an inlet side of the heater. When an electric motor (not shown) as a driving source drives the shaft **131**, each rotor **141** revolves around the cam portion **132** in the cylinder **121**, whereby the rotor **141** draws liquid-phase refrigerant from the condenser into the operation chamber V and pumps the liquid-phase refrigerant to the heater.

[0030] In this operation, the high-pressure chamber in the operation chamber V is communicated with the vane groove **124** through the backpressure feed port **125**, so that urging force is exerted to the vane **151** to urge the vane **151** onto the rotor **141** correspondingly to differential pressure between pressure in the low-pressure chamber and pressure in the high-pressure chamber. The urging force is steadily applied to the vane **151** to bias the vane **151** toward the rotor **141**, so that a tip end of the vane **151** is steadily maintained in contact with the outer circumferential periphery of the rotor **141**. In the present structure, the vane **151** is maintained in contact with the rotor **141**, whereby liquid-phase refrigerant in the high-pressure chamber can be restricted from leaking into the low-pressure chamber.

[0031] Next, the urging force exerted to the vane **151** is further specifically described with reference to FIG. **3**. Here, the spring **152** is provided in the vane groove **124**. The spring **152** exerts biasing force to the vane **151** to maintain the state where the vane **151** is in contact with the rotor **141** mainly in a state where the refrigerant pump **100** stops. In this structure, the spring **152** biases the vane **151** to restrict liquid-phase refrigerant in the high-pressure chamber from leaking into the low-pressure chamber when the refrigerant pump **100** starts operation. The biasing force of the spring **152** is significantly small and negligible compared with the urging force, which correlates with the differential pressure between the high-pressure chamber and the low-pressure chamber. Therefore, in the following description, the biasing force of the spring **152** is disregarded.

[0032] In FIG. **3**, a thickness direction of the vane **151** is defined as a horizontal direction (left and right direction), and a direction perpendicular to the sheet of FIG. **3** is defined as a depth direction of the vane **151**. The operation chamber V on the right side of the vane **151** is a low-pressure chamber directly communicating with the inlet port **122**. The operation chamber V on the left side of the vane **151** is a high-pressure chamber directly communicating with the outlet port **123**.

[0033] In an initial state, the tip end of the vane **151** is substantially in convex. The tip end has a center portion with respect to the thickness direction, and the center portion has the maximum projected portion relative to the rotor **141**. The tip end of the vane **151** and the rotor **141** therebetween have a contact portion in which the tip end of the vane **151** defines a contact point **151***a*. The contact point **151***a* is at a distance Lhigh from an end of the vane **151** on the high-pressure side, i.e., on the side of the high-pressure chamber. The contact point **151***a* is at a distance Llow from an end of the vane **151** on the low-pressure side, i.e., on the side of the low-pressure chamber. Backpressure Pb is applied to the vane **151** toward the rotor **141**. The high-pressure chamber is at pressure Phigh. The low-pressure chamber is at pressure Plow.

[0034] The vane **151** has a depth DP with respect to the depth direction orthogonal to the page of FIG. 3. The vane **151** is exerted with back pressure force Fb, which is calculated by the following equation 1.

$$Fb=Pb\times(L\mathrm{high}+L\mathrm{low})\times DP \quad (1)$$

[0035] The vane **151** is also exerted with pushback force Fhigh from the high-pressure chamber.

$$F\mathrm{high}=P\mathrm{high}\times L\mathrm{high}\times DP \quad (2)$$

[0036] The vane **151** is further exerted with pushback force Flow from the low-pressure chamber.

$$F\mathrm{low}=P\mathrm{low}\times L\mathrm{low}\times DP \quad (3)$$

[0037] Here, the backpressure Pb is equal to the pressure Phigh, and according to the above equations (1) to (3), urging force Fu, with which the vane **151** is exerted toward the rotor **141**, can be calculated by the following equation (4).

$$Fu=Fb-(F\mathrm{high}+F\mathrm{low})=(P\mathrm{high}-P\mathrm{low})\times L\mathrm{low}\times DP \quad (4)$$

[0038] Thus, the urging force Fu is determined correspondingly to the difference between the pressure Phigh in the high-pressure chamber and pressure Plow in the low-pressure chamber. The urging force Fu becomes large as the distance Llow increases.

[0039] As an operation period of the refrigerant pump **100** increases, abrasion between the vane **151** and the rotor **141** develops. Specifically, as shown in FIG. **4**, the convex portion of the tip end of the vane **151** is scraped substantially to be a flat surface extending along an envelope defined by revolution of the rotor **141**. In the initial state, the tip end of the vane **151** is in contact with the rotor **141** via a contact surface of a contact width A. As the vane **151** is scraped and worn out as the rotor **141** revolves, the contact width A increases to a contact width B. Therefore, at a time point in the operation of the refrigerant pump **100**, the contact point **151***a* largely moves toward the low-pressure chamber, and the distance Llow decreases. As a result, the urging force Fu exerted to the vane **151** decreases. Consequently, liquid-phase refrigerant may leak from the high-pressure chamber into the low-pressure chamber, and such leakage results in decrease in a pump performance of the refrigerant pump **100**. In addition, the vane **151** may fluctuates to cause further ablation, breakage in the components, and abnormal noise between components.

[0040] Therefore, in the present embodiment, the hardness ratio Hr is defined by the value of: (hardness Hvv of the vane **151**)/(the hardness Hvr of the rotor **141**), and the hardness ratio Hr is determined to be greater than or equal to 1.6. In the present structure, ablation between the vane **151** and the rotor **141** can be steadily reduced.

[0041] Specifically in the present embodiment, as shown in FIG. **5**, the base material of the vane **151** is selected from four kinds of tool steel including chrome molybdenum steel (SCM415), high-speed tool steel (SKH51), and alloy tool steel (SKD11). The surface of the vane **151** is applied with heat treatment including carburizing, quenching, and tempering (GC), softnitriding (TFG) with quenching and tempering (QT), chromium nitride coating (CrN), and titanium nitride coating (TiN). Surface roughness Rz of the vane **151** is predetermined to 3.2 or 0.3.

[0042] Here, the surface roughness Rz is, for example, a ten points average height as mean roughness depth calculated by measuring distances between highest five peaks and lowest five bottoms, then averaging the distances.

[0043] In the present embodiment, the base material of the rotor **141** is chrome molybdenum steel (SCM415) as case-hardened steel. The surface of the rotor **141** is applied with carburizing, quenching, and tempering (GC). Surface roughness Rz of the rotor **141** is predetermined to 3.2.

[0044] Vickers hardness (Hv) of each material applied with heat-treating is obtained. The hardness ratio Hr is defined by the following equation (5) using hardness Hvv of the vane **151** and hardness Hvr of the rotor **141**.

$$Hr=Hvv/Hvr \quad (5)$$

[0045] A specific wear rate WR [m2/N] of each of the vane **151** and the rotor **141** is defined by the following equation (6) using abrasion ABR [m3], load L [N], and sliding distance D [m].

$$WR[\mathrm{m2/N}]=ABR[\mathrm{m3}]/L[\mathrm{N}]\times D[\mathrm{m}] \quad (6)$$

[0046] As shown in FIG. **5**, a result indicating the specific wear rate WR of each of the vane **151** and the rotor **141** relative to each hardness ratios Hr is obtained by combining relations between the materials, the heat treatment, and the surface roughness Rz of the vane **151** and the rotor **141** at four levels. According to the result, as the hardness ratio Hr increases, the specific wear rate WR of the vane **151** substantially decreases, and the specific wear rate WR of the rotor **141** substantially increases. It suffice to determine the hardness ratio Hr to be greater than or equal to 1.6 so as to restrict the specific wear rate WR of the vane **151** to be less than the specific wear rate WR of the rotor **141**.

[0047] It is confirmed in advance that it suffices to restrict the specific wear rate WR of each of the vane **151** and the rotor **141** to be less than or equal to about 10 to the minus 16th power [m2/N] so as to restrict leakage of liquid-phase refrigerant. To satisfy the present condition of the specific wear rate WR, it suffice to restrict the hardness ratio Hr to be less than or equal to about 2.5. Thus, according to the result shown in FIG. **5**, both the levels **2**, **3** are determined to be preferable.

[0048] As described above, the specific wear rate WR of the vane **151** relative to the rotor **141** can be steadily suppressed by determining the hardness ratio Hr between the vane **151** and the rotor **141** to be greater than or equal to 1.6. Therefore, by determining the hardness ratio Hr to be greater than or equal to 1.6, the contact width of the vane **151** relative to the rotor **141** can be restricted from increasing. Thereby, the contact point between the vane **151** and the rotor **141** can be restricted from moving toward the low-pressure chamber as described above with reference to FIG. **4**. Thus, the pump performance of the refrigerant pump **100** can be maintained for a long period. The specific wear rate WR of, in particular, the rotor **141** can be suppressed by determining the hardness ratio Hr to be less than or equal to 2.5, and the pump performance of the refrigerant pump **100** can be significantly maintained.

Second Embodiment

[0049] The specific wear rates WR of the vane **151** and the rotor **141** can be further reduced by appropriately determining the surface roughness Rz of the vane **151** and the rotor **141**, in addition to the determination in the first embodiment.

[0050] According to the first embodiment, when the surface roughness Rz of the rotor **141** is reduced from 3.2 to 0.08 in the level **2** in FIG. **5**, the specific wear rate WR of the vane **151** can be reduced to a value of 8.63×10 to the minus 19th power [m2/N], and the specific wear rate WR of the rotor **141** can be reduced to 2.22×10 to the minus 18th power [m2/N]. In the present determination of the surface roughness Rz of the rotor **141**, the specific wear rate WR of the vane **151** is reduced by two orders of magnitude, and the specific wear rate WR of the rotor **141** is reduced by one order of magnitude. The specific wear rate WR is considered to be presently reduced, since abrasion between the vane **151** and the rotor **141** is reduced by the determination of the surface roughness Rz of the rotor **141**.

[0051] Therefore, it suffices to determine the surface roughness Rz of the vane **151** to be less than or equal to 0.3 and the surface roughness Rz of the rotor **141** to be less than or equal to 0.1 so as to reduce the specific wear rates WR of both the vane **151** and the rotor **141** to be less than or equal to 10 to the minus 18th power [m2/N].

Other Embodiments

[0052] In the first embodiment, the above structure is applied to the refrigerant pump **100** for the waste-heat recovery Rankine cycle provided to the vehicle. Alternatively, the above structure may be applied to various hydraulic pumps for pumping fluid. For example, the above structure may be applied to a hydraulic pump for a Rankine cycle provided to a stationary power generator or the like.

[0053] Various modifications and alternations may be diversely made to the above embodiments without departing from the spirit of the present invention.

What is claimed is:

1. A hydraulic pump comprising:
a cylinder block having a cylinder;
a rotor rotatable in the cylinder; and
a vane movable substantially in a radial direction of the rotor and configured to be biased to the rotor;
wherein the vane, the cylinder, and the rotor thereamong define an operation chamber,
the rotor is configured to draw fluid into the operation chamber and configured to send the fluid outside the operation chamber,
the vane is configured to be urged to the rotor according to differential pressure between pressure of fluid at high-pressure in the operation chamber and pressure of fluid at low-pressure in the operation chamber,
the vane and the rotor define a hardness ratio being calculated by dividing hardness of the vane by hardness of the rotor, and
the hardness ratio is greater than or equal to 1.6.

2. The hydraulic pump according to claim **1**, wherein the hardness ratio is less than or equal to 2.5.

3. The hydraulic pump according to claim **1**,
wherein the vane is formed from tool steel being quenched and tempered, and
the rotor is formed from case-hardened steel being carburized, quenched, and tempered.

4. The hydraulic pump according to claim **3**, wherein the vane is formed from the tool steel nitrided after being carburized, quenched, and tempered.

5. The hydraulic pump according to claim **1**, wherein the vane and the rotor therebetween define a sliding portion, in which:
a surface roughness Rz of the vane is equal to or less than 0.3, and
a surface roughness Rz of the rotor is equal to or less than 0.1.

6. The hydraulic pump according to claim **1**, the hydraulic pump being used for a vehicle.

7. The hydraulic pump according to claim **6**, the hydraulic pump being used for a waste heat recovery cycle for the vehicle.

8. The hydraulic pump according to claim **7**, wherein the waste heat recovery cycle is a Rankine cycle.

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