

- [54] **VANE CONTROLLING SYSTEM FOR
ROTARY SLIDING VANE COMPRESSOR**
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- [63] Continuation-in-part of Ser. No. 160,694, July 8, 1971, abandoned.
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- [58] Field of Search 418/82, 84, 88, 93;
184/6.16, 31

- [56]
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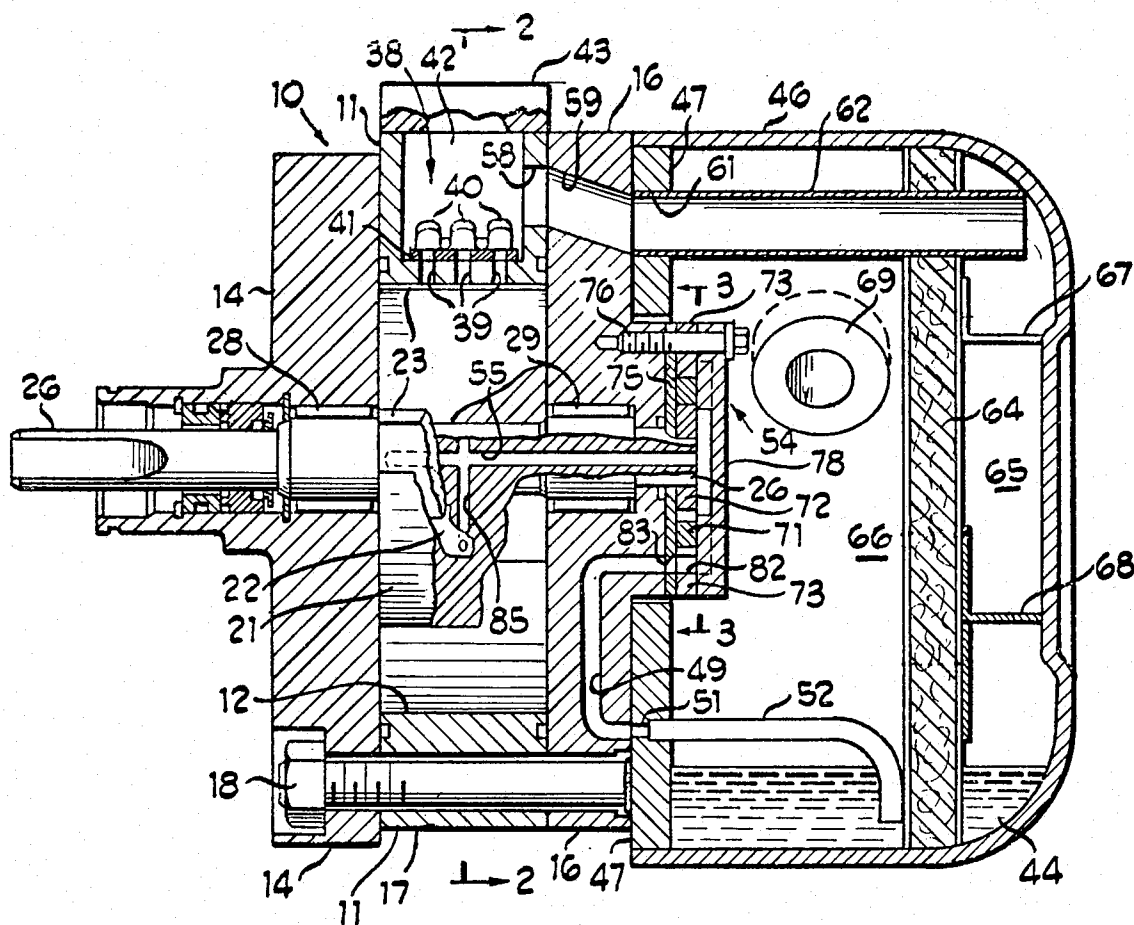
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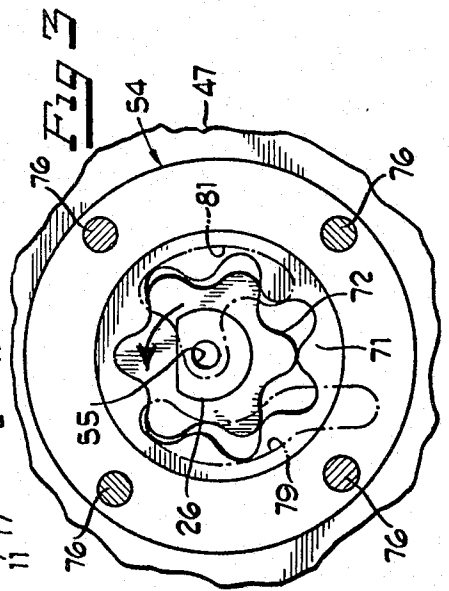
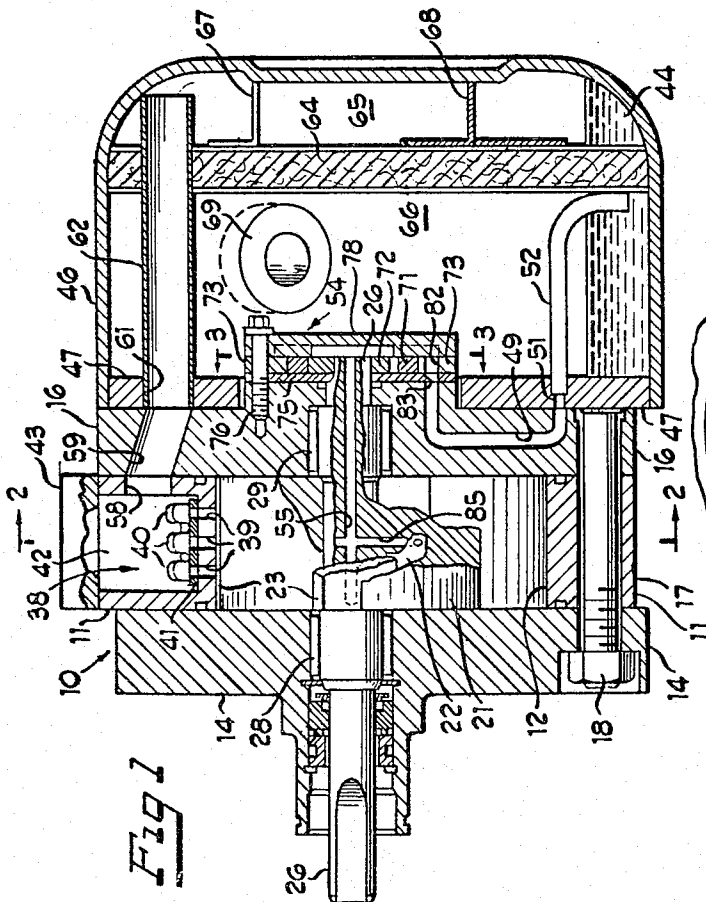
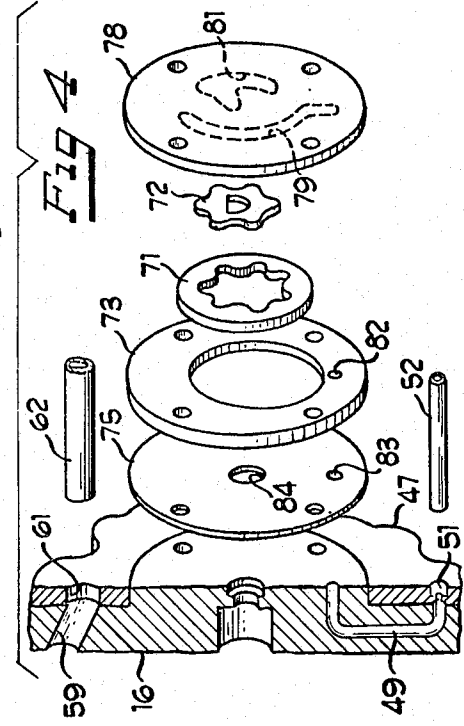
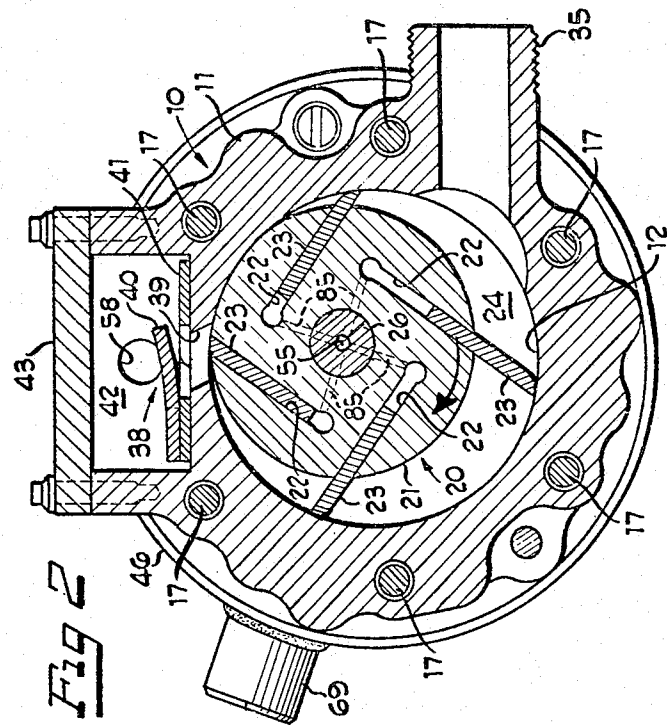
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[57] ABSTRACT

An oil pump feeds pressurized oil into the bottom of the vane-retaining slots of the rotor of a rotary sliding vane compressor in order to urge the vanes out of the slots and into sealed contact with the cylindrical wall of the cavity in which the rotor is eccentrically rotated. The oil pump and rotor are both driven by the same drive shaft and as their speed changes the oil pressure on the vanes tends to increase. At the same time, centrifugal action also tends to increase the force impelling the vanes outwardly. The total force on each vane, however, is maintained within desired limits regardless of rotor speed by dimensioning the oil pump supply to cause increasing cavitation, and consequently a reduction in the rate at which the oil pressure would otherwise increase, in response to increasing speed.

5 Claims, 4 Drawing Figures





VANE CONTROLLING SYSTEM FOR ROTARY SLIDING VANE COMPRESSOR

CROSS REFERENCE TO RELATED APPLICATION

This is a continuation-in-part application of application Ser. No. 160,694, filed July 8, 1971, now abandoned.

BACKGROUND OF THE INVENTION

This invention relates generally to a rotary compressor of the sliding vane type and more particularly to an arrangement for controlling and positioning the vanes of the compressor. It is especially attractive when incorporated in a rotary sliding vane compressor adapted for use in an automotive air-conditioning system and will be described in that environment.

In a typical rotary sliding vane compressor, vanes or blades are slidably received in slots of a rotor revolving in a cylindrical cavity about an axis offset or eccentric from the cavity's axis, a crescent-shaped compression chamber being formed between the rotor and cylindrical cavity wall. Springs interposed between the bottom of the slots and the vanes urge or bias the vanes to constantly bear against the cylindrical wall as the rotor rotates, with the result that suction gas introduced into one portion of the compression chamber is compressed and discharged from another part of the chamber at a higher pressure. Unfortunately, since the spring load is fixed it must be designed for the most severe counteracting force to be experienced at each vane tip. This means that the spring force must be higher than actually needed under most operating conditions. Centrifugal forces, which vary directly with rotor speed, add to the constant force imparted to the vanes by the springs. During start-up and at low speeds, when centrifugal forces are either absent or of negligible effect, the spring forces alone must hold the vanes against the cylindrical wall. As a consequence, during normal operation the strain on the vanes, the vane tip friction and wear, and the power requirements for the compressor are all greater than that necessary. Moreover, it is also a severe application for the springs, breakage often-times occurring and this decreases the compressor's reliability.

Another method employed in the past for holding the vanes out of their slots involves the use of oil pressure differentials. Ordinarily, this is done by directing high pressure oil from a differential lubricating system to the bottom of the slots and under the vanes. One major disadvantage of that approach is that the vanes must be extended to create the required differential pressure. Furthermore, the vanes are subject to an undue strain during normal operation when centrifugal forces are of substantial magnitude.

The present invention is calculated to overcome the disadvantages and shortcomings of previous vanecontrolling arrangements by providing a system in which the force imparted to each vane from start-up to maximum speed always falls within a relatively narrow range, the lower limit of which is only that necessary to engage the cylindrical wall and establish a fluid-tight connection, while the upper limit of the range is less than that which would cause undue strain, friction and wear on the vanes.

It is, therefore, an object of the invention to provide a new and improved rotary sliding vane compressor in

which the force pushing each vane outwardly of its slot is confined at all times to a desired narrow range.

It is another object to provide a vane-controlling system wherein the force imparted to each vane is limited, under all operating conditions and regardless of rotor speed, to a level which will not cause undue strain, friction and wear on the vanes.

SUMMARY OF THE INVENTION

The rotary sliding vane compressor of the invention comprises a casing which defines a closed cavity having a cylindrical wall and a pair of spaced parallel end walls. A slotted rotor is eccentrically positioned within the cavity to define with the cylindrical wall and end walls a crescent-shaped compression chamber. The rotor has a drive shaft journaled in the end walls. A plurality of vanes are slidably mounted or retained in the slots of the rotor. Suction and discharge ports communicate with the compression chamber. A lubricating system, including an oil pump, supplies pressurized oil into the slots behind the vanes to urge them toward and in sealed engagement with the cylindrical wall when the drive shaft is driven, thereby to compress a gaseous fluid introduced through the suction port and to discharge that fluid through the discharge port at a higher pressure. Control means are included in the lubricating system for effectively controlling the oil pressure delivered to the slots so that the total force, as contributed by both the oil pressure and by centrifugal action, holding each of the vanes against the cylindrical wall is maintained within desired limits as the drive speed changes.

More particularly, the oil pump tends to vary the oil pressure directly with the speed at which the drive shaft is driven. The control means effectively reduces the rate at which the oil pressure would otherwise increase as speed increases.

DESCRIPTION OF THE DRAWING

The features of the invention which are believed to be novel are set forth with particularity in the appended claims. The invention, together with further objects and advantages thereof, may best be understood, however, by reference to the following description in conjunction with the accompanying drawing in which like reference numbers identify like elements, and in which:

FIG. 1 is a side view, with portions broken away and partly in cross-section, of a rotary sliding vane compressor constructed in accordance with the present invention;

FIG. 2 is a sectional view taken along the plane of section line 2—2 in FIG. 1 and primarily illustrates the casing and rotor assembly of the compressor;

FIG. 3 is a fragmentary sectional view taken along the plane of section line 3—3 in FIG. 1 and provides an end view of the oil pump; and,

FIG. 4 is an exploded view in perspective illustrating the various parts of the oil pump.

DESCRIPTION OF THE ILLUSTRATED EMBODIMENT

The disclosed compressor has a casing 10 which includes a cylinder structure 11 having a cylindrical bore or wall 12 extending therethrough a front bearing plate 14, and a rear bearing plate 16, all secured together by a series of six bolts 17 and nuts 18, only one of which nuts is shown in FIG. 1. Casing 10 provides a closed

cavity formed by cylindrical wall 12 and bearing plates 14 and 16 which serve as spaced parallel end walls for the cavity. The rotor assembly 20, eccentrically positioned within that cylindrical cavity, includes a slotted rotor 21 having a series of four slots 22 arranged circumferentially and each extending along a plane parallel to the rotor's axis. The closed end of each slot, for convenience, may be referred to as the bottom end. Each of a series of four reciprocating vanes 23 is slidably mounted in a respective one of slots 22. The eccentric positioning of rotor assembly 20 within cylindrical wall 12 is obtained by rotatably mounting rotor 21 on an axis offset with respect to the axis of wall 12. Such eccentric mounting creates a crescent-shaped compression chamber 24 between rotor 21, wall 12, and the two end walls or bearing plates 14 and 16.

Rotor 21 has a drive shaft 26 journaled in bearings 28 and 29 affixed to plates 14 and 16 respectively. The left end of shaft 26 (as viewed in FIG. 1) projects outwardly of front bearing plate 14 to facilitate driving of that shaft. Since the illustrated embodiment is especially adapted for automotive use, it is contemplated that a V-belt pulley and clutch mechanism (not shown) would be coupled to the left end of shaft 26 to permit the compressor to be driven by the engine fan belt or accessory drive belt of the automobile. Of course, the disclosed compressor may be employed in many different environments and may be used in other than refrigeration or air-conditioning systems to compress a variety of different gaseous fluids. Whatever the driving means, it may conveniently be coupled to drive shaft 26.

The compressor is designed to operate when rotor assembly 20 revolves in a clockwise direction as viewed in FIG. 2. In a manner to be explained, under all operating conditions vanes 23 will be forced outwardly to their positions shown in FIG. 2 in order to firmly bear against cylindrical wall 12 and establish a fluid-tight, sealed connection thereto. In operation, suction gas from the evaporator of the automotive air-conditioning system is admitted to an inlet 35 formed in cylinder structure 11. As is illustrated in FIG. 2, this suction gas flows into the suction portion of compression chamber 24. As the rotor is driven clockwise, the suction gas is trapped between two adjacent vanes 23 and carried forward toward the discharge area. As this occurs, the volume between the adjacent vanes is reduced thereby resulting in a corresponding increase in pressure of the gas. A discharge valve assembly 38 is located in the discharge zone for assuring proper compression of the gases issuing from a series of three outlet or discharge ports 39, bored in cylinder structure 11, and for preventing reverse flow of gases back into compression chamber 24. The valve assembly 38 is of the reed type comprising the three valve reeds 40 and the valve seat 41. The compressed gas emanating from ports 39 flows into a chamber 42 defined by cylinder structure 11 and a cover plate 43.

The processing of the gas after it is delivered to chamber 42 will be considered later. At this juncture it is desirable to describe briefly the lubricating system in the compressor. A flow of lubricating oil to the various bearing surfaces and moving components is, of course, required to provide proper lubrication and to seal the high and low pressure sides of the compressor from each other. Furthermore, in accordance with the present invention the lubricating system also produces pres-

surized oil for controlling vanes 23. More particularly, a reservoir of oil or sump 44 is provided in the lower portion of a shell 46, the open end of which is attached and hermetically sealed to a mounting ring 47 in turn affixed and sealed to rear bearing plate 16. Oil passages 49 and 51, formed in plate 16 and mounting ring 47 respectively, and pick-up tube 52 establish a flow path between the oil sump and the inlet of oil pump 54, the details of which will be described hereinafter. Suffice it to say at this point that the oil pump delivers pressurized oil into the axially extending bore 55 of drive shaft 26 and to all of the areas requiring lubrication and sealing.

Due to the use of oil for sealing, the discharge gas flowing through valve assembly 38 and into chamber 42 is heavily laden with oil. This entrained oil must be removed from the gas because substantial quantities of oil in the discharge gas reduces the heat transfer in the condenser and evaporator. In addition, it is much more difficult to supply a sufficient amount of oil to the compression chamber to attain the necessary sealing between the rotor and chamber surfaces.

Oil separation in the disclosed compressor takes place within shell 46. A passageway formed by bores 58, 59 and 61 in cylinder structure 11, bearing plate 16 and mounting ring 47, respectively, together with tube 62 communicates chamber 42 to the extreme end of shell 46. Tube 62 extends through an oil separating filter screen 64 comprised of gas permeable material, such as coarse mesh metal fibres as in a scouring pad. The periphery of separator 64 has the same contour as that of the shell so that its edges fit against the internal diameter of the shell. In this way, separator 64 constitutes a partition to define two different chambers 65 and 66 within shell 46. Element 67 serves as a support bracket for separator 64, while element 68 constitutes a baffle.

In operation of the oil separator, the discharge gas together with the entrained oil flows out of chamber 42 and into chamber 65 through the conduit provided by bores 58, 59 and 61 and tube 62. At that point the velocity of the gas is greatly reduced as it expands into a much larger volume. In expanding, the gas strikes the end of shell 46 and reverses direction as a consequence of which most of the oil separates on the rear surface of the shell and flows down into sump 44. The gas with the remaining oil, after striking the shell and reversing its flow direction, now heads toward the front of the compressor passing through oil separator 64 where the residual oil coalesces and runs down into reservoir 44. The discharge gas flowing into chamber 66 will thus be oil-free. A discharge outlet 69 mounted on shell 46 permits the gas to flow out of chamber 66. Baffle 68 prevents the turbulent gas from reaching and stirring up oil pool 44 and re-entraining oil back into the gas.

Consideration will now be given to the manner in which the lubricating system functions to provide variable pressure oil flow to effect the desired control of vanes 23. Oil pump 54 is of the conventional internal gear-type having a freely rotatable internally-toothed outer gear 71 (see especially FIGS. 3 and 4) driven by an externally-toothed inner gear 72 eccentrically positioned within gear 71. More specifically, the right end of drive shaft 26 projects outwardly from casing 10 and is keyed to inner gear 72 to drivingly connect the shaft to the gear. Outer gear 71 is mounted for free rotation on an axis spaced or offset from that of shaft 26 and

inner gear 72 by means of an eccentric bearing ring 73 rigidly affixed to, but separated by a spacer 75 from, rear bearing plate 16 by means of four cap screws 76. As best seen in FIG. 3, inner gear 72 has one less tooth than gear 71, and a plurality of separate pumping chambers are effectively defined between those gears. Cover plate 78, also held in place by cap screws 76, has a pair of arcuate cutaway or recessed cavities 79, 81 to provide inlet and outlet passages for the oil pump. For convenience, the recessed passages are shown in FIG. 3 in dashed construction to illustrate most clearly the fluid connections to and from the oil pump. Cavity 79 has a vertically depending leg which communicates through holes 82 and 83, in ring 73 and spaced 75 respectively, to the upper end of passage 49 formed in bearing plate 16. Hence, a flow path exists between sump 44 and the inlet of the oil pump. Cavity 81, on the other hand, has a laterally projecting portion extending to the central area of cover plate 78 to fluidly connect the pump outlet to axial bore 55 via bore 84 formed in spacer 75.

In operation of oil pump 54, as drive shaft 26 is driven inner gear 72 likewise rotates (in a counter-clockwise rotation as viewed in FIG. 3) and one portion of each inner gear tooth is always in sliding sealing contact with a portion of an outer gear tooth to maintain the pumping chambers sealed from each other. The chambers effectively revolve about the axis of shaft 26 and each progressively increases to a maximum volume and then decreases to a minimum volume. Passage or inlet port 79 is located so that it communicates with the chambers that are increasing in volume and thus at a relatively low pressure, while outlet port 81 fluidly couples to those chambers that are decreasing in volume and hence at a high pressure.

A series of four radially extending passages 85 are provided in rotor 21 to fluidly couple axial bore 55 to the bottom ends of slots 22. With this arrangement, the high pressure oil exiting from the oil pump is delivered to the slots behind vanes 23 thereby to impel the vanes toward and in sealed engagement with cylindrical wall 12. The magnitude of the oil pressure is set so that during start-up the pressurized oil along will be sufficient to cause the vanes to move out of their slots and establish fluid-tight connections with the cylindrical wall. Preferably, the pressure level will be just adequate to make the required sealed contact between the vane tips and cylindrical wall, but yet will not cause undue strain on the vanes and needless wear.

Since the oil pump is driven by drive shaft 26, the oil pressure fed to slots 22 would normally tend to increase linearly as the rotor speed increases. In accordance with a salient feature of the invention, the rate at which the oil pressure increases with RPM is reduced so that the pressure tends to level off to a constant magnitude. This is desirable inasmuch as centrifugal force, which also acts to push each of the vanes against wall 12, increases with RPM. By effectively limiting the extent to which the oil pressure increases from start-up to maximum rotor speed, the total force (as contributed by both the oil pressure and by centrifugal action) holding each of the vanes against the cylindrical wall may be maintained within a relatively narrow acceptable range and will never be excessive. The upper limit of the range, which prevails at maximum speed, will be less than that which would cause undue strain, friction and wear on the vanes.

The rate, at which the oil pressure would otherwise increase as speed increases, is reduced by decreasing the efficiency of oil pump 54 with RPM. This is achieved by sizing or dimensioning the passageway extending between oil sump 44 and the pump inlet in order to cause cavitation, the amount of which increases with rotor speed. In other words, during high speed operation oil cannot flow into the pump fast enough, thus effecting a vacuum or void in the pump which gives rise to a reduction in the rate of oil pressure increase as speed increases. The faster the operation, the greater will be the vacuum created and the smaller will be the rate of pressure increase.

Proper dimensioning of the oil pump supply passageway, in order to obtain the desired cavitation, may most easily be accomplished by providing an adjustable orifice in the oil pick-up tube of a compressor constructed in accordance with the invention and designed to have a given size and for a given application. While operating the compressor at design conditions, the orifice would be adjusted to introduce the desired cavitation which would exist when the oil flow to the oil pump is optimized to attain the maximum compressor operating efficiency. This is determined by monitoring compressor discharge gas, horsepower and capacity. After attaining the optimum oil flow, the adjustable orifice would be removed and the orifice size would be determined. An orifice of that specific size could then be made an integral part of the oil pick-up tube.

As a variation of the invention, a bypass line controlled by a series-connected pressure relief valve may be coupled from the oil pump's outlet back to its inlet. The valve would be adjusted to block the bypass line during start-up and at low speeds, while at high speeds the resulting pressure would be sufficient to open the valve and bypass or short circuit at least some of the output oil flow directly back to the pump inlet. In this way, the maximum oil pressure on the vanes is reduced even further.

The invention provides, therefore, a unique and more reliable rotary sliding vane compressor in which each of the vanes is subject to a force lying within desired limits under all operating conditions so that the vanes are impelled outwardly instantly upon start-up and so that during high speed operation no excessive vane-pushing force develops.

While a particular embodiment of the invention has been shown and described, modifications may be made, and it is intended in the appended claims to cover all such modifications as may fall within the true spirit and scope of the invention.

I claim:

1. A rotary sliding vane compressor comprising:
 - a casing defining a closed cavity having a cylindrical wall and a pair of spaced parallel end walls;
 - a slotted rotor eccentrically positioned within said cavity to define with said cylindrical wall and said end walls a crescent-shaped compression chamber, and having a drive shaft journaled in said end walls;
 - a plurality of vanes slidably mounted in the slots of said rotor;
 - suction and discharge ports communicating with said compression chamber;
 - and a lubricating system including an oil reservoir which supplies oil to an oil pump which then delivers pressurized oil into said slots behind said vanes

to urge said vanes toward and in sealed engagement with said cylindrical wall when said drive shaft is driven, thereby to compress a gaseous fluid introduced through said suction port and to discharge that fluid through said discharge port at a higher pressure,

the inlet of said oil pump communicating with said oil reservoir via an oil supply passageway sized to create cavitation in an amount which increases as drive shaft speed increases thereby controlling the oil pressure delivered to said slots so that the total force, as contributed by both the oil pressure and by centrifugal action, holding each of said vanes against said cylindrical wall is maintained within desired limits as the drive speed changes.

2. A rotary sliding vane compressor according to claim 1 in which one end of said drive shaft projects outwardly of one of said end walls to facilitate driving of said shaft, and wherein the other end of said shaft projects outwardly of the other end wall and is drivingly

and directly connected to said oil pump.

3. A rotary sliding vane compressor according to claim 1 in which said lubricating system includes an axially extending bore in said drive shaft and a plurality of radially extending passages in said rotor for fluidly connecting said bore to said slots.

4. A rotary sliding vane compressor according to claim 3 in which said oil pump is driven by said drive shaft and has its outlet communicating with the axially extending bore in said drive shaft for conveying high pressure oil to said slots.

5. A rotary sliding vane compressor according to claim 1 in which said oil pump is of the internal gear-type having a freely rotatable internally-toothed outer gear driven by an externally-toothed inner gear eccentrically positioned within said outer gear, and in which said drive shaft is drivingly connected to said inner gear.

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