

SCREW COMPRESSOR

The present invention relates generally to screw compressors.

Compressors are employed in refrigeration systems
5 to raise the pressure of a refrigerant gas from a
suction pressure to a higher discharge pressure which
permits the ultimate use of the refrigerant to
accomplish the cooling of a desired medium. Many
types of compressors, including reciprocating, scroll
10 and screw compressors, are used in refrigeration
applications. Screw compressors employ complementary
male and female screw rotors disposed within the
working chamber of a rotor housing to compress gas.
The working chamber can be characterised as

a volume generally shaped as a pair of parallel intersecting cylindrical bores closely toleranced to the outside length and diameter dimensions of the screw rotors disposed therein. The screw rotor housing has low and high pressure ends which include suction and discharge ports respectively. Both the suction and discharge ports are in flow communication with the working chamber of the rotor housing.

Refrigerant gas at suction pressure enters the compressor working chamber via the suction port at the low pressure end of the rotor housing and is there enveloped in a pocket formed between the rotating complementary screw rotors. The volume of this chevron-shaped pocket decreases and the pocket is displaced toward the high pressure end of the compressor as the rotors rotate and mesh within the working chamber. The gas within such a pocket is compressed by virtue of the decreasing volume in which it is contained, until the pocket opens to the discharge port at the high pressure end of the compressor. As the pocket opens to the discharge port, the volume of the pocket continues to decrease and the compressed gas is forced through and out of the discharge port of the rotor housing.

Due to the extremely close tolerances between the rotors of the screw rotor set and the elements in the compressor which cooperate to define the working chamber in which the rotors are disposed, the bearing arrangement by which the rotor set is mounted within the working chamber is critical to compressor operation and life. The bearings in a screw compressor are subject to high axial and radial loads which can vary greatly from the low pressure end of the compressor to the high pressure end. Protection and lubrication of the rotor bearings is therefore of paramount concern in the design of rotary screw compressors. Since the suction and discharge ports of screw compressors are

valveless and are essentially unobstructed openings into and out of the working chamber of the compressor, the rotor set within the working chamber is exposed, in operation, to the high pressure gas downstream of the compressor discharge port. Additionally, the gas undergoing
5 compression in a pocket between the rotors bears against the high pressure end wall of the working chamber to create additional thrust on the rotors in a direction toward the low pressure end of the compressor. Therefore, a large axial thrust is developed, in operation, on the rotor set of a screw compressor in a direction from the high pressure
10 end of the compressor to the low pressure end. This axial force must be compensated for by the compressor's bearing arrangement.

At compressor shutdown, the backflow of high pressure gas from the high pressure side of a refrigeration system through the open discharge port of the compressor toward the low pressure side of the
15 system, if allowed to occur, would cause the high speed reverse rotation of the no longer driven screw rotors within the working chamber. Such freewheeling of the rotors could occur at speeds greater than the maximum design RPM of the rotor set and rotor bearings. Additionally, the resulting rush of downstream high pressure gas back through the
20 compressor to the low pressure side of the system could result in a surge of pressure into the low pressure side of the system such that a higher pressure might momentarily develop at the suction end of the compressor than exists at the discharge end of the compressor. This situation would result in a re-surge of pressure and gas from what is
25 normally the low pressure side of the system to what is normally the high pressure side of the system in attempt to equalize system pressures and would further result in inordinate and uncommonly large axial

forces acting on the screw rotor set and rotor bearings in a direction opposite that normally expected and compensated for in operation. That is, axial force will be brought to bear on the screw rotor set and the bearings in which the rotors are mounted in a direction toward the high pressure end of the working chamber of the compressor. Several un-
5 ward results can occur if such high-speed reverse direction rotor rotation and the resulting pressure transients are allowed to occur. Among these results are the aforementioned development of axial thrust on the rotor set in a direction which is not compensated for to the degree
10 normal axial thrust is compensated for within the compressor. Further, possible mechanical failure due to the achievement of rotor speeds exceeding design RPM might occur. Additionally, most, if not all, screw compressor bearing lubrication schemes are predicated on the development of pressure downstream of the compressor to drive lubricating oil
15 to the rotor bearings. The high speed reverse rotation of the rotor set and momentary development of high pressures upstream of the working chamber, if allowed to occur, could theoretically cause oil to be sucked from the bearings or, in any event, not to be delivered to the bearings with possibly catastrophic results.

20 The number and complexity of patented screw compressor bearing and/or bearing lubrication schemes illustrates the ongoing need for an uncomplicated, inexpensive device by which bearing and/or bearing lubrication arrangements in screw compressors can be protected and simplified.

It is an object of this invention to prevent the high speed reverse rotation of the screw rotors in a screw compressor due to the backflow of previously compressed gas into and through the working chamber of the compressor subsequent to compressor shutdown.

The invention includes a screw compressor provided with apparatus for preventing the reverse rotation of screw rotors thereof upon compressor shutdown, said apparatus comprising:

10 a body located downstream of the discharge port of said compressor and movable with respect to said discharge port between a first position in which the flow of compressed gas from said discharge port is unimpeded while said compressor is in operation and a
15 second position in which the backflow of gas through said discharge port from downstream of said compressor is prevented, said body having a concavely curved contoured surface; and

means for mounting said body for movement between
20 said first and said second positions, said body being mounted on said mounting means so that said concavely curved contoured surface faces into the flow of gas discharged from said compressor such that said concavely curved contoured surface diverts the
25 discharged gas which impacts it.

The invention also includes a screw compressor assembly in a refrigeration system comprising an oil-injected compressor section, said compressor section including a screw rotor housing defining a working
30 chamber and a discharge port in flow communication with said working chamber, said compressor section further including a pair of complementary screw rotors disposed for rotation in said working chamber, said compressor discharge port receiving a mixture of oil
35 and gas compressed by the rotation of said rotors in a

predetermined direction within said working chamber when said compressor section is in operation and said rotors being free to freewheel in a direction opposite said predetermined direction when said motor is de-energized;

an oil separator section in flow communication with the discharge port of said compressor section; and

means, disposed downstream of the discharge port of said compressor section, for preventing the rotation of said screw rotors in said direction opposite said predetermined direction due to the backflow of previously compressed gas from downstream of said discharge port back to and through said working chamber upon compressor shutdown, said means for preventing the rotation of said rotors upon compressor shutdown comprising a body having a symmetrical concavely curved contoured surface facing into the gas-oil mixture discharged through said discharge port when said motor is energized, said body being positionable to said first position under the impetus of the gas-oil mixture discharged from said compressor when said motor is energized and positionable to said second position under the impetus of the initial backflow of gas from downstream of said discharge port which occurs back toward said discharge port when said motor is de-energized.

Briefly, in an embodiment of the invention described hereinafter the above-mentioned body is positioned downstream of and in line with the discharge port or discharge passage within a screw compressor assembly in a refrigeration circuit and has an aerodynamically-shaped surface which faces into the

mixture discharged from the compressor. In a first or retracted position, the body allows for the unimpeded discharge of the compressed gas-oil mixture from the screw compressor. The body is carried away from the discharge port or passage under the impetus of the mixture discharged from the compressor until it seats on a stop. The aerodynamically-shaped surface on the body deflects the discharged mixture with the result that the mixture is imparted a radial velocity vector. This change in direction itself facilitates disentrainment of liquid from the mixture. Additionally, such deflection aids in the delivery from the mixture in an advantageous manner to a centrifugal oil separator while minimizing the pressure drop in the compressed refrigerant gas portion of the mixture.

The body is slideably disposed on a rod so that upon compressor shutdown, the initial backflow of high pressure gas downstream of the compressor section to the low pressure side of the system, via the compressor discharge port and working chamber, carries the valve body into the discharge port or passage to essentially stop such backflow at the outset. System pressures will then equalize at a much slower rate and the rush of high pressure gas back through the working chamber of the compressor and the resultant high speed reverse rotation of the rotors will be prevented. In systems where the slide valve actuating piston is exposed to discharge pressure and is biased to the unload position by such pressure, the maintenance of pressure in the high pressure side of the system subsequent to compressor shutdown facilitates and insures the movement of the slide

valve to the unload position within the compressor assembly. Further, the maintenance of pressure downstream of the compressor insures the continued
5 delivery of oil to critical compressor locations as the rotors coast to a stop.

In order that the invention may be well understood, the above-mentioned embodiment thereof, which is given by way of example only, will now be
10 described, with reference to the accompanying drawings, in which:

Figure 1 is a cross-sectional view of a screw compressor operating at part-load conditions in a refrigeration circuit;

15 Figure 2 is a partial cross-section of the compressor assembly of Figure 1 illustrating the position of compressor assembly components when the compressor is both shut down and unloaded;

20 Figure 3 is a partial cross-sectional view of the inlet area of the oil separator section of the compressor assembly of Figure 1;

Figure 4 is a perspective view of an anti-rotation body of the assembly; and

25 Figure 5 is a sectional view taken along line 5-5 of Figure 1.

Referring first to Figure 1, a refrigeration system 10 includes a screw compressor assembly 12 which is comprised of a compressor section 14 and an oil separator section 16. Refrigeration system 10
30 further includes, typically, a condenser 18, an expansion device 20 and an evaporator 22. Compressed refrigerant gas, from which oil has been separated, is directed from oil separator section 16 of compressor

assembly 12 to condenser 18 where it is condensed and becomes a low temperature, high pressure liquid. From condenser 18 the refrigerant is directed to expansion device 20 where it becomes a low temperature, low pressure liquid by the process of expansion. The low pressure, low temperature liquid refrigerant next enters evaporator 22 and is there vaporized and becomes a low pressure low temperature gas prior to being returned to compressor section 14.

Compressor section 14 includes a rotor housing 24 which defines a suction area 26, into which vaporized low pressure refrigerant gas is communicated from evaporator 22. Rotor housing 24 also defines a suction port 28 through which such gas is admitted to compressor working chamber 30. Screw rotors 32 and 34 are housed in working chamber 30. Attached to the driven one of rotors 32 and 34 is motor 36 which drives shaft 38 on which the driven rotor is mounted. Suction area 26, in the preferred embodiment, includes suction subareas 40 and 42 all of which are in flow communication within rotor housing 24. Rotor housing 24 also defines an opening 44 into suction subarea 42, the purpose of which will later be described.

Rotor housing 24 further includes a discharge port 46 through which compressed refrigerant gas is discharged from working chamber 30. Disposed within rotor housing 24 and cooperating therewith to define working chamber 30 is a slide valve 48. Slide valve 48 is axially movable with respect to rotors 32 and 34 within rotor housing 24. In the position illustrated in Figure 1, working chamber 30 is in flow communication with suction subarea 40 of suction area 26 as well as suction area 26 through suction port 28. Slide valve 48 is positionable between a first position in which low pressure end face 50 of the valve abuts stop 52 of rotor housing 24 and a second position, illustrated in

Figure 2, in which the degree to which rotors 32 and 34 are exposed to suction subarea 40 is at a maximum. When low pressure end face 50 of valve 48 abuts stop 52 of rotor housing 24, direct flow communication between working chamber 30 and suction subarea 40 is prevented and the
5 compressor operates at full load. In the position illustrated in Figure 1, slide valve 48 is at an intermediate position representing a position where the compressor is operating at part-load conditions. The degree to which rotors 32 and 34 are exposed to suction subarea 40 is determinative of the volume of gas which will be compressed between
10 the rotors and therefore, the load on the compressor.

Referring now to both Figures 1 and 2, oil separator section 16 includes a centrifugal oil separator element 54 disposed within a sealed oil sump housing 56. In the preferred embodiment, a bearing housing 58 defining a discharge passage 60 is disposed between dis-
15 charge port 46 of rotor housing 24 and separator element 54. Separator element 54 defines an inlet 62 in flow communication with passage 60 of bearing housing 58 and includes a permeable wall 64 which cooperates with inner cylindrical housing 66 and ramp 68 to define a helical passage between inlet 62 and outlet 70 of sump housing 56.

20 Inner cylindrical housing 66 includes a pressure housing 72 in which piston 74 and spring 76 are disposed. Piston 74 and pressure housing 72 cooperate to define a pressure chamber 78 which is capable of selective flow communication with opening 44 in rotor housing 24 or with sump area 80 of oil separator 16 through opening 82 in sealed sump
25 housing 56. Pressure chamber 78 is put into flow communication with opening 44 and suction subarea 42 by the opening of solenoid valve 84 or with sump area 80 by the opening of solenoid valve 86. Housing 66

has an end cap 88 which defines an opening 90 through which the face of piston 74 opposite the face which cooperates to define chamber 78 is constantly maintained in flow communication with the remainder of the interior of oil separator element 54.

5 Also disposed interior of separator element 54 are swirl vanes 92 and anti-rotation body 94. Body 94, as will more thoroughly be discussed, is slidably mounted on connecting rod 96 which connects piston 74 within oil separator section 16 and slide valve 48 within rotor housing 24. It will be appreciated that when piston 74 moves
10 within pressure housing 72, slide valve 48 is correspondingly moved within rotor housing 24 and further, that the movement of rod 96 does not of itself affect the movement of body 94.

Referring to Figures 3, 4 and 5, body 94 is restrained from rotation around connecting rod 96 by pins 98 which are attached to body
15 94 and which are slidably disposed for movement through holes 100 in end cap 88. Further, it will be seen that flange portion 102 of body 94 is prevented from moving toward discharge passage 60 any further than is illustrated in Figures 2 and 3 by seating surface 104, which as illustrated in the Figures, is a flat surface on bearing housing 58.
20 Likewise, body 94 is prevented from moving toward opening 90 of end cap 88 any more than is illustrated in Figure 1, by the abutment of back surface 106 of body 94 with seats 108 on swirl vanes 92. In this way a gap is maintained between back surface 106 of body 94 and end cap 88 to ensure that opening 90 of end cap 88 is unobstructed at all times so as
25 to maintain flow communication between one side of piston 74 and the remainder of the interior of separator element 54. Back face 106 of body 94 is flat while face 110 of body 94, which faces into discharge passage 60, has a symmetrical concavely curved contoured surface.

In operation, refrigerant gas is sucked into working chamber 30 through suction port 28 by the rotation and meshing of rotors 32 and 34, one of which is driven in a predetermined direction by motor 36. When motor 36 is in operation, at least a portion of the refrigerant gas sucked in through suction port 28 into working chamber 30 is compressed and discharged through discharge port 46 no matter what the position of slide valve 48. As illustrated in Figure 1, compressed refrigerant gas is discharged from working chamber 30 through discharge port 46 and into discharge passage 60 of bearing housing 58. Although not illustrated, oil from sump 80 is injected into working chamber 30 while the compressor is in operation. Oil in sump 80 is essentially at discharge pressure when the compressor assembly is in operation due to the permeability of wall 64 of separator element 54. The oil from sump 80 is further employed to lubricate the bearings and the bearing areas in which the ends of the shafts of rotors 32 and 34 are mounted in the compressor assembly. Such lubricating oil is vented into the working chamber of the compressor after it passes through the bearings and bearing areas. Additionally, sump oil is selectively directed out of sump 80 through solenoid valve 86, when valve 86 is opened, and into pressure chamber 78 to cause the movement of piston 74 and the corresponding movement of slide valve 48 in rotor housing 24. When it is desired that the slide valve should be moved so as to unload the compressor, pressure chamber 78 is vented through solenoid valve 84 into suction subarea 42 of rotor housing 24. It will readily be appreciated that what is discharged from discharge port 46 of rotor housing 24 is compressed refrigerant gas heavily laden with the oil which makes its way into the working chamber from the many locations described above.

The mixture of oil and refrigerant gas discharged from compressor section 14 enters oil separator section 16 through inlet 62 and immediately impinges on contoured face 110 of anti-rotation body 94. Such impact initially serves to carry body 94 away from discharge port 46 until back face 106 contacts seats 108 of swirl vanes 92. While the discharge of the oil-refrigerant gas mixture continues, body 94 will remain seated against seats 108 and in the position illustrated in Figure 1 under the influence of the force of the mixture being discharged from the working chamber of the compressor section. The area around inlet 62 of oil separator element 54 will be saturated with oil as the discharge mixture impacts contoured face 110 of body 94. Consequently, body 94 slides easily on rod 96. A small amount of leakage past body 94 through the gap between rod 96 and the hole defined by body 94 through which rod 96 passes is not detrimental. A somewhat loose fit between rod 96 and body 94 is therefore permissible and insures the easy sliding of body 94 on rod 96.

The mixture of refrigerant gas and oil entering inlet 62 of oil separator element 54 is forced by interaction with contoured face 110 of body 94 to undergo a smooth transition from essentially axial flow to a combination of axial and radial flow within separator element 54. The mixture is thus fed into swirl vanes 92, which impart a rotational or swirling motion to the mixture in a predetermined direction, already having been imparted a radial velocity vector by body 94. This pre-swirled mixture is next fed into the helical passage defined within separator element 54 by ramp 68, permeable wall 64 and inner housing 66. The gradual and smooth directional changes imparted to the mixture are purposeful and minimize pressure drop in the compressed refrigerant

gas during the oil separation process. As the high pressure mixture moves through separator element 54, the centrifugal force developed within the mixture due to its flow path causes the heavier liquid portion of the mixture to migrate radially outward within the separator element and to pass through permeable wall 64. The gas from which the oil has been separated continues to move through the separator element and out of sealed housing 56 through outlet 70. The separated oil is deposited in sump 80 of sealed housing 56.

Upon shutdown of the compressor, that is, when motor 36 is de-energized and the rotors are no longer driven, the high pressure gas within and downstream of separator element 54 will tend to rush back through inlet 62, discharge passage 60 and discharge port 46 of rotor housing 24 and into working chamber 30. Such backrush of gas will, unless prevented, cause the rotors to be driven in a direction opposite that in which they are driven by motor 36 and at possibly unacceptably high speeds. The rotors are free to freewheel in any direction whenever they are no longer driven by motor 36. The initial backrush of such gas out of oil separator element 54 will, however, carry body 94 along rod 96 toward the compressor section until flange portion 102 of body 94 seats against seat 104 effectively plugging inlet 62 and preventing further backflow from occurring. It will be appreciated that the length of time it takes for the refrigeration system high pressure and low pressure sides to equalize will thus be prolonged while the high speed reverse rotation of the rotors is prevented. The prolonged maintenance of pressure downstream of inlet 62 of oil separator element 54 further insures that the delivery of oil to the aforementioned

rotor shaft bearings and working chamber of the compressor continues to occur immediately subsequent to compressor shutdown since it is the high pressure developed and maintained within oil separator section 16 which, in operation, drives oil from sump 80 to its places of employment within compressor assembly 12.

The maintenance of pressure in oil separator section 16 at shutdown also assists spring 76 in biasing the slide valve 48 to the unload position within rotor housing 24 and as illustrated in Figure 2: Upon compressor shutdown, solenoid valve 84 is automatically opened so as to vent pressure chamber 78. The pressure within oil separator 16 bears on the same side of piston 74 as does spring 76. When body 94 plugs inlet 62 so as to prevent the rush of gas back through inlet 62 to the low pressure side of system 10, the pressure trapped within oil separator element 54 will act on piston 74 in conjunction with spring 76 with the result that slide valve 48 will be moved to the unload position in rotor housing 24. Once body 94 plugs inlet 62, valve 48 will no longer itself be acted upon by discharge pressure at the discharge port and the movement of the valve to the unload position shown in Figure 2 will be assured.

From the foregoing it will be understood that by preventing high speed reverse rotation of the screw rotors due to the backflow of previously compressed gas into and through the working chamber of the compressor subsequent to compressor shutdown, axial thrust is prevented on the rotor set and the bearings in which the rotor set is mounted, in a direction opposite the direction of axial thrust developed on the rotor set in normal loaded operation of the compressor. This facilitates and economizes the design of a bearing arrangement in a screw compressor by eliminating the need for apparatus dedicated to the compensation of axial thrust in a direction opposite the direction in which axial thrust is developed in normal operation within a screw compressor.

CLAIMS:

1. A screw compressor provided with apparatus for preventing the reverse rotation of screw rotors thereof upon compressor shutdown, said apparatus
5 comprising:

a body located downstream of the discharge port of said compressor and movable with respect to said discharge port between a first position in which the flow of compressed gas from said discharge port is
10 unimpeded while said compressor is in operation and a second position in which the backflow of gas through said discharge port from downstream of said compressor is prevented, said body having a concavely curved contoured surface; and

15 means for mounting said body for movement between said first and said second positions, said body being mounted on said mounting means so that said concavely curved contoured surface faces into the flow of gas discharged from said compressor such that said
20 concavely curved contoured surface diverts the discharged gas which impacts it.

2. A compressor according to claim 1, wherein said body is mounted for slidable movement on said mounting means and is positionable in said first position under
25 the impetus of gas discharged from said compressor when said compressor is in operation and is



positionable in said second position under the impetus of the flow of discharge gas back toward the discharge port of said compressor which occurs when said compressor is shut down.

5 3. A compressor according to claim 2, wherein said mounting means comprises a rod and wherein said contoured surface of said body is symmetrical about an axis, said body defining a bore coincident with the axis of symmetry of said contoured surface and said
10 bore being penetrated by said rod.

4. A compressor according to claim 3, further comprising means for preventing said body from rotating on said rod.

5. A compressor according to claim 4, wherein said
15 rotation preventing means comprises at least one pin attached to and movable with said body as said body moves slidably on said rod.

6. A compressor according to any one of claims 1 to 5, wherein said body has a seating surface on the same
20 side of said body as said contoured face.

7. A screw compressor assembly in a refrigeration



system comprising an oil-injected compressor section,
said compressor section including a screw rotor
housing defining a working chamber and a discharge
port in flow communication with said working chamber,
5 said compressor section further including a pair of
complementary screw rotors disposed for rotation in
said working chamber, said compressor discharge port
receiving a mixture of oil and gas compressed by the
rotation of said rotors in a predetermined direction
10 within said working chamber when said compressor
section is in operation and said rotors being free to
freewheel in a direction opposite said predetermined
direction when said motor is de-energized;

an oil separator section in flow communication with
15 the discharge port of said compressor section; and

means, disposed downstream of the discharge
port of said compressor section, for preventing
the rotation of said screw rotors in said direction
opposite said predetermined direction due to the
20 backflow of previously compressed gas from
downstream of said discharge port back to and
through said working chamber upon compressor
shutdown, said means for preventing the rotation of
said rotors upon compressor shutdown comprising
25 a body having a symmetrical concavely curved
contoured surface facing into the gas-oil



mixture discharged through said discharge port when said motor is energized, said body being positionable to said first position under the impetus of the gas-oil mixture discharged from said compressor when said motor is energized and positionable to said second position under the impetus of the initial backflow of gas from downstream of said discharge port which occurs back toward said discharge port when said motor is de-energized.

8. A compressor assembly according to claim 7, wherein said oil separator section includes a centrifugal oil separator element, said contoured surface of said body cooperating with said oil separator element so that the impact of said gas-oil mixture discharged by said compressor on said contoured surface of said body directs said mixture into said separator element in a manner facilitating the separation of oil from said mixture in said separator element.

9. A compressor assembly according to claim 8, wherein said centrifugal oil separator element defines an inlet and includes a plurality of swirl vanes juxtaposed said inlet and wherein in said first position said body seats in said separator element so



that said mixture discharged from said compressor section is imparted a radial velocity vector by said body prior to entering said swirl vanes.

10. A compressor assembly according to claim 7, 8 or 5 9, wherein said body defines a bore coaxial with the axis of symmetry of said contoured surface, said body being slidably disposed for movement between said first and said second positions on a rod which passes through said bore.

10 11. A compressor assembly according to claim 10, further comprising means for preventing said body from rotating around said rod.

12. A compressor assembly according to claim 11, wherein said body rotation preventing means comprises 15 at least one pin attached to and movable with said body as said body moves slidably on said rod, the movement of said at least one pin being constrained to movement parallel to said rod.

13. A compressor assembly according to claim 10, 11 20 or 12, further comprising a slide valve disposed for movement in said compressor section and a slide valve actuator piston disposed for movement in said oil



separator section, said valve and said piston being connected by said rod on which said body is disposed so that movement of said piston in said oil separator section results in the corresponding movement of said slide valve in said compressor section, the movement of said body on said rod being independent of the movement in said rod which occurs in response to the movement of said actuator piston.

14. A compressor assembly according to claim 13, wherein one face of said actuator piston is in flow communication with the interior of said separator element irrespective of whether said body is in said first or said second position whereby upon compressor shutdown pressure within said oil separator section and on said piston is maintained while system pressures equalize other than through the discharge port of said compressor.

15. A compressor assembly substantially as hereinbefore described with reference to the accompanying drawings.

