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GENERAL STABILITY AND DESIGN SPECIFICATION OF THE
BACK-PRESSURE SUPPORTED AXIALLY COMPLIANT ORBITING SCROLL

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ABSTRACT

A linear, first order relationship may be used to approximate the stability characteristic of the back-pressure supported orbiting scroll. The resulting average tip loads may be similarly represented. Operating zones of incipient orbiting scroll instability are found to approximately follow zones of constant compression ratio. A simple design procedure is presented which may be used to specify axial compliance parameters for stability over an arbitrary operating range and to estimate resulting tip loads.

INTRODUCTION

The scroll compressor commonly consists of one scroll orbiting with respect to a second, typically fixed, scroll. Hence the terms "orbiting scroll" and "fixed scroll". These scrolls each have a flat floor portion on which is an involute-shaped wrap. As the scrolls intermesh, a series of trapped pockets are formed which decrease in size as they travel towards the center, compressing the gas within. Ideally the tips and flanks of the wraps would always be in light contact for truly sealed pockets.

During compression, gas pressure acts against the scrolls to separate them both axially and radially. These forces must be countered by some mechanism or structure. In so-called compliant designs, the scrolls are brought into positive contact to properly seal pockets for efficient compression. The scrolls are held in place by external forces, usually gas-induced, which allow the scrolls to "float" and determine their own geometric relation to each other. This nonrigid support also allows the scrolls to separate in response to liquids or small debris which may be ingested.

Compliance in the radial direction is usually provided by mechanical means in the orbiting scroll drive. In this paper we focus on compliance in the axial direction and discuss the theory and design considerations for stable operation over an arbitrary range of conditions.

FORCES ON THE SCROLLS

The forces on the orbiting scroll resulting from the compression process are shown in Figure 1. The radial gas force, \( F_{rg} \), acts along the line between the centers of the two scrolls and tries to push them apart to a common center. The tangential gas force, \( F_{tg} \), acts at the midpoint of and perpendicular to the line between the centers of the two scrolls. This is the force against which the actual work of compression is performed. The axial gas force, \( F_{ag} \), acts midway along the line between the two scrolls and normal to the plane of orbiting motion. This force tends to separate the scrolls axially. All forces act equally and symmetrically on both scrolls.

For axial compliance design, we must, as a beginning, overcome the influence of the axial gas force. For the back-pressure supported orbiting scroll, gas pressure from the sealed
compression chambers is admitted to sealed zones on the back of the orbiting scroll. The resulting force overcomes the axial separating force and pushes the orbiting scroll into contact with the fixed scroll. In addition to this, it is necessary to add an additional force increment to overcome a characteristic overturning moment which acts on the orbiting scroll.

In Figure 2 is a diagram of the forces acting on the orbiting scroll in the plane which is parallel to the axial and tangential gas forces. For this analysis, there is also an influence from the radial gas force, but it is very small and will be neglected for this first-order approximation. In addition to the gas forces, there is a force applied to the drive bearing of the orbiting scroll in response to the tangential gas force and a reaction force acting axially between the orbiting and fixed scrolls. Force summations in the axial and tangential directions are:

\[ \Sigma F_a = 0 = F_{bc} - F_{sg} - F_r \]  \hspace{2cm} (1)

\[ \Sigma F_t = 0 = F_{tg} - F_b \]  \hspace{2cm} (2)

where \( \Sigma F_a \) is the axial force summation, \( F_{bc} \) is the net back chamber force, \( F_r \) is the scroll-to-scroll axial reaction force, and \( F_b \) is the scroll drive bearing force.

Since the tangential gas force and bearing reaction do not typically act on a single line, there is also an overturning moment associated with them:

\[ \Sigma M = 0 = F_{tg} l - F_b r \]  \hspace{2cm} (3)

where \( l \) is the distance between the midpoint of the scroll vane and the drive bearing (the distance between \( F_{tg} \) and \( F_b \)). The value of the reaction radius, \( r \), will vary according to the particular back-chamber design and operating condition.

**ORBITING SCROLL STABILITY**

If the theoretical value of the reaction radius should exceed the physical size of the orbiting or fixed scroll, whichever is smaller, it will in fact be confined to the physical edge of the part. Since a sufficient radius to balance the moment equation is unavailable, the moment summation will no longer be zero, the orbiting scroll will no longer be in static equilibrium, and it will start to overturn until it comes into contact with some other mechanical restraint. This action, coupled with the orbital movement of the scroll, results in a sort of wobbling motion with all the axial contact occurring along the edge of the part.

This wobbling, or instability, results in leakage through the gap opened by the separated tips, edge loading of the scroll surfaces, and angular misalignment of the scroll drive bearing. All these can quickly lead to loss of performance and premature failure of the compressor.

**GAS FORCE EQUATIONS**

The axial and tangential forces, whether expressed in peak or average terms, consist of two components. Part of each force is derived from the pressure in the sealed pockets, which is a function of scroll geometry and suction pressure only. The other part is derived from the pressure in the discharge pocket and is a function of scroll geometry and the suction to discharge pressure difference only. These forces can be written in the linear form:
\[ F_{tg} = C_1 P_s + C_2 P_d \]

\[ F_{ag} = C_3 P_d + C_4 P_d \]

where \( P_s \) and \( P_d \) are the compressor suction and discharge pressures respectively. The constants \( C_1 \) through \( C_4 \) are functions of the particular scroll geometry. These force equations are derived to be "gage" forces, i.e., resulting from pressures above the "ambient" pressure \( P_0 \). The constants \( C_1 \) through \( C_4 \) may be derived to represent either average or peak forces. The timing of the peak force, whether axial or tangential, is dependent on whether the compressor is operating above or below the design pressure ratio. The peak force occurs just before the discharge porting point for operation below the design pressure ratio and just after porting for operation above the design pressure ratio. For this reason, the value of the four constant coefficients for peak loads will be different for the two zones of operation.

Back chamber pressure is typically provided by a combination of pressure from the discharge and from the sealed compression pockets. In this manner, the back chamber force may be made up of two components which behave in a similar manner, with respect to operating pressures, as do the internal gas forces. A common method is to provide two separate chambers for these force components. In that case, the back chamber force may be written as

\[ F_{bc} = (C_0 - 1) P_d A_2 + (P_d - P_s) A_d \]

where \( C_0 \) represents the time-average normalized pressure seen by a vent communicating between an intermediate pressure back chamber and a sealed compression pocket and \( A_1 \) and \( A_2 \) respectively are the areas of the intermediate and discharge back chambers. This method is most commonly applied in the high-side compliant orbiting scroll design and in the compliant fixed scroll design, in both of which it is convenient to provide discharge pressure at the center of the axially compliant scroll.

Another method especially suited to the low side compliant orbiting scroll design is to use a single back chamber whose vent is exposed to intermediate pressure for a portion of the time and to discharge pressure for the remainder. Figure 3 illustrates how such a vent hole sees the compression process. In this example \( P_d \) happens to be equal to the pressure in the sealed pockets as they open to discharge, but may vary independently. The vent is located at an angle \( v \) from the inner end of the wrap. In one revolution (or orbit) the vent sees the pressure in the sealed pocket for a period of \( v/2\pi \), and it sees \( P_d \) for a period of \( (2\pi - v)/2\pi \). The average pressure seen in the sealed pockets is calculated by assuming a polytropic compression process and solving an averaging integral which will result in the form \( C_0 P_s \), where, similar to the two chamber design, \( C_0 \) is a function of the scroll geometry and now \( v \). The back chamber pressure equation can be written as:

\[ P_{bc} = \left( -\frac{v}{2\pi} \right) (C_0 - 1) P_s + \left( 1 - \frac{v}{2\pi} \right) (P_d - P_s) \]

where \( P_{bc} \) is the average back chamber pressure. Note that this equation is of the same linear form as the back chamber force equation for the two chamber case. The back chamber force is found by simply multiplying the pressure by the chamber area:

\[ F_{bc} = P_{bc} A_{bc} \]
STABILITY EQUATIONS

Rearranging the force and moment summation equations above to solve for \( r \) gives

\[
r = \frac{F_T r}{(F_{bc} - F_{sg})}
\]  

(1)

Substituting the force equations for \( F_{tg} \), \( F_{sg} \), and \( F_{bc} \) gives:

\[
r = \frac{1}{A_{bc}} \left[ \frac{v}{2\pi} (C_0 - 1) P_s + \frac{2\pi - v}{2\pi} (P_a - P_s) \right] - C_3 P_s - C_4 P_d
\]  

(2)

Simplifying and grouping terms for \( P_s \) and \( P_d \) results in:

\[
r = \frac{[LC_1] P_s + [LC_2] P_d}{[A_{bc} \left( \frac{v}{2\pi} C_0 - 1 \right) - C_3] P_s + [A_{bc} \frac{2\pi - v}{2\pi} - C_4] P_d}
\]  

(3)

Letting \( K_1 \), \( K_2 \), \( K_3 \), and \( K_4 \) represent the above bracketed terms, respectively, and dividing the numerator and denominator both by \( P_s \), we have:

\[
r = \frac{K_1 + K_3 \text{OPR}}{K_3 + K_4 \text{OPR}}
\]  

(4)

where \( \text{OPR} \) is the operating pressure ratio of the compressor. The reaction radius \( r \) is found to be constant for a given operating pressure ratio of the compressor.

A similar derivation and simplification for the "tip" or reaction force \( F_r \) gives:

\[
F_r = P_r (K_3 + K_4 \text{OPR})
\]  

(5)

DESIGN PROCEDURE

Lines of constant reaction radius are superimposed on an operating envelope in Figure 4. They correspond to constant operating pressure ratio lines. The design procedure is to choose two extreme design points in the operating envelope, for example the maximum and minimum expected operating pressure ratio points, and fix \( r \) equal to the radius of the orbiting scroll floor portion. This results in two linear equations which are solved simultaneously for the two unknowns of \( Abc \) and \( v \). As a result, all other \( r \) values within the operating envelope will be less than the radius of the floor. This assures stability over the entire envelope. The final values of \( Abc \) and \( v \) must be chosen within the physical limits of the scroll compressor's layout.

In practice, use of peak loads will result in a very conservative design with higher tip loading than may be really required. When the compressor reaches an unstable operating condition, based on peak load calculation, it will begin to wobble for a small portion of the cycle while the peak loads exist. The resulting tip leakage causes higher pocket pressures which in turn feed back to the back chamber, raising its pressure as well. This positive feedback will hold the scroll in a quasi-stable condition beyond the expected stable operating range. On the other hand, use of average loads only will likely be insufficient to assure stability. Designers should
apply a correction factor based on experience with the particular compressor with which they work.

EXPERIMENTAL VERIFICATION

Figure 5 contains results of an experiment investigating the stability of a compressor with a back chamber supported orbiting scroll over a range of operating conditions and back chamber pressures. Scroll motion was monitored with internal proximity probes and the back chamber pressure was manually regulated from an external pressure source. For each condition, the pressure was slowly reduced until a rise in scroll wobble occurred. The reaction radius was calculated for the last pressure reading before instability and for the first reading after instability. These two radii are plotted with respect to operating pressure ratio. The marginally stable points are found to lie close to the line representing the physical radius of the scroll floor R. The lack of pressure feedback to the back chamber generally allowed for clear distinction between stable and unstable conditions, though in some cases, the transition from stable to unstable operation did not always occur instantaneously. This hints that the scroll may still have some limited self-stabilizing capability or simply that it is slow to respond to extremely brief excursions into an unstable operating zone.

CONCLUSION

The relationship derived above is a useful tool for designing stability into the axially compliant back-pressure supported orbiting scroll. It may be applied with equal effectiveness to either the single or dual chamber designs.

When peak reaction radii, calculated using peak loads, are greater than the radius of the floor for a portion of a revolution, static equilibrium does not exist and the orbiting scroll starts to experience a wobble or vibration that can quickly lead to loss of performance and premature failure of the compressor. However, positive pressure feedback from the sealed pockets to the back pressure chamber will cause the scroll to remain stable at conditions somewhat beyond those that these first order relationships predict.

REFERENCES

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Figure 3

Wrap Angle (Volume)

Figure 4

Constant Pressure Ratio and Reaction Radius

Design Pressure Ratio

Operating Envelope

OPR increasing
Figure 5