Nov. 11, 1952 F. H. BEALL 2,617,582

ROTARY GAS COMPRESSOR

Filed May 6, 1947 2 SHEETS—SHEET 1

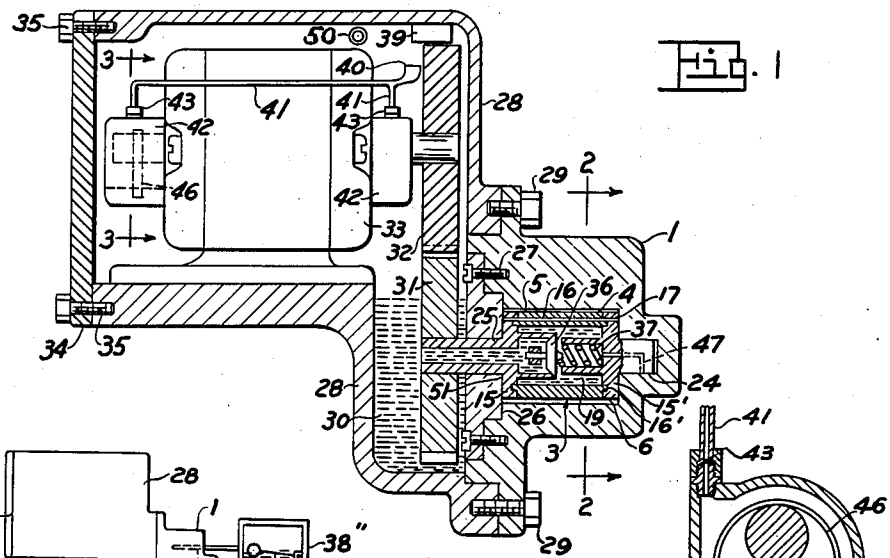


Fig. 1

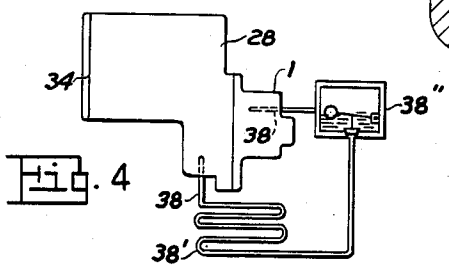


Fig. 4

Fig. 3

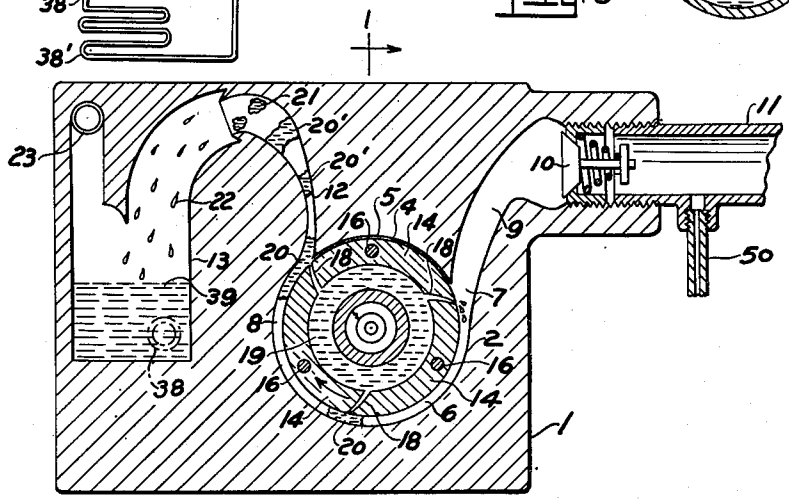


Fig. 2

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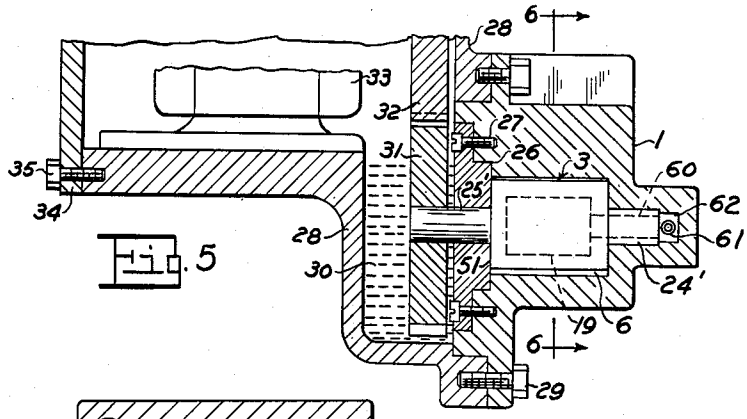


Fig. 5

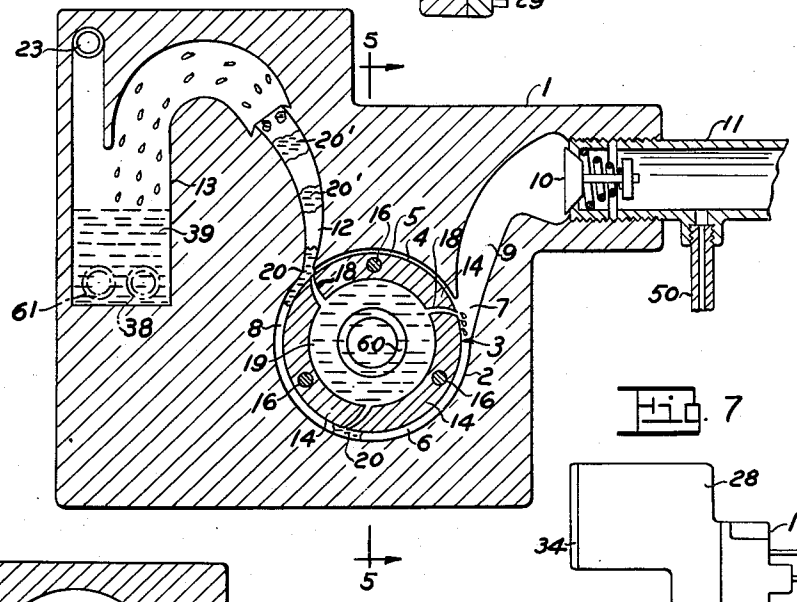
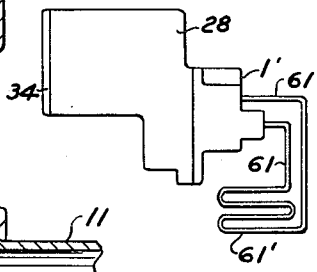


Fig. 6

Fig. 7



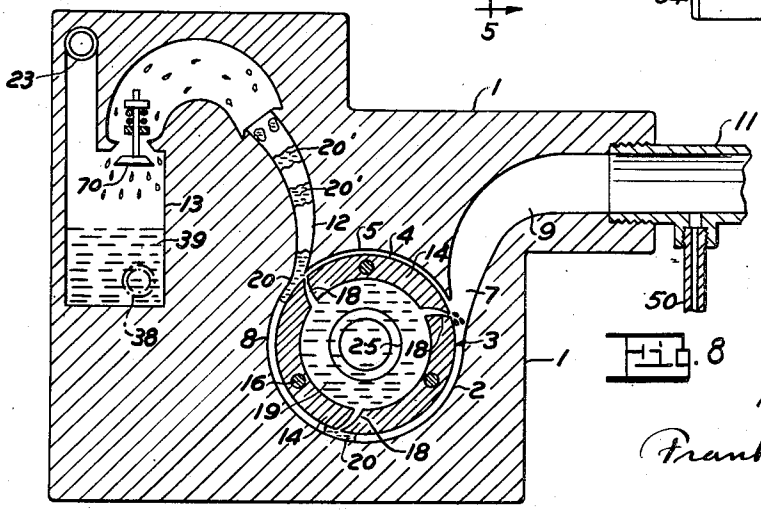


Fig. 8

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UNITED STATES PATENT OFFICE

2,617,582

ROTARY GAS COMPRESSOR

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20 Claims. (Cl. 230—58)

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This invention relates to compressors for the cooling gas or fluid used in refrigeration and primarily applies to small refrigeration units such as ice boxes for home use and soda fountains or small refrigerating units used for any purpose.

It is well known that the compressors for large refrigerating units are relatively efficient, but that the efficiency of compressors decreases rapidly as the volume of refrigerating gas decreases until the efficiency of the ordinary household refrigerator compressors is very low.

The primary object of this invention is to provide an efficient, self lubricating unit containing an oil storage compartment which will contain sufficient lubricant for maintaining proper lubrication over a period of years.

In accordance with my invention I provide a hollow cylindrical rotor provided with a plurality of nozzles for ejecting a limited quantity of oil or other operating fluid per revolution into an annular space extending substantially about two-thirds around the circumference of the rotor.

The refrigerating gas enters one side of this annular space and is carried around the rotor by the oil entering this space through the nozzles in the rotor and which is exhausted at the opposite side of the annular space into an expansion passage where the velocity of the operating fluid is utilized to compress the refrigerating gas to the desired pressure.

A further object of the invention is to provide proper seals so that, when inoperative, leakage is prevented from taking place from the high pressure to the low pressure side of the refrigerating unit.

Further objects of the invention will appear in the hereinafter description of a preferred embodiment of the invention.

The invention consists in novel parts and combination of parts to be described hereinafter, all of which contribute to produce an efficient refrigerating pump.

In the drawing:

Fig. 1 is a vertical section of my compressor taken on line 1—1 of Fig. 2.

Fig. 2 is a cross section taken on line 2—2 of Fig. 1.

Fig. 3 is a cross section taken on line 3—3 of Fig. 1.

Fig. 4 is a diagrammatic showing of the intercooler system for the compressor system shown in Figs. 1 and 2.

Fig. 5 is a vertical section of an alternative design of my compressor taken on line 5—5 of Fig. 6.

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Fig. 6 is a cross section taken on line 6—6 of Fig. 5.

Fig. 7 is a diagrammatic showing of the intercooler system for the alternative design of my compressor shown in Figs. 5 and 6.

Fig. 8 is a cross section of a third form of my compressor system.

In the compression of refrigerating gases by present methods, such as using the piston and cylinder, little heat is lost in the compression cycle and the compression takes place according to the adiabatic equation

$$PV^y=\text{Constant}$$

Where y is the ratio of specific heat at constant pressure to the specific heat at constant volume. Where air is used as the compressive medium,

$$y=1.405$$

However, due to the loss of heat to the cylinder walls during the compression stroke the practical value of y has been found to be about 1.35 for large cylinders and about 1.33 for small cylinders.

Considering the conditions at the beginning of the compression stroke and at the completion of the compression stroke the equation becomes as follows:

$$PcVc^y=PiVi^y$$

or:

$$Pc=\frac{PiVi^y}{Vc^y}$$

Giving

$$Pc=PiV^y$$

Where

Pi=the absolute initial pressure
Pc=the absolute final pressure
Vi=the initial volume
Vc=the volume at completion of the stroke
V=the volumetric compression ratio
y=the exponent

Using a compression ratio of 5, Pc=14 and y=1.33:

$$Pc=14\times5^{1.33}$$
$$=119 \text{ lbs. absolute}$$
$$=104 \text{ lbs. gauge}$$

In contrast with the adiabatic compression, an isothermal compression would follow the following equation:

$$Pc=PiV$$

Where:

$$119=14\times V$$

or

$$V=\frac{119}{14}=8.5$$

Thus for the same final pressure and volume the ratio of initial volumes are

$$8.5:5=1.70$$

A second important advantage of isothermal compression over adiabatic compression is the temperature of the gases at the beginning of the compression stroke. Where adiabatic compression is used the cylinder becomes highly heated and as a result the incoming gases during the intake stroke of the piston become heated before the compression stroke commences. In compressors a temperature of 125 to 150° F. is not uncommon at the beginning of the compression stroke. The equation for the temperature at the completion of the compression stroke is as follows:

$$Tc=Ti\ V^{y-1}$$

Thus using the above experimentally determined value of $y=1.33$ this becomes:

$$Tc=Ti\ V^{0.33}$$

where Tc and Ti are temperatures above absolute zero. Thus using 135° F. as the initial temperature and V=5 the equation becomes:

$$Tc=(135+460)\ 5^{0.33}-460$$
$$=560°\ F.$$

It is this temperature at the completion of the compression stroke that produces the 125 to 150° F. temperature at the beginning of the compression stroke. In comparison with this temperature at the completion of the compression stroke the temperature at the completion of the compression stroke of the isothermal compression is the same as the initial temperature, say about 75 to 80° F.

Since the weight of gas per unit volume varies as the absolute temperature the ratio of the weight of gases per unit volume becomes:

$$\frac{135+460}{80+460}=\frac{595}{540}=1.10$$

Thus the resultant advantage of the use of isothermal compression as against adiabatic compression for temperatures and pressures normal to refrigerating cycles is expressed by the following equation:

$$1.70\times1.10=1.87$$

While applicant's process does not produce complete isothermal compression the tremendous advantage of using a compressive medium that flows through an intercooler between compression strokes is obvious.

In the figures, 1 represents a casing in which is formed a cavity 2 adapted to closely fit a hollow rotor 3 with a minimum of clearance 4 between the rotor 3 and casing 1 throughout an arc 5 forming an upper portion of the periphery of cavity 2, and with a compressor clearance volume 6 between the casing 1 and rotor 3 throughout the remainder of the periphery of the cavity. Clearance 6 is greater on one side of the arc of minimum clearance as indicated at 7 and gradually decreases in the direction of rotation as indicated at 8 until a minimum clearance is attained just prior to the arc of minimum clearance 5.

Leading into the clearance 7 is the inlet port 9 for drawing in the refrigerating gas through check valve 10 from the pipe 11 connected with the low pressure side of the refrigerating system.

Leading from the clearance 8 is the outlet port 12 leading into the oil separator 13.

Rotor 3 is formed of a plurality of segments 14 fastened to end pieces 15—15' by a plurality of rods 16 which may be welded as at 17. Segments 14 are so formed that abutting segments form nozzles 18 so directed as to eject the oil from the cavity 19 in the hollow rotor 3 in the direction of rotation as indicated by the arrow in Fig. 2. To retain the segments against the centrifugal force they are keyed into end pieces 15—15' by the circular keys 16'.

In operation, rotor 3 rotates in the direction of the arrow, drawing in low pressure refrigerating gas from pipe 11 through check valve 10 and around clearance 6. During rotation, oil from cavity 19 is ejected into clearance 6 to form slugs 20 which increase in size during rotation and partially compress the refrigerating gas and expel it into outlet port 12.

Outlet port 12 is formed as an expanding nozzle so that the velocity of the oil slugs 20' gradually slow down and impart their energy to the refrigerating gases to compress them to the desired pressure. At a certain point 21 in the expanding nozzle the oil slugs disintegrate and form an oil spray 22, which through surface friction continue to impart their energy to the gases until the outlet port 12 leads into the oil separator 13. After separation the compressed gases pass into the high pressure side of the refrigerating system through outlet 23.

For efficient operation the nozzles 18 are so restricted that the oil expelled from cavity 19 into clearance 6 is limited so that the kinetic energy of the oil is sufficient to compress the gases entering from the low pressure side of the refrigerating system through inlet passage 9 to the pressure necessary on the high pressure side of the refrigerating system in oil separator 13. To prevent the oil from flowing along the walls of the expanding nozzle 12, the wall is undercut as shown so that the rapidly moving oil is injected into the gas stream.

Rotor 3 is mounted on stub shaft 24 journaled in casing 1 and hollow shaft 25 journaled in cover plate 26 which is bolted in casing 1 by bolts 27. Casing 1 is bolted to housing 28 by bolts 29. Formed in housing 28 adjacent casing 1 is an oil well or storage compartment 30. Shaft 25 projects into this compartment or reservoir and has fixed thereon gear 31 which meshes with gear 32 which is driven by motor 33. Housing 28 is sealed by cover plate 34 which is bolted to the housing by bolts 35.

Oil from the storage compartment 30 enters the rotor cavity 19 through hollow shaft 25 and passes through the check valve 36 which is retained in closed position by the spring 37 and thus through the nozzles 18, clearance 6 and expanding nozzle 12 into the oil separator 13. From the separator the oil leaves by pipe 38 and is returned to the oil storage compartment 30 by means of float valve 38'' so as to maintain a constant oil level 39 in oil separator 13. Pipe 38 includes an oil cooler 38' so that the oil is cooled before entering the storage compartment 30.

Oil for the motor bearings is scraped from the top of gear wheel 32 by scraper 39 and fed into oil cup 40 which is connected by pipes 41 to bearings 42 by couplings 43. As shown in Fig. 3, the oil is retained at a constant level

44 by vent 45 and oil is constantly fed to the bearings by oil ring 46.

The oiling of the rotor bearing for shaft 25 results from the centrifugal force of the rotor which draws in oil from the storage compartment, along the bearing and end piece 15 and out into clearance 6. Oil is fed to the bearing for stub shaft 24 by drilled passage 47 and to clearance 6 in the same way. Due to the very small clearances the quantity of oil so circulated is small, but sufficient for lubricating purposes.

A feature of this invention is the seal between the high pressure and low pressure side of the refrigerating system. In all small refrigerating units, such as this system is particularly directed, the unit is inoperative a substantial portion of the time. It is therefore particularly important that there be no leakage through the bearing of shaft 25 in cover plate 26 into housing 28 which is connected by tube 50 to pipe 11 so as to be subject to the pressure on the low pressure side of the refrigerating system.

As constructed rotor 3, when in the static condition, is subjected to the high pressure side of the refrigerating system as represented by the gas pressure in separator 13. As a result rotor 3 will be pressed to the left, as shown in Fig. 1, due to the pressure on the right hand side of the rotor which must be supported by the pressure on the left hand side of the rotor. The supporting area 51, however, is less than the total of the right hand area by the unsupported area of shaft 25 and therefore the hydrostatic pressure in the oil film in area 51 is greater than the difference in pressure between the high and low side of the refrigerating system and no leakage can occur through the bearing for shaft 25 when the pump is in a static condition. Due to capillary attraction the oil film is maintained in area 51 under all reasonable differential of pressure occurring in a refrigerating system.

Figs. 5 and 6 show an alternative form of the invention. The construction of my pump is the same and like parts have been given the same numbers. In this form, however, shaft 25' is solid and stub shaft 24' is hollow, as indicated at 60, and oil from the separator 13 leaves by pipe 61 and enters chamber 62 and thence through hollow shaft 24' to cavity 19 in hollow rotor 3. In this case the oil cooler 61' is located in connection 61 and will have to be constructed to withstand the pressure of the high pressure side of the refrigerating system. The connection 38 between the separator 13 and the oil storage compartment would still be required since there would be a small leakage of oil along shaft 25' and area 51 during pump operation. However, no oil cooler is required in this connection. When the pump is inoperative the seal for a return flow of this oil along shaft 51 would be complete, as explained above for Figs. 1 and 2. Oil for lubricating stub shaft 24' would come from chamber 62. Also check valve 36 would not be required in this form of the invention.

Fig. 8 shows a third alternative form of the invention. In this form check valve 10 is omitted and a check valve 70 is added, so that the compressed refrigerating gas from the low pressure inlet 11, and the oil, passes through this valve after leaving the outlet port 12. Since the rotor 3 and clearance 6 is subject to the pressure of the low pressure side of the refrigerating system, when the pump is not operating, check valve 36, shown in Fig. 1, is not required and the oil enters cavity 19 of rotor 3 through hollow shaft 25 from storage compartment 30 as shown in Fig. 1. Oil from separator 13 is conducted back to storage compartment 30 through passage 38, as shown in Figs. 1, 2 and 4, after passing through oil cooler 38'.

What I claim is:

1. A compressor for a gaseous fluid, said compressor comprising a casing, said casing having a generally cylindrical cavity formed therein, a cylindrical rotor in said cavity, a pair of shafts mounted in bearings for supporting said rotor, one of said shafts projecting from said casing for rotating said rotor, said rotor containing a hollow compartment, said rotor and said casing being spaced apart and forming a compressor clearance volume therebetween extending around a substantial portion of the periphery of said rotor, said casing having an inlet passage for said gaseous fluid leading into one side of said compressor clearance volume, said casing having an outlet passage leading from the other side of said compressor clearance volume, said cylindrical cavity being imperforate between said inlet and outlet passages, said inlet and outlet passages being separated by an arc of minimum clearance between said cylindrical rotor and said casing throughout the remaining portion of the periphery of said rotor, nozzles in said rotor connecting said hollow compartment with said compressor clearance volume and minimum clearance, a compressive fluid in said hollow compartment, a reservoir for said compressive fluid and means connecting said hollow compartment with said reservoir whereby fluid from said reservoir and hollow compartment is ejected from said nozzles and compresses said gaseous fluid as it travels around said compressor clearance volume to said outlet passage, a separator for said gaseous fluid and said compressive fluid connected to said outlet passage, said outlet passage being of sufficient length to permit the momentum of said compressive fluid to further compress said gaseous fluid between said compressor clearance volume and said separator.

2. The combination of claim 1 further characterized whereby the area of said outlet passage increases from its connection with said compressor clearance volume to said separator.

3. The combination of claim 1 further characterized wherein the area of said compressor clearance volume decreases from said inlet side of said compressor clearance volume to the outlet side thereof.

4. The combination of claim 1 further characterized by a check valve located in said inlet passage.

5. The combination of claim 4 further characterized wherein said inlet passage decreases in area from said check valve to its junction with said compressor clearance volume.

6. The combination of claim 1 further characterized wherein said inlet passage and compressor clearance volume decrease in area until the junction with said outlet passage and wherein said outlet passage increases in area between said junction and said separator.

7. The combination of claim 1 further characterized wherein said reservoir is associated with said separator and subject to the pressure of said gaseous fluid after compression and said connecting means connects said reservoir with said hollow compartment, whereby the pressure

in said hollow compartment is substantially the same as that of said compressed gaseous fluid.

8. The combination of claim 7 further characterized by cooling means in said connecting means.

9. The combination of claim 7 further characterized wherein one of said shafts is a hollow shaft and said connecting means includes said hollow shaft.

10. The combination of claim 1 further characterized wherein said reservoir is separate from said separator and connected thereto by additional connecting means.

11. The combination of claim 10 further characterized to include cooling means in said additional connecting means.

12. The combination of claim 10 further characterized by valve means in said additional connecting means to permit the flow of compressive fluid therethrough, but to maintain a higher pressure in said separator than in said reservoir.

13. A compressor unit for a gaseous fluid, said compressor unit comprising a casing, said casing having a generally cylindrical cavity, an oil well exterior to said cavity, a cylindrical rotor in said cavity, a pair of shafts mounted in bearings for supporting said rotor, one of said shafts projecting from said casing into said oil well for rotating said rotor, said rotor containing a hollow compartment, said rotor and said casing being spaced apart forming a compressor clearance volume therebetween extending around a substantial portion of the periphery of said rotor, said casing having an inlet passage for said gaseous fluid leading into one side of said compressor clearance volume, said casing having an outlet passage leading from the other side of said compressor clearance volume, said cylindrical cavity being imperforate between said inlet and outlet passages, said inlet and outlet passages being separated by an arc of minimum clearance between said cylindrical rotor and said casing throughout the remaining portion of the periphery of said rotor, nozzles in said rotor connecting said hollow compartment with said compressor clearance volume and minimum clearance, a compressive fluid in said hollow compartment, a separator for said gaseous fluid and said compressive fluid connected to said outlet passage and connecting means between said separator and hollow compartment to return said compressive fluid from said separator to said hollow compartment whereby fluid from said separator and hollow compartment is ejected from said nozzles and compresses said gaseous fluid as said nozzles travel around said compressor clearance volume to said outlet passage, a housing mounted on said casing, said projecting shaft projecting into said housing, said oil well being located in said housing, a gear wheel mounted on said shaft located in said oil well in said housing, and means within said housing to drive said gear wheel.

14. The combination of claim 13 further characterized by an electric motor in said housing, said motor being provided with bearings and oil means therefore and said drive means being a second gear wheel driven by said motor and means for automatically supplying oil from said oil well to said motor bearings.

15. The combination of claim 13 further characterized wherein said compressive fluid is oil and said oil well constitutes an oil reservoir located in said connecting means between said separator and said hollow compartment and said projecting shaft is hollow to provide the connecting means between said reservoir and said hollow compartment.

16. The combination of claim 15 further characterized by the provision of a check valve in said connecting means between said reservoir and said hollow compartment.

17. A compressor system for a gaseous fluid, said compressor system including a high and low pressure side, comprising a casing, said casing having a generally cylindrical cavity formed therein, a cylindrical rotor mounted for rotation in said cavity, said casing being provided with bearings at opposite ends of said cavity, a pair of shafts mounted in said bearings for supporting said rotor, one of said shafts projecting from its bearing and said casing for rotating said rotor, said rotor containing a hollow compartment, said rotor and said casing being formed to provide a compressor clearance volume therebetween extending around a substantial portion of the periphery of said rotor, said casing having an inlet passage for said gaseous fluid leading into one side of said compressor clearance volume, said casing having an outlet passage leading from the other side of said compressor clearance volume, said cylindrical cavity being imperforate between said inlet and outlet passages, said inlet and outlet passages being separated by an arc of minimum clearance between said cylindrical rotor and said casing throughout the remaining portion of the periphery of said rotor, nozzles in said rotor connecting said hollow compartment with said compressor clearance volume and minimum clearance, a compressive fluid in said hollow compartment, a separator for said gaseous fluid and compressive fluid connected to said outlet passage and connecting means between said separator and hollow compartment to return said compressive fluid from said separator to said hollow compartment whereby fluid from said separator and hollow compartment is ejected from said nozzles and compresses said gaseous fluid as said nozzles travel around said compressor clearance volume, a check valve in said inlet passage, said shaft, projecting through said casing, being of substantially less diameter than said rotor, said rotor having a side wall extending from said projecting shaft to the perimeter of said rotor and said casing having a side wall surrounding the bearing of said projecting shaft and registering with said rotor side wall whereby the area of said registering side walls is less than the area of said registering side walls and said shaft, oiling means for said one bearing and said registering side walls whereby the resulting hydrostatic pressure in said oil film on said registering side walls is greater than the pressure on the high pressure side of said system when said compressor is not operating, thus preventing leakage of said gaseous fluid past said bearing for said projecting shaft.

18. A compressor for a gaseous fluid, said compressor comprising a casing having a generally cylindrical cavity formed therein, a cylindrical rotor in said cavity having a hollow compartment therein containing a compressive fluid, said rotor and said casing being formed to provide a compressor clearance volume therebetween extending around a substantial portion of the periphery of said rotor, said casing having an inlet passage for said gaseous fluid leading into one side of said compressor clearance volume, said casing having an outlet passage leading from the other side of said compressor clearance volume, said cylindrical cavity being imperforate between said inlet and outlet passages, said inlet

and outlet passages being separated by an arc of minimum clearance between said cylindrical rotor and said casing throughout the remaining portion of the periphery of said rotor, nozzles in said rotor connecting said hollow compartment with said compressor clearance volume and minimum clearance, a separator for said gaseous fluid and compressive fluid connected to said outlet passage and connecting means between said separator and hollow compartment whereby fluid from said separator and hollow compartment is injected from said nozzles and compresses said gaseous fluid as said nozzles travel around said compressor clearance volume, a check valve in said outlet passage and valve means in said connecting means to permit the flow of compressive fluid therethrough, but adapted to maintain a higher pressure in said separator than in said hollow chamber when said rotor is inoperative.

19. A compressor for a gaseous fluid, said compressor comprising a casing having a generally cylindrical cavity formed therein, a cylindrical rotor in said cavity, a pair of shafts mounted in bearings for supporting said rotor, one of said shafts projecting from said casing for rotating said rotor, said rotor containing a hollow compartment, said rotor and said casing being formed to provide a compressor clearance volume therebetween extending around a substantial portion of the periphery of said rotor, said casing having an inlet passage for said gaseous fluid leading into one side of said compressor clearance volume, said casing having an outlet passage leading from the other side of said compressor clearance volume, said cylindrical cavity being imperforate between said inlet and outlet passages, said inlet and outlet passages being separated by an arc

of minimum clearance between said cylindrical rotor and said casing throughout the remaining portion of the periphery of said rotor, nozzles in said rotor connecting said hollow compartment with said compressor clearance volume and minimum clearance, a compressive fluid in said hollow compartment, a reservoir for said compressive fluid and means connecting said reservoir and said hollow compartment, whereby said compressive fluid from said reservoir and hollow compartment is ejected from said nozzles and compresses said gaseous fluid as it travels around said compressor clearance volume to said outlet passage and a separator for said gaseous fluid and said compressive fluid connected to said outlet passage.

20. The combination of claim 19 further characterized wherein said reservoir is associated with said separator and said nonprojecting shaft is hollow and said connecting means includes said hollow shaft.

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