1988

On Transfer Slot Design in Rotary Sliding Vane Compressors with Special Attention to Gas Pulsation Losses

V. Yee
Purdue University

W. Soedel
Purdue University

ON TRANSFER SLOT DESIGN IN ROTARY SLIDING VANE COMPRESSORS WITH SPECIAL ATTENTION TO GAS PULSATION LOSSES

V. Yee
Graduate Research Assistant [1]
and
W. Soedel
Professor of Mechanical Engineering
Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907 USA

INTRODUCTION

The discharge port region of the rotary sliding vane compressor, as shown in Figure 1, is of primary interest in this study. In locating the angular position of the entrance to the discharge port region, commonly referred to as the throat section, two contradicting requirements must be balanced. From one point of view it is desirable to have the angular position of the throat section as close as possible to the axial seal so that the volume of oil and refrigerant trapped between the contact (here contact and axial seal are used synonymously) and end of the port is small. Once the sliding vane passes the end of the port, the trapped working fluid must be forced through the contact into the intake side of the compressor. If the trapped volume of working fluid is of such size that the flow area at contact is insufficient to accommodate smooth flow of this fluid to the intake side, damaging effects to the drive motor and compressor parts could occur.

Viewing this requirement, one may perhaps conclude that placing the end of the port as close to contact as possible is the ideal situation. This is not correct because a sufficient angular interval between the end of the port and the contact must be maintained to prevent excessive leakage of high pressure working fluid from the discharge to the suction side of the compressor. An excessive leakage past contact is highly undesirable since this will reduce the volumetric efficiency.

Because of the basic geometric configuration of the compressor, it is noted that the closer the angular position of the throat section to the contact point - the smaller the throat section area. A small throat section area can lead to a condition known as overcompression. Overcompression is a term describing the condition wherein the cylinder pressure rises well above the pressure in the discharge line as a result of flow resistance in the throat region. This rise in pressure becomes an undesired increase in the work of compression.

Therefore, the designer is required to satisfy the two requirements of small volume of trapped working fluid and sufficiently large throat section area to prevent overcompression. In order to satisfy both requirements, designers have included a small crescent shaped volume into the wall of the cylinder in the vicinity of the discharge port as shown in Figure 1. From a manufacturing standpoint this is accomplished by use of a circular milling cutter.

It was desired in this study to examine the influences of the transfer slot on the work of compression in rotary sliding vane compressors. This was accomplished by development of a mathematical simulation routine that predicted the pressure histories within the chambers that are formed during rotor rotation and the corresponding valve displacement history. Knowing the pressure histories across a given vane, the work and power required for compression and discharge of the working fluid was obtained. After verification of the model, geometric parameters associated with the transfer slot were varied to study the influences of these variations on the work of compression.

Previous simulation attempts of rotary vane compressors have lumped the discharge port, transfer slot, and volume following the throat section with the volume preceding it. Coates [3] did consider the throat area as a restriction during the discharge process, but he still did not make a distinction between the two regions as being separate during re-expansion and compression. For re-expansion of gases as a vane passes the throat section, Coates assumes an instantaneous mixing of working fluid. Ucer and Aksel [4] considers two separate volumes during re-expansion until equilibrium of pressures occur; then, once the pressures are equalized the two regions again lose their separate identities.

THEORETICAL APPROACH

The basic elements of computer simulation for positive displacement compressors were outlined in a set of short course notes presented in conjunction with the 1972 Compressor Technology Conference at Purdue University [5]. As stated there, the basic elements are four sets of coupled equations. They are (1) the volume equation, relating chamber volumes to input shaft angle or time, (2) thermodynamic equations, giving the instantaneous gas masses, pressures, and temperatures in the chambers, (3) the mass flow equations, describing the mass flow through restrictions at any instant of time, and (4) the valve dynamics equations, which describe the valve motion at a given instant of time.

Previous compressor simulations used a quasi-steady flow equation to describe flow through restrictions. This quasi-steady flow equation possesses a shortcoming in its inability to account for the inertia of the gas in the restrictions. In compressors, pressure oscillations may be initiated by the sudden opening of chambers and the discharge of gas through restrictions into adjoining chambers. In this work, an alternative to the quasi-steady flow model traditionally used was developed. This nonlinear model considers the inertia of the gas in restrictions and, thereby, can model pressure oscillations resulting from sudden opening of the pressurized chambers.

In the rotary compressor, as a vane passes the throat section, the working fluid that is trapped in the transfer slot, discharge port, and volume between cylinder and rotary piston re-expands in an oscillatory fashion into the trailing chamber. During and after re-expansion the pressure of the working fluid is increased as a result of the decreasing size of the trailing compartment and, subsequently, working fluid is again forced into the transfer slot-discharge port region. When the pressure within the discharge port-transfer slot region is sufficient to open the discharge valve, working fluid from this region flows past the valve while being resupplied by working fluid from the trailing compartment.

Basically, a time dependent control volume approach was used. The control volumes were selected to be closely related to Helmholtz resonator combinations of chambers and necks.

Figure 2(a) shows the chamber-neck combinations into which the system was divided. The first chamber is composed of the volume enclosed by the cylinder and rotor walls, throat section, and approaching vane. Its corresponding neck is composed of the region enclosed by the cylinder and rotor after the throat section to contact.

The second chamber is composed of the region consisting of the transfer slot and discharge port. The small volume between the valve and the plane of the seat is neglected. The corresponding neck is composed of the imaginary doughnut-like ring between the valve and seat. The extent of this ring is imagined to terminate at the closest point of clearance of the valve reed.

There is a variation to the above situation when a vane is present between the throat section and contact. This is shown in Figure 2(b). In this situation the neck for the first chamber will consist of the region enclosed by the vane, throat section, rotor wall, and the exposed entrance to the transfer slot. The second volume, in addition to the volume previously described, will include a small volume enclosed by the blade, rotor wall, contact, and the entrance to the transfer slot.

Figure 3 shows the flow restrictions encountered by the working fluid in moving from the throat section to the entrance of the transfer slot. This is represented in terms of a fluid flow diagram. From this diagram it can be seen that A_2 and A_3 are in series. The combination of A_2 and A_3 is in parallel with A_1. Finally, the overall combined orifice for A_1, A_2, and A_3 is in series with the entrance to the transfer slot, A_4.

For the case when a vane is present in the region between the throat section and contact, A_3 consists of the two side panel areas from the throat section to the vane and A_4 consists of the slot opening exposed by the vane, as shown in Figure 4.

It is assumed that the flow restrictions act like time dependent incompressible inertia elements, while the gas in the chambers is compressible but of negligible inertia. Considerable judgement is required to make this distinction properly. More work needs to be done to reduce the subjectivity of this approach.

The inertia of the gas trapped in the volume between valve seat and valve reed was also considered, similar to work by Trella and Soedel [6].

While the assumptions are of the Helmholtz type, the resulting nonlinear equations are relatively complicated [2].

RESULTS AND COMPARISON TO EXPERIMENTAL MEASUREMENTS

The pressure measurements will not be discussed in this paper in detail, except to point out that they are more intricate to perform than for reciprocating compressors since the volumes between vanes travel. Several transducers are necessary. Details can be found in reference [2].

Comparison of theory and experiment during the compression process typically indicates that the simulation model predicts a compression pressure level slightly below the experimental one as shown in Figure 5. The pressure differential predicted from the simulation model across the 1st neck is negligible. What appears to be a single curve are actually two curves overlaid. The experimental curves show a slight separation during initial compression with eventual merging of the curves prior to valve opening. The lower theoretical curves during compression were also seen by Coates [3]. The slight discrepancy is believed to be the result of neglecting leakage flow in the simulation routine. This also contributes to the slightly lower pressure in the chamber following the compressing vane.

During the discharge process the initial pressure peak and the subsequent steady state discharge pressure level are at higher levels than seen in the experimental curves. This correlates also with the difference seen between theoretical and experimental valve curves. The theoretical valve curve shows a slower rise rate and lower peak amplitude. This would result in an increased flow restriction and higher peak pressure. Although the drop rate of the valve curves agree, the amplitude is at a lower level, possibly explaining the higher theoretical steady state discharge level predicted. The relatively small discrepancies seen in the valve motion are believed to be partially attributed to neglecting the contributions of the higher modes in the overall valve motion and slight errors in experimentally determined mode shapes.

During the re-expansion process, the theoretical model seems to approximate the pressure drop during the initial phase well. The experimental curve drops at a slower rate than that predicted by the simulation during the latter part of re-expansion. However, the oscillations seen at the completion of re-expansion in the experimental curve are now in principle predicted, even while agreement could be better.

VARIATION OF TRANSFER SLOT GEOMETRY

The first item varied was the angular position of the throat section. The purpose of this variation was to simulate manufacturing differences between cylinders that may occur as a result of positioning of the cutter. This variation may be visualized as the transfer slot sliding along the profile of the cylinder cavity. As the location of the transfer slot moves, the throat section area, discharge port volume, and underslot area are all affected. The results of this variation are shown in Figure 6. The figure in general indicates that the angular position of the transfer slot in the prototype design was essentially at the optimum position.

The second item varied was the depth of cut of the transfer slot. As before, from a manufacturing viewpoint this can be thought of as differences between slot cylinders as a result of fluctuation in feed of the milling cutter. Again, the throat section area, the discharge port volume, and the underslot area are all affected. The results of this variation are shown in Figure 8. The figure shows that a slight improvement could possibly be obtained by decreasing the depth of cut.

The third item varied associated with the transfer slot is width of the slot. This variation will affect the volume of the transfer slot and the underslot area. The results are shown in Figure 7. The figure indicates that the width of the slot of the prototype compressor is essentially at the optimum. Decreasing the slot width may decrease the power consumption only minutely.

CONCLUSION

It is felt that the simulation model, in general, gives an adequate approximation of trends in the actual compressor. The formulation has the capability of modeling gas oscillation, which was lacking in previous simulations. Another capability added was a more realistic approximation of re-expansion than the instantaneous mixing used prior to this work. Using this simulation, a qualitative idea of the results of variation of the various parameters associated with the compressor may be obtained. The use of the simulation in studying certain variation of the geometric parameters associated with the transfer slot was demonstrated.

ACKNOWLEDGEMENT

The support of this research by Whirlpool Corporation was appreciated.

REFERENCES AND FOOTNOTES

1. Now with Naval Ocean System Center, San Diego, California.
2. Victor Yee, "Analytical and Experimental Study of High Speed Rotary Sliding Vane Compressor Dynamics with Application to Transfer Slot Design," Ph.D. Thesis, Purdue University, 1982.
3. D.A. Coates, "Design Technique for Performance Optimization of a Small Rotary Vane Compressor," Ph.D. Thesis, Purdue University, January 1970.
4. A.S. Ucer and H.M. Aksel, "Simulation Studies on Rotary Type Compressor Systems," Proceedings of the Purdue Compressor Technology Conference, Purdue University, July 1980.
5. W. Soedel, "Introduction to Computer Simulation of Positive Displacement Compressors," Short Course Text, Editors Office, South Campus Courts, Purdue University, 1972.
6. T. Trella and W. Soedel, "Lumped Parameter Modeling of a Nonlinear Pneumatic-Mechanical System," 71-VIBR-41, American Society of Mechanical Engineers, 1971.
7. V. Yee and W. Soedel, "Pressure Oscillations During Re-Expansion of Gases in Rotary Vane Compressors by a Modified Helmholtz Resonator Approach," *Journal of Sound and Vibration*, Vol. 91, No. 1, 27-36, 1983.

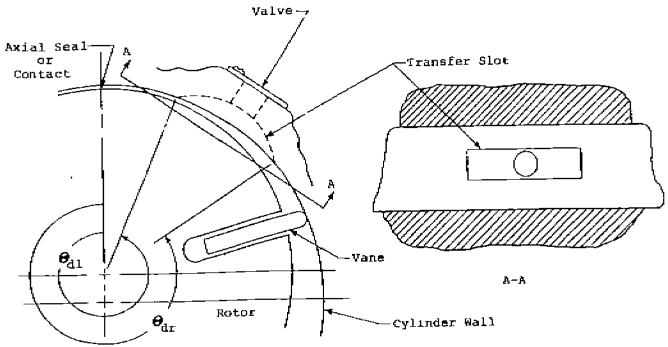


Figure 1 Transfer slot in rotary vane compressor

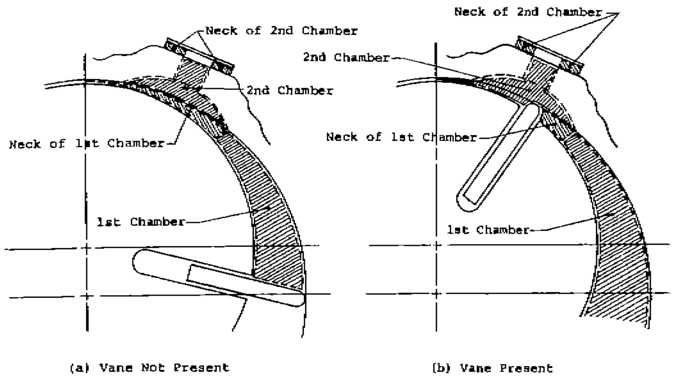


Figure 2 Control volumes

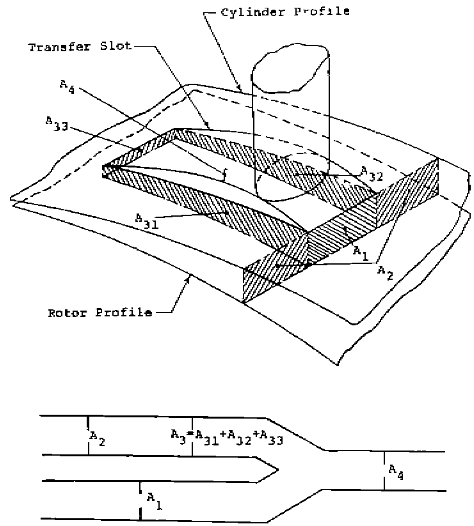


Figure 3 Flow restrictions when vane is not present

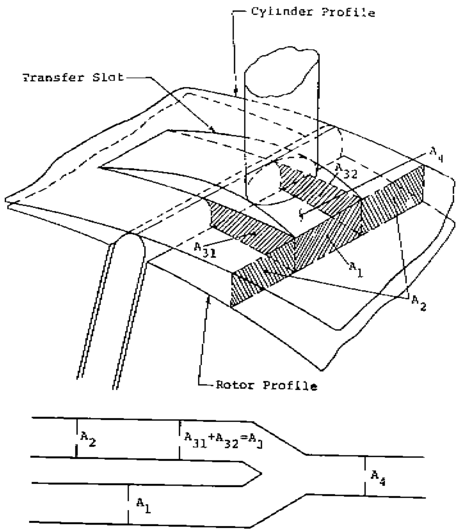


Figure 4 Flow restrictions when vane is present

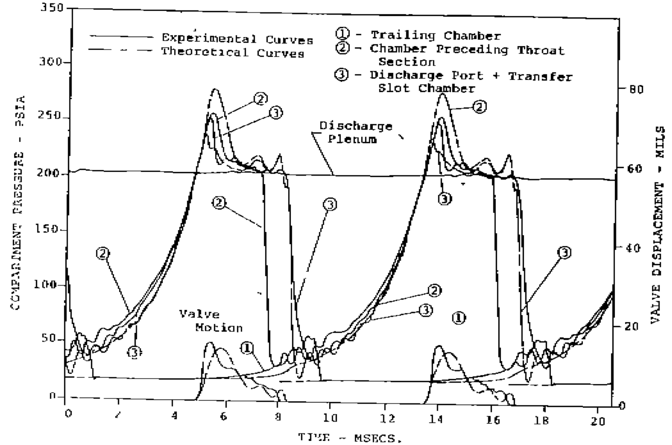


Figure 5 Theoretical and experimental pressures and valve displacements

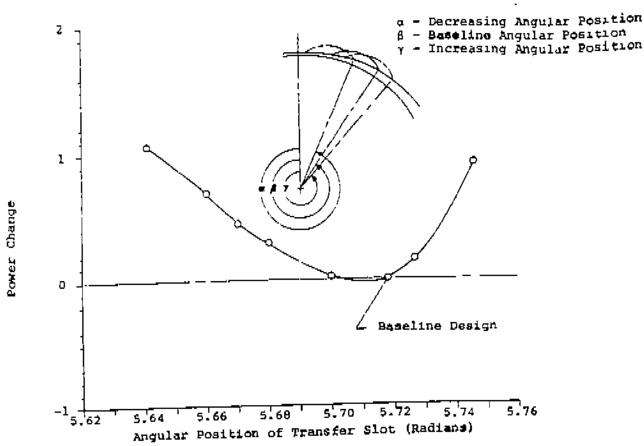
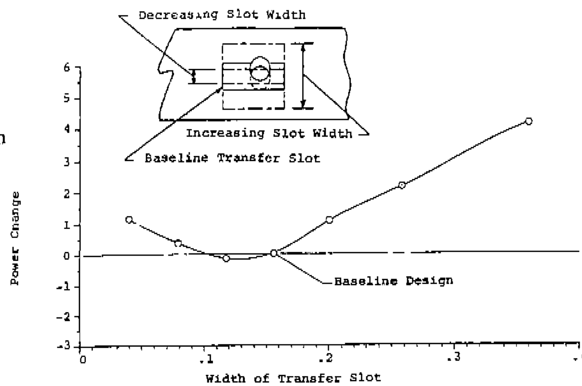


Figure 6 Power requirement change as function of angular position of transfer slot

Figure 7

Power requirement change as function of transfer slot depth



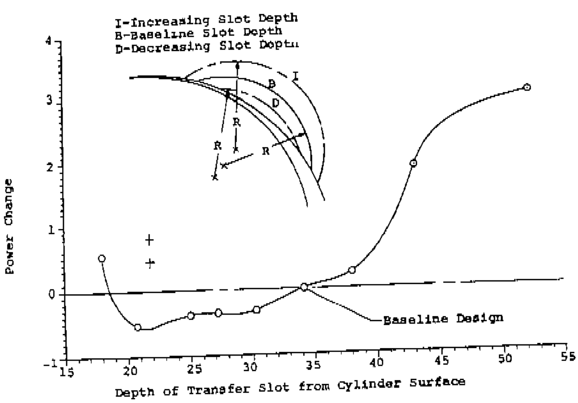


Figure 8

Power requirement change as function of transfer slot width