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Novel Design Technique for Extending Operating Range of Axially Compliant Scrolls

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ABSTRACT

A scroll compressor displacement is often limited by the maximum attainable height of a scroll compressor wrap. This occurs, as the wrap height cannot exceed a certain threshold value without compromising axial stability. Axial stability also limits the scroll compressor operating envelope. Loss of axial stability leads to degradation of compressor performance due to internal leakage. New design technique improves axial stability by maximizing pressure in a back chamber that is located behind the axially compliant scroll element. In the conventional arrangement, the back chamber pressure is controlled by a vent hole machined through an orbiting or fixed scroll floor, while the new technique relies on the vent hole machined through the scroll wrap. Additional indentations are placed on the floor of the opposing scroll to time the opening and closing of the vent hole to maximize the back chamber pressure. Design features, including photographs of scroll wraps using the standard and new vent hole arrangement are shown. Analysis to estimate the benefits of applying the new design technique is also presented.

NOMENCLATURE

\[ A_{bc} \] – back chamber area
\[ F_{tg} \] – tangential gas force
\[ F_{ax} \] – axial gas force
\[ F_{bc} \] – back chamber force
\[ L_{ov} \] – overturning distance
\[ H \] – wrap height
\[ \theta \] – crank angle
\[ B \] – distance from bearing to scroll floor
\[ P_s \] – suction pressure
\[ P_{bc} \] – back chamber pressure
\[ P_d \] – discharge pressure
\[ P_{int} \] – intermediate pocket pressure
\[ R_{ov} \] – distance from center to contact point

BACKGROUND

A scroll compressor compression element consists of two intermeshing scroll components with one orbiting in relation to the other. The orbiting scroll motion creates compression pockets. Gas compression takes place as the volume of the pockets reduces toward the discharge. Most of the commercially available scroll compressors rely on axial compliance mechanism to provide sealing between the fixed and orbiting scroll. The axial compliance is achieved by allowing a slight axial movement of one of the intermeshing scroll with respect to the other scroll. Gas forces in the compression pockets act to separate the scrolls in axial direction. As long as these forces are kept below certain value the scrolls do not separate. Figure 1 shows a schematic of the gas forces acting on the scroll compressor pump-set components, when the orbiting scroll already separated from the fixed scroll. The magnitude of the axial separation distance is greatly exaggerated in this Figure as the actual separation is typically under 0.1 mm. As shown, the pump-set assembly consists of the axially compliant orbiting scroll, fixed scroll and supporting crankcase structure.
As per common convention, $F_{tg}(\theta)$ is the tangential overturning force acting on the orbiting scroll wraps; $F_{ax}(\theta)$ is the axial gas force acting over the floor and tips of the orbiting scroll. As described by Bush, et al. [1], the tangential gas force $F_{tg}$ acts at the mid-point of the scroll wrap and is parallel to the plane of the scroll motion. This force creates an overturning (tipping) moment that separates the scrolls axially. The axial gas force $F_{ax}$ acts in the direction perpendicular to the plane of the scroll motion and also separates the scrolls axially. $F_{bc}$ is so called back chamber force that counteracts these two forces and acts in the direction opposite to $F_{ax}(\theta)$. The back chamber force is a “gauge” force that is found by multiplying the gage pressure (difference between the back chamber pressure and suction pressure) by the back chamber area $A_{bc}$:

$$F_{bc} = (P_{bc} - P_s)A_{bc}$$  \hspace{1cm} (1)

The vent hole is machined through one of the intermeshing scrolls, connecting the back chamber to a compression pocket. One end of the hole is exposed to pressure in the scroll compression pocket and the end of the vent is exposed to the back chamber. The two circular seals installed in the crankcase isolate the back chamber from the vapor surrounding the pump-set. The vapor pressure above the vent hole alternates as the hole passes from the intermediate compression pocket to the discharge pocket. Pressure in the back chamber is controlled by pressure the vent “sees” when it is being exposed to the alternating pressure in these two compression pockets. When pressure above the vent hole is higher than the pressure in the back chamber, then vapor is injected into the compression chamber. When pressure above the vent hole is below the back chamber pressure then the vapor is ejected out of back chamber into the compression pocket. When the opposing scroll covers the vent hole, the communication between the vent hole and the compression pocket ceases and vapor does not leave or enter the back chamber. On average cycle basis, neglecting the leakage through the back chamber seals, the amount of vapor entering the back chamber is the same as leaving the back chamber (otherwise, there would be continuous refrigerant build up or rarefaction in the back chamber). Back chamber pressure must be sufficiently high to counteract the opposing forces to avoid the scroll separation.

**STANDARD VENT HOLE ARRANGEMENT**

In the standard arrangement, the vent is located on the floor of the axially compliant scroll element. Depending on a scroll compressor design, either a fixed or orbiting scroll is axially compliant. This paper deals with the latter, that is Carrier’s design; though the derivations presented below can be similarly applied to the axially compliant fixed scroll. As shown schematically in Figure 1, the back chamber is positioned behind the axially compliant orbiting scroll element. Figure 2 is a photograph of the axially compliant orbiting scroll with the vent hole located on the scroll floor.

As the vent hole is located on the surface of the orbiting scroll floor, the vent hole follows the orbiting motion of the orbiting scroll in relation to the fixed scroll. During its orbit the vent hole spends part of the time being exposed to the discharge pressure, part of the time being exposed to the intermediate pressure, and the tips of the fixed scroll cover the vent hole for the remaining portion of the time. Figure 3 shows the vent hole exposure within its 360-degree ($2\pi$) orbit to varying pressure in compression pockets in relation to the orbiting scroll crank angle. As can be seen from this Figure, the average 225 PSIA pressure to which the vent hole is exposed while passing through the intermediate pressure pocket is well below the maximum pressure in the intermediate pocket.

**BACK CHAMBER PRESSURE AND SCROLL WOBBLE**

further accounting for wrap thickness, compressibility effects and coupling of the back chamber pressure to scroll wobble. In our analysis we will keep the simplicity of approach by Bush, et al. [1], while also considering the effect of wrap thickness as described in [3]. The back chamber pressure $P_{bc}$ is then calculated based on the time weighted average the vent hole spends in the intermediate and discharge compression pocket:

$$P_{bc} = \frac{(P_{\text{int-average}} \Delta \theta_{\text{int}} + P_{\text{dis-average}} \Delta \theta_{\text{dis}})(\Delta \theta_{\text{int}} + \Delta \theta_{\text{dis}})}{\Delta \theta_{\text{int}} + \Delta \theta_{\text{dis}}}$$  (2)

$P_{\text{int-average}}$ is an integrated average pressure seen by the vent hole while it is open to the intermediate pocket and $P_{\text{dis-average}}$ is an integrated average discharge pressure seen by the vent hole while it is open to the discharge compression pocket. Where:

$$P_{\text{int-average}} = \int_{\theta_{\text{int,1}}}^{\theta_{\text{int,2}}} P_{\text{int}}(\theta) \, d\theta / \Delta \theta_{\text{int}}$$  (3)

$$P_{\text{dis-average}} = \int_{\theta_{\text{dis,1}}}^{\theta_{\text{dis,2}}} P_{\text{dis}}(\theta) \, d\theta / \Delta \theta_{\text{dis}}$$  (4)

The crank angle interval the vent hole is uncovered in the intermediate and discharge pocket is calculated as:

$$\Delta \theta_{\text{int}} = \theta_{\text{int,2}} - \theta_{\text{int,1}}$$  (5)

$$\Delta \theta_{\text{dis}} = \theta_{\text{dis,2}} - \theta_{\text{dis,1}}$$  (6)

The time the vent hole is covered by the opposing scroll is ignored in calculating the back chamber pressure (this is a valid assumption as essentially no vapor enters or exits the vent hole during the time period when it is covered by the opposing scroll).

If the scroll wobble becomes excessive (scrolls separate from each other), the scroll compressor performance deteriorates, as a leakage path develops between the floor and tips of the fixed and orbiting scroll, Lifson [3]. As covered in detail by Bush, et al [1] the orbiting scroll is on the verge of separating from the fixed scroll (start of wobbling) when:

$$R_{ov} = F_{tg}(\theta) \cdot L_{ov} / (F_{bc} - F_{ax}(\theta))$$  (7)

As shown in Figure 1, $R_{ov}$ is a distance from the center to the last point on the outer periphery of the orbiting scroll that is in contact with the fixed scroll and $L_{ov}$ is an overturning arm equal to the distance from the center of the scroll wrap to the center of the orbiting scroll bearing. $L_{ov}$ is calculated as the sum of the wrap height $H$ and the distance $B$ from the center of the orbiting scroll bearing to the orbiting scroll floor:

$$L_{ov} = H + B$$  (8)

For a given scroll wrap geometry and operating condition, $F_{tg}(\theta)$, $F_{ax}(\theta)$, $L_{ov}$, and $A_{bc}$ are fixed. Therefore, the only variable left to enhance the scroll axial stability is the back chamber pressure. The back chamber pressure $P_{bc}$ CAN be increased if the average pressure $P_{\text{int-average}}$ of equation (3) can be brought near the maximum pressure in the intermediate compression pocket. As shown in Figure 3, pressure in the intermediate pocket reaches maximum just before porting.
EFFECT OF SCROLL WRAP HEIGHT ON AXIAL STABILITY

Increase in the fixed and orbiting scroll wrap height is often the most desirable way to extend scroll compressor displacement, as it results in the least disruption to manufacturing processes and minimizes the compressor cost as compared to other options such as for example increase in the scroll compressor outer shell diameter. However, as the scroll wrap height increases it becomes more difficult to assure axial stability at extremes (high and low) of scroll compressor pressure ratio operating range. Bush, et al [1]. As shown by equation (7), the overturning moment $F_{tg}(\theta)L_{ov}$ is a controlling factor in determining the scroll stability. The overturning moment increases rapidly with the growth in the scroll wrap height. This occurs as BOTH $F_{tg}(\theta)$ and $L_{ov}$ increase for taller scrolls, with $F_{tg}(\theta)$ being directly proportional to the scroll wrap height and $L_{ov}$ as defined by equation (8). On the other hand, both $F_{bc}$ and $F_{ax}(\theta)$ forces of equation (7) are being independent of the scroll height. Thus, as the scroll height increases, the back chamber force must increase accordingly to maintain the scroll axial stability.

Enlarging the surface area of the back chamber and/or elevating the vapor pressure in the back chamber increases the back chamber force. The increase in the back chamber area is straightforward and is normally accomplished by extending the back chamber area to its maximum value allowed by the geometrical constraints of the scroll compressor (where the main geometrical constraint is the diameter of the scroll compressor outer shell). After the area of the back chamber is increased to the maximum, then the scroll compressor designer needs to adjust the back chamber pressure to maintain adequate axial stability. The increase in the back chamber pressure is more subtle than the increase in the back chamber area, as the back chamber pressure is affected by three interdependent factors: pressure in the intermediate and discharge compression pocket, amount of time the vent hole is uncovered in each pocket, and crank angle position of the vent hole in each pocket. To maximize the scroll axial stability at high pressure ratio condition it is necessary for the vent hole to spend more time being exposed to the discharge pressure than to the intermediate pressure, as at this condition the discharge pressure is always higher than the intermediate pressure – this is the so called under-compression condition. However, to maximize the scroll axial stability at low pressure ratio condition it is necessary for the vent hole to spend more time being exposed to the intermediate pressure than to the discharge pressure, as at this condition the maximum intermediate pressure inside the scroll compression pockets is normally higher than the discharge pressure – this is the so called over-compression condition. A. Lifson and J. Bush [4] present a detailed discussion on this topic.

Thus, to assure the scroll stability at extremes of low and high pressure ratio operation, the position of the vent hole is selected to “pick-up” vapor in strategic locations in both the discharge and intermediate compression pocket. However, as scroll height increases and the back chamber area cannot be increased any further, we reach a point where no matter what the percentage of time the vent hole spends being exposed to the discharge pressure vs. intermediate pressure, the scrolls can not be made stable at the extremes of its operating range.

NEW VENT HOLE ARRANGEMENT

A novel vent hole arrangement as described in a US patent by A. Lifson and J. Bush [5] achieves the goal of substantially extending the scroll compressor operating range or increasing scroll compressor displacement for a given operating envelope. In the new arrangement, one end of the vent hole originates on the tip of the orbiting scroll wrap, rather than on the base (floor) as it is the case in the standard vent hole arrangement. As machining of the vent hole through thin wraps is problematic, the application of this new technique became possible and is also most advantageous for the case of thick scroll wraps, representative of refrigeration applications requiring operation over a broad range of pressure ratio conditions. Figure 4 is a photograph of an orbiting scroll designed for the refrigeration application, showing the vent hole being machined through the wrap tip and into the base of the orbiting scroll.

Two corresponding shallow grooves are machined on the floor of the fixed scroll. Figure 5 is a photograph of the fixed scroll of the same compressor showing the locations of these two grooves. The
vent hole crosses over each of these grooves during its orbit. When the vent hole does not cross over the
grooves, it is covered by the fixed scroll floor. However, when the vent hole crosses over the first groove it
is open to the intermediate pocket pressure and when it crosses over the second groove it is open to the
discharge pocket pressure. The cross over area of each groove defines the amount of time the vent hole is
exposed to corresponding vapor pressure in each pocket. The crank angle position of the vent hole when it
crosses over the groove defines where in the intermediate compression process (first groove intersection) or
where in the discharge compression process (second groove intersection) the grooves and vent hole
communicate. As pressure in the discharge process is fairly uniform (except for a short period of time
following porting), see Figure 6, the crank angle position when the vent hole is exposed to the discharge
pressure is not critical. Though the duration of the vent hole exposure to the discharge pressure is, of
course, important. However, the crank angle position when the vent hole is exposed to the intermediate
pressure is very critical as pressure in the intermediate pocket changes substantially (often by a factor of
four or more from start to end of compression in this pocket). Thus, to maximize the pressure in the back
chamber, we position the first groove so that the vent hole intersects it as close to the end of the
intermediate compression process as possible; this is a point in the compression cycle where the
intermediate pressure is at its maximum. This is illustrated in Figure 6, where \( p_{int\_average} \) equals 275 PSIA.

Note that in a standard vent hole arrangement we do not have a flexibility of finding an appropriate
vent hole position on the orbiting scroll floor, so that the vent hole will be exposed to the maximum or near
maximum intermediate compression pocket pressure while still spending sufficient time in the discharge
pocket to assure stability at high pressure ratio operation. For a typical scroll wrap geometry a new vent
hole arrangement would allow for a scroll wrap height increase of approximately 20% as compared to the
standard vent hole arrangement. If there is a sufficient margin in other scroll compressor design constraints
such as bearing load, wrap thrust load or motor strength, this height increase translates into ability to
increase the maximum scroll displacement by the same 20% without having a noticeable effect on the
scroll compressor production cost.

**CONCLUSION**

A patented new vent hole arrangement has been introduced into a design of commercially available
scroll compressor. This vent hole arrangement enhances the scroll axial stability, increasing the maximum
achievable displacement without the need to employ larger diameter scroll compressor; thus avoiding
substantial cost penalty. Conversely a compressor of the same displacement can operate at substantially
broader operating range if the new venting technique is utilized. This new technique is especially
applicable to scroll compressors with thick wraps typical of refrigeration application.

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Figure 1 Schematic of Forces Acting on Axially Compliant Orbiting Scroll

Figure 2 Photograph of Orbiting Scroll with Standard Vent Hole Arrangement
Figure 3  Vent Hole Exposure to Pressure in Compression Pockets for Old Arrangement

Figure 4  Photograph of Orbiting Scroll with New Vent Hole Arrangement
Figure 5  Photograph of Fixed Scroll with Grooves Located on Scroll Floor

Intermediate Pressure Groove

Discharge Pressure Groove

Figure 6  Vent Hole Exposure to Pressure in Compression Pockets for New Arrangement