

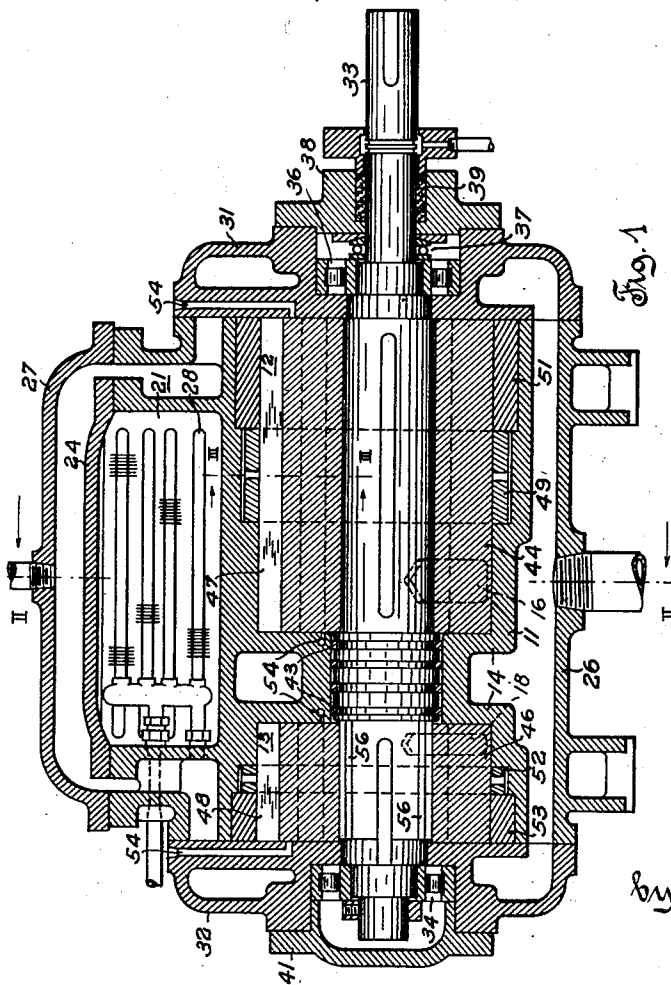
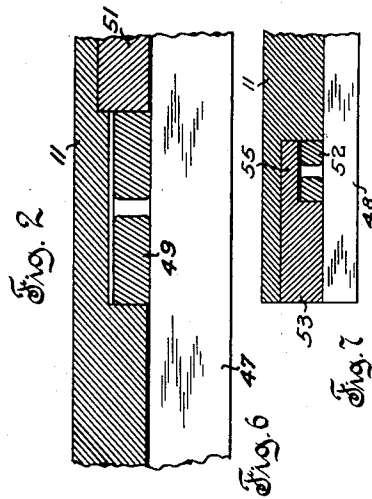
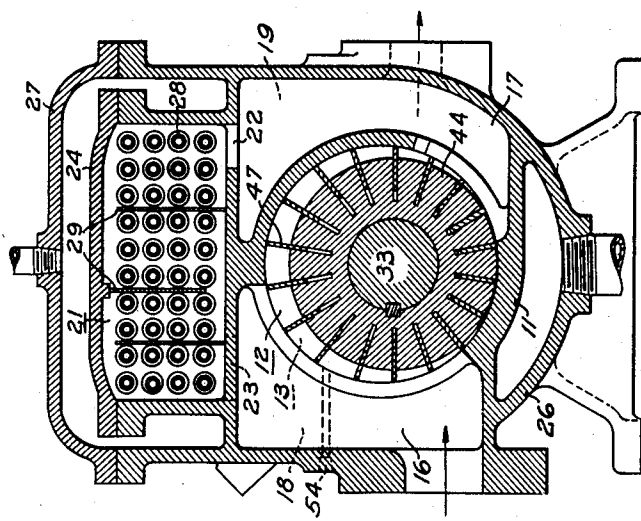
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SLIDING VANE COMPRESSOR

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SLIDING VANE COMPRESSOR

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This invention relates in general to improvements in compressors of the sliding vane type, and more particularly to an improved multistage compressor having a unitary casing including therein the compressor cylinders and an interstage reservoir or intercooler.

Each cylinder of a sliding vane compressor must be provided with so-called floating rings for supporting the rotor vanes and preventing rubbing thereof against the cylinder wall under the action of centrifugal force, as it is very difficult to avoid excessive and uneven wear of the cylinder when the vanes are allowed to apply themselves directly against the surface thereof. When floating rings are used, however, appreciable clearances must be provided between the rings and the walls of the slot containing them, in order that the rings may easily be driven by the vanes of the rotor simply by frictional engagement therebetween. Due to such clearances, leakage takes place between the rings and the casing, which leakage reduces the efficiency of the compressor. It has been found that the vanes of a rotor can be properly supported by a single ring having a width not exceeding four-tenths of the length of the rotor, and leakages between the cells formed by the vanes of each cylinder are thus materially reduced.

Other means, and suitable methods of construction described hereinafter, are also utilized to aid in substantially preventing gas leakages from points of higher pressure to points of lower pressure, which tend to occur at different points of the compressor.

Multistage compressors of the sliding vane type, and more particularly compressors of the two-stage type, usually comprise a plurality of independent coaxially arranged cylinders each containing a rotor provided with vanes, the shafts of the rotors being provided with suitable couplings connecting such shafts and connecting the shafts with a suitable driving means such as a motor or prime mover. The cylinders usually rest on a common base, which may enclose or may form an interstage reservoir or a cooler. Such arrangement, however, results in an installation of considerable weight, requiring a considerable amount of floor space and necessitating complicated piping between the stages of the compressor and the intercooler and for cooling purposes, such piping being arranged externally of the compressor and its base and requiring an undesirably large number of pipe joints. Such compressor also requires stuffing boxes at each point of issue of the shafts from the cylinders so that it becomes difficult to maintain the units sufficiently gas tight. If the intercooler is utilized as a base for the cylinders, such intercooler must be made excessively heavy and is not conveniently accessible for cleaning and repair. The above

recited disadvantages are all obviated by the use of a compressor comprising a unitary casing forming two coaxial cylinders and an intercooler arranged above the cylinders and directly within the casing itself.

It is, accordingly, one of the objects of the present invention to provide a compressor of the sliding vane type in which the danger of gas leakage from the compressor and between the several cells formed by the vanes, is reduced to a minimum.

Another object of the present invention is to provide a compressor of the sliding vane type in which all parts are provided with adequate lubrication for preventing wear between relatively movable parts and for substantially preventing leakage of gas from points of higher to points of lower pressure within the compressor.

Another object of the present invention is to provide a compressor of the multistage sliding vane type comprising a unitary casing for the several cylinders.

Another object of the present invention is to provide a compressor of the two-stage sliding vane type comprising a unitary casing enclosing an interstage reservoir or intercooler.

Another object of the present invention is to provide a compressor of the multistage sliding vane type requiring only a single stuffing box for sealing the extension of the shaft from the casing.

Objects and advantages other than those above set forth will be apparent from the following description when read in connection with the accompanying drawing, in which:

Fig. 1 is an axial, vertical, cross-section through a multistage compressor having a unitary casing enclosing and forming a portion of an intercooler;

Fig. 2 is a transverse cross-section taken along a plane on the line II—II in Fig. 1;

Fig. 3 is an enlarged fragmentary cross-section taken along a plane on the line III—III in Fig. 1;

Fig. 4 is a view of a low pressure rotor vane for a compressor utilized for compressing gas above atmospheric pressure, in which the curvature of the outer edge of the vane was accentuated;

Fig. 5 is a view of a low pressure rotor vane for a compressor utilized as a vacuum pump, in which the curvature of the outer edge of the vane was accentuated.

Fig. 6 is an enlarged portion taken on a plane parallel to the axis of the compressor and accentuating the clearances between the wall of one of the cylinders and the associated floating ring and rotor vanes; and

Fig. 7 is an enlarged portion taken on a plane parallel to the axis of the compressor and accentuating the clearances between the wall of one

of the cylinders and the associated floating ring, and showing a modified form of locking ring construction.

Referring more particularly to the drawing by characters of reference, reference numeral 11 designates a unitary casing which may be cast of suitable material, such as cast iron, or which may be fabricated from laminated material to define a pair of adjacent coaxial cylinders 12 and 13 separated by a partition 14 which is provided with passages for the flow of a cooling medium therethrough. Casing 11 is shaped to form an inlet passage 16 and an outlet passage 17 for cylinder 12, which is the low pressure cylinder of the compressor, the casing also forming an inlet passage 18 and an outlet passage 19 for cylinder 13, which is the high pressure cylinder of the compressor. The casing further defines or encloses an interstage reservoir or intercooler 21 joining outlet passage 17 with inlet passage 18, reservoir 21 and the passages having a common wall provided with ports 22 and 23 providing for direct communication between the reservoir and passages 17 and 18. The reservoir is arranged above the cylinders and is closed by means of a cover 24. All of the connections between the several cylinders and the reservoir are thus arranged internally of the casing and are thus minimized to as great degree as is possible. Reservoir 21 is provided with cooling means to thereby constitute an intercooler, and the cylinders and the intercooler may be arranged to be cooled by air circulating thereabout by convection or by forced draft. It is however preferred to shape casing 11 to form a water jacket 26 surrounding the cylinders and the intercooler, such jacket being provided with a cover 27 to give access to the intercooler cover 24. The intercooler then contains a cooling coil 28 preferably provided with a water inlet separate from the inlet of water jacket 26 and with an outlet discharging into the water jacket thus placing the cooler and jacket in series and avoiding failure to cool any part and providing better temperature conditions in the compressors. Coil 28 may be of any of the types known in the art, but a coil made of copper tubing provided with fins is to be preferred as a coil of such type requires the least amount of space for a given rate of heat transfer. The air or other gas propelled by the compressor through the intercooler is preferably caused to follow a sinuous path within the intercooler by means of baffles so as to permit a compact arrangement of the cooling coil. The cooling coil is connected with the inlet and with the outlet thereof by means of unions, so that the coil may be easily removed from the intercooler for cleaning or for replacement.

It must be noted that the dimensions of both the inlet and discharge ports into the several cylinders is highly important, that is, the ratio of the width of the ports to the length of the cylinder must be kept between the values .125 to .162 for cylinder 12 compressing to the lower pressure and between the values .141 to .147 for cylinder 13 compressing to the higher pressure. The above ratios cannot be widely varied without too greatly reducing the internal cylinder surface or too greatly increasing the velocity of the air or gas through the ports and the several compressor cells which would result in unequal pressures on opposite ends of the vanes or increased air friction losses. Unequal pressures on the vane ends would tend to cause the vanes to rock

in the rotor slots resulting in uneven wear of the vanes and cylinder walls.

Casing 11 is provided with a pair of cylinder heads 31, 32 preferably provided with passages communicating with the water jacket of the casing. The cylinder heads are provided with recesses for a plurality of aligned bearings which are arranged eccentrically with respect to cylinders 12 and 13 for supporting a shaft 33 eccentrically in the cylinders. The bearings are preferably of the antifriction type and may comprise a radial bearing 34 arranged in cylinder head 32, a radial bearing 36 arranged in cylinder head 31, and a thrust bearing 37 acting between a shoulder on the shaft 33 and casing 11 through cylinder head 31 and through a stuffing box 38. Bearings 36 and 37 may also be replaced by a combined thrust and radial bearing if so desired. The end of shaft 33 supported in bearing 34 is confined within casing 11 by a bearing cover 41 bolted on cylinder head 32 and sealed thereagainst, the other end of the shaft being sealed against the casing by a packing 39 arranged in stuffing box 38. Such packing is the only packing required for the compressor, shaft 33 being sealed against partition 14 by means of snap rings 43 arranged within grooves of the shaft. The several snap rings are provided with shallow grooves formed radially in the sides only of such rings toward the low pressure cylinder 12 to allow sufficient oil to pass through the grooves with air from the high pressure cylinder to provide for proper lubrication of the rings in the grooves.

Shaft 33 carries a pair of cylindrical rotors, of which the low pressure rotor 44 is mounted on the shaft with a tight fit and is keyed thereon, while the high pressure rotor 46 is splined on the shaft or provided with a plurality of keys in such manner as to be axially movable with respect thereto. The ratio of the diameters of the cylinder and the rotor should be kept between the values .9 to .904 which determines the ratio of eccentricity of the shaft mounting to the diameter of the cylinder which should be kept between the values .05 to .0473. In no case should the above values vary greatly from .9 and from .05. The value of the first mentioned ratio should be kept as low as possible to maintain volumetric efficiency and keep the dimensions of the machines as small as possible but should also be made as large as possible to obtain a sufficiently stiff shaft. A very small change in such ratios rapidly affects the space available for the shaft within the rotors.

Rotor 44 is provided with a plurality of substantially radial slots each containing a movable blade or vane 47, the vanes being made of heat treated steel, aluminum, artificial resins, fiber composition or other suitable material, the choice of the material depending upon the speed of shaft 33 and on the conditions of pressure and temperature and composition of the gas to be propelled through the compressor. Rotor 46 is also provided with slots containing a plurality of movable vanes 48. The use of artificial resin, fiber composition or similar material for the vanes has been found to be very advantageous for the reason that adequate lubrication and sealing can then be obtained with the use of water soluble oil such as the coolant oils used for metal cutting purposes. Such oils are ideal lubricants for the above type of vanes, may be used in very large quantities to secure the most effective sealing and cooling and are readily separable from air thus allowing delivery of oil free air. A reduction in the moisture

content of the delivered air as well as in the degree of corrosion of the compressor are also obtained with the use of such oils.

Cylinder 12 is formed to present a single annular groove in the interior surface and having the center thereof at or near the center of the cylinder and containing a single floating ring 49. Ring 49 cannot conveniently be made of segments, and it is accordingly necessary to bore casing 11 to form a stepped cylinder fully opened at the end thereof adjacent cylinder head 31, an end ring 51 being fitted at such end of the cylinder into the step of largest diameter and cooperating with the other steps thereof to define a cylinder of uniform diameter having a single annular groove substantially equidistant from the ends thereof. Cylinder 13 is similarly provided with a single floating ring 52 arranged in a groove formed by casing 11 and an end ring 53. As shown in Figs. 1 and 3, ring 49 is preferably made of rectangular cross-section substantially completely filling the complementary groove. The periphery of the ring is generally cylindrical but so shaped as to form, with the bottom of the groove, a plurality of sector shaped chambers, whereby lubricating oil is forced between the ring and the bottom wall of the groove during rotation of the ring. The ring is provided with a plurality of radial passages distributed along the periphery thereof to equalize the pressures exerted on the inner and outer peripheries thereof. Rings 52 and 53 may be similar as shown in Fig. 1 or may be provided with a reduced projecting portion 55 and fitting into a single step in the cylinder to provide a groove formed by the complementary action of the cylinder 11 and the retaining ring 53, 55 as shown in Fig. 7. Such structure of the retaining rings allows the cylinders 12 and 13 to be made with only a single diameter step rather than with two diameter steps as shown in Fig. 1.

A plurality of oil ducts 54 are provided in casing 11 for introducing lubricating oil under pressure at the bottom of the cylinder grooves and between the end surfaces of the rotors and the complementary surfaces of the cylinders, the oil being supplied by any suitable oil pump (not shown). It will be observed that the oil ducts 54 formed in the end heads 31 and 32 extend only to a point adjacent the periphery of the rotors 44 and 46 respectively to discharge the oil flowing through such ducts into the clearance space between the end heads and the rotors closely adjacent to the periphery of such rotors. Such locations of the ducts is necessary to provide for proper lubrication of the rotor ends and sealing of the clearance spaces due to the tendency of the air pressure to force the oil toward the shaft thus leaving the peripheral edges of the rotors unlubricated and unsealed.

The construction of the compressor will be examined more in detail in connection with the conditions of pressure and temperature prevailing in the compressor during operation thereof. During such operation, rotors 44 and 46 rotate at a speed such that vanes 47 and 48 are urged against the walls of the associated cylinders by centrifugal force. The rotors being eccentrically supported within the cylinders, the vanes thereof cooperate with the cylinder walls in defining cells of variable volume carrying air or any other gas from one pressure to a higher pressure if the machine is used as a compressor or as a vacuum pump, the gas passing from one pressure to a lower pressure if the machine is utilized as a motor. Assuming the machine to operate as an

air or gas compressor, shaft 33 being driven by a suitable motor (not shown), air or gas is drawn at substantially atmospheric pressure through inlet passage 16 into cylinder 12 and is delivered from cylinder 12 at a higher pressure into passage 17. The air passes through port 22 into intercooler 21 and through port 23 into inlet passage 18 from which it is drawn through cylinder 13 and delivered into outlet passage 19 and through the outlet port of the compressor, into a suitable compressed air delivery system (not shown).

The air compressing operation causes the evolution of a considerable amount of heat in the air, such heat being transmitted to the walls of the casing and through coil 28 to the water circulating through the water jacket and through the coils. While the walls of the cylinders are maintained at a relatively low and substantially uniform temperature by the water jacket, the moving parts of the compressor are subjected to relatively high and non-uniform temperatures. In particular, considering the operation of cylinder 12, ring 49 reaches a temperature which is materially greater than the temperature of casing 11. The ring accordingly increases in inner and outer diameters by thermal expansion while the cylinder retains substantially its initial diameter. Under such conditions, ring 49 must have an inner radius such that vanes 47 apply themselves thereagainst and are supported thereby out of frictional engagement with the cylinder wall to thereby avoid wear which would result from such frictional engagement. These vanes are then sealed against the cylinder by a film of lubricating oil. The inner radius of ring 49 must thus be smaller than the radius of the cylinder by substantially the thickness of the oil film to be maintained between the vanes and the cylinder, such radii being measured at the respective operating temperatures of the ring and of the cylinder. When the compressor is being started all parts thereof being at the same temperature ring 49 is slightly too small in inner diameter so that the clearance between the vanes and the cylinder is slightly too large. The lubricating oil present on the wall of the cylinder has then, however, a higher viscosity than when such oil has reached normal running temperature, so that an oil film having a thickness equal to the actual clearance of the vanes may be maintained on the cylinder wall.

To avoid the occurrence of wear between the vanes and the ring, such ring must be easily driven by the vanes by frictional engagement therewith to cause the ring to rotate at substantially the average speed of the outer edges of the vanes, such edges having only a very slight combined sliding and oscillating movement with respect to the ring. To insure such operation the outer radius of the ring must be less than the radius of the complementary groove by the thickness of a substantial oil film, the maintenance of such film being insured by the presence of the sector shaped chambers defined by the ring and the bottom wall of the groove. A cross-section taken through the cylinder and the ring when such elements are at room temperature then has the appearance indicated in Fig. 3, in which radial clearances are, however, accentuated to clarify the drawing. It will be understood that the inner surface of ring 49 is subjected to different pressures at different points thereof due to different degrees of air compression in the several cells and that a side thrust tends to be exerted on the whole ring as a result of the unbalance of such

pressures. Such thrust is avoided by the provision of the radial passages through the ring whereby the pressures exerted outwardly at every point of the inner surface of the ring are also exerted inwardly on corresponding points of the outer surface, the inward and outward pressures thus balancing each other.

The width of ring 49 necessary for providing adequate support for vanes 47 is determined by the length of the vanes, but must always be made less than four-tenths of the length of the cylinder 13. The dimensions of ring 52 are determined by the same considerations as those of ring 49. In order to provide sufficient room beside ring 52 for the ports joining cylinder 13 with passages 18 and 19, it is generally necessary to reduce the width of ring 52, such width however, being not less than two-tenths of the length of cylinder 13. The above ratios of floating ring width to cylinder length is highly important because the single floating ring must be sufficiently wide to balance the vanes without permitting rocking thereof in the rotor slots in an axial plane. The single floating ring used must be placed substantially centrally in the cylinder and cannot be made so wide as to encroach on the space required between the cylinder port edges and the edge of the floating ring groove.

During operation of the compressor, shaft 33 reaches a temperature higher than that of casing 11, and the shaft accordingly expands axially to a material extent. The end of shaft 33 adjacent cylinder 12 and carrying thrust bearing 37 remains in fixed relation to casing 11 and the end surface of rotor 44 adjacent bearing 37 is not materially displaced with respect to the casing upon heating of the shaft. Rotor 44 however also expands axially and is accordingly made shorter than cylinder 12 by an amount such that the clearance between the rotor and the partition 14 remains within the range of values suitable for the establishment of an oil film therein at all operating temperatures. The end of shaft 33 carrying rotor 46, being remote from thrust bearing 37, is displaced axially to a material extent by thermal expansion of the shaft and it is therefore necessary to leave rotor 46 axially movable with respect to the shaft. Rotor 46 is made shorter than cylinder 13 by an amount such that the clearances between the end surfaces of the rotor and of the cylinder are always within the range of values suitable for the establishment of an oil film therein. Such oil films are the sole means utilized for centering rotor 46 axially in cylinder 13, such result being produced by the slight thrust exerted by the oil films on the end surfaces of the rotor by capillarity. Satisfactory oil films are not however obtainable unless the oil is supplied at points closely adjacent to the periphery of the rotor.

The two oil films established on the surfaces of partition 14 by the introduction of oil through oil ducts 54 are jointly subjected to the difference of the pressures exerted by the air in cylinder 12 and in cylinder 13. Under the action of such pressure difference, oil tends to flow from cylinder 13 through the bore of partition 14 into cylinder 12 and, if the oil films do not extend over the entire surfaces of partition 14, air may also leak between the two cylinders through the bore of the partition. Snap rings 43 cooperate with the bore of the partition in forming a seal of the labyrinth type, such seal substantially preventing the flow of oil and of air between the cylinders under the action of the air pressure.

An absolutely complete prevention of such flow would not materially improve the operation of the compressor and would require the use of a packing, which could be installed and maintained only with considerable difficulty.

The slight flow of oil and air within the clearance between rotor 46 and partition 14 tends to cause the end surface of the rotor adjacent the partition to be subject to pressures differing from the pressures exerted by the oil against the other end surface of the rotor which is not subject to such oil flow. To obtain an accurate axial centering of rotor 46 within cylinder 13, it is necessary that the pressures exerted by the oil films on the two end surfaces thereof be substantially balanced. Such result is obtained by providing grooves 56 in the periphery of shaft 33 or in the rotor whereby the two end clearances of rotor 46 are put in direct communication and are therefore maintained at the same pressure.

During operation of the compressor, the rotor and the vanes thereof, which are not directly water cooled, are unable to transfer an appreciable amount of heat from the air passing through the compressor to the water jacket. The rotor and the vanes of each stage accordingly assume temperatures intermediate between the inlet and outlet temperatures of the air passing through such stage. In general, the inner edge of each vane, which is deeply embedded in the associated rotor, is at substantially rotor temperature while the outer edge of the vane, which generally protrudes from the rotor, assumes another temperature. As a result of such non-uniformity of temperature, the two edges of each vane expand by different amounts, thus causing the vane to assume the shape of a fragment of an annulus with the result that the outer edges of the vanes become curvilinear and no longer present a uniform clearance with the wall of the cooperating cylinder. The curvature of the outer edges of the vanes is usually material only in the long low pressure vanes, such as vanes 47, and may become sufficiently pronounced to render it necessary to make such edges curvilinear when the vanes are of uniform temperature so that such edges may become rectilinear by thermal expansion of the vanes at the non-uniform operating temperatures thereof. When the compressor is utilized for compressing air from substantially atmospheric pressure to a higher pressure, the outer edges of vanes 47 which protrude from rotor 44 mostly on the inlet side of cylinder 12, are cooled by the air to a temperature lower than the temperature of rotor 44, with the result that the distortion of vanes 47 causes the outer edges thereof to become concave. Such occurrence may be prevented by making such edges convex as shown in Fig. 4, the convexity of the edge when the vane is at uniform temperature being such that the edge becomes substantially rectilinear at the non-uniform operating temperature of the vane.

When the compressor is utilized as a vacuum pump to compress gas from a pressure materially below atmospheric pressure to substantially atmospheric pressure, the cooling effect of the air in the inlet passage on the outer edge of the vane 47 is negligible, and such outer edges are then heated by the air being compressed in cylinder 12 to a temperature higher than the temperature of rotor 44. Such outer edges accordingly tend to become convex and are preferably made concave when the vanes are at uniform temperature so as to become rectilinear when the vanes reach the non-uniform operating temper-

ature thereof. The curvature of the vanes is always very slight, so that when the compressor is started and the vanes are at uniform temperature an oil film may still be maintained between the outer edges of the vanes and the wall of the cylinder.

It will be apparent that many of the novel structural features herein disclosed such as the single floating ring in each cylinder, the manner of supplying oil to the cylinder end heads, the composition blades and water soluble oil lubrication, etc. are applicable to both multistage and single stage compressors of the sliding vane type and as so suitable, have been separately claimed herein.

Although but one embodiment of the present invention has been illustrated and described, it will be apparent to those skilled in the art that various changes and modifications may be made therein without departing from the spirit of the invention or from the scope of the appended claims.

It is claimed and desired to secure by Letters Patent:

1. In a rotary compressor, a casing defining a cylinder having a groove formed in the inner surface thereof, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, a plurality of vanes movably mounted in the slots of said rotor, said vanes being urged toward the wall of said cylinder upon rotation of said rotor, and a ring mounted in the groove in said cylinder for supporting said vanes out of frictional engagement with the inner surface of said cylinder, said ring being provided with apertures therethrough and having the outer periphery formed with plane surfaces about the apertures to form sector shaped chambers with the surface of the groove.

2. In a rotary machine of the sliding vane type, a casing defining a cylinder, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder and beyond the periphery of said rotor upon rotation thereof, the longitudinal edges of said vanes extending beyond the periphery of said rotor being made curvilinear relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine.

3. In a rotary machine of the sliding vane type, a casing defining a cylinder, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder and beyond the periphery of said rotor upon rotation thereof, the edges of said vanes extending beyond the periphery of said rotor being made convex relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine as a compressor.

4. In a rotary machine of the sliding vane type, a casing defining a cylinder, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being

urged toward the wall of said cylinder and beyond the periphery of said rotor upon rotation thereof, the edges of said vanes extending beyond the periphery of said rotor being made concave relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine as an exhauster.

5. In a rotary machine of the sliding vane type, a casing defining a cylinder formed to provide a single substantially centrally located groove therein, a ring mounted in the groove, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder into engagement only with said ring and beyond the periphery of said rotor upon rotation thereof, the edges of said vanes being made curvilinear relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine.

6. In a rotary machine of the sliding vane type, a casing defining a cylinder formed to provide a single substantially centrally located groove therein, a ring mounted in the groove, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder into engagement only with said ring and beyond the periphery of said rotor upon rotation thereof, the edges of said vanes being made convex relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine as a compressor.

7. In a rotary machine of the sliding vane type, a casing defining a cylinder formed to provide a single substantially centrally located groove therein, a ring mounted in the groove, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, and a plurality of vanes of rectangular cross section severally movably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder into engagement only with said ring and beyond the periphery of said rotor upon rotation thereof, the edges of said vanes being made concave relative to the axis of said vanes by a sufficient amount to provide a rectilinear edge of said vanes due to thermal expansion thereof during operation of the machine as an exhauster.

8. In a rotary compressor of the sliding vane type, a casing defining a cylinder having a groove formed in the inner surface thereof and having ports thereinto, a rotor eccentrically supported in said cylinder and having a plurality of slots therein, a plurality of vanes slidably mounted in the slots in said rotor, said vanes being urged toward the wall of said cylinder upon rotation of said rotor, and a ring mounted in the groove in said cylinder for supporting said vanes out of frictional engagement with said cylinder, the width of the ports and of said ring being between the limits of one-sixth and one-eighth and between the limits of two-tenths and four-tenths, respectively, of the length of said cylinder.

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