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3,130,902

REFRIGERATOR COMPRESSOR

Filed Aug. 28, 1961

2 Sheets-Sheet 1

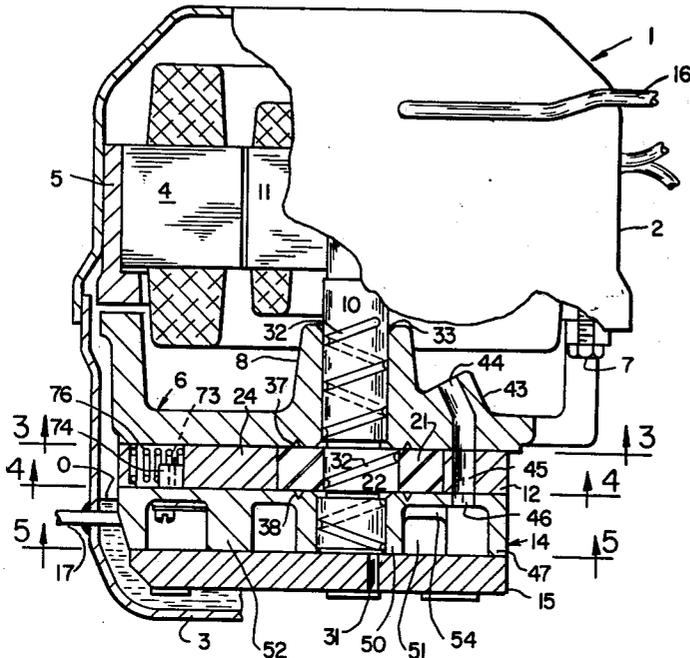


FIG. 1

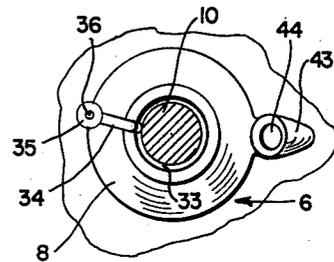


FIG. 2

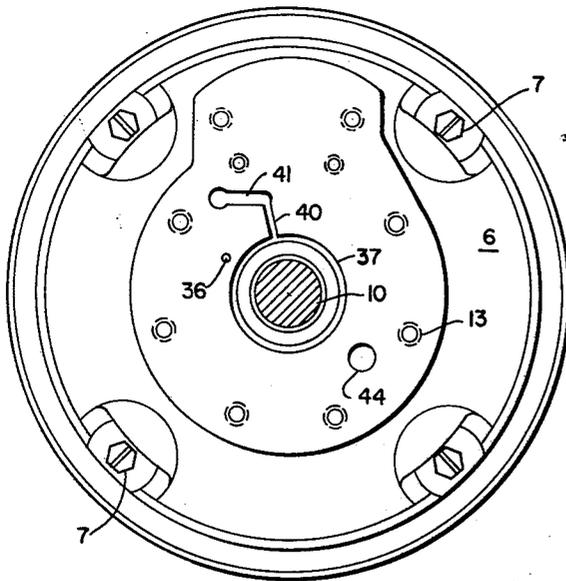


FIG. 3

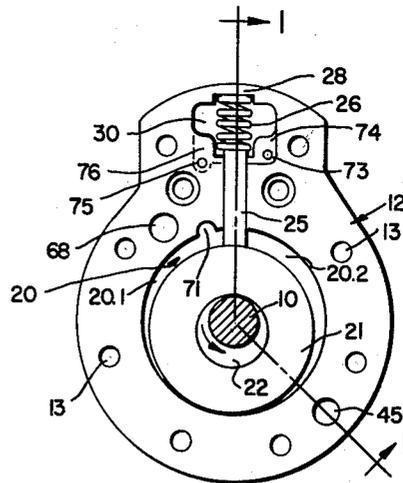


FIG. 4

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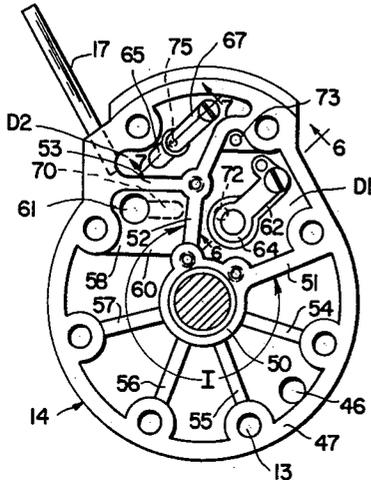


FIG. 5

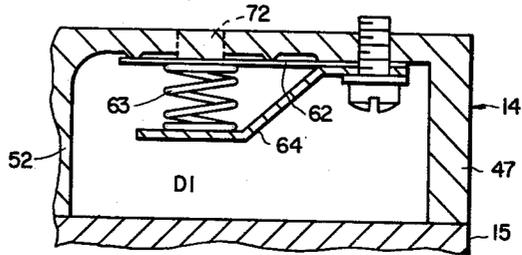


FIG. 6

FIG. 8

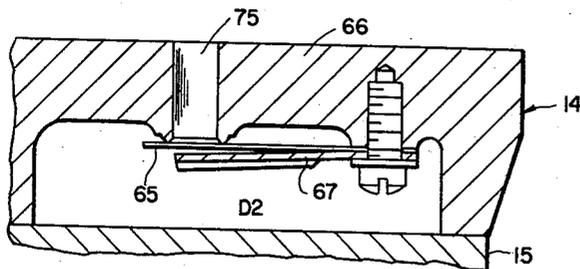
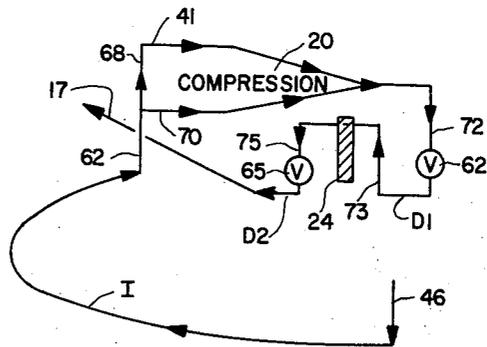


FIG. 7

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1

2

3,130,902

REFRIGERATOR COMPRESSOR

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 4 Claims. (Cl. 230-207)

This invention relates to refrigerant compressors, and in particular, to refrigerant compressors of the type which are hermetically sealed within a casing which forms a part of the low-pressure intake system of the compressor.

Although it is not restrictive thereto, the invention is particularly advantageous in a compressor of the rotary-piston type, for a wall structure comprising one of the cylinder closure elements may be adapted, in accordance with the invention, to cooling the compressor cylinder by heat transfer to the incoming low-pressure refrigerant gas.

It is an object of the invention to improve the thermodynamic efficiency of the refrigerator compressor by exposing large areas of the compressor structure to the cooling effect of low-pressure suction gases whereby these relatively cool gases will absorb the heat of compression at its source.

It is a further object of the invention to provide a refrigeration compressor in which temperature differences between the end plates of the cylinder are minimized, thereby tending to equalize the expansion and contraction of said plates relative to the cylinder.

It is another object of the invention to provide a refrigerant compressor in which means are provided for efficiently reducing suction gas noises.

In a presently preferred form, a refrigerant compressor embodying the invention comprises a rotary compressor of the type in which the suction gases enter the hermetic casing within which the compressor is sealed, and flow from said casing to the inlet of the compression cylinder. Broadly, such a mechanism is disclosed in Tarleton U.S. patent application, Serial No. 662,441, filed May 29, 1957 (now U.S. Patent 3,003,495 granted October 10, 1961) for "Refrigeration Apparatus" and assigned to my assignee herein. In the Tarleton construction, one of the cylinder end plates is exposed to the cooling effect of the low-pressure gaseous refrigerant, and the other comprises a muffler for the hot discharge gas, whereby the lubricating oil within which this latter plate is immersed, comprises substantially the sole cooling medium.

I have found it very advantageous to pass the incoming low-pressure refrigerant through a muffler chamber of large area in said latter plate, thereby not only reducing the compressor valve noises attending the inflow of refrigerant, but also providing for absorption of the heat of compression by direct heat transfer from the cylinder itself.

Pursuant to my invention, the cool low-pressure gas flows about and through one of the cylinder end plates, and then directly into an enlarged muffler mechanism in the opposite cylinder end plate. From this muffler mechanism, it passes into the inlet area of the cylinder. Because the muffler mechanism comprises an end wall of the compressor cylinder, it is therefore in direct heat transfer relation therewith. In a preferred construction, the intake gas muffler has an effective heat transfer area which comprehends not only a substantial portion of the area of the cylinder cavity, but a substantial portion of the surrounding cylinder wall as well. It will be apparent that the wall structure comprising the first-named cylinder end plate is arranged for optimum heat transfer from the cylinder to the incoming low-pressure refrigerant within the casing, whereupon each of the cylinder end plates is maintained at substantially equivalent temperatures, thus minimizing variances in expansion of one wall structure

relative to the other and consequent improvement in the sealing of the cylinder against leakage of high-pressure gas. In addition, the substantial reduction in operating temperatures resulting from the application of the present invention to compressors of the rotary-piston type, minimizes the disintegration or carbonization of the lubricating oil at the discharge valve system.

A further advantage deriving from the present invention is the reduction of the operational noise of the compressor with particular reduction of the "popping" noise usually attributed to the valve action at the inlet side of a rotary compressor. The present invention contemplates subdividing a cylinder closure wall structure into inlet and discharge muffler systems in which the inlet muffler system is substantially the larger and provides the primary heat exchange facility, as above noted.

Other features and advantages of the invention will be apparent from the following detailed description of a presently preferred form thereof, read in connection with the accompanying drawings in which:

FIG. 1 is a partial side sectional elevation of a rotary-piston type refrigeration compressor taken on lines 1-1 of FIG. 4;

FIG. 2 is a fragmentary top plan view of the main frame of the compressor, showing the gas passage and lubricating oil entry port;

FIG. 3 is a plan view of the main frame looking in the direction of the lines 3-3 of FIG. 1;

FIG. 4 is a plan view of the gas compression mechanism looking in the direction of lines 4-4 of FIG. 1;

FIG. 5 is a plan view of the valve plate looking in the direction of the lines 5-5 of FIG. 1;

FIG. 6 is an enlarged side sectional elevation through one of the discharge muffler chambers showing a preferred arrangement of the discharge valve structure;

FIG. 7 is an enlarged sectional elevation taken through another of the discharge gas muffler chambers showing a preferred form of discharge check valve; and

FIG. 8 is a schematic diagram showing the path taken by the incoming and the compressed refrigerant during operation of the compressor.

Referring now to FIG. 1 a refrigerant compressor 1 embodying the invention includes the hermetically sealed upper and lower housing parts 2, 3 forming a sealed casing within which the motor and compressor operate. The motor 4 is advantageously of the two-pole induction type having a free-running speed of about 3600 r.p.m. The motor is conventional, having its stator secured within a supporting structure 5 fastened securely to the inner wall of the upper housing 2. A main frame structure 6 of the compressor is fastened to the stator-supporting structure 5 as by a suitable plurality of machine screws 7. Said main frame structure provides a bearing 8 for the compressor shaft 10 to the upper end of which the rotor 11 of the motor is affixed. It will be understood that the foregoing parts and the description thereof are by way of illustration only and merely typify a conventional arrangement, well known in the domestic refrigerator art.

The lower surface of the main frame 6 provides the top of the cylinder 12. The so-termed valve plate 14 provides the bottom closure for the cylinder; and the basic structure of the compressor is completed by the relatively heavy thrust plate 15. As is obvious, the structural elements are rigidly secured together by a suitable number of bolts (not shown) passing through the several bolt holes, such as 13, in the component parts. These conventional fastenings hold the assembly tight under the substantial pressures generated by the compressor. The lower portion of the compressor is immersed in lubricating oil, the usual level O of which is such that the valve plate 14 may be almost totally within the body of

3

oil. Low-pressure refrigerant from the evaporator (not shown) is introduced into the housing through the intake tube 16 and compressed refrigerant is discharged from the compressor to the condenser (not shown) through the discharge tube 17.

Referring now to FIG. 4, the cylinder 12 has the circular cavity 20 accommodating the rotor 21 which occupies the full depth of the cylinder. That is, the upper and lower surfaces of the rotor are in rubbing contact with the lower and upper surfaces of the main frame 6 and valve plate 14 respectively. The rotor accommodates the eccentric 22 of the shaft 10, whereupon rotation of the shaft causes the familiar planetating action of the rotor which traverses the cavity 20. An oil film of minute thickness builds up between the walls of the rotor and that of the cavity 20 to lubricate and to seal, as is well known in the art. Also, and as more fully explained in the aforementioned Tarleton application, an oil sealing and lubricating film is generated on the top and bottom surfaces of the rotor and the eccentric. By means of the reciprocating blade 24, which is slideable within the slot 25, and in slideable contact with the surfaces of the main frame and the valve plate, the cavity 20 is divided into a low pressure or intake portion 20.1 and a high pressure or discharge portion 20.2, it being understood that the blade 24 is in constant engagement with the rotor 21. This engagement is maintained by the effort of the spring 26, bottomed within a pocket in the rear wall 28 of a high pressure gas chamber 30, and by the pressure of the high-pressure gas against the rear wall of the blade 24, as later explained.

The rotor and other rotating and reciprocating parts are lubricated by the metered introduction of oil which is at the low pressure of the incoming refrigerant. Briefly stated, the oil reaches the lower end of the shaft 10 through the hole 31 in the thrust plate 15 and it is then pumped by way of the continuous screw 32 to the cup-like depression 33 at the top of the bearing 8. On the way to the top of the bearing, it is obvious that lubrication of the rotary shaft is accomplished and, of course, the oil lubricates the engagement of the eccentric 22 and the rotor. From the depression 33 the oil flows along the groove 34 (FIG. 2) entering the cup 35 where it fills the passage 36 communicating between the bottom of the cup and the low pressure side 20.1 of the cylinder cavity. As explained in detail in Tarleton, the point of entry of the passage 36 is such that the rotor 21 sweeps over it and uncovers it for the entry of oil into the cylinder cavity only for a brief interval during each rotation of rotor 21. Also according to Tarleton, the main frame (FIG. 3) and the valve plate (FIG. 1) are provided with the circular intercepting grooves 37 and 38. Groove 37 communicates by way of the branch 40 with a channel 41 along which low pressure gas enters the cavity portion 20.1, as presently described. It will be understood that the valve plate has a similar branch and channel, later identified. The intercepting grooves 37 and 38 prevent a high-pressure gas condition from reaching the pumping screw 32, for if this occurred, the high-pressure gas would blow through the groove, and among other things, interfere with the pumping of lubricant.

It is well recognized in the refrigeration art that low-pressure refrigerant gas can be used to cool the motor windings, for in a low-pressure crank case system, the motor operates in the low-pressure gas atmosphere. In prior art compressors, however, the gas is then brought directly into the low-pressure side of the cylinder. I have found that this does not use the cooling capacity of the gas to maximum effect; and the present invention therefore departs in an important way from the prior art by introducing the low-pressure gas into the cylinder after first causing it to pass through the valve plate where it is in direct contact with the wall forming the bottom closure plate of the compressor. The gas is therefore in a position to abstract heat from the hottest portion of the

4

compressor system and transfer the heat of the valve plate to the thrust plate and thence to the body of lubricating oil in which the plates are submerged. Additionally, I modify the conventional valve plate to constitute first and second muffler systems in which the muffler system of largest area and effect comprises a part of the low-pressure gas inlet system. Among other advantages, this enlarged low-pressure side and muffler very materially reduces the "popping" and other noises which usually accompany the intake movement of the low-pressure gas. The second muffler system is of less area and volume, but as presently explained, the directions of passage of the gas from the cylinder to the exterior of the casing are such as to introduce a substantial muffling effect.

Referring again to FIGS. 1 and 2, it will be seen that the main frame 6 is formed with a boss 43 which rises angularly from the base of the frame, and that the boss is drilled to provide the angular passage 44. This passage is of the order of .375 inch in diameter in a compressor of the usual 1/8 horsepower size common in domestic refrigeration. This passage therefore accommodates flow of refrigerant with minimum friction and wire-drawing losses. The passage 44 communicates directly with a passage 45 in the cylinder wall and this in turn with a passage 46 in the valve plate 14.

FIG. 5 shows the underside of the valve plate to illustrate the muffler areas. Muffler area I is defined by the marginal wall portion 47 and the inner wall 50, and by the walls 51, 52 and 53, all of which are full depth so as to seal against the upper surface of thrust plate 15 (FIG. 1). The intake muffler I is also characterized by the ribs 54, 55, 56, 57 (FIG. 5) which extend partially down from the roof of the muffler chamber. The intake muffler chamber terminates in a chamber 58 which has a ceiling 60 which is substantially at the level of the ribs 54, 55 etc.; whereby the chamber 58 is of less height than the remainder of the muffler area of the intake muffler mechanism. The incoming gas leaves the muffler area by flow through the passage 61, presently explained. The discharge muffler is actually a two-compartment structure identified in FIG. 5 as D1 and D2, which are respectively defined by portions of the peripheral wall 47, wall 50 and the walls 51 52 and 53, it being noted that the reference character 52 comprehends the portion of the wall projecting upwardly and to the right of wall 53, as viewed in FIG. 5. Chamber D1 accommodates a reed valve 62 (FIG. 6) which is spring-biased to closed position by a coil spring 63 bottomed on the right bracket 64, screw-fastened to the upper wall of the valve plate. Said valve 62 is the discharge valve and is relatively heavily loaded. Similarly, chamber D2 accommodates a reed valve 65, self-biased to closed position and screw-mounted on the relatively thicker ceiling 66 of the valve plate 14 by means including the rigid back-up plate 67. The valve 65 is a check-valve preventing return flow of high-pressure gas from the chamber D2. The ceiling 66 of chamber D2 is at the level of the ceiling 60 of the intake muffler compartment I, whereupon the height of the muffler chamber D2 and the chamber 58 of the intake muffler I have the same relationship to the height of the muffler chamber D1 and to that of the remainder of the intake muffler I. This height relationship is apparent from comparison of FIGS. 6 and 7 which are drawn to the same enlarged scale.

The path of gas from the intake muffler I to the high-pressure discharge tube 17 can best be understood from FIGS. 3, 4 and 5 plus the schematic of FIG. 8. The passage 61 (FIG. 5) communicates directly with a passage 68 through the wall of the cylinder 12 (FIG. 4). The upper end of passage 68 communicates with the channel 41 formed in the bottom wall of the main frame 6 shown in FIG. 3; and as previously indicated, said channel 41 is duplicated as channel 70 in the upper surface of the valve plate 14, whereupon a part of the inlet gas leaving passage 61 flows through the last named channel 70 and

the remainder through channel 41. Each of the channels 41 and 70 terminates at the groove 71 in the vertical wall of the cylinder (FIG. 4), whereupon gas is fed into the low-pressure cylinder area 20.1 throughout the depth of the cylinder. As the eccentric 22 rotates in counterclockwise direction indicated in FIG. 4, the gas is compressed until at maximum pressure, determined by the resistance of discharge valve spring 63, the gas passes from the high-pressure cylinder portion 20.2 through the port 72 in the valve plate 14. Said port 72 is substantially tangential to the wall of the divider plate 24 for maximum usage of the cylinder space. The compressed gas thus enters muffler chamber D1, where it expands because of the relatively large volume of said chamber. The gas then passes upwardly through passage 73 (FIGS. 4 and 5) to enter the chamber 30 within which a portion of the blade 24 reciprocates. The high-pressure gas therefore exerts a positive effort on the rear of the blade to supplement spring 26 in maintaining the blade tightly against the rotor 21. It will be noted from FIGS. 1 and 4 that the passage 73 extends upwardly through a wall 74, whereupon the gas is caused sharply to change its direction at the upper portion of the chamber 30. The gas leaves the chamber through the passage 75 in a similar wall 76, whereupon it enters the muffler chamber D2 after traversing the check-valve 65. As earlier shown the chamber D2 is of less volume than D1 and there is less expansion of the gas within chamber D2. The expansion within the chambers D1 and D2 however, and the abrupt changes in direction of movement of the gas, makes said chambers effective as mufflers. Further, the abrupt changes in direction of the gas within the chamber 30 effects a separation of oil which the gas will have absorbed during the compression cycle. This oil accumulates within the chamber 30 and provides a reservoir of lubricant for the blade 24.

It will be noted from a comparison of FIGS. 4 and 5 that the intake muffler mechanism I comprehends more than 75 percent of the area of the cylinder cavity 20 and is in optimum position to absorb heat from all but the final compression operation. Also, the intake muffler area extends into heat transfer relation with a substantial portion of the cylinder wall itself, whereupon the relatively cool incoming gas can absorb a substantial portion of the heat of said wall. Also, of course, the thick thrust plate 15 provides a substantial heat sink, for it as well as the side wall of the valve plate, is submerged in the lubricating oil. The lubricating oil in turn is in heat transfer relationship with a very large area of the lower casing structure 3 which provides a large and efficient heat transfer structure. It will also be remembered that the upper frame member 6 is cooled by the incoming gas, whereupon the cylinder is exposed to an efficient cooling system at its top and bottom. A very practical effect of this cooling effort is a lessening of expansion and contraction differences between the various structural components resulting in better sealing of the parts against loss of compression.

Of the discharge muffler system only a comparatively small portion of the chamber D1 is in heat exchange relation with the cylinder and rotor; chamber D2 is not in a heat transfer situation therewith. There is, of course, a heat flow from the discharge muffler D1 to the relatively cooler walls of the intake muffler mechanism, resulting in a cooler valve port and valve reed at the discharge valve area. It has been noted that this cooler valve condition prevents a breakdown or disintegration of the lubricating oil content of the high-pressure gas, whereupon there is no accumulation of oil residue about the valve seat.

While there has been described what is at present considered to be the preferred embodiment of the invention, it will be understood that various modifications may be made therein, and it is intended to cover in the appended claims all such modifications as fall within the true spirit and scope of the invention.

What is claimed is:

1. Refrigerating apparatus comprising a sealed casing having an inlet for gaseous refrigerant to be compressed, said casing having a crankcase openly communicating therewith for containing a quantity of lubricating oil, a refrigerant compressor arranged in said casing and having a cylinder, a rotary piston in said cylinder, a crankshaft for planetating said piston, blade means in said cylinder engaging said piston to divide said cylinder into intake and discharge portions, upper and lower wall members in covering relation to said cylinder for sealing the same, said lower wall member being substantially submerged in lubricating oil and having a plurality of cavities respectively forming mutually independent low-pressure and high-pressure gas sound muffling mechanisms in its lower portion, said low-pressure sound muffling mechanism having an area substantially greater than said high-pressure mechanism and being in heat transfer relation with a large proportion of the cylinder area traversed by said piston, said high-pressure gas sound muffling mechanism including first and second compartments each communicating with said cylinder rearwardly of said blade means whereby said blade means is urged against said piston by the pressure of said refrigerant, passage means including said upper wall means and said first-named sound muffling mechanism for conducting gaseous refrigerant from said casing to said cylinder for compression therein, passage means including a valve mechanism for discharging compressed refrigerant from said cylinder to the first compartment of said second-named muffling mechanism, passage means for conducting said compressed refrigerant to said cylinder rearwardly of said blade means, passage means including a valve mechanism for conducting said compressed refrigerant therefrom to the second compartment of said second-named muffling system, passage means for conducting said refrigerant from said second compartment to the exterior of said casing, and motor means connected to said crankshaft for operating said rotary piston.

2. Refrigerating apparatus comprising a sealed casing having an inlet for gaseous refrigerant to be compressed, said casing having a crankcase openly communicating therewith for containing a quantity of lubricating oil, a refrigerant compressor arranged in said casing and having a cylinder, a piston in said cylinder, a crankshaft for actuating said piston, upper and lower wall members in covering relation to said cylinder for sealing the same, said lower wall member being substantially submerged in lubricating oil and having a plurality of cavities forming mutually independent first and second sound muffling mechanisms in its lower portion, said first sound muffling mechanism having an area substantially greater than said second mechanism and being in heat transfer relation with a large proportion of the cylinder area traversed by said piston, said second muffling mechanism including first and second compartments each communicating with a portion of said cylinder laterally displaced from the area thereof traversed by said piston, passage means including said upper wall means and said first sound muffling mechanism for conducting gaseous refrigerant from said casing to said cylinder for compression therein, passage means including a valve mechanism for discharging compressed refrigerant from said cylinder to the first compartment of said second muffling mechanism, passage means for conducting said compressed refrigerant to said cylinder portion, passage means including a valve mechanism for conducting said compressed refrigerant from said cylinder portion to the second compartment of said second muffling system, passage means for conducting said refrigerant from said second compartment to the exterior of said casing, and motor means connected to said crankshaft for operating said piston.

3. Refrigerating apparatus comprising a casing having an inlet for gaseous refrigerant to be compressed, said casing having a crankcase openly communicating therewith

7

for containing a quantity of lubricating oil, a refrigerant compressor arranged in said casing and having a cylinder, upper and lower wall members in covering relation to said cylinder for sealing the same, said upper wall member having substantially its entire area exposed to said incoming gaseous refrigerant and said lower wall member being in heat transfer relation with said oil, said lower wall member, further, having a plurality of cavities forming mutually independent first and second sound muffling mechanisms in its lower portion, said first muffling mechanism comprehending substantially the entire area of said cylinder, a piston within said cylinder, passage means including said first sound muffling mechanism for conducting gaseous refrigerant from said casing to the intake side of said cylinder, means including a motor drivingly connected to said piston for operating said piston to effect a compression of said refrigerant, passage means including a valve mechanism for conveying said compressed refrigerant to said second muffling mechanism, and passage means for conducting said compressed refrigerant therefrom to the exterior of said casing.

4. A refrigeration compressor comprising a sealed casing, inlet means adapted to connect said casing to the low pressure side of a refrigerant evaporator whereby said casing is arranged solely to receive low pressure gaseous refrigerant to be compressed, said casing having a sump

8

containing a quantity of lubricating oil, a refrigerant compressor within said casing and having a cylinder, wall means including first and second plate structures enclosing said cylinder, said first plate structure being wholly exposed to the incoming refrigerant and the second plate structure being immersed in lubricating oil, means within said second plate structure providing a flow passage having an area substantially equal to the area of the cylinder cavity enclosed thereby, passage means for introducing said low pressure gaseous refrigerant from said sealed casing into said second plate structure flow passage, passage mean for said refrigerant communicating between said flow passage and said cylinder, a compressed gas discharge passage communicating between said cylinder and a second flow passage within said second plate structure, valved passage means for conducting compressed gas from said second flow passage to the exterior of said casing, a piston in said cylinder, and a motor mechanically connected to said piston for operating said piston to compress the gas therein.

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