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HERMETIC COMPRESSOR WITH ROTARY ROLLING PISTON

The present invention relates to a hermetic compressor with rotary rolling piston and of high internal pressure, usually applied to small refrigerating machines and air conditioning equipments. A rotary rolling piston hermetic compressor of high internal pressure should be understood to be one the shell of which is submitted to the condensing pressure of the system in which it is used.

In rotary rolling piston hermetic compressors with high internal pressure in the shell, a phenomenon occurs of passage (penetration) of lubricating oil to the interior of the cylinder, into the suction and the discharge chambers.

The oil contained at the bottom of the shell and submitted to high gas pressures inside the shell, is elevated by oil pumping means till it reaches the crankshaft. From there the oil is carried along oil grooves to reach the rolling piston and is radially displaced through the gaps between the annular end faces of the rolling piston and the bearing covers, so as to enter the cylinder internal chambers.

This penetration of oil at a high temperature into the cylinder causes the compressor to function and perform as discussed hereinafter.

On penetrating into the suction chamber, the oil which is at a high temperature, warms up the incoming suction gas, causing its specific volume to increase, thereby reducing the filling up capacity of the suction chamber. Thus, the gas mass which fills the cylinder suction chamber is reduced by the effect of the increase in the gas specific volume. Besides this inconvenience, it should be noted that the oil volume itself which penetrates in the suction chamber takes gas filling space; however, this effect is of quite secondary importance in relation to the heating effect.

The above mentioned problem causes a decrease in the pumping capacity of the compressor as a result of the penetration of lubricating oil into the cylinder.

On penetrating into the cylinder compression chamber, the oil is, during a great part of the compression period, at a temperature higher than the gas temperature under compression. This also causes heating of such gas and increase in its specific volume. This phenomenon results in an increase in the work required for compressing the gas and, consequently, an increase in the energy consumption of the compressor. This fact can be verified in figure 2 of the accompanying drawings, where a diagram of pressure x angle of rotation is presented to demonstrate that the compression pressure rises faster when the oil leakage to the interior of the compression chamber increases.

These effects in combination, contribute to considerably reduce the volumetric and energetic efficiency of the compressor.

On the other hand, the presence of the lubricating oil flow carries out two favourable functions which are fundamental to the compressor functioning:-

- The first and the most obvious one is the lubrication of the movable parts involved.
- The second one is the sealing of all clearances between the movable parts thus avoiding the direct leakage of gas from the interior of the cylinder to the interior of the shell. Such leakage if it occurs, can be even more prejudicial to the compressor, in terms of capacity drop, than the gas over-heating by the oil.

This property of the oil to seal gaps between the movable parts, affects both the cylinder internal leakages (from the compressor chamber to the suction chamber at low pressure), and the leakages from the compression chamber to the interior side of the shell.

In the specific case of the oil which penetrates radially into the cylinder through the rolling piston end faces, the lubricating oil prevents leakage of gas from the compression chamber to the internal parts of the crankshaft and from there to the interior of the shell.

Therefore, the amount of oil which gets into the cylinder must be controlled at an optimum level i.e., at a minimum level in

order to allow gas leakages to be sealed and, at the same time, result in minimal gas heating in the interior of the cylinder.

A well-known way to control the amount of oil that penetrates into the cylinder, via the gaps of the the rolling piston end faces, is to reduce such gaps up to a minimum level at which the losses due to friction between the rolling piston end faces and the bearing cover faces do not reach such a value to completely cancel the gains produced by oil flow reduction through said gaps.

Despite the possibility of reducing the rolling piston end gaps as a way to get an advantageous reduction of the amount of lubricating oil which penetrates into the cylinder, the obtained gain, in terms of energy efficiency of the compressor, will always, be inferior to that which would be reached by just the reduction of the oil flow, because of greater friction losses as a result of reduction of the said gaps.

Thus, it is an object of the present invention provide a rotary rolling piston hermetic compressor which presents a lubricating oil flow to the interior of the cylinder, considerably reduced in relation to the known solutions, without provoking any considerable increase of friction between the compressor movable parts, more specifically between the rolling piston end faces and the bearing covers and, between the rolling piston and the eccentric portion of the camshaft.

The invention provides a hermetic compressor having a back pressure, comprising a hermetic shell that is submitted to a higher internal pressure than the back pressure of the compressor, a cylinder block which is housed inside the shell, having an internal cylindrical chamber; a crankshaft having an eccentric portion; a rolling piston which is assembled around the eccentric portion of the crankshaft, in order to rotate inside the cylindrical camber; end walls that close the opposite axial ends of the cylindrical chamber, said chamber being internally divided by the rolling piston into a suction chamber, whose internal pressure is substantially lower than the internal pressure of the shell, and into a compression chamber, presenting an internal pressure that is substantially lower than the internal pressure of the shell during

most of the compression cycle; and an axial gap for the passage of lubricant oil between an end wall of the cylindrical chamber and the opposite face of the rolling piston, characterized in that the crankshaft presents an end to the cylinder block, the end being defined by the eccentric portion, the axial end wall of the cylinder chamber, adjacent to said end of the crankshaft, being defined by a plate which is attached to the adjacent face of the cylinder block, whereas the opposite axial end wall of the cylindrical chamber is defined by a plate of a bearing which is fastened to the cylinder block (4), the rolling piston having the following dimensional relation:

$$1.63 \leq \frac{RR}{Rr} \leq 2.22 \quad (1)$$

where RR is the external diameter and Rr is the internal diameter to set the radial path of the lubricant oil through said axial gaps.

The rolling piston is built in order to present an external/internal diameter relation of about 1.63 up to about 2.22 in order to cause an increase in the radial path of the lubricating oil flow through the opposite axial gaps of the rolling piston and towards the suction and compression chambers, sufficient to reduce by at least 10% the oil flow inward the discharge and suction chambers without increasing, in a corresponding way, the friction losses between the movable parts.

For an equation that models the radial flow of oil through the rolling piston faces (viscous flow between parallel discs) it can be noted that the flow is controlled by the gap (δR) and by thickness of the rolling piston wall or, more precisely, by the relation:

$$\frac{1}{\ln \frac{\phi_{ext}}{\phi_{int}}}$$

The behavior of such function can be observed in the graph of fig. 3. The rolling piston dimensions found in known compressors present an external diameter (ϕ ext.)/internal diameter (ϕ int.) relation between 1.40 and 1.55, defining rolling pistons of thin walls.

The invention provides a rolling piston having a thick wall which have a ϕ ext./int. relation ≥ 1.63 , approximately.

It can be noticed by the graph of fig. 3 that, below about the value of ϕ ext./int. = 1.63 the slope of curve Ln^{-1} . $\phi_{\text{ext.}}/\phi_{\text{int.}}$ is quite accentuated, becoming softer gradually after the said value reaches 1.6.

It should be remembered that the behaviour of the curve that represents the oil flow toward the interior of the cylinder is proportionally reflected in the compressor performance as already explained, i.e., the higher the function $\text{Ln}^{-1} (\phi \text{ ext.}/\phi \text{ int.})$, the greater the oil flow, and the worse is the volumetric and energetic

efficiency of the compressor. Therefore, it is important to notice that the penetration of oil inward the cylinder can be reduced by at least 10% with a dimensioning of the rolling piston diameters in a way to get a relation (ϕ ext./ ϕ int.) above (1.63) as compared to the range commonly used; the rolling piston being dimensioned in order to get a relation ϕ ext./ ϕ int. ≥ 1.63 approximately. It is also important to mention that diameters relation of 1.63 up to nearly 2.22 are perfectly feasible in the production processes normally used for rotary rolling piston compressors and yet, there is no impediment or disadvantage in terms of the compressor performance when using such relations.

The only disadvantage that one could quote would be the increase of friction losses between the piston faces and the cylindrical chamber end walls due to the increase of the contact surface. However, the increase of the friction losses does not effectively occur because with the increase of the contact surface it tends to have a reduction of the angular velocity of the piston over its own axle which compensates the loss. Besides, such loss by friction is of the order of magnitude at least one time smaller than the losses caused by the heated oil. Therefore, such loss could be neglected anyway.

There are still other favourable indirect consequences in terms of the compressor energetic performance when the greater relation between the piston external and internal diameters, through the reduction of the internal one, is used. Said advantages are the following:

- With the reduction of the piston internal diameter, obviously the diameter of the shaft eccentric portion is also reduced, thereby reducing the important loss of viscous drag between the piston contact surfaces and the eccentric shaft.

- Usually, when the diameter of the shaft eccentric portion is reduced, the diameter of the edge of the minor shaft can also be reduced and, depending on the circumstances, even the sub-bearing can be eliminated, adding a support or axial bearing (see figures of preferential configurations). This solution reduces the friction losses in such a bearing.

- With the reduction of the eccentric diameter and the, consequent reduction of the contact surface between the eccentric and the piston, the piston angular velocity on its own shaft becomes smaller, which considerably reduces one of the compressor's most important friction losses, that is the friction loss between the top of the vane and the piston external diameter.

Therefore, it is possible to conclude that it is more advantageous to increase the relation between the roller diameters by means of a decrease of the internal diameter than through an increase of the external one.

In the accompanying drawings:

Figure 1A represents a partial longitudinal sectional view of a rotary rolling piston compressor of the type used in the present invention;

Figure 1B represents a cross-sectional view taken according to line B-B of figure 1A;

Figure 2 represents a diagram of the compression pressure produced in terms of the rotation angle of the rolling piston;

Figure 3 illustrates a representative graph of the function of oil radial flow through the rolling piston faces X the thickness of the piston angular wall;

Figure 4A illustrates a side view of the crankshaft rolling piston set of the prior art;

Figure 4B illustrates a side view of the crankshaft rolling piston set built according to some features of the invention;

Figure 4C represents a similar view to the figure 4B but illustrating said set according to another configuration of the invention; and

Figure 5 represents an enlarged view of part of the set of figure 4C when disposed inside the bearing and the cylinder block.

According to figures 1A and 1B, the preferred compressor includes a shell 1, fastening suction 2 and discharge 3 tubes, and houses a cylinder block 4 in the interior of which a cylindrical chamber houses a rolling piston 5 which is mounted on a crankshaft 6 driven by an electric motor composed by a stator 7 and a rotor 8. This compressor is of high internal pressure in the shell, to which

interior an inlet end of the discharge tube 3 is opened.

The crankshaft 6 is supported on a main bearing 10 and a sub bearing 20, each having a plate or flange 10a and 20a fixed against one of the axial end faces of the cylinder block 4, in such a way as to define the cylindrical chamber walls in the interior of which the rolling piston 5 is displaced.

In the illustrated example, a discharge muffler chamber 13 is provided next to the sub bearing 20 external face to receive the compressed gas inside the cylinder. The sub bearing plate 20a (or the cylinder block wall 4 in case of absence of such bearing) is provided with a discharge orifice 22 with its outlet end defining an annular valve seat against which is seated a known reed valve 13 is internal to the discharge muffler chamber 30.

Still according to the basic construction illustrated in figures 1A and 1B, the main bearing plate 10a is provided with a radial channel 11 with its inferior end immersed in the lubricating oil OL stored at the bottom of the shell 1 and, with its superior end opened to an oil pump or another pumping device, defined around the crankshaft 6 and in fluid communication with longitudinal superficial grooves 14 provided along the crankshaft 6, in order to conduct the lubricating oil to the bearing and to the end axial gaps of the rolling piston 5.

As illustrated in figure 1B, the cylinder block 14 presents windows 4a for internal pressure balance in the shell and for lubricating oil passage and also, a slot in which a sliding vane 9 is incased which, together with the external cylindrical face of the rolling piston 5, divides the cylindrical chamber into a suction chamber and a discharge chamber, being the first one fed through a channel 4b made on the cylinder block and maintained in communication with the internal end of the suction tube 2.

Figure 4A illustrates a conventional construction of a crankshaft 6 including an eccentric portion 6a, of the rolling piston 5 drive and, an end portion 6b for journalling inside the sub bearing 20; usually both these crankshaft portions have the same diameter as the rest of the crankshaft body. In this prior art, which corresponds to the arrangement illustrated in figure 1B,

the rolling piston 5 presents an annular wall thickness that leads to a relation of $\phi \text{ ext.} / \phi \text{ int.} < 1.60$, making the oil radial path, represented by the arrows in figure 1B, short enough to allow a prejudicial amount of lubricating oil to penetrate inside the cylinder.

Figure 4B illustrates a crankshaft 60 with its eccentric portion 60a driving a rolling piston 50 with annular wall thickness according to the relation $\phi \text{ ext.} / \phi \text{ int.} > 1.63$. This relation including the limits defined in the present invention was reached by decreasing the internal diameter of the rolling piston 50 and, consequently, the diameter of the eccentric portion 60a of the crankshaft 60. With such a diametrical reduction of the eccentric portion 60a of the crankshaft 60, its end portion 60b can also have a reduced diameter, compared to the rest of the shaft body, such as it is illustrated in figure 4B.

The additional advantages resultant from the above mentioned diametrical reductions of the crankshaft 60 portions 60a and 60b are mentioned in the initial part of the present disclosure.

Figure 4C illustrates a preferred construction for the crankshaft 600, which as a result of the large thickness of the rolling piston wall 500, presents an eccentric portion 600a of the very reduced diameter, the end portion being also completely eliminated and, consequently, the sub bearing 20 as well. In this case, the corresponding axial end wall of the cylinder chamber can be defined by a simple closing plate fastened to the adjacent face of the cylindrical block.

In the case of the constructive solution illustrated in figure 4C, an axial stop 601 in the shape of an annular flange incorporated to the crankshaft 600 body is provided; it is placed next to its eccentric portion 600a in a way to be housed in a respective recess 10b made on the end face of the main bearing 10 plate 10a, in order to limit the crankshaft 600 axial displacement towards the motor, caused by its rotor magnetic strength actuating upon said shaft. The recess 10b works as axial bearing to the stop 601, allowing due to its deepness, a big gap to be formed between the axial stop 601 and the adjacent end wall of the rolling piston.

CLAIMS

1. Hermetic compressor having a back pressure, comprising a hermetic shell that is submitted to a higher internal pressure than the back pressure of the compressor; a cylinder block which is housed inside the shell, having an internal cylindrical chamber; a crankshaft having an eccentric portion; a rolling piston which is assembled around the eccentric portion of the crankshaft, in order to rotate inside the cylindrical chamber; end walls that close the opposite axial ends of the cylindrical chamber, said chamber being internally divided by the rolling piston into a suction chamber, whose internal pressure is substantially lower than the internal pressure of the shell, and into a compression chamber, presenting an internal pressure that is substantially lower than the internal pressure of the shell during most of the compression cycle; and an axial gap for the passage of lubricant oil between an end wall of the cylindrical chamber and the opposite face of the rolling piston, characterized in that the crankshaft (600) presents an end to the cylinder block (4) the end being defined by the eccentric portion (600a), the axial end wall of the cylindrical chamber, adjacent to said end of the crankshaft (600), being defined by a plate which is attached to the adjacent face of the cylinder block (4), whereas the opposite axial end wall of the cylindrical chamber is defined by a plate (10a) of a bearing (10) which is fastened to the cylinder block (4), the rolling piston having the following dimensional relation:

$$1.63 \leq \frac{RR}{Rr} \leq 2.22 \quad (1)$$

where RR is the external diameter and Rr is the internal diameter to set the radial path of the lubricant oil through said axial gaps.

2. Hermetic compressor, according to claim 1, wherein the crankshaft (600) incorporates, next to its eccentric portion



(600a), a peripheral annular flange (601) which is housed in a recess (10b) provided on the end face of the plate (10a) of the bearing (10).

3. A hermetic compressor comprising a rotary rolling piston of annular section enclosing and rotatable around an eccentric portion of a crankshaft;

the piston and eccentric portion being held within a cylindrical chamber of a cylinder block;

the piston and an externally abutting sliding vane cooperating to separate the cylindrical chamber into suction and discharge regions;

the cylindrical chamber having at least one end separated from a neighbouring annular end-face of the piston by a gap through which in operation lubricating oil flows, the piston having a ratio of its external to its internal diameter of at least 1.63, the crankshaft comprising a body portion which is rotatably mounted in a bearing, the bearing comprising a plate which faces the piston and through which the crankshaft passes.

4. A hermetic compressor according to claim 3, in which the crankshaft further comprises an annular flange adjacent to the eccentric portion, and there is a recess for the flange in the plate of the bearing to limit axial movement of the crankshaft.

