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(54) REFRIGERATION COMPRESSOR

(71) We, JOHN E. MITCHELL COMPANY, of 3800 Commerce Street, Dallas, Texas, United States of America, a Corporation of the State of Missouri, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to compressors and more particularly, to refrigerant compressors of the type having a lubrication system.

In automotive air conditioning systems comprising a refrigerant compressor having a so-called positive lubricating system, lubricating oil and refrigerant circulate together through the entire conditioning system. The concentration of oil in circulation depends on the initial charge of oil, the characteristics of the compressor in retaining oil therein, the compressor speed, and the refrigerant flow conditions. Normally, the amount of oil circulation in the system is directly proportional to compressor speed and inversely proportional to flow rate of the refrigerant.

Too much oil in the system impedes the cooling effect of the refrigerant and the above-described relationship regarding oil circulation in the system conflicts with optimized efficiency of operation.

The specific type of multi-cylinder wobble plate type compressor to which this invention will be related in a preferred embodiment described below is disclosed in U.S. Patent No. 3,552,886, to John W. Olson, Jr., issued January 5, 1971, which disclosure is incorporated by reference herein.

In such compressors, oil enters a rotor case, which encloses the wobble plate and a rotor for wobbling the latter, during the compression stroke of each piston because of normal blow-by of the oil past the piston rings.

Oil can leave the rotor case in two primary ways. Firstly, at some point in the suction stroke of each piston, the pressure in the rotor case is momentarily greater than that in the compression space, the latter be-

ing in the process of filling up with a charge of refrigerant oil mixture from the refrigerant system. Accordingly, a reverse blow-by is achieved from the rotor case into the cylinder and subsequently the oil is pumped out into the system by the scouring action of the piston rings. Secondly, blow-by equalizing holes are normally provided in an attempt to equalize the rotor case and suction pressures, by connecting the rotor case to the suction side of the compressor. Oil thus escapes from the rotor case through these equalizing holes whenever a pressure differential exists between the rotor case and the suction side of the compressor.

It is an object of the present invention to provide a new and novel refrigerant compressor and lubrications system therefore.

According to this invention there is provided a compressor comprising a housing, including a compression chamber; an oil sump chamber in the housing; a rotor disposed in the sump chamber and rotatable to drive a working member for compressing gas in said compression chamber; suction and discharge chambers; and means permitting communication of compressed oil-bearing gas from the compression chamber into the sump chamber to produce a pressure in the sump chamber above that of the suction chamber; a first oil flow passage extending radially inwardly of the rotor from a position within the sump chamber and being coupled at its radially inner end with the suction chamber, for transmission of oil from the sump chamber against the centrifugal action of the rotor tending to resist the transmission of the oil and thereby cause some of the oil introduced to the sump chamber to collect therein, and a second flow passage including a calibrated restrictor directly interconnecting the sump chamber with said suction chamber for transmitting oil from the sump chamber into the suction chamber, under the action of said pressure in the sump chamber, for preventing excessive accumulation of oil in the sump chamber at a predetermined rotor operating speed.

In order that the invention and its various

other features may be more easily understood an embodiment thereof will now be described by way of example only with reference to the drawings.

In the drawings:

Figure 1 is an end view of a compressor constructed in accordance with the invention illustrating the compressor drive shaft and adjacent oil porting;

Figure 2A is a cross-section taken along line 2A—2A of Figure 1 illustrating the internal structure and oil flow paths;

Figure 2B is a partial cross-section taken along line 2B—2B of Figure 1; and

Figure 3 is an end view of the opposite end of the compressor to that of Figure 1.

Referring in detail to the drawings, the compressor 10 is shown as including a one piece, substantially cylindrical cylinder block and rotor case assembly 12. The assembly 12 comprises a cylinder block 12A at one end and a hollow housing portion 12B at the other as can be seen in Fig. 2A.

The hollow portion 12B mounts a front cover plate 14 by means of a plurality of cap screws 14A; and by means of a plurality of bolts 15, the cylinder block 12A mounts a cylinder head 16 and a valve seat plate 18 to complete a closed housing assembly of the compressor 10.

The front plate 14 covers one open end of the assembly 12 while the cylinder head 16 and the valve seat plate 18, the latter positioned substantially coextensively with the inward surface of the former, cover the other open end of the assembly 12.

The interior of the assembly 12 comprises an oil sump chamber or reservoir 20 and serves further as a rotor case for the input rotor 22 and its co-operating wobble plate 24 which serve to reciprocate pistons 26 in cylinders 28 in the cylinder block portion 12A of the assembly 12 in the manner fully described in the herein before referenced U.S.A. Patent 3,552,886.

The pistons 26 co-operate with conventional reed or vane type suction valves (not shown) in a suction valve plate 30 juxtaposed between the assembly 12 and the valve seat plate 18, the latter being ported to respectively connect the cylinders 28 with peripherally disposed suction or input chambers 16A in the cylinder head 16 on the respective suction strokes of the pistons 26. All of the suction chambers 16A are in open communication with a refrigerant-oil input (suction) coupling 16B (Fig. 3) on the cylinder head 16 by means of appropriate conventional porting (not shown).

The cylinders 28 are selectively connected through respective conventional open ports in a suction valve plate 30 and valve seat plate 18, on respective compression strokes of the pistons 26, with a centrally disposed output or discharge chamber 16C within the

cylinder head 16 through conventional vane or reed type discharge valves 32, schematically shown in Figure 2A.

The discharge chamber 16C is directly connected to a discharge coupling 16D on the exterior of the cylinder head 16.

As is known in the art, the wobble plate 24 nutates upon a fixed bevel gear assembly 34 mounted centrally within the interior of the assembly 12, substantially symmetrically with the central axes of all the cylinders 28 therein.

The rotor 22 is keyed for rotation to an input drive shaft 36, substantially coaxially with the fixed bevel gear assembly 34, rotor 22 and wobble plate 24, the drive shaft 36 being journaled in a first needle bearing assembly 38 housed within the front cover plate 14. The front cover plate 14 also includes a shaft housing 14B which defines a shaft seal cavity 40 housing a shaft seal 42 both concentrically disposed about the drive shaft 36.

A first lubricating gap 44 is provided between the rotor 22 and the interior of front cover plate 14, about the drive shaft 36 by means of a first thrust bearing assembly 46 therebetween; illustrated as being one of the roller or needle bearing type.

A second lubricating gap 48 is provided between the rotor 22 and the wobble plate 24 by means of a second thrust bearing assembly 50 therebetween, illustrated as being of the roller or needle bearing type.

An axial lubricating passage 52 is provided in the drive shaft 36 and is connected by intersecting radial cross bores 54 in said shaft to the shaft seal cavity 40 in the front cover plate 14.

The axial passage 52 extends inwardly from the radial cross bores 54 into communication with the second lubricating gap 48 intermediate the rotor 22 and wobble plate 24, thus establishing a first path for the flow of oil from the sump chamber 20 through the said second thrust bearing assembly 50, second gap 48, the said axial passage 52 and radial cross-bores 54 into the shaft seal cavity 40.

Furthermore, a second path for the flow of oil exists from sump chamber 20, through the first thrust bearing assembly 46, first gap 44 and journal bearing 38 into the shaft seal cavity 40.

A third path for the flow of oil is established by a substantially radial flow passage 56 in the front cover plate 14 extending from the shaft seal cavity 40 into intersecting communication with a bore 58 in the peripheral shell of the assembly 12 which communicates directly through the valve seat plate 18 with the suction chambers 16A in the cylinder head 16 to complete the third path for the flow of oil from the input shaft seal cavity 40 to the said suc-

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tion chambers 16A. The radial flow passage 56 is closed at its outer end by a closure plug 56A.

Referring now to Figure 2B, a fourth path for the flow of oil is provided by means of a second longitudinal flow passage 60 in the peripheral shell of the assembly 12 which extends directly from the rotor case and oil sump chamber 20 to one of the suction chambers 16A in the cylinder head 16. A dumper orifice 60A is provided in the suction chamber adjacent the end of the second flow passage 60 and comprises a calibrated flow restrictor therein to permit sufficient oil to pass from the sump chamber 20 into the suction chambers 16A at relatively high compressor speeds of rotation to preclude overheating of the compressor 10. In prior art-compressors this overheating is caused by the efficient centrifugal scouring of oil from the rotating parts of the compressor 10, causing such a build up of oil in the sump chamber 20 that the oil physically interferes with the moving parts of the compressor 10. As a result, fluid friction between the moving parts and the oil in the sump 20 chamber causes overheating of the compressor 10 at speeds, for example, of the order of 5,000 r.p.m.

By means of the calibrated dumper orifice 60A, sufficient oil will be by-passed from the sump chamber 20 to the suction chambers 16A via the second flow passage 60 to preclude the occurrence of an excess (interfering) oil level in the sump chamber and rotor case 20 at such high compressor speeds.

A return path for oil from the suction chambers 16A is through the cylinders 28 by the well known phenomenon of blow-by past the pistons 26 (and their piston ring assemblies, not shown) onto the wobble plate 24 and bevel gear assembly 34, from whence this oil will flow downward into the sump chamber 20.

Assuming rotation of the input or drive shaft 36 of the compressor 10, and with particular reference to Figure 1 and 2, the operation of the present invention is as follows:

Oil and refrigerant are drawn into the cylinders 28 from suction chambers 16A by the pistons 26 during the suction strokes thereof. The charges of oil and refrigerant are compressed in the cylinders 28 and discharged into the discharge chamber 16C. During the compression strokes of the pistons 26 in the cylinders 28, oil is scraped off the walls of the cylinders 28 and returns to the oil sump chamber 20 as part of the blow-by that occurs past the rings (not shown) and the ring gaps (not shown) of the pistons 26 which are conventionally known.

The blow-by gas and a small quantity of oil is made to leave the sump chamber 20 via the first and second paths of oil flow described above.

To recapitulate, the first flow path for lubricating oil mixed with refrigerant gas is from the sump chamber 20 through the first thrust bearing assembly 46, first lubricating gap 44, journal bearing assembly 38 and onto the shaft seal cavity 40, thus lubricating both the first thrust bearing assembly 46 and the journal bearing assembly 38.

The second lubricating path achieved is from the sump chamber 20 through the second thrust bearing assembly 50, second lubricating gap 48, axial passage 52 in the drive shaft 22, radial cross bores 54 in the drive shaft 22, and thence into the shaft seal cavity 40. Therefore, the thrust bearing assembly 50 is adequately lubricated.

Gas and oil flowing into the shaft seal cavity 40 flows via the radial passage 56 in the front cover plate 14 and thence passes through the axially disposed bore 58 in the assembly 12 to the suction chambers 16A.

Thereafter, on an intake or suction stroke of the pistons 26 in the cylinders 28 this gas and oil mixture mixes with the incoming charge of refrigerant in the suction chambers 16A and thence is drawn into the cylinders 28 and a portion of oil which escapes the cylinders 28 via the blow-by action is thus recycled within the compressor 10 through the said cylinders.

It can now be readily seen that this method of lubrication traps an adequate quantity of oil in the sump chamber 20, where the action of the rotating-wobbling mechanism comprising the rotor 22 and wobble plate 24 causes the other components of the compressor 10, which are in need of lubrication, to be splash-lubricated adequately. The reason that this system traps oil is that in order for the refrigerant gas to escape through the lubricating paths described above, it must first get past the first and second thrust bearing assemblies 46 and 50, respectively, which are rotating and also which come in contact with the surface of the rotor 22 which is itself rotating.

Therefore, some particles of oil trying to enter the passages or lubricating gaps 44 and 48 between the foregoing elements are slung back into the sump chamber 20 through the action of centrifugal force imparted by the needle bearings in the thrust bearing assemblies 46 and 50 as well as the surface of the rotor 22. Notwithstanding this accumulation of the oil in the sump chamber 20, a small quantity of oil remains entrained in the refrigerant gas which is adequate for lubrication and does escape through the drive shaft 36 and shaft seal cavity 40 to be recycled into the suction chambers 16A as previously described.

As the speed of rotation of the rotor 22 and its related parts increases, the level of the oil in the sump chamber 20 is precluded from increasing beyond a proper optimized level for cool running of the compressor 10 by the escape of excess oil from the sump chamber 20, through the second flow passage 60 and the calibrated dumper orifice 60A into the suction chamber 16A, from whence a portion thereof is recycled.

As can be readily seen the above-described embodiment of the present invention provides refrigerant compressors with an oil lubricating system wherein the lubricating oil and refrigerant are mixed, and wherein the amount of oil in the oil sump of the compressor is constrained to a proper level and the amount of oil in circulation in the system with the refrigerant gas is minimized. Therefore, the above-described exemplary compressor provides maximum lubrication and maximum cooling efficiency which, heretofore, have been incongruous joint characteristics in the art.

A compressor incorporating the feature of circulation of the lubricant against centrifugal action and blow-by return of lubricant through the cylinders of a wobble plate compressor is described and claimed in our copending British Patent Application No. 45788/72 (Serial No. 1,412,776).

WHAT WE CLAIM IS:—

1. A compressor comprising a housing, including a compression chamber; an oil sump chamber in the housing; a rotor dis-

posed in the sump chamber and rotatable to drive a working member for compressing in said compression chamber; suction and discharge chambers; and means permitting communication of compressed oil-bearing gas from the compression chamber into the sump chamber to produce a pressure in the sump chamber above that of the suction chamber; a first oil flow passage extending radially inwardly of the rotor from a position within the sump chamber and being coupled at its radially inner end with the suction chamber, for transmission of oil from the sump chamber against the centrifugal action of the rotor tending to resist the transmission of the oil and thereby cause some of the oil introduced to the sump chamber to collect therein, and a second flow passage including a calibrated restrictor directly interconnecting the sump chamber with said suction chamber for transmitting oil from the sump chamber into the suction chamber, under the action of said pressure in the sump chamber, for preventing excessive accumulation of oil in the sump chamber at a predetermined rotor operating speed.

2. A compressor as claimed in Claim 1 incorporating a calibrated flow passage for lubricating oil substantially as described with reference to Figure 2B of the drawings.

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2. A compressor as claimed in Claim 1 incorporating a calibrated flow passage for lubricating oil substantially as described with reference to Figure 2B of the drawings.

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