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1. ABSTRACT

The presence of leakage triangles (blow-holes) in screw compressors is an inevitable consequence of the rotor profile geometrics. Leakage, from one closed cavity to another through the leakage triangle, affects performance by raising specific power and discharge gas temperature. Its effect on compressor capacity is usually minimal. While the knowledge of leakage triangle area is critical in performance prediction with the use of computer models, methods to calculate such area are lacking in the published literature. This paper presents three methods, of varying complexity and accuracy, to compute leakage triangle area from rotor and housing geometry. Also presented is the relative contribution of blow-hole leakage to the overall performance loss for a selected compressor. These results indicate that the loss increases rapidly at lower compressor speeds. The leakage triangle area can be reduced to near-zero by adjusting the female rotor addendum and by properly shaping both male and female rotor profiles around the pitch circle diameter. However, these changes affect torque transfer from one rotor to the other and care should be exercised that the reduced blow-hole design does not result in torque reversals and unstable operation.

2. INTRODUCTION

Screw compressor performance to a great extent, depends on the geometry of the rotor profile and on various operating clearances. Singh and Patel [1] and Singh and Bowman {2} have quantified the influence of these variables on compressor performance and have shown that the effect is quite significant. A major share of the performance losses can be attributed to various leakage paths within the compressor (mechanical and viscous losses are the other culprits). Leakage losses are directly proportional to the effective leakage areas.

Screw compressors, because of the complex, helical shape of their rotors, have many leakage paths which influence performance in different ways. For example, leakage through interlobe (rotor-to-rotor) clearance takes place from the cavity in compression directly to the compressor inlet. The result is both substantial power and capacity loss. On the other hand, leakage through the blow-hole or rotor-tip clearance takes place from one cavity to the adjacent cavity and the resulting pressure drop, as in a labyrinth seal, is gradual. While such leakage has very little influence on machine capacity, continuous recompression of the leaked flow does cause power loss and increase in the discharge gas temperature.

From this discussion, it becomes apparent that reliable performance prediction requires a good knowledge of leakage losses and therefore, leakage areas. Of all the possible leakage areas, the calculation of the blow-hole area poses the greatest challenge. In fact, most designers use approximate and often crude techniques to estimate this area. Despite many publications on the performance prediction methods {1-4}, computational techniques to determine blow-hole area are singularly lacking. This paper presents three such techniques of varying complexity and accuracy. Also presented is a discussion of the blow-hole's contribution to loss in performance.

We may point out that blow-hole is not unique to twin-screw compressors discussed here. A single-screw compressor with one main rotor and two gate rotors also has similar leakage areas.

3. DEFINITION

Blow-hole area can be defined as the smallest triangular area bounded by the housing and the two rotors, in a plane normal to the primary leakage flow path between two adjacent cavities. The so-called leakage triangle does not lie in a cartesian plane. It can only properly be defined in curvilinear coordinates. Figure 1 shows the leakage triangle in a typical screw compressor. The leakage triangle can actually be visualized in a simpler way. Figure 2 shows a cross-section of the housing and the two rotors in a plane normal to the rotors' axes. In the Figure, one of the male rotor lobes is contacting the bottom housing cusp and the mating rotor, thus creating a leakage path from cavity I to cavity II. Figure 3 is a reproduction of Figure 2 except the rotors have been rotated to a position such that one of the male rotor lobes is contacting the top housing cusp. The tip of the matching female rotor lobe now is quite a bit away from the cusp, implying a large blow hole. However, nearby cavities, III and IV, are at nearly identical suction pressure, and negligible leakage takes place from one cavity to the other. Thus, we have leakage triangles in both the suction zone and the compression zone of the compressor but the former are negligible in their effect on performance.

Another method to view and qualitatively estimate the size of the leakage triangle is to look at the end view of the contact line as shown in Figure 4. The figure shows that the bottom end of the projected contact line does not meet the bottom housing cusp. Farther apart these two points, larger the leakage triangle area. Again, the corresponding two points on the top of the Figure represent the suction leakage triangle.

Before we proceed to the calculation of leakage triangle areas, one more observation should be of interest. All leakage areas except the leakage triangle depend on both the rotor profile shape (which determines the length of the leakage line) and on the operating clearance (a function of manufacturing tolerances and thermal growth). The leakage triangle are on the other hand is determined by the rotor profile shape, and other factors such as tighter manufacturing tolerances or thermal expansion have only minor influence.

4. BLOW-HOLE AREA CALCULATION

4.1 Exact Method

By exact method, we imply the use of a completely analytical representation, described in {5}, to calculate the leakage triangle area. In this section, only a framework for the calculation procedure is provided so as not to confuse the basic logic with mathematical details.

In this method, the blow-hole area is treated as a curvilinear triangle, with two vertices as the points where adjacent male and female rotor crest lines meet the rotor housing cusp respectively, and the third vertex as one of the rotor-to-rotor contact points. In the computer program written to calculate this area, the third vertex is selected out of the contact points array such that its use leads to the minimum area compared to all other feasible points. This procedure ensures correct selection for all possible profile shapes. In Figure 5, A and B are two housing cusp vertices, and C is the contact point vertex of the blow-hole triangle. Lines BC and AC follow the contact paths as the rotors are rotated and do not lie in the same plane. Therefore, area ABC is calculated by,

$$S_{bh} = \left|\vec{S}_{bh}\right| = \left|1/2(\vec{I}_{AB} + \vec{I}_{BC} + \vec{I}_{CA})\right|$$

$$\text{where } I_{AB} = \int_A^B \vec{r} \times \vec{dr} \text{ , etc}$$

and $\vec{r}$ is a position vector with respect to a fixed coordinate system. The evaluation of these integrals is briefly described in {5}.

The actual blow-hole area of interest for leakage flow calculations is an area normal to the flow direction. From visualization of the compressor geometry, we infer that most likely flow direction is along the helical surface of female rotor in the blow-hole vicinity. Thus, the desired projection of the blow-hole area, normal to the nominal direction of the leakage flow, is given by,

$$S_T = \vec{S}_{bh} \cdot \vec{T}$$

Where $\vec{T}$ is a unit vector tangent to the female rotor helix near the cusp.

4.2 Approximate Methods

METHOD 1

The simplest approximation is obtained by determining the area indicated in Figure 5. This end plane area is then divided by the cosine of the rotor helix angle at the pitch diameter. In Figure 6 the rotors are positioned such that any backward rotation will result in separation of the trailing flanks. The end-plane area can be found by numerically integrating Greens' Theorem in the plane.

$$A = 1/2 \oint (XdY-YdX)$$

where X and Y are coordinates defining the perimeter of the area of interest. Of course this is a very crude approximation and should only be used when precise results are not required.

METHOD 2

A more accurate approximation can be obtained by examining the manner in which the leakage triangle is formed. One tip of the leakage triangle lies at the point where the trailing flanks of the rotor are on the verge of separating (point 1 in figure 7). The Z coordinate for this point is obtained by multiplying the lead per degree by the angle of rotation of the rotor (Figure 8). A second point considered to be on the leakage triangle is defined by the point at which the radius to point H aligns with the lower cusp.

At this point (point 2) the leakage path between the rotor cells is effectively cut off. A third point can be located by finding the point on the helical line traced out by the female rotor pitch circle which is closest to point 2. These three non-collinear points in space define a plane. We take the intersection of this plane with the rotors and housing bore as the leakage triangle. The equation of the plane can be obtained by expanding the determinate

$$\begin{vmatrix} (X-X_1) & (Y-Y_1) & (Z-Z_1) \\ (X-X_2) & (Y-Y_2) & (Z-Z_2) \\ (X-X_3) & (Y-Y_3) & (Z-Z_3) \end{vmatrix} = 0$$

where X_i, Y_i, Z_i etc. are the coordinates of point i, i= 1 to 3. The expansion of the determinate can be written as

$$\alpha X + \beta Y + \gamma Z = C$$

where the unit surface normal is

$$\vec{N} = \alpha\vec{i} + \beta\vec{j} + \gamma\vec{k}$$

The area can be obtained by numerically integrating the line integral

$$A = 1/2 \oint (\beta Z - \gamma Y)dX + (\gamma X - \alpha Z)dY + (\alpha Y - \beta X)dZ$$

Where X, Y, Z are the coordinates of the curve obtained from the intersection of the rotors and housing bore with the plane.

4.3 Method Verification

While it is difficult to calculate the leakage triangle area exactly, experimental measurements are not much easier. Leakage triangle area of an SRM-A profile compressor was measured by inserting "Silly Putty" through the blow-hole and shaping it to conform with the leakage triangle configuration. The putty was then sliced in a plane normal to the assumed flow direction and the crosssectional area traced and measured. This process was repeated three times and an average was calculated. Measured values were consistent with the computed values (exact method) within the estimated measurement accuracy range of +4%.

Approximate method 1 has been compared to the exact calculation. It gives results which are within 8% of the exact method. Approximate method 2 gives results that are within 1.5-2% of the exact method.

5. EFFECT ON PERFORMANCE

Figure 9 shows the effect of change in blow-hole area on compressor performance for a small equal-diameter rotor compressor. The performance calculations are derived from the use of the computer model described in {1}. The calculations were run by arbitrarily changing the blow-hole area area in the program input. In practice, this kind of freedom is not possible since any attempt to change blow-hole area must also influence other design parameters such as seal line lengths, and inlet and discharge port areas. Thus, the data presented here should only be used as a measure of relative contribution of the blow-hole to capacity and power losses. Since contact line leakage length is likely to increase with the reduction in blow-hole area, these losses can be thought of as an upper limit.

Data plotted in Figure 9 shows that VE (volumetric efficiency) loss and % specific power (BHP/100CFM) increase relative to the zero blow-hole area case. Therefore blow-hole losses for a given area can be directly read from these plots. The standard blow-hole area for this compressor is .0176 sq. in . These cases were run for atmospheric inlet and 105 psig discharge pressure. From the results, it is evident that blow-hole's contribution to losses decreases significantly with speed but still remains relatively large. Also, % capacity losses are much smaller than % power losses.

Figure 10 shows similar results for a large compressor with same profile as above (nominal blow-hole area = .114 sq. in.). These results indicate that blow-hole's effect on performance does not diminish with size. Thus a balanced profile design with a small blow-hole area can be applied across a wide range of machine sizes.

6. EFFECT OF ROTOR PROFILE GEOMETRY

Leakage triangle size is essentially defined by the rotor profiles, but other geometrical parameters such as wrap angle, number of lobes, etc., also have an influence. The triangle area is influenced by the size of the female rotor addendum, (rotor radius - pitch circle radius). A larger addendum, generally creates a larger blow-hole. A zero addendum can eliminate the leakage triangle completely if a point-generated surface is used. For a given addendum, leakage triangle size can be varied by shaping the profile in different ways.

If a leakage triangle is a source of controllable performance loss, why aren't all profiles designed to eliminate blow-hole completely? Screw compressor rotor profiles have evolved from symmetric profiles with very large blow-hole areas to asymmetric profiles with small blow-hole areas. The switch from symmetric to asymmetric (e.g., SRM-A) profiles showed a gain of 8 to 10% in performance despite an increase in rotor-to-rotor contact leakage area. Further gains have not been so dramatic, but in optimizing compressor performance blow-hole losses can't be ignored.

A zero blow-hole profile may not necessarily be ideal in practice. Generally, only a very small fraction (5 to 10%) of total torque is transferred by the male (drive) rotor to the female (driven) rotor.

7. CONCLUSIONS

Three techniques, of varying accuracy and complexity, to calculate blow-hole area in twin-screw compressors have been presented. Accurate knowledge of blow-hole area is important in computing compressor performance, particularly at low speeds. This paper brings out the following significant points:

o Blow-hole area is essentially defined by the rotor profile geometry. Therefore, careful attention should be paid to computation of blow-hole area during the profile generation process. One visual check would be the distance between housing cusp and the closest point on the projection of rotor-to-rotor contact line on a plane normal to the rotor axis.

o During the profile design process, the male rotor addendum should be kept to a minimum to reduce the blow hole area.

o The so-called "approximate methods" (Method 2, in particular) presented here are sufficiently accurate for all practical purposes.

o The primary attributes of performance loss due to blow-hole leakage are increase in specific power and high discharge temperature. The impact on volumetric efficiency is minimal.

8. ACKNOWLEDGMENTS

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9. REFERENCES

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FIGURE 1 BLOW-HOLE

LOOKING ALONG TRAILING FLANK OF FEMALE ROTOR

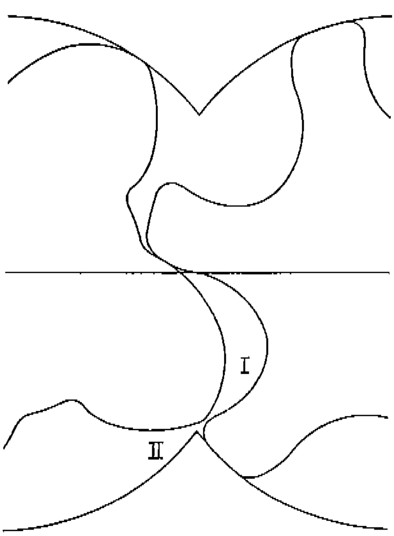


FIGURE 2

ENDVIEW SECTION THROUGH BLOW-HOLE

HIGH PRESSURE SIDE

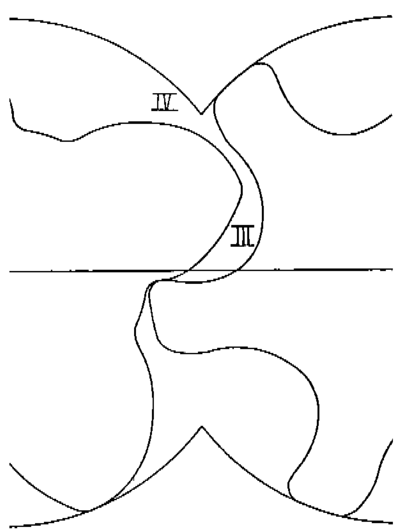


FIGURE 3

ENDVIEW SECTION THROUGH BLOW-HOLE

LOW PRESSURE SIDE

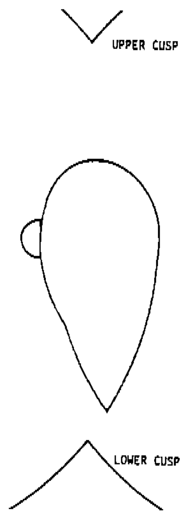


FIGURE 4

ROTOR CONTACT LINE

VIEWED FROM INLET END PLANE

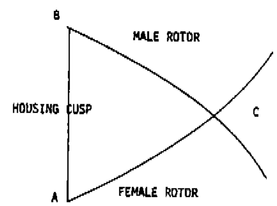


FIGURE 5

BLOW-HOLE TRIANGLE

EXACT METHOD

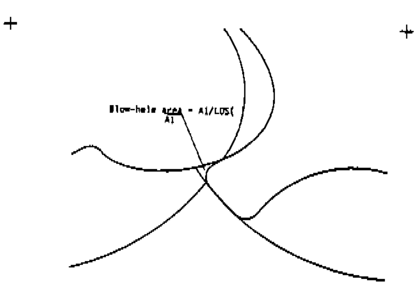


FIGURE 6

BLOW-HOLE AREA APPROXIMATION

METHOD I

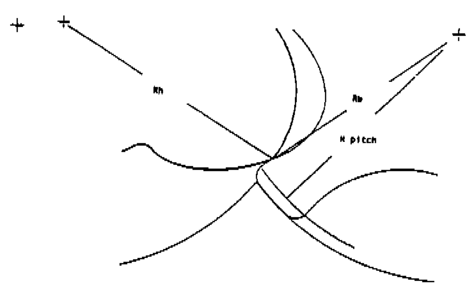


FIGURE 7

BLOW-HOLE AREA APPROXIMATION

METHOD 2

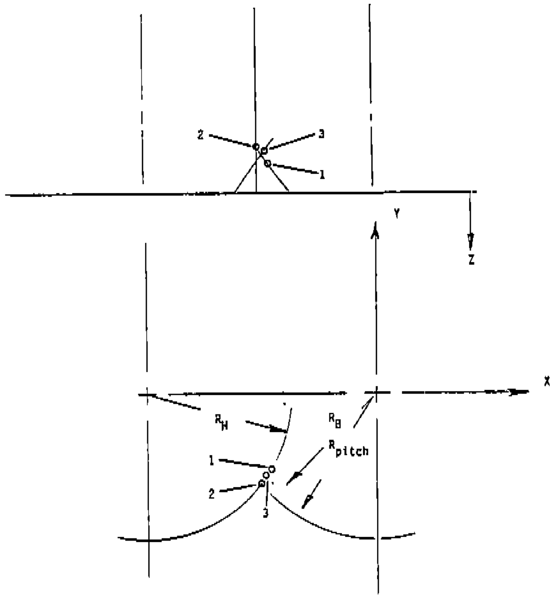
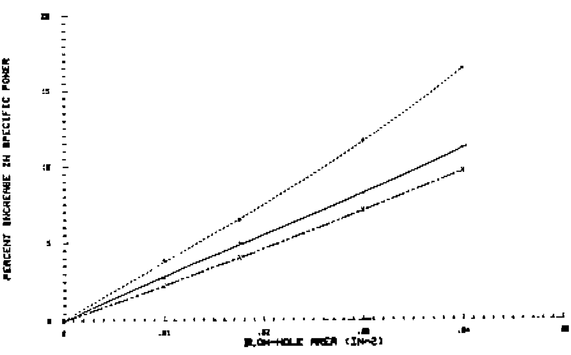


FIGURE 8

LOCATION OF VERTICLES

APPROXIMATE BLOW-HOLE TRIANGLE



* ------ 20 M/SEC. + ——— 30 M/SEC. x ——-- 40 M/SEC.

FIGURE 9

EFFECT OF BLOW-HOLE SIZE ON PERFORMANCE
100 mm x 1.9 L/D

DISCHARGE AIR PRESSURE = 105 PSIG

* ------ 20 M/SEC. + ——— 30 M/SEC. X ——--- 40 M/SEC.

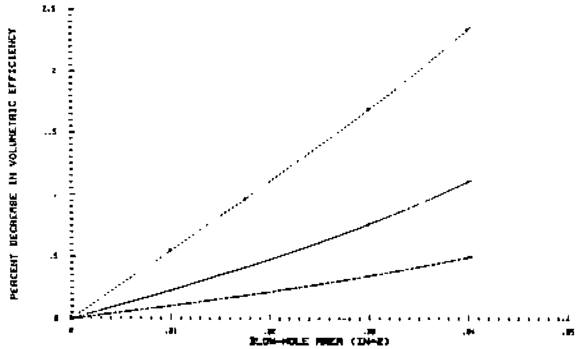


FIGURE 9 (CONTINUED). EFFECT OF BLOW-HOLE SIZE ON PERFORMANCE.

100 mm x 1.9 L/D. DISCHARGE AIR PRESSURE = 105 PSIG

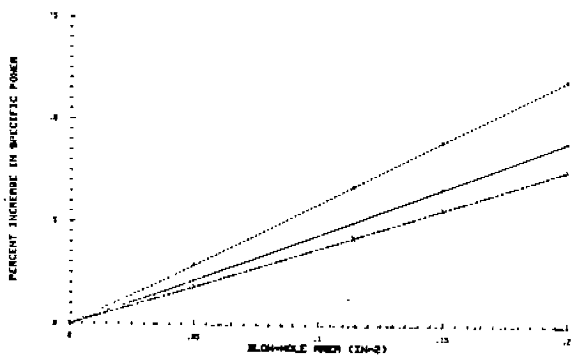


FIGURE 10. EFFECT OF BLOW-HOLE SIZE ON PERFORMANCE.

255 mm x 1.65 L/D. DISCHARGE AIR PRESSURE = 105 PSIG

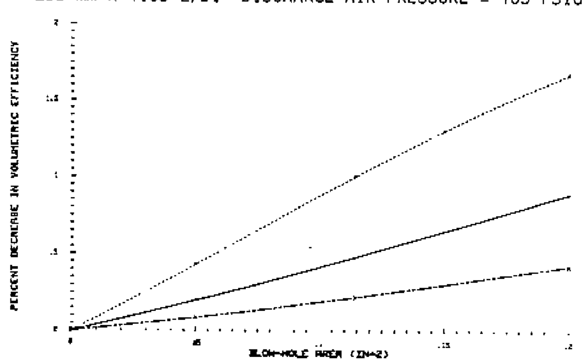


FIGURE 10 (CONTINUED). EFFECT OF BLOW-HOLE SIZE ON PERFORMANCE.

255 mm x 1.65 L/D. DISCHARGE AIR PRESSURE = 105 PSIG