1994

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Analysis Of Clearance Volume Equalization And Secondary Pressure Pulse In Rolling Piston Compressors

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ABSTRACT

A common characteristic to rolling piston compressor operation is the re-expansion of compressed fluid from the clearance volume back into the cylinder chamber. Another characteristic which occurs is the formation of a secondary pressure pulse in the cylinder chamber near the end of the discharge process. These characteristics are modeled as dynamic processes in a comprehensive simulation. Results predicted by the models compare well with measurements. Sensitivity of the clearance volume re-expansion and secondary pulse formation to clearance volume geometry is demonstrated.

NOMENCLATURE

A	Area		
C	Flow coefficient	$\dot{m}$	Mass flow rate of equalization
EOD	Crank angle at end (trailing edge) of discharge port	SOD	Crank angle at start (leading edge) of discharge port
g_c	Gravitational acceleration constant	θ	Crank angle
h	Enthalpy of fluid	ρ	Density of fluid

Subscripts

c	Cylinder chamber	e	Equalization
cv	Clearance volume	i	Index referring to cylinder chamber 1 or 2
do	Downstream	up	Upstream

INTRODUCTION

Many papers [1,2] over the past two decades have discussed various aspects of rolling piston (rotary) compressor operation. These discussions have covered the range of topics from thermofluid performance such as leakage, heat transfer, compression losses, etc. to machine dynamics such as shaft oscillations and compressor vibration. Another characteristic common to rotary compressors is the backflow (re-expansion) of fluid from the clearance volume back into the cylinder chamber. The clearance volume consists of any space between the discharge valve and cylinder chamber not swept by the rolling piston inside the cylinder chamber. This space includes the discharge port, as well as any passage leading to the discharge port. The re-expansion of high pressure fluid in the clearance volume back to low pressure in the cylinder chamber generally occurs as a back flow pulse which can excite pressure oscillations in the cylinder chamber. A related phenomenon is the formation of a secondary pressure pulse in the cylinder chamber near the end of the discharge process. This characteristic is caused by the simultaneous reduction of discharge port area and cylinder volume as the piston approaches and passes the discharge port.

The effect of clearance volume on rotary compressor performance has been investigated by Yanagisawa [3] experimentally and using an analytical model limited to the re-expansion phenomenon. Some aspects of his model have been adopted in our approach. Though our model predicts the back flow pulse, the resulting pressure oscillations are not predicted. An approach for modeling the oscillations was demonstrated by Kawaguchi [4] using a model focused only on the re-expansion produced pressure oscillations in the cylinder chamber. A discussion of the formation of the secondary pressure pulse in the cylinder chamber has not been found in the open literature. In this paper, a detailed description is given of the modeling approach used for the clearance volume equalization process, as well as for the formation of the secondary pressure pulse.

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ANALYTICAL MODELING

Geometric Relationships

Equalization of the clearance volume and formation of the secondary pressure pulse are two of a number of models we have included in a comprehensive simulation of the rotary compressor. To model the pressure equalization process in the clearance volume, as well as the coupled processes of suction, compression, and discharge, geometric descriptions are required for cylinder chamber volume and areas open for flow between the various processes. Analytical expressions describing these geometric regions will not be included here as most have been well documented in previous papers [3-8]. However, a qualitative description of the geometric regions used in our current model will be provided for clarification of the approach.

Figures 1 and 2 show crank angle positions of the piston and associated labeling which assist in this discussion. In Fig. 1, the seal point (actually a line) between the piston and the cylinder is shown dividing the cylinder space into two chambers: the leading chamber ahead of the seal point, and the trailing chamber behind the seal point. Since two crankshaft revolutions are required for a fluid particle to pass from suction through compression and discharge, these processes are modeled by following a moving control volume through the cylinder chambers for two revolutions. During the first revolution, the trailing cylinder chamber is formed as the space between the piston and cylinder enlarges behind the seal point. This trailing chamber control volume is referred to as V_{c1}. At the transition point from the first revolution to the second revolution (360 deg. crank angle, Fig. 2), the previous trailing chamber becomes the leading chamber. Then during the second revolution, this leading cylinder chamber undergoes volume reduction as the space ahead of the seal point between the piston and cylinder decreases. This leading chamber control volume is referred to as V_{c2}. All the while the clearance volume V_{cv} is fixed by the volume in the discharge port plus any connecting passage cut out of the cylinder wall. One such cutout passage opening the discharge port more to the cylinder chamber is shown in Fig. 3 and discussed below. Typically, the formation of the secondary pressure pulse occurs nominally during the time in which the piston moves from the 270 deg. crank angle in Fig. 1 to the zero crank angle in Fig. 2. In going from Fig. 1 to Fig. 2, the piston passes over the discharge port, allowing high pressure fluid in the clearance volume V_{cv} to re-expand back to the low pressure fluid in the trailing chamber V_{c1} (and V_{c2}). This backflow

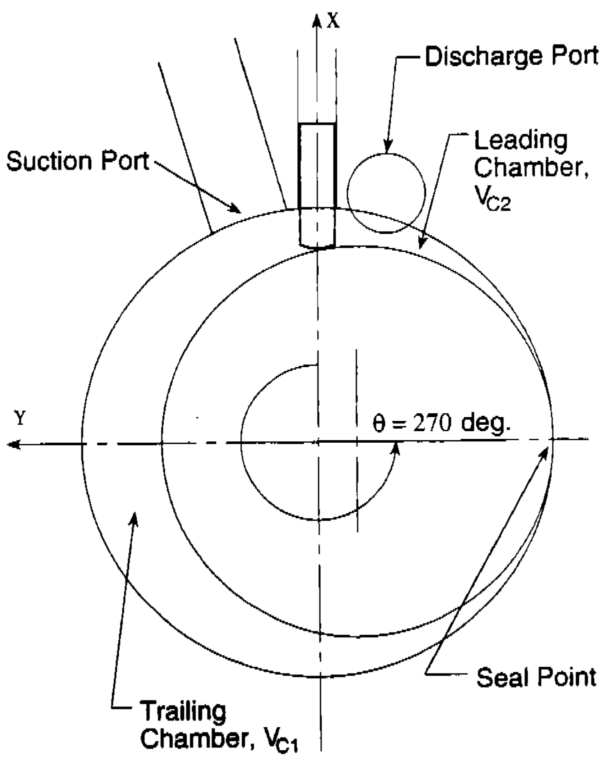


Fig. 1 Cylinder chambers for moving control volume with piston at 270 deg. crank angle.

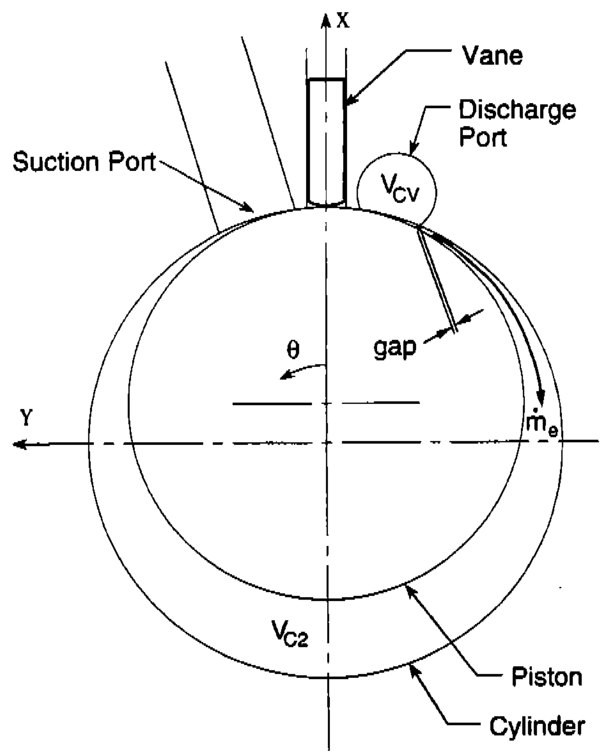


Fig. 2 Moving control volume V_{c2} and stationary control volume V_{cv} with re-expansion flow $\dot{m}_e$ through gap area.

typically occurs as a short duration pulse relative to the period of shaft revolution. The start of re-expansion occurs when the seal line between the piston and cylinder passes the leading edge of the discharge port. From Fig. 2, it should be clear that the suction port is open to the moving control volume V_{c2} while the discharge port (clearance volume) is also open to V_{c2}. Thus, the re-expansion of fluid can cause subsequent backflow out the suction port. However, due to dynamics of the fluid, less backflow occurs than might be expected [3]. Also, some backflow can occur out the suction port due simply to the reduction of volume V_{c2} while the suction port is still open. This latter effect is similar to that found in the scroll compressor and can be modeled similarly [9].

The effective area, A_e, open for equalization flow between the clearance volume and the cylinder chamber volumes can get rather complicated. Our current approach uses an effective area which is the combination of three different geometric areas that vary during each revolution. In Fig. 3, two of these areas are shown and consist of the intersection of the discharge port and a cutout passage with the endwall and inner cylinder surfaces, respectively. The area due to the intersection of the cutout passage is clearly shown in Fig. 3c, while the area due to the discharge port that is not covered by the piston is shown cross-hatched in Fig. 3a. These first two types of geometric areas represent the available area open to the clearance volume at the boundary of the cylinder chamber and, therefore, are added together to get the total geometric area open at this boundary. The third type of geometric area is shown in Figs. 2 and 3a and consists of the three-dimensional (3-D) area formed by the radial gap between the cylinder wall and piston around the edge of the cutout passage in the cylinder wall [3]. This gap area is the most restricted open area for equalization flow when the piston is close to the discharge port. The combination of these three areas as an effective area for equalization flow is shown in Fig. 4. The sum of the first two areas, at the boundary of the cylinder chamber, produces the constant area in the middle of the curve. This is the maximum area which can be open for equalization flow. The other regions of the curves are produced by the third type of area due to the 3-D gap between the edge of the cutout passage and the piston.

Control Volume Thermodynamic Process

The clearance volume equalization process involves the interaction of a stationary control volume (clearance volume) with both of the moving control volumes (leading and trailing cylinder chambers). The moving control volumes are employed to model the suction and compression processes in the cylinder chambers. The stationary control volume not only controls clearance volume equalization, it also controls the discharge process of flow past the valve. In these control volumes, the following models and assumptions were used to govern the processes.

1) conservation of mass
2) polytropic process
3) real properties of pure refrigerant for working fluid
4) uniform properties throughout control volumes

These models governing control volumes have been well documented in the literature [1-9] and, therefore, will not be repeated here.

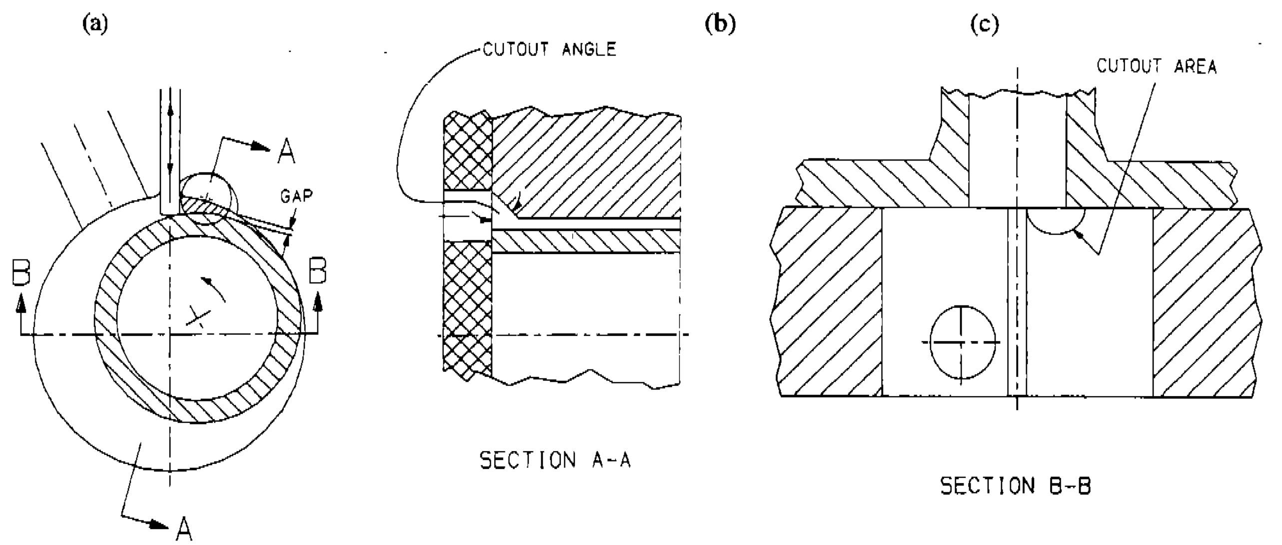
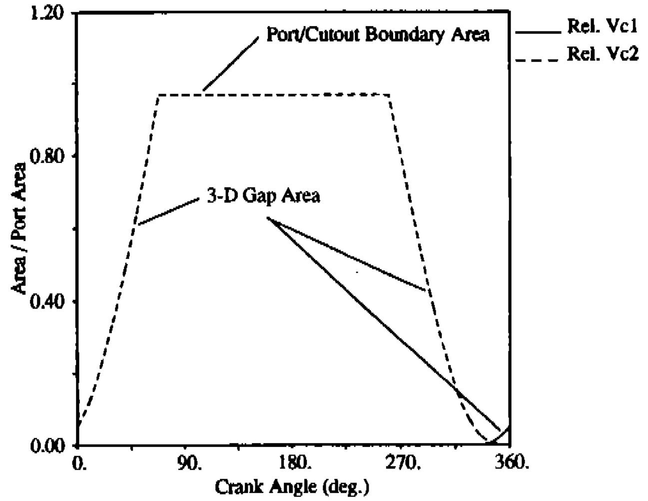


Fig. 3 Clearance volume geometry consisting of discharge port and cutout passage; (a) area at gap and port/endwall interface, (b) cutout passage angle, and (c) area at cutout/cylinder interface.



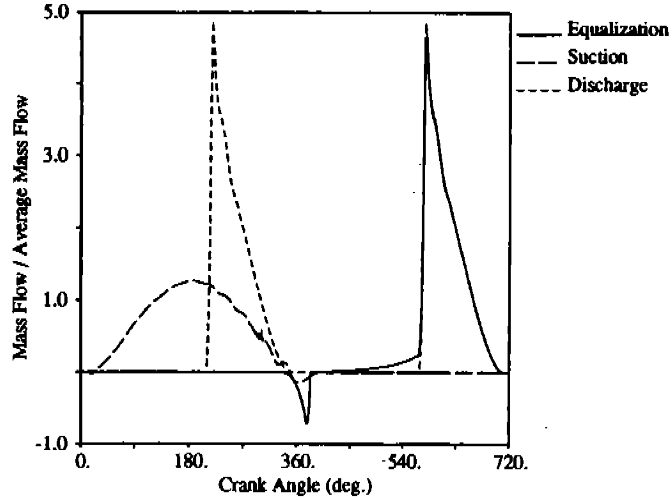


Fig. 4 Effective equalization flow area between clearance volume and moving control volume; relative to V_{c1} during first revolution, relative to V_{c2} during second revolution.

Fig. 5 Mass flow rates predicted for equalization flow, suction flow, and discharge flow; equalization and suction relative to moving control volume, discharge relative to stationary clearance volume.

Flow Equalization Process

The instantaneous mass flow rate of fluid exchange between the clearance volume and cylinder chambers was modeled using the steady, one-dimensional, isentropic flow equation,

$$\dot{m}_{ei}(\theta) = C_{ei}(\theta)\, A_{ei}(\theta)\, \rho_{ei}(\theta) \sqrt{2g_c(h_{up}-h_{do})} \tag{1}$$

where C_{ei} is the flow loss coefficient through area A_{ei}. Both C_{ei} and A_{ei} are functions of crank angle; A_{ei} varies with crank angle as was shown in Fig. 4, and C_{ei} varies with crank angle since it is dependent upon the upstream and downstream area ratios with A_{ei}. The index is either 1 or 2 to differentiate between flow equalization with cylinder chamber 1, V_{c1}, or chamber 2, V_{c2}.

Other Models

The above models were employed in a comprehensive rotary compressor simulation which couples together the effects of many significant processes occurring during steady-state operation. A list of these other models which are important for predicting compressor operation, but are not specific to the clearance volume equalization process and secondary pressure pulse formation, are as follows [1-8].

1) leakage
2) valve dynamics
3) pressure drop in suction and discharge manifolds
4) pressure pulsation in suction and discharge manifolds
5) kinematic and dynamic loads
6) shaft rotational dynamics

DYNAMIC CHARACTERISTICS

Mass flow rates predicted by our simulation which are associated with both the moving and stationary control volumes are shown in Fig. 5. In the first shaft revolution, the moving control volume in cylinder chamber V_{c1} experiences primarily suction mass flow. Near the end of the first revolution, as the piston passes over the discharge port, control volume V_{c1} starts to receive re-expansion flow from the clearance volume. Soon thereafter, near the transition of the moving control volume from V_{c1} to V_{c2} at 360 deg. (Fig.2), backflow occurs in the suction port as the clearance volume re-expansion peaks. As the piston passes over the trailing edge of the suction port, sealing off suction flow, the re-expansion flow approaches zero as the clearance volume pressure equalizes with the cylinder chamber. Thereafter, as the fluid in the moving control volume V_{c2} is compressed, fluid is forced into the clearance volume to keep pressure equalized. This process continues until the pressure in the clearance volume is sufficient to force the discharge valve open, whereupon mass flow out past the valve occurs and is matched by equalization flow into the clearance volume from the cylinder chamber.

The pressures corresponding to the above mass flow rates are shown in Fig. 6. Here the pressure in the moving control volume (in cylinder chambers) is shown for suction and compression covering two shaft revolutions while the pressures in the stationary locations of suction port, clearance volume, and discharge valve plenum are shown repeating every shaft revolution. The suction and compression pressure curve is fairly typical except that a significant pressure rise occurs shortly after 360 deg. This rise in suction fluid pressure is met by a rapid drop in clearance volume pressure, due to the re-expansion of high pressure fluid in the clearance volume, until equalization occurs. Thereafter, a typical compression curve is followed until discharge commences. During discharge, the cylinder chamber pressure is higher then the clearance volume due to pressure loss from flow between the cylinder chamber and discharge port. Near the end of discharge, the pressure in the cylinder chamber V_{c2} increases to form a secondary pressure pulse as flow area between the cylinder chamber and clearance volume is reduced to zero while the volume in V_{c2} is reduced to near zero. Thus, while the fluid within the cylinder chamber can experience the secondary pressure pulse due to flow restriction simultaneous with volume reduction, the fluid in the discharge port (clearance volume) cannot experience this effect.

EXPERIMENTAL VERIFICATION

Verification that these models are working correctly is demonstrated in Figs. 7 - 10 where comparisons are made between measured (solid curve) and predicted (dashed curve) pressures in the cylinder chamber V_{c2} and clearance volume during the second shaft revolution. Figures 7 and 8 are for the ARI condition (45 F saturated evaporator / 130 F saturated condenser) at 60 Hz and show very good agreement between measured and predicted pressures. One notable difference is that the pressure fall off in the clearance volume around 360 and 720 deg. due to re-expansion is not predicted to occur as soon as is measured. This is due to a small region of re-expansion flow area which opens earlier than what is currently modeled. Also, the pressure oscillations in the compression part of the measured curve are not modeled and, therefore, are absent in the predicted pressures. Further, these predictions were performed with an acoustic model of the discharge manifold which was limited to low frequency pressure oscillations, thus the high frequency oscillations during discharge in the measured pressure are not predicted. Figures 9 and 10 are for the 55 F / 90 F condition at 60 Hz and exhibit fair agreement. The same differences discussed previously are also true here. In addition, the second pressure pulse is under-estimated at this high flow condition. Some further refinement of the model is needed to better address the flow effect on this secondary pressure pulse.

PARAMETER VARIATION STUDY

As discussed previously, the formation of the secondary pressure pulse in the cylinder chamber results from two simultaneous events: a rapidly decreasing flow area between the cylinder chamber and clearance volume, and cylinder chamber volume reduction approaching zero as crank angle approaches 360 deg. This effect is dramatically demonstrated in the results of Fig. 11, where cylinder pressure variation is shown for the four cutout angles of 30, 40, 50, and 60 deg. at the ARI condition and 60 Hz. All other geometry remained unchanged for these values of cutout angle. The impact of increasing the cutout angle is twofold: increased flow area for clearance volume equalization, and increased clearance volume. As expected, these combined effects result in a decreasing secondary pressure pulse in the cylinder chamber as cutout angle is increased from 30 to 60 deg. Also noted is the general reduction of chamber pressure during the discharge process for increasing cutout angle. The overall effect can be demonstrated by examining the variation of input power, capacity, and EER with cutout angle as shown in Fig. 12. Capacity decreases slightly (0.3%) over the range while input power decreases significantly by 2.0%. This decrease in power is achieved principally by reducing the secondary pressure pulse and, consequently, the over-pressure power, resulting in an increase in EER. Over the range of cutout angle, the clearance volume increases 52 % (this is a 1.1% increase in volume relative to the swept volume), but only results in a decrease in volumetric efficiency of 0.4%. These trends indicate some improvement in performance can be gained by increasing the cutout passage angle.

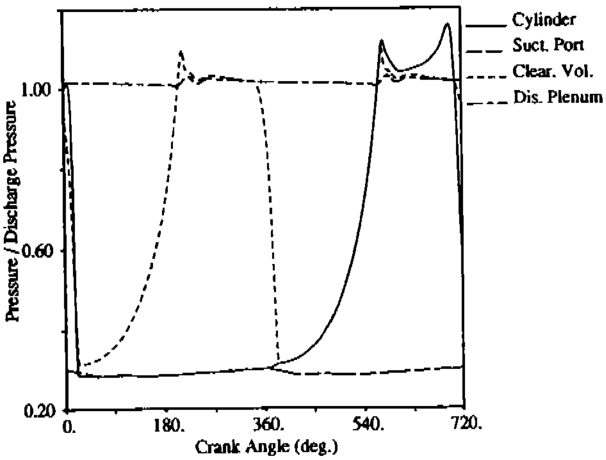


Fig. 6 Pressures predicted in cylinder moving control volume, suction port, clearance volume, and discharge valve plenum.

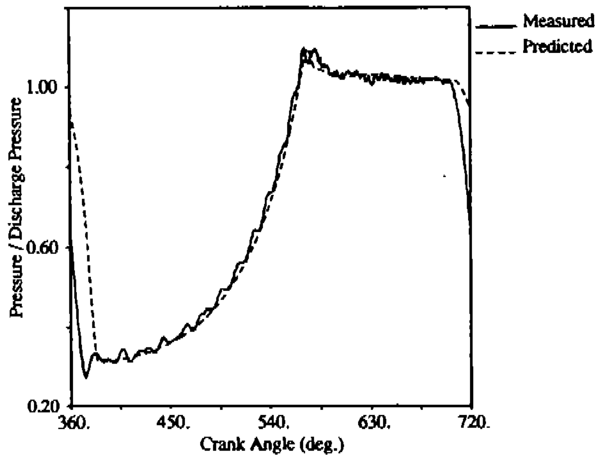


Fig. 7 Comparison of predicted with measured clearance volume pressure for ARI, 60 Hz.

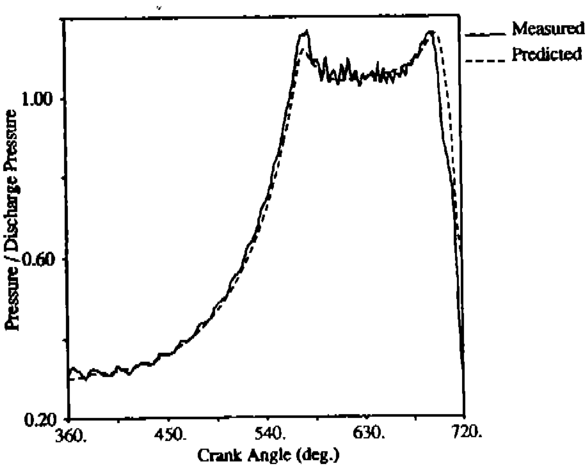


Fig. 8 Comparison of predicted with measured cylinder chamber pressure for ARI, 60 Hz.

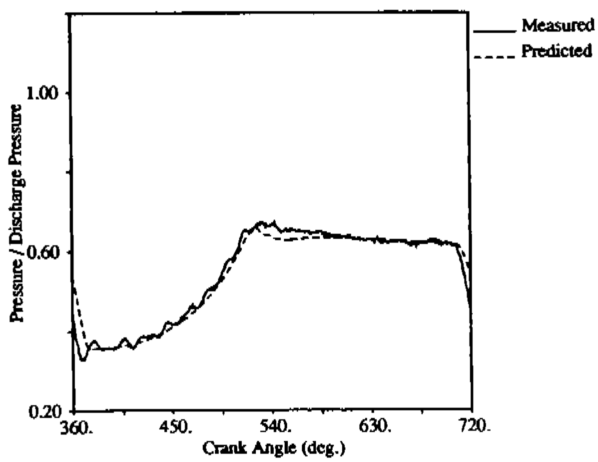


Fig. 9 Comparison of predicted with measured clearance volume pressure for 55/90, 60 Hz.

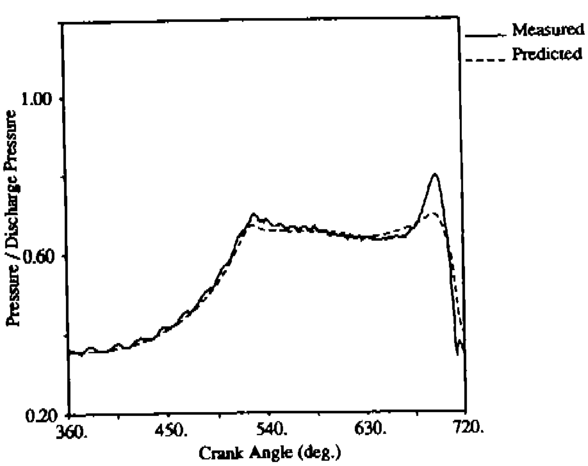


Fig. 10 Comparison of predicted with measured cylinder chamber pressure for 55/90, 60 Hz.

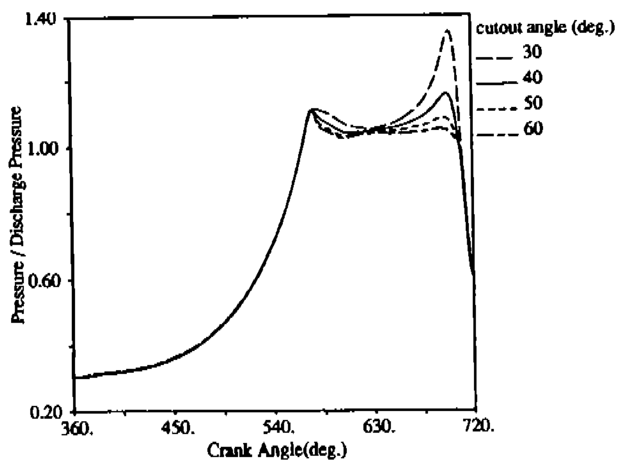


Fig. 11 Predicted variation of cylinder chamber pressure with cutout angle.

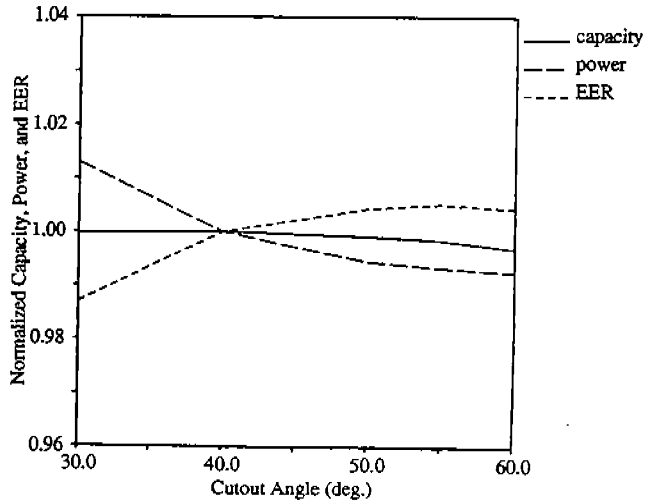


Fig. 12 Predicted variation of input power, capacity, and EER with cutout angle at ARI and 60 Hz; values normalized to those for 40 deg. cutout angle.

CONCLUSIONS

1) The approach of modeling the clearance volume as a stationary control volume separate from the moving control volume in the cylinder chambers has been shown to work well for a rolling piston compressor. The important characteristic of clearance volume re-expansion and equalization is predicted accurately.

2) The occurrence of a secondary pressure pulse has been modeled and shown to be formed in the cylinder chamber by the simultaneous reduction in area open to the discharge port with reduction in cylinder chamber volume.

3) The geometry of the cutout passage (in particular, cutout angle) in the cylinder wall, which is designed to improve flow to the discharge port, has been shown to have a strong effect on the formation of the secondary pressure pulse and, consequently, on over-pressure power.

ACKNOWLEDGEMENT

We would like to thank Carrier Corporation for funding this modeling effort. Also, thanks are due to Dave Cutts[1] for assistance with the experimental data.

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