United States Patent [19]

Adams, Jr. et al.

[11] **3,927,889**

[45] **Dec. 23, 1975**

[54] **ROTATING ELEMENT FLUID SEAL FOR CENTRIFUGAL COMPRESSOR**

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[73] Assignee: **Westinghouse Electric Corporation**, Pittsburgh, Pa.

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[21] Appl. No.: **398,351**

[52] **U.S. Cl.** **277/25**; 277/29; 277/67

[51] **Int. Cl.²** **F16J 15/44**

[58] **Field of Search** 277/17, 67, 25, 29, 53, 277/55–57, 70–79, 12; 415/111, 115, 118, 170, 143, 122, 144, 222; 416/244; 417/360, 423

[56] **References Cited**

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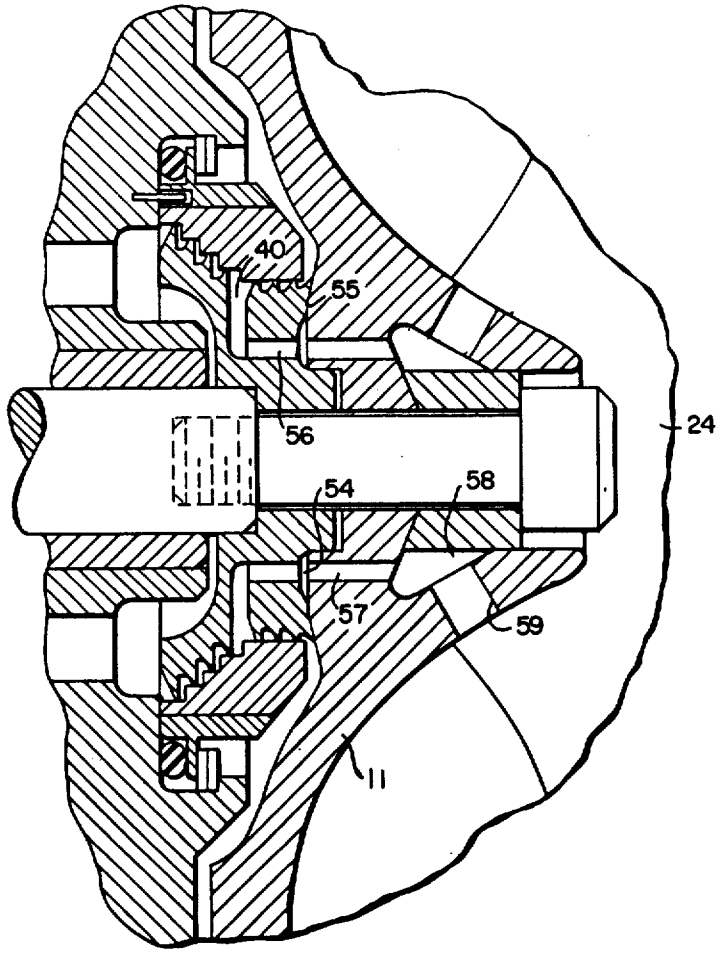
Primary Examiner—Robert I. Smith

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[57] **ABSTRACT**

A vertical dynamic seal such as may be used between the high pressure discharge chamber of a centrifugal gas compressor and a chamber in which the driving machinery for the impeller of the compressor is contained, and formed by the opposing faces of a seal stator and a seal rotor vented at an intermediate location along the axial extent of the seal to the space at the inlet or suction side of the impeller, is provided with outwardly-directed bore means in the venting passage to obtain a pumping effect to the inlet side of the impeller.

7 Claims, 5 Drawing Figures



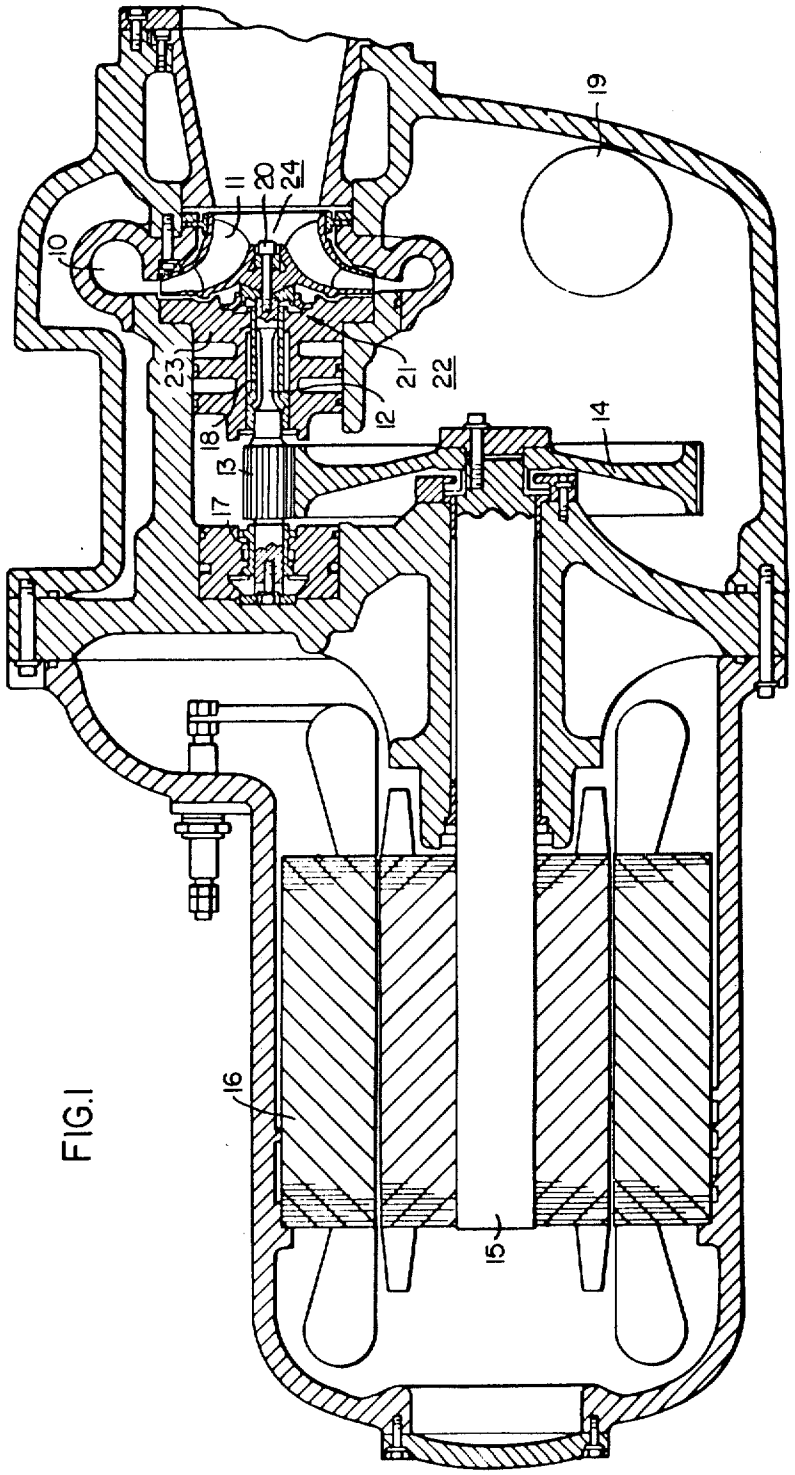


FIG.1

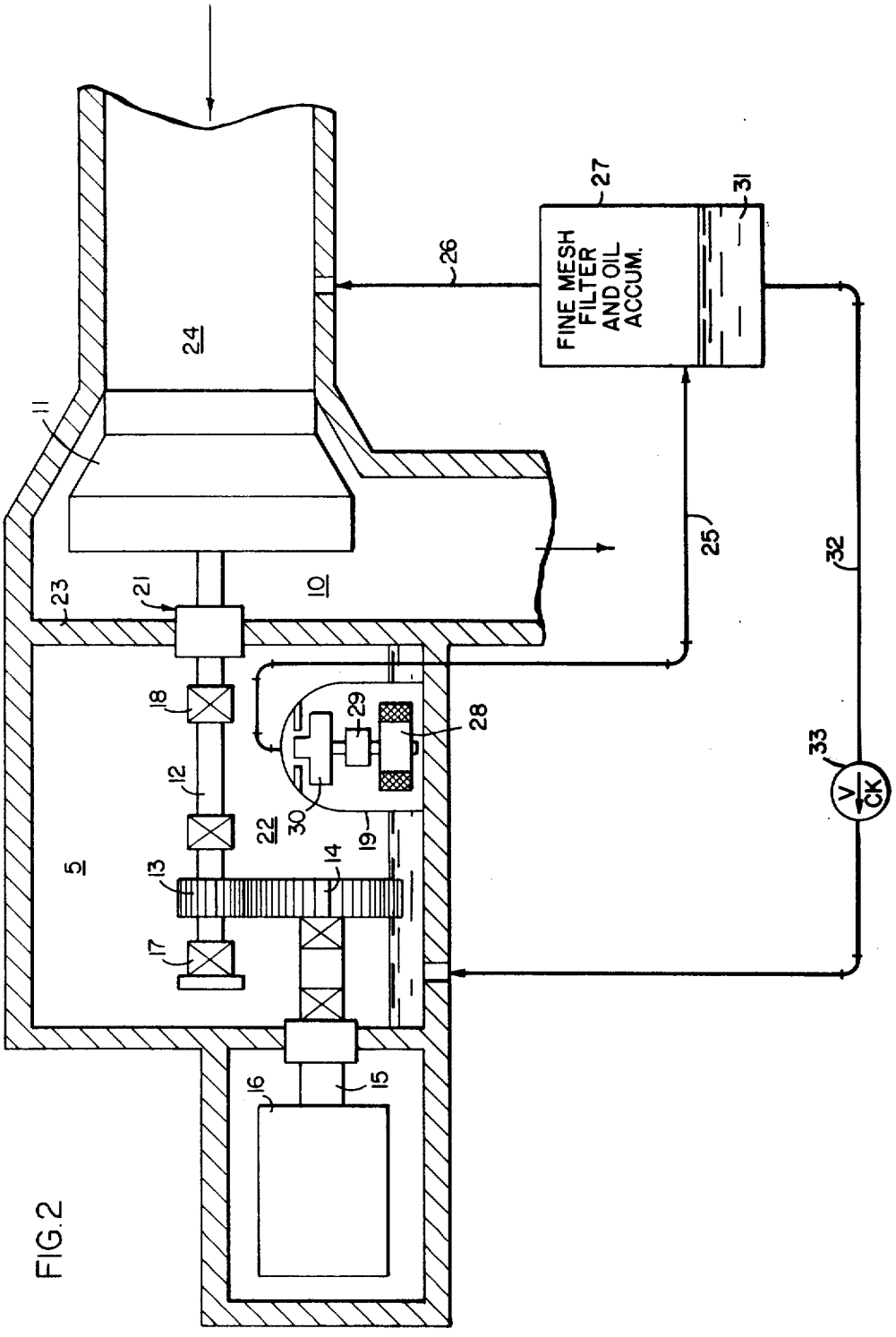

FINE MESH FILTER AND OIL ACCUM.
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FIG.2

FIG. 4

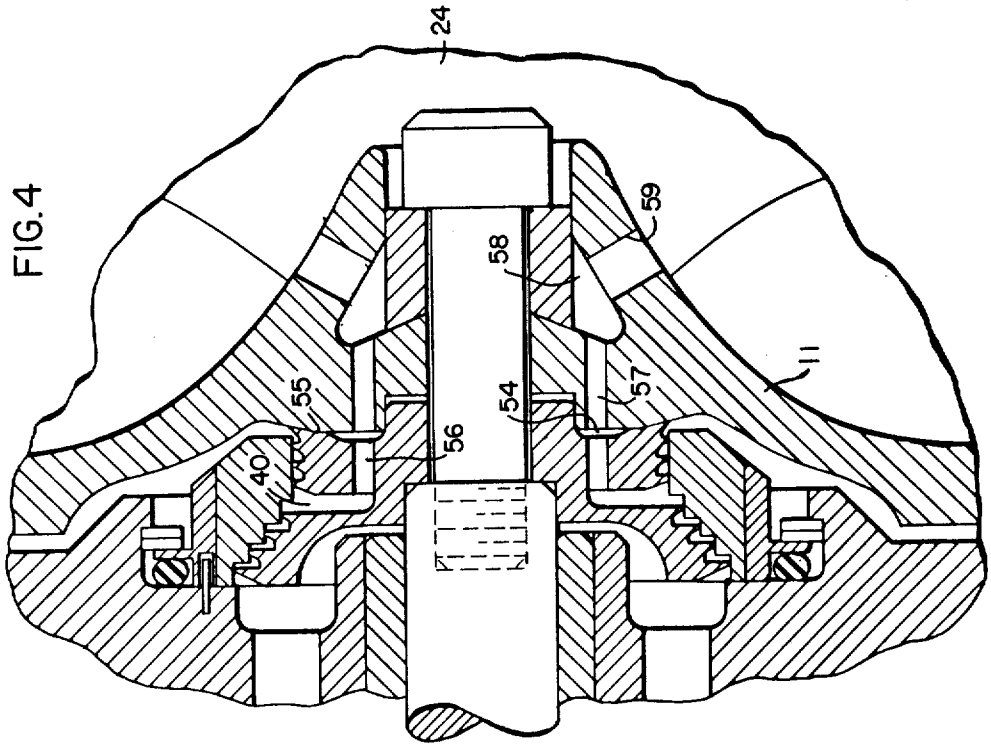

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FIG. 3

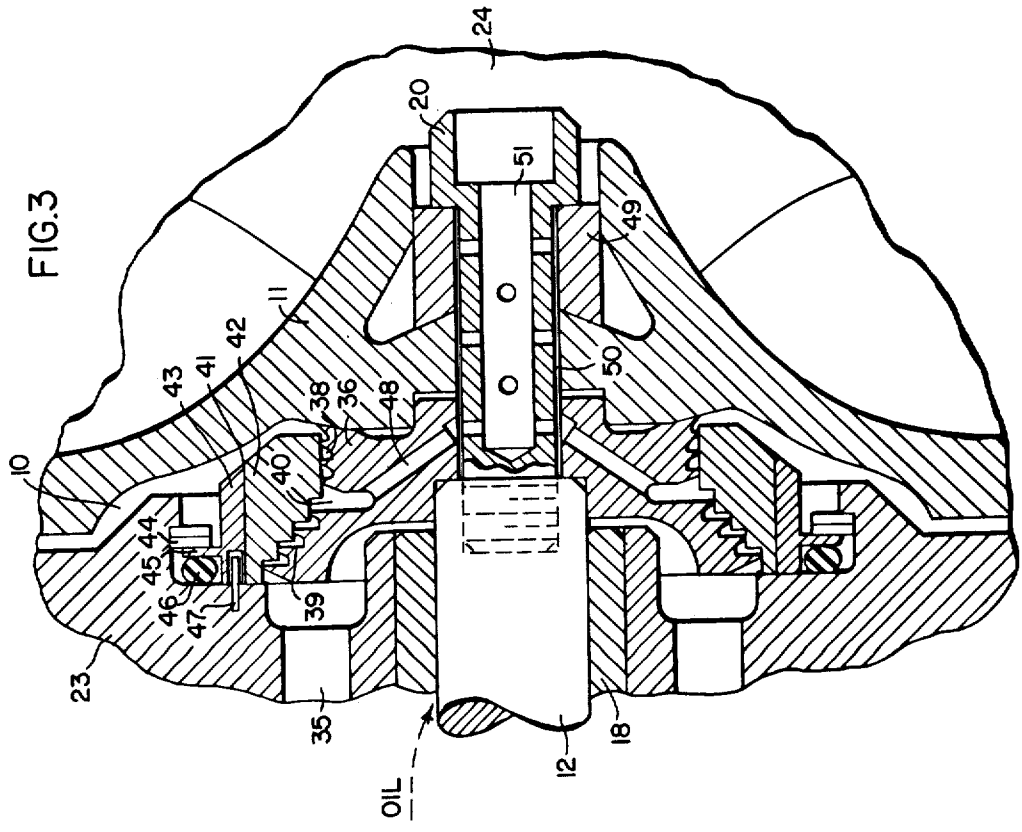
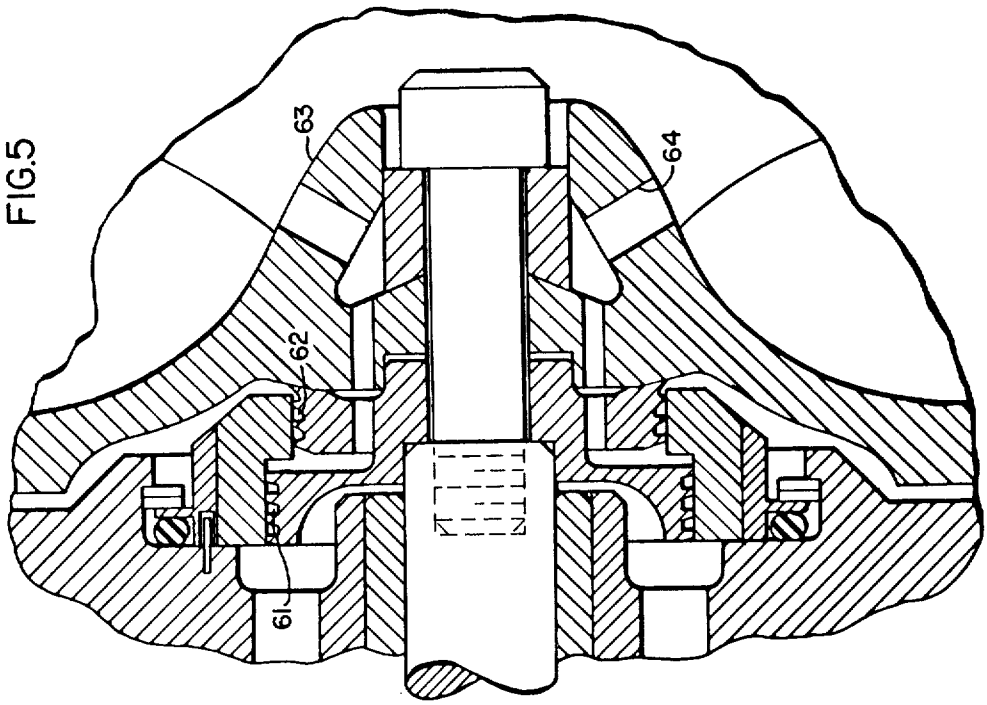

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FIG.5



ROTATING ELEMENT FLUID SEAL FOR CENTRIFUGAL COMPRESSOR

CROSS REFERENCE TO RELATED APPLICATION

Adams U.S. patent application Ser. No. 398,352 filed concurrently herewith is a related application in that it discloses and claims the basic invention from which the present invention arose.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention pertains to the art of rotary or dynamic seals particularly useful in the field of centrifugal refrigerant gas compressors.

2. Description of the Prior Art

U.S. patents of which applicants are aware and which have some similarities with respect to seals and/or venting passages are: U.S. Pat. No. 838,744; U.S. Pat. No. 3,392,910; U.S. Pat. No. 3,480,284; U.S. Pat. No. 3,493,169; U.S. Pat. No. 3,647,313. It is our view that the teachings in these patents do not anticipate or make obvious the invention claimed herein.

From the standpoint of structural similarities of certain of the parts, the closest prior art which we are aware is the centrifugal refrigerant compressor units currently produced by applicants' assignee. These devices provide an environment in which our invention finds one of its most useful applications. The general arrangement of the centrifugal compressor impeller and shaft assembly and the assembly including the driving motor, sump chamber in which the drive train connecting the motor to the impeller is located and the general casing arrangement, are shown in U.S. Pat. Nos. 3,601,501 and 3,619,086, to which reference should be had for a full understanding of one general construction of a centrifugal refrigerant compressor in which the present invention is incorporated.

The present seal between the high pressure discharge space and the sump chamber on the opposite side of the wall separating these spaces is provided by a seal rotor and seal stator providing an ordinary serrated bushing type seal. The leakage of refrigerant gas from the high pressure discharge chamber side through such a seal to the sump chamber, which is maintained at substantially the same pressure as the suction pressure to the compressor inlet by venting the sump chamber to the inlet side of the impeller through an oil-refrigerant separator and oil filter arrangement, is sufficiently high with the relatively small volume sump chamber used on the relatively compact centrifugal compressor assembly, that periodically the system must be shut down to permit oil which has accumulated in the oil filter accumulator to be returned to the sump chamber. The excessive rate of refrigerant gas leakage from the high pressure side to the sump chamber creates the problem which this invention is intended to solve. A further explanation of the arrangement as a whole and the advantages accruing from the improved seal arrangement according to this invention will be set forth hereinafter.

SUMMARY OF THE INVENTION

In accordance with the invention, the seal arrangement is provided in the described environment by a seal rotor arranged to rotate with the impeller shaft and an annular seal stator around the rotor, with opposing faces of the rotor and stator forming a dynamic seal having one end of the axial extent of the seal exposed to the compressor discharge pressure space and the other end of the axial extent of the seal exposed to the sump chamber space, and a venting passage from an intermediate location along the axial extent of the seal, to the inlet side of the impeller is provided, the venting passage having a considerably lower resistance to fluid flow than the resistance to fluid flow between the intermediate location along the axial extent of the seal and either end of the seal, and including means to scavenge the venting passage in a direction toward the impeller inlet side.

DRAWING DESCRIPTION

FIG. 1 is a longitudinal sectional view of a centrifugal gas compressor assembly of one type in which the invention may be incorporated;

FIG. 2 is a generally schematic view of the parts of a refrigerant system in which the invention is particularly applicable;

FIG. 3 is a sectional view, enlarged relative to the showing of FIG. 1, of one form of seal arrangement as incorporated in the impeller, shaft, and casing construction according to the noted Adams invention;

FIG. 4 is a view similar to FIG. 3 and illustrating one form of our invention and which is believed to be preferable in some systems to the noted Adams invention; and

FIG. 5 is a view similar to that of FIG. 4 and showing a modification of the arrangement of FIG. 4.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

A description of the disclosed embodiment of the noted Adams patent application, and the background relating thereto, is reproduced herein for the purpose of lending understanding to the embodiments of our invention.

Background and the Adams Invention

The centrifugal compressor assembly illustrated in FIG. 1 is of a design in which compactness is considered an important feature. This is achieved in part by the use of a relatively small, but high speed, compressor arrangement. The compressor impeller in the particular arrangement illustrated is driven currently at a nominal speed of 34,000 rpm. After a brief description of the main parts of the compressor assembly of FIG. 1, the significance of the compactness relative to the invention will be discussed.

The illustrated assembly includes a compressed gas collecting scroll 10 with a shrouded centrifugal impeller 11 concentric therewith secured to the end of a rotatable shaft 12 rotated by its pinion gear 13 which is in turn driven by bull gear 14 on the shaft 15 of motor 16.

The shaft 12 for driving impeller 11 is journaled in bearings 17 and 18, which are hydrodynamic journal bearings providing full film lubrication and are fed with oil derived from an oil pump-separator 19 (which will be described later) through oil feedlines not shown. The impeller 11 is attached to the shaft 12 by an impeller retaining bolt 20 and other parts which friction key the impeller to the shaft in accordance with the general arrangement described in U.S. Pat. No. 3,601,501.

The seal arrangement according to the invention is located in the area designated 21 in FIG. 1 and the importance of a reasonably good seal at this location

separating the high pressure or scroll chamber area **10** from the interior space **22** of the sump chamber will be described in connection with FIG. **2.**

In FIG. **2,** the sump chamber space **22** and any space therein in communication with the space **22** is shown as being separated from the high pressure discharge space **10** by a casing wall **23** provided with the seal **21** in the opening in the wall through which the driving shaft **12** for the impeller **11** projects.

In the particular compressor assembly design used as an example for purposes of description, the compressor discharge space **10** (FIGS. **1** and **2**) in normal operation will normally have a pressure of say 65 p.s.i.g. (the undiffused static pressure at the impeller discharge), and to which the one side of the seal **21** is exposed, while the pressure in the sump chamber **22** will nominally be 35 p.s.i.g., with the other side of the seal **21** exposed to this pressure. With these values, a pressure differential of 30 p.s.i. is available to drive refrigerant gas from the discharge pressure space **10** to the sump chamber space **22.** The sump chamber space **22** is maintained at substantially the same pressure as the suction pressure at the inlet **24** at the eye of the impeller through the connecting lines **25** and **26** which connect the oil-refrigerant separator to a fine mesh filter and oil accumulator **27** and the filter-accumulator to the suction side of the compressor, respectively. The entire refrigerant loop is hermetically closed and the lowest pressure in this loop is that at the compressor impeller inlet or suction pressure. Therefore, the leakage of refrigerant gas through the seal **21** is required to be returned to the refrigerant system and this is accomplished by the oil pump-separator **19** which includes an electric motor **28** driving a gear pump **29** for pumping oil to the bearings in the sump chamber, and a driven oil—refrigerant separator **30** which is intended to remove large particles of oil from the refrigerant and vent the refrigerant through the line **25** to the filter and oil accumulator **27.**

In normal operation of the compressor assembly the gear arrangement of the drive train generates voluminous oil mist particles which are in part entrained in refrigerant leakage into the sump chamber **22.** The larger oil particles are prevented from being vented along with the refrigerant back to the refrigerant system by the centrifugal oil—refrigerant separator **30** but some of the small oil mist particles, such as those which are 2 microns or smaller, are not separated but escape through the line **25** to the filter and oil accumulator **27.** This filter is provided with a filter medium which for the most part prevents the passage of those particles greater than 0.4 microns through the vent line **26** to the refrigerant system. However, these separated oil particles are stored in the accumulator portion **31** and can be drained back to the sump chamber only during compressor shutdown through the drain line **32** containing a check valve **33.** In that connection it is noted that FIG. **2** is a schematic view and that the filter and accumulator **27** is located to permit gravity drain back to the sump chamber through the line **32.**

The desirability of having an effective seal **21** arises in large part from the compactness of the sump chamber. With the relatively compact sump chamber, which is desirable in connection with the overall compactness of the compressor assembly, the oil capacity in the sump chamber is limited. For example with a centrifugal compressor of the compact size having approximately a 100-ton refrigeration producing capacity, in the particular illustrated arrangement the oil capacity is limited to approximately 2 gallons. The limited size sump chamber also imposes a disadvantage in that the internal surface area in the sump chamber upon which oil mist can coalesce is relatively limited as contrasted to units with larger sump chambers. In the compact size units the total amount of oil present in the chamber is so limited that the oil which is not coalesced on surfaces is subject to being entrained by the refrigerant gas being vented back to the refrigerant system. Since the rate of oil carry-out is generally proportional to the refrigerant venting rate the result is that in a shorter time period than is desirable sufficient oil has been moved into the accumulator **27** by the venting refrigerant gas that a shutdown of the system is required to permit drainage of the oil from the accumulator back to the sump chamber. While in larger units of this general design a sump pump and venting is also provided, there is no requirement for the filter **27** because the larger oil capacity in the sump chamber permits a greater venting loss of oil and larger periods of time between shutdown of the compressor. Thus, in addition to the disadvantage of the limited time intervals between shutdowns, the compact centrifugal compressor has the further disadvantage of requiring the use of the fine mesh filter and oil accumulator device **27.** From this it will be appreciated that if refrigerant leakage from the high pressure chamber to the sump chamber can be substantially reduced, the noted disadvantages can be substantially avoided. The seal arrangement of the invention is intended to accomplish this, particularly in the noted environment of a compact centrifugal refrigerant compressor. The seal arrangement is also of a reasonably low cost and does not disturb the basic design parameters of the centrifugal compressor assembly having the general impeller and shaft arrangement disclosed in U.S. Pat. No. 3,601,501.

In the illustrated seal arrangement of FIG. **3** the shaft **12** is supported by the hydrodynamic journal bearing **18** to which oil is fed in the direction indicated by the dash line arrow under pressure from the pump **29** (FIG. **2**) to provide a full film lubrication, with the oil then passing into the oil cavity **35** which is in turn connected by means not shown to drain to the sump of the sump chamber **22.** Accordingly the oil cavities are at the same pressure as the interior of the sump chamber, which in the example is approximately 35 p.s.i.g.

The seal rotor part **36** (FIG. **3**) has the general shape of a circular washer with a stepped opening therein. The shoulder of the counter bored part of the stepped opening seats on the end of shaft **12.** The outer circumference of the seal rotor **36** includes one portion **38** extending in a generally axial direction and provided with labyrinth teeth thereon and an outwardly, obliquely directed portion **39** also provided with labyrinth teeth thereon, these two portions being separated at an intermediate location along the total axial extent of the outer circumference of the rotor by a circumferential groove or channel **40.**

The seal stator **41** (FIG. **3**) has the general shape of a ring which encircles the seal rotor **36** and includes in its currently preferred form an inner ring **42** of carbon graphite which is shrunk fit in a steel outer retaining ring **43.** The inner surface of the ring **42** is shaped to accommodate the profile formed by the tips of the teeth on the seal rotor. The seal stator is secured in position to the casing **23** by retaining rings **44** engaging a circumferential flange **45** of the ring **43,** the flange in

turn compressing an O-ring secondary seal **46** in the currently preferred construction. An anti-rotation pin **47** locking the outer section of the seal stator against rotation relative to the casing **23** may also be provided.

It will be appreciated from the description and from FIG. **3** that the dynamic seal thus provided by the seal rotor and seal stator has an axial extent which includes the one portion **38** and the other portion **39** separated by the intermediate location circumferential channel **40.**

The rotor **36** (FIG. **3**) has a plurality of bores **48** which extend from the channel **40** to an annular clearance space **50** defined between the shank **37** of the retaining bolt and the surrounding structure including the seal rotor **36,** the hub of impeiler **11,** and a ring-shaped spacer **49.** It is noted that the impeller and shaft assembly as a whole is of the general arrangement described in U.S. Pat. No. 3,601,501 which results in the impeller being friction keyed to the shaft **12** and permits the clearance space **50** noted. While the size and shape of the illustrated rotor **36** dictates that the bores **48** extend obliquely, in rotors of other shapes or larger sizes they could be directly radial.

An axial bore **51** (FIG. **3**) is provided in the retainer bolt **20** and a plurality of radially directed transverse bores **52** place the axial bore in communication with the clearance space **50** and with the rotor bores **48.** Thus, a venting passage for refrigerant gas leaking into the seal from the high pressure space is provided from an intermediate location along the leakage path of the dynamic seal to the space **24** at the inlet or suction side of the impeller.

It is considered important in the particular centrifugal compressor assembly in which the seal arrangement is described as being incorporated that the labyrinth seal be manufactured with a relatively close clearance. Satisfactory results have been obtained by constructing the seal stator with an inside diameter of the carbon graphite ring being such that, as assembled to the seal rotor, a 0 to 0.002 inch diametral clearance exists between the stator and rotor. During the first few seconds of operation the sharp metal labyrinth teeth rapidly cut from the carbon what is needed, but no more. This wear-in occurs because the seal stator is clamped (non-floating) at assembly with the seal rotor as a guide. In addition the characteristics of the particular machine are such that the shaft will move radially a few mils in going from its position at assembly to its normal position at operating speed. This radial shift of the rotational centerline arises from differential thermal expansions occurring in the main bearings and from a shift of position of the rotating shaft within the journal bearings reflecting oil film thickness changes. The relatively close tolerances of the seal arrangement are considered to be important to insure that each portion of the labyrinth seal, **38** and **39** (FIG. **3**) has a higher gas flow resistance than the internal vent passage leading from the intermediate location along the seal to the inlet space **24.**

The way in which the seal arrangement works is as follows. In normal operation a pressure of, say, 65 p.s.i.g. exists at the impeller or high pressure side **10** (FIG. **3**) of the seal arrangement while a pressure of, say, 35 p.s.i.g. exists at the other side of the seal arrangement exposed to the oil cavity **35.** The pressure in the annular groove **40** at the intermediate location will be only slightly greater than the pressure at the suction side of the impeller, which as was noted, is substantially the same as that in the oil cavity **35.** For example the pressure at the intermediate location **40** at the annular groove may be in the order of 36 or 37 p.s.i.g., and with the relatively low flow resistance of the vent passage, most of the refrigerant gas leakage which reaches the intermediate location is returned to the suction side of the impeller rather than going through the relatively high resistance leakage path of the portion **39** of the seal leading to the oil cavity **35.** In testing it has been determined that the ratios of reduction of leakage into the oil cavity with the illustrated arrangement, over the seal presently in use, ranged from about 1 to 10 to 1 to 20 depending upon the clearances between the rotor teeth and the stator.

It is currently believed preferable that the portion **39** (FIG. **3**) of the seal be obliquely disposed as illustrated, rather than extending axially as the other portion **38.** The reason for this will be better understood in light of the following remarks. In normal operation of the compressor the pressures are the noted nominal pressures of 65 p.s.i.g. discharge and 35 p.s.i.g. at the inlet to the impeller and in the sump chamber. However, when the compressor is shut down for a sufficient period to equalize the pressures throughout this hermetically closed system, the pressures throughout the system will be in order of about 80 p.s.i.g. Thus upon start-up of the compressor the discharge pressure is lower than the sump pressure for a short time, such as 10 seconds. This is due to the lesser restriction of the refrigeration system as a whole relative to the restriction of the communicating lines from the sump chamber to the refrigeration system. As a result, upon start-up there can exist a reverse pressure gradient across the seal which could tend to permit oil leakage back through the seal into the refrigerant system. By providing the oblique disposition of the seal portion **39,** centrifugal force in enlisted to "throw-back" oil which might tend to leak back through the seal.

While the seal of the noted Adams patent application and illustrated in FIG. **3** provides the noted significant reduction in leakage in the range of operating speeds currently used, with possible increases to higher operating speeds in the future, greater rates of leakage than occur with the current operating speeds can be expected. The projected increased leakage will result from the generally radially directed bores **48** (FIG. **3**) in the seal rotor **36** producing a pumping action as if they formed a small centrifugal compressor. While this pumping action may be considered, on balance, to be advantageous to prevent reverse leakage in an installation operated normally with an appreciable number of starts and stops, this does not apply to a system intended for mostly continuous operation with infrequent shut-downs. In the illustrated arrangement the pumping action can produce a pressure of say 5 p.s.i. higher than the compressor suction pressure at the current nominal operating speeds. Because the low pressure side of the seal is at the same compressor suction pressure, there is then a 5 p.s.i. pressure difference available to drive refrigerant leakage from the annular groove **40** through the second portion **39** of the seal and into the sump chamber. While this leakage is not objectionably large at the nominal 30,000 to 32,000 rpm operating speed of the compressor, the leakage will increase with increased speeds and thus would become significantly higher with higher operating speeds. This stems from the leakage increasing with compressor speed changes from mid-range speeds, such as 10,000 to 20,000 rpm,

up to and above the nominal operating speed.

This Invention

The constructions in accordance with our invention, and illustrated in FIGS. 4 and 5, are such that the adverse rising-leakage/speed characteristic at such speeds is substantially eliminated, and also yields a further reduction in leakage at the current operating speeds. Further, the constructions are such that all portions of the venting passage formed in the retaining bolt 20 are eliminated so there is no chance of weakening of the retaining bolt. It is noted that those parts of the assembly in FIGS. 4 and 5 which are of the same structure as in FIG. 3 are given identical numerals to the extent they are identified by numerals.

The general concept of the arrangements in FIGS. 4 and 5 incorporates the elimination of radial bores in the seal rotor and the addition of what are termed scavenging vent holes provided in the vent passage from the seal to the inlet space of the compressor.

The stator construction of FIG. 4 is the same as that of FIG. 3. The rotor 55 differs in that the radially inner portion of the annular channel 40 is connected through a plurality of axially-extending bores 56 spaced around the rotor, rather than radially-extending bores as in the rotor 36 of FIG. 3. These axially extending bores communicate with axially extending bores 57 in the hub of impeller 11, through the small annular cavity 54 between the two sets of axial holes, i.e. between the seal rotor and impeller hub. These bores 57 also communicate with the annular cavity 58 inside the impeller hub which in turn communicates with a plurality of radially outwardly directed bores 59 which open to the suction or inlet side 24 of the impeller. These bores 59 serve as a small centrifugal compressor which in effect pump down the vent passageway back to the annular groove 40 and thus lower the gas pressure in the annular groove 40. Thus this arrangement may be logically termed as a scavenging-vent passage arrangement in that the leakage gas is drawn rather than simply vented.

The arrangement shown in FIG. 5 differs from that shown in FIG. 4 in two main respects. First, the seal portions 61 and 62 both extend axially only. The expected advantage of this is that the seal is less sensitive to axial motions of the shaft which can occur during operation and enables looser control of machine tolerances which affect axial positioning. The second labyrinth stage 62 however is located at a larger radius than the first labyrinth stage 61 of the seal to thereby use centrifugal acceleration to prevent ingress of oil during start-up. Another advantage of the parallel, two stage seal, as distinguished from the seals of FIGS. 3 and 4 which include at least one oblique stage, is that the number of diameters to be machined are reduced. Since the machining must be to relatively close tolerances, such as 1 mil, the reduction in the number of diameters provides a cost advantage.

The other difference of the arrangement of FIG. 5 is that the upstream portion of the hub of the impeller 63 is thickened so that the pumping bores 64 therein may be increased in radial extent. This provides a stronger pumping action and is believed to be possibly useful in an arrangement in which it is intended that the seal have zero or reverse leakage during normal operation. It is noted that this arrangement of reverse leakage is not considered desirable in a centrifugal refrigerant compressor of the general type described since this would encourage oil leakage from the sump chamber back through the seal. However, such a reverse leakage arrangement may be desirable in the pumping of toxic gases or gases subject to being radioactively contaminated.

We claim:

1. In a seal arrangement for a centrifugal compressor having an impeller and in which the seal formed by the surfaces of an inner seal rotor and outer seal stator includes vent passage means formed in part at least in said rotor and extending from an intermediate location along the leakage path of the seal to an outlet located at the inlet side of the impeller, the improvement comprising:

said vent passage means includes as a part thereof radially outwardly directed bore means between said rotor and said outlet of said passage means at the inlet side of said impeller to promote movement of fluid through said passage means by providing a centrifugal pumping action of said fluid in said radially outwardly directed bore means.

2. In an arrangement according to claim 1 wherein:

said seal rotor includes an annular groove extending throughout the circumference of said rotor at said intermediate location to form at least said part of said vent passage in said rotor;

said rotor includes a plurality of substantially axially directed bores connected to the inner circumference of said groove to complete said vent passage means in said rotor.

3. In an arrangement according to claim 1 wherein:

said seal formed by said rotor and said stator includes one and another portion on opposite sides of said intermediate location, said one portion being of reduced diameter relative to said another portion.

4. In an arrangement according to claim 3 wherein:

said another portion of said seal is inclined obliquely and radially outwardly from said intermediate location to the low pressure end of said leakage path.

5. In an arrangement according to claim 3 wherein:

said leakage path formed by said one and said another portions of said seal extends substantially axially.

6. The combination of a seal arrangement according to claim 1 with apparatus in which the pressure at the end of said leakage path opposite the inlet side of said impeller is normally maintained at a pressure equal to or in excess of the pressure existing at the inlet side of said impeller, wherein:

said outwardly directed bore means have a radial dimension sufficient to provide a pumping action creating sufficient suction pressure in said vent passage means to draw fluid through said part of said seal between the low pressure end of said leakage path to said vent passage means.

7. In a centrifugal compressor having a rotatable impeller, an inner seal rotor rotatable with said impeller and mounted on the impeller shaft, an outer seal stator, a vented seal arrangement for conducting fluid leakage along the leakage path of said seal to a location upstream of said impeller, comprising:

a circumferential groove located intermediate of the axial extent of said seal and having its depth extending radially inwardly into said rotor;

a plurality of generally axially extending bores in said rotor, each bore having one end in communication with the radially inner part of said groove and the other end in communication with an annular chamber in the hub of said impeller; and

a plurality of bores in the hub of said impeller connecting said annular chamber with the space at the inlet side of said impeller, said last mentioned bores having a direction including a radially outwardly-directed component to provide a centrifugal pumping action tending to scavenge fluid leakage in said seal to said space at the inlet side of said impeller.

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