1982

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OPTIMIZATION OF A SPECIAL SHAPED ROTARY VANE COMPRESSOR-COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS

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ABSTRACT
In cooperation with a West German company and the University of Hannover a special shaped rotary vane compressor was designed and developed. Purpose of investigation was to minimize friction and compression losses. Theoretical research in this case is based upon experimental results of the compressor gained at a special installed test apparatus. The rotary vane compressor was indicated in particular manner from the rotor. By this means a representative p-V diagram could be obtained for one compression cycle. These results were used for verification of the simulation model of compression. Additional results of another simulation program useful to calculate friction forces, losses and bearing capacities under the vane tip could be checked. A detailed description of measurement techniques will be made. Moreover a discussion of parameters which were found to be of significant influence on a theoretical optimized rotary vane compressor shall be presented.

SPECIFIC CHARACTERISTICS OF COMPRESSOR
The rotary vane compressor this paper deals with is designed as a two flow rotary machine /1/. Besides possible application in the fields of electromotor driven heat pumps this compressor chiefly is thought to be installed in automotive air conditioning plants. The compressor is designed for various displacement volumes at a range between 120 · 10^{-6} and 170 · 10^{-6} m^3.

With special regard to this response the rotary vane compressor had to be developed as small as well as efficient as possible in competition to other usual air conditioning compressors installed for this purpose. Advantages of a rotary machine in contrast to conventional reciprocating compressors are to be seen in the improved momentum- and delivery characteristic. The specific compressor discussed in this paper is equipped with leaf-type valves at the discharge side in contrast to other common rotary vane compressors. By this a certain capacity controlled running of the compressor is possible irrespective of a built in pressure ratio.

For this special application a two flow rotary vane compressor fitted with four vanes was thought to be the best solution (Fig. 1). Because of minimized number of vanes small mechanical losses are to be expected.

Volumetric efficiency of the compressor will be mainly influenced by throttling losses in the region of pressure valves as well as by leakage losses. Losses in the region of suction channels can be avoided by sufficient dimensioned flow areas. As this special compressor is not designed with discharge valves variable working conditions are possible without losses as caused by a fixed built-in pressure ratio. Furtheron a capacity control by means of speed regulation is possible. Additional because of special constructive design it is possible to retract two vanes in pair into the slots so the four chamber compressor is changed to a two chamber one. Thus a further effective capacity

Figure 1. Sectional view of the rotary vane compressor
A higher number of vanes would provide a much better delivery characteristic but besides of increased friction losses much higher production costs must be expected. The oil sump of the rotary machine is arranged within the discharge chamber. Caused by available pressure difference the oil is transported via slots to the rotor and to the compression chamber. Thus lubrication of vanes is provided. The refrigerant oil mixture is transported afterwards via discharge bores to the oil separator integrated into the compressor housing. From here the oil is lead to the oil sump. Because of this construction no oil pump is necessary.
control is given. The outer cell contour of the actual compressing chamber is described by a specific mathematical function. By means of a complex variable

\[ Z = x + i \cdot y \quad (i^2 = -1) \]

This function is defined by the following equation /1/:

\[ Z = R \cdot e^{(i \cdot k_1 \cdot \kappa)} \cdot L_2 \cdot e^{(i \cdot \lambda_2 \cdot (k_1 + l) \cdot \kappa)} \cdot D_3 \cdot e^{(i \cdot (k_1 + 1) \cdot (k_2 + 1) \cdot \kappa)} \]

\[ \text{Eq. 1} \]

This equation involves the independent variable \( \kappa \) and other variables like \( R_1, L_2, D_3, \lambda_2, k_1 \) and \( k_2 \). Because of demand the cylinder contour has to fit in a harmonic function where the maximum results of polar radii are situated on the symmetric lines. The symmetry of function is of second order. Thus the only changeable parameter will be \( R_1 \) when radius of rotor and maximum vane lift are given. This variable is optimized in sense of minimized mass forces generated by minimum relative acceleration and minimized angle between acceleration and polar radius. Because of even number of vanes complete balance of gas forces is obtained. This is given because for each position of the rotor two comparable compression chambers are opposed. For this reason a taper bore mounted rotor can be installed in this rotary vane machine.

ANALYTICAL INVESTIGATION ON THE ROTARY VANE COMPRESSOR

The above described compressor was tested and theoretically optimized at the University of Hannover. Measured performance data and therefore used test rig assembly is not discussed more detailed in this paper. Common performance measurements were made which have to be carried out with special regard to well known standards (ISO 917).

![Figure 2. Plan of test rig](image)

More detailed the indication of the rotary vane compressor will be discussed as well as evaluation of measured p-V diagrams. The theoretical part of this paper will deal with friction and friction losses as caused under the vane tip. The model used here is based on earlier published simulation models by Kruse /2/ and Platts /3/. Furthermore throttling- and leakage losses in the region of discharge valves were examined. Therefore a compressor simulation model presented by Soedel /4/ at the 1972 Compressor Technology Conference was modified and adjusted for the rotary compressor this paper deals with.

INDICATION OF ROTARY VANE COMPRESSOR

Indication of piston machines is thought to obtain information about compression work. Knowing this compression work a statement of isentropic efficiency can be given as well as of mechanical efficiency. In contrast to usual examined reciprocating compressors a particular problem has to be solved in this case. The compression chamber cannot be indicated from a stationary point since the control volume is rotating with respect to the rotating rotor. To obtain a continuous p-t diagram by use of only one pressure pick up obvious will not be possible.

The problem can be solved by installing more than one pressure pick up into the stationary parts of the compressor.

By this a complicated generation of a p-t-diagram is a disadvantage because of necessary overlapping of the individual pressure signals. Therefore discussions had been made to install pressure pick ups into the rotor of the rotary compressor /5/. Because of the low number of vanes two pressure pick ups have to be applied in this particular case. Therefore measurement of one complete compression cycle was possible.

![Figure 3. Sectional view of the rotor with installed pressure pick ups](image)

To avoid extension of clearance volume of the compressor by application of pressure pick ups into the rotor employment of special adapters was necessary. For this purpose the installed pick up was connected to the indicated cell by a minimized borehole (fig. 4) layed out with regard to the frequency by using the Helmholtz resonator model.

Further at a range of 1000 rpm to 7000 rpm no influence of rotational speed on the measured signals could be found in test runs. These tests were run under unloaded conditions.
Figure 4. View of the rotor and adapter with pressure pick up

By means of an extended rotor shaft measured test data were transmitted from the rotor to a mercury transposer. From there these data could be picked up from stationary tools.

Figure 5. Arrangement to transmit measured signals

The mini pressure pick ups were fitted into the rotor next to the leading and next to the trailing vane. Hereby the leading pick up will register complete suction period and beginning of compression. The trailing one will register signals from the complete compression and discharge period. Supposing those two signals a complete p-t diagram can be obtained.

Figure 6. Measured p-t diagrams

The trailing pressure pick up registrates gaspulsations, what are not measured by the leading pick up. Because of the large amount of oil in this region high damping of the measured signal in case of the first pressure pick up must be responsible for this effect. Obviously these gaspulsations are caused by reexpansion after the leading vane has passed this discharge valve region. To state the registrated signals measured by the leading pressure pick up a stationary piezo quarz was fitted to the rotary vane compressor.

EVALUATION OF P-V DIAGRAMS

Because of known function what defines the cylinder contour of the rotary vane compressor the cylinder volume can be related to different rotor positions. Hereby it is possible to transform the measured p-t diagram to a p-V diagram.

Figure 7. p-V diagrams obtained at several speeds
The planimetered diagram divided by the swept volume of one camber enables to predict values of indicated and mechanical efficiency for the compressors.

MEASUREMENT OF VALVE-LIFT

To control valve lifts of the reed type valve installed at the discharge bore contactless inductive pick ups were used. Disadvantage of this measure technique is a non linear signal what has to be transformed. The application of other measure techniques were found to be less favorable for the following reasons. The capacitive method cannot be chosen since dielectric of refrigerant oil mixture what especially must be expected in the discharge bore region is not known. Therefore no representative results can be expected. An attempt to measure the discharge valve lift by use of strain gauges failed. Measured signals were found not to be representative because of superposed disturbances caused by valve reed impact, valve twisting etc. Non active arrangement of inductive pick ups was realized. To eliminate the influence of temperature the nonactive pick up was applied next to the active one within the compressor housing.

THEORETICAL INVESTIGATIONS WITH RESPECT TO AN OPTIMIZED COMPRESSOR DESIGN

Theoretical investigations concerning this particular rotary vane compressor were made to analyse the vane tip friction and friction losses caused under the vane tip. Furtheron another program was used to examine throttling-, leakage and reexpansion losses in the region of discharge valves.

Optimization of Vane Tip Geometry with Regard to Minimized Friction Losses

Based on the Reynold's equation

\[ \frac{\delta}{\delta x} \left( h^3 \frac{\delta p}{\delta x} + \frac{\delta}{\delta z} \left( h^3 \frac{\delta p}{\delta z} \right) \right) = 6 \cdot n \cdot U \frac{\delta h}{\delta x} + 12 \cdot n \cdot V \]

\[ \eta - \text{viscosity of lubricant} \]
\[ h - \text{film thickness} \]
\[ p - \text{pressure of lubricant} \]
\[ U - \text{rel. velocity between sliding surfaces} \]
\[ x - \text{coordinate vane thickness} \]
\[ z - \text{coordinate vane width} \]
\[ V - \text{rectangular velocity} \]

which describes the pressure profile, created in the lubricant, a simplified model can be diverged as follows:

\[ \frac{\delta}{\delta x} \left( h^3 \frac{\delta p}{\delta x} \right) = 6 \cdot \eta \cdot U \frac{\delta h}{\delta x} \]

Hereby the pressure profile caused in the lubricant is described by a two dimensional model assuming stationary conditions. A flow of lubricant in z-direction was neglected in this case. This assumption can be made because of large ratio of vane width to vane thickness \((W/W_{th})\). The non stationary term of eq. 2 was neglected as lubricant flow is to be assumed as laminar. The influence of inertia forces is of second order compared to friction forces. Solving eq. 3 the load capacity of lubricant under the vane tip can be estimated. Associated friction forces can be analyzed, where Newton's law of shearing stresses must be applied.

The thickness of lubrication film \(h\), which is used in eq. 2 and 3, must be described as function of vane compressor, the wedge contur will change as function of rotor position. Thus a mathematical modelling of actual wedge contur is necessary.

Based on a theoretical model /3/ vane tip geometry and cylinder contur can be seen as sliding subjects with different shaped radii. The vane tip radius \(R_v\) remains constant and the actual radius of curvature of the cylinder wall \(R_c\) will change with respect to associated rotor position. By means of so defined radia the wedge contur can be described by the reduced parameter \(r\):

\[ r = \frac{R_v}{R_c} \]

Parameters describing the wedge profile are to be defined as follows:

\[ \gamma = \frac{\gamma}{Z} \]

\[ \theta = \pi - (\gamma_1 - \gamma_p) \]

\[ \gamma_p - \text{gradient of tangent at the cylinder wall} \]

\[ \gamma = \arctan \left( \frac{\delta x}{\delta y} \right) \]

The width of lubricant film \(t\), which can be described as follows will change with respect to vane position. Therefore it must be seen as a further parameter influencing the friction losses.

\[ t = R_v \cdot \sin (\alpha_3 + \delta) \]

\[ \theta = \frac{\pi}{2} - \phi \]

\[ AA = (R_v + h_0)^2 + Z^2 - 2 \cdot (R_v + h_0) \cdot Z \cdot \cos (\gamma) \]

\[ R_v - \text{vane tip radius} \]
\[ h_0 - \text{minimum thickness of lubrication film} \]
\[ Z - \text{polar radius} \]

Using

\[ \alpha_1 = \arcsin \left( \frac{E_x}{AA} \right) \] and \[ \alpha_2 = \pi - \arcsin \left( \frac{Z}{AA} \cdot \sin (\gamma) \right) \]

requested angle \(\alpha_3\) is defined by

\[ \alpha_3 = \pi - \alpha_1 - \alpha_2 \]

Thus the width of lubrication film will be

\[ t = R_v \cdot \sin (\alpha_3 + \delta) \]

\[ \delta - \text{opening angle at vane tip} \]

The minimum film thickness of the lubrication film has to be iterated with regard to a demanded accuracy:

- load capacity of lubricant for any value of \(h_0\) must be calculated.
An earlier study dealt with arrangement of vanes in the rotor. This optimization was made to achieve minimum acceleration-, load- and momentum characteristics. Therefore only parameters describing the vane tip geometry (R_y - vane tip radius, EX - eccentric position of the center line of vane tip radius, W_v - width of vane) should be optimized with respect to minimum friction losses. These parameters only can be varied within given restrictions. The vane tip radius must not be smaller than half of the vane thickness. Otherwise no uniform curvature can be achieved. On the other hand the vane tip radius must be smaller than curvature of the cylinder wall for any rotor position. An opening wedge and therefore a badly affected pressure buildup in the lubricant would be the consequence in this case.

The limit of the eccentricity EX must be discussed with regard to vane tip radius R_y and the positioning of vane slots in the rotor. For the same reason as shown out before the sum of half vane thickness additioned by the amount of noncentric position of slots must be smaller than the value for eccentricities EX. On the other hand summed values of the amount of noncentric position of the slots and of the vane tip radius must be greater than the sum of amount of noncentric position of the slots plus half vane thickness. Otherwise no uniform vane tip geometric is possible. The minimum vane thickness is limited by manufacturing requirements. Otherwise when the thickness is enlarged undesired increasing mass forces are to be expected. Theoretical analysis was made for rotor speeds in a range between n = 1000 and 7000 rpm. Results ware shown in fig. 10.
Each parameter was varied whereas the other two remained constant as given by the actual compressor construction. The most significant influence on the friction force is obtained by enlarging the eccentricity \( EX \). This statement already was made in another paper \( /5/ \) published by Kruse at the 1980 Purdue Compressor Technology Conference, dealing with rotary vane compressors in general.

Decreasing the vane tip radius will minimize the friction force further. The improved lubrication wedge under the vane tip will be responsible for this effect. But as visible the influence of vane tip radius is not as significant as that one shown before.

Minimizing the vane thickness must not result into improved friction forces as not show here in a diagram. Although decreased mass forces are obtained friction losses possible will be more unfavorable because of a poor shaped lubrication wedge. As result can be said enlarged ratios of vane thickness to vane tip radius \( (V_{TH}/R_v) \) and therefore optimized eccentricity \( EX \) will show optimal lubrication conditions under the vane tip.

A comparison of simulated mechanical efficiencies at the vane tip and measured overall mechanical efficiencies shown in fig. 11 states the validity of the simulation model.

Figure 11. Comparison of measured and simulated mechanical efficiencies

Additional mechanical losses caused elsewhere in the compressor are responsible for the different levels of the curves. Nevertheless this figure shows the overwhelming portion of vane tip friction compared to the overall friction losses, especially with increasing rotor speeds. Therefore a optimization of vane tip lubrication can contribute remarkably to minimize overall friction of a rotary vane compressor.

Optimization of the Discharge Valve Geometry

An analytical study of losses in the discharge valve region was made by use of a compressor simulation program what was adjusted to this special compressor. This program was diverted from a simulation model, published by Soedel \( /4/ \) at the 1972 Purdue Compressor Technology Conference, describing a common rotary vane compressor. The working gas \( R_12 \) in this case is viewed as an ideal gas. One compression cycle is defined as follows:

\[
\theta_{S,A} < \gamma_1 < \theta_{D,A} + \frac{\pi}{2}
\]

\( \theta_{S,A} \) - opening edge of suction channel
\( \theta_{D,A} \) - closing edge of discharge bore
\( \gamma_1 \) - rotor position with regard to the leading vane of the viewed compression chamber

Suction period will be limited by

\[
\theta_{S,A} < \gamma_1 < \theta_{S,E} + \frac{\pi}{2}
\]

\( \theta_{S,E} \) - closing edge of suction channel

and the discharge period by

\[
\theta_{D,A} < \gamma_1 < \theta_{D,E} + \frac{\pi}{2}
\]

\( \theta_{D,A} \) - opening edge of discharge bore

Most of the input data for the program must be estimated by adjusting simulated p-V diagrams to measured ones.

Figure 12. Measured and calculated p-V diagrams and valve lift curves

Furtheron an adjustment of calculated valve lifts is necessary to assure a correct simulation of the valve behaviour. Here this was done for an operation point of \( n = 3000 \text{ rpm} \) and a pressure ratio of \( \text{p}_c/\text{p}_o = 13,5 \text{ bar}/1,82 \text{ bar} \). Worth mentioning is the high damping
factor obtained. Because of the high amount of oil in the discharge valve region calculations had to be run with a factor of \( D = 0.5 \). Using a factor of \( D = 0.1 \) which was found for a free damped valve plate no corresponding results could be obtained.

Fig. 13 shows compared simulated and measured values of isentropic efficiencies at a range of rotor speeds between \( n = 1000 \) and \( n = 7000 \) rpm. Sufficient correlation is visible. From this figure it can be seen that the simulation model was adjusted to the rotary compressor for \( n = 3000 \) rpm. An influence of speeds on this adjustment is obvious since diverging results are achieved for other speeds.

![Figure 13. Measured and calculated isentropic efficiencies](image)

Leaving the compression chamber the refrigerant has to pass three different flow areas:

1. Area between rotor and cylinder wall
2. Area in the discharge bore
3. Area under the reed valve

To obtain enlarged flow areas between rotor and cylinder additional grooves are cut into the cylinder wall next to the discharge bores. The geometry shown in fig. 14 describes these grooves. Of course these grooves will provide improved flow areas. But on the other hand reexpansion and related losses will be affected. The larger the grooves the larger the discharge volume will be from were compressed gas can reexpand. Furtheron the point where reexpansion begins will be reached earlier when the grooves are not positioned proper.

Since the smallest flow area will be responsible for the delivered mass flow equalized and optimized flow areas are desirable.

![Figure 14. Discharge bore geometry](image)

To value the compressor the compression work

\[ W_i = \int p \cdot dv \]

or the specific work

\[ w_i = \frac{W_i}{m_R} \]

must be estimated. Losses caused by reexpansion can be defined as follows:

\[ W_L = m_L \cdot \Delta h \]

Mainly two mass flows causing losses must be viewed:

1. the leading vane reaches the region of discharge bores, reexpansion of compressed gas will start
2. because of manufacturing possibilities a minor flow area will remain at the sealing slot, so caused by the pressure difference a leakage mass flow must be present.

Therefore the following parameters influence the energetic characteristics of this rotary vane compressor:

- valve bore diameter
- volume in the discharge bores
- maximum valve lift
- arrangement of grooves next to the discharge valves
- natural frequency of valve reed

For the compressor examined here it was found, that by halving the discharge volume (fig 15) specific compression work can be improved up to 2%.

An enlarged valve lift with regard to equalized flow areas only resulted in an improvement of about 0.5% specific compression work and this only was found at rotor speeds above \( n = 3000 \) rpm. This is because the valve does not open completely at lower speeds.
For this reason the natural frequency of the valve reed was changed in order to obtain an improved characteristic (fig. 16). But only small improvements were found at low speeds. Because of non proper closing of the weakened valve reed especially at high rotor speeds increased losses were dominant.

Changing the geometry of grooves was only of small success in this case. A later start of reexpansion could be reached by changing these parameters. Therefore compression losses could be minimized a little bit. But no significant improvement of specific work could be obtained for this specific compressor. Nethertheless the importance of proper arrangement of these grooves must be pointed out in general.

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CONCLUSION

A two flow rotary vane compressor designed with special regard to automotive air conditioning applications was examined experimentally and theoretically. By use of special measuring techniques, indicating the compressor from the rotor side, experimental data were found. Based hereon representative simulation models for theoretical optimization could be established. Using two independant models on the one hand lubrication conditions under the vane tip could be optimized and on the other hand throttling and leakage losses in the region of discharge valves were minimized.