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NUMERICAL CALCULATION OF PNEUMATIC LOSSES IN A ROTARY VANE COMPRESSOR

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Abstract: *Positive displacement pneumatic compressors rely on suitable sealing in order to achieve the required pressure and work efficiently. Most rotary vane compressors rely on a thin film of hydraulic oil that forms between the vanes, stator, rotor, and the end plates. The oil film has the role of sealing to minimize air losses, but also to reduce the wear between the components. If the clearance between the different components is too large a significant amount of air escapes and the compressor does not achieve sufficient outlet pressure. If it is too small, machining costs increase and it is also possible for the vanes to get locked and damage the compressor. The paper calculates the amount of air lost through internal leakage in a vane compressor based on the clearances between the components. An algorithm that uses the equation for the flow of a fluid through a straight tube of constant cross-sectional area is proposed and proprietary software is developed for a numerical transient simulation.*

Key words: *pneumatic rotary vane compressor, compressor sealing, pneumatic loss, leakage, numerical simulation.*

1. INTRODUCTION

Compressed air plays an important part in most industries where it can be used as energy air or active air. Energy air is often required for storing and transmitting energy in order to do mechanical work in industrial equipment. It can also be used as active air where it is a direct part in the process and is in contact with the product. Active air is used in industries such as chemical, pharmaceutical, food and beverage, aeration and mixing, medical applications, etc.

Vane compressors are an important part of the positive displacement compressor category. In the industry the most used are single stage and usually provide air at 8, 10 or 12 bar. They normally use a lot of oil for internal lubrication and sealing, but recover it and offer almost oil-free air. The air can be further dried and cooled if necessary. Even though their purchase cost is higher, they offer many advantages over traditional piston compressors. Compressors of this type can work non-stop to provide a constant mass of air that has small pressure fluctuations, and they are also characterized by the fact that they need little maintenance, requiring only oil and oil filter changes. They have a long lifespan, many of them exceeding 100 000 operating hours.

Sealing is accomplished in one of two ways, depending on the type of compressor. Normal use vane compressors use a film of oil between all moving components. The viscosity of the oil is very important so that it not only lubricates the components well, but also seals the space between them. The play between the parts of the compressor is usually calculated based on thermal expansion and the oil viscosity is chosen based on the play.

The other type of compressors, the dry type, doesn’t use oil for sealing since this would contaminate the resulting air and it would render it unsuitable for certain uses (medical, food etc.). These compressors use graphite vanes which release small particles as they wear. The graphite absorbs water from the atmospheric air and works as a sealing paste between the components. The air can be dehydrated following compression, but the air that is taken in should have a certain amount of moisture for this method to work.

Paper [1] investigated the leakage in a lubricant free revolving vane compressor. Even though this type of compressor is different than the rotary vane type presented in the current paper, it shares many similar aspects. The paper identified 4 types of leakage of which 3 are common with the rotary vane compressor: leakage around the vane tip, vane end face leakage (both end faces) and rotor end face leakage (both end faces).

They calculated the leakage for this type of compressor using the mass flow rate through an orifice equation and used a CFD software to confirm the calculations. The mechanical design was used to develop a prototype for bench tests.

The measurements showed very poor compression ratios and volumetric efficiency [1]. However, the predicted and the measured mass flow rates were within satisfactory tolerance of ±15%. The volumetric efficiency of their prototype ranged between 17% and 45%, depending on the speed and flow of air. This was due to the large clearance gaps at the vane and rotor end-faces .

Paper [2] analyzed the leakages in rotary vane refrigerant compressors. They identified multiple leakage paths including through the lubrication system and past the vane edges. They determined that the leakage past the vane tip was negligible. In order to calculate the flow past the vane edges it used the mass flow through an orifice equation.

2. AREAS OF PNEUMATIC LEAKAGE FOR ROTARY VANE COMPRESSORS

Vane compressors work as most other compressors with a certain amount of leakage and the smaller this amount is, the better the pneumatic efficiency of the machine is. The pneumatic efficiency is the largest contributor to the overall efficiency of the machine. The total energy efficiency of a vane compressor consists of many factors including, but not limited to: the aforementioned pneumatic efficiency, the friction between the vanes and the rotor, the friction between the vanes and the stator, the friction of the bearings and the efficiency of the electric motor.

In order to assess the leakage, it is important to understand where it takes place. Each chamber of the compressor works like a mini-compressor by itself. Its volume grows when passing in front of the intake, it reaches its maximum volume at the end of the intake and then starts to decrease and compress the air. Towards the end of the compression phase, it meets the exhaust port and delivers compressed air. It reaches its smallest volume at the end of the exhaust phase. The leaks relevant to the machine’s efficiency occur during the compression phase.

Each chamber of a vane compressor is defined by the leading vane, the trailing vane, the stator towards the outside, and the rotor towards the inside. After the vanes are machined and installed in the rotor slots, there is a run-in period when the vane's upper surface and bottom surface get a small amount of wear and start to develop an excellent seal above and below. If the machining tolerances are correctly selected, the top and bottom of a vane can be considered to not lose any air.

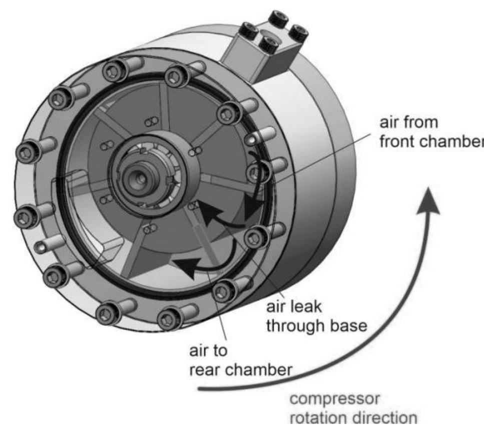

Fig. 1. Diagram of leakage flow

However, there needs to be some play between the vanes’ end faces and the stator, or otherwise the rotor would lock due to thermal expansion generated by the heating of the compressed air. This same play is also accounted for by the rotor to avoid the same problem of jamming. This means that air leaks between adjacent chambers through the end face of the part of the vane that sticks out of the rotor, but also through the bottom of the chamber, namely between the face of the rotor and the inner face of the housing, according to figure 1. The air that escapes through the bottom of the chambers flows into this space, from where it mostly escapes to the chamber or chambers that have access to the intake and then flows outside to the atmosphere.

3. ALGORITHMS FOR APPROXIMATING PNEUMATIC LOSSES

One of the most comprehensive solutions for calculating the pneumatic losses is using a 3D model of the compressor and CFD software. A time-dependent simulation should be run to see the gradual build-up of pressure in the compressor and the leakage flow paths. However, this is a difficult process since firstly, few CFD software packages can simulate the changing geometry of the compressor with the vanes moving up and down. Secondly, the air that escapes the compressor's chambers has to go through very narrow passages on the order of 0.01 mm. This requires a very small mesh and combined with the time-dependent character of the simulation would make it very resource intensive in terms of memory and computational power.

A different route was chosen for the approximation of these losses. If a single chamber is analyzed it can be observed that it receives air from the chamber that is in front of it, it loses air to the chamber behind it, and it also loses air through the bottom part, between the rotor and housing. The vanes have the same width as the rotor, but there is some play between the rotor's sides and the sides of the housing. This play is equal on both sides of the rotor and has an important role during functioning: it prevents the sides of the stator from contacting the housing due to thermal expansion. This phenomenon would lead to either the rotor or the vanes locking up and damaging the compressor.

The losses through these 3 areas can be approximated using the equation for the flow of a fluid through a straight tube of constant cross-sectional area:

$$\dot{m} = C_d \cdot A \cdot \sqrt{2\rho \cdot \Delta P} \qquad (1)$$

Where,

m: mass flow rate;

Cd: discharge coefficient;

A: orifice surface area;

ρ: density of air;

ΔP: pressure difference before and after orifice.

Because the rotor and the vanes are constantly moving, the algorithm for calculating the pneumatic losses should be time-dependent. If the time increment used is denoted by Δt then the mass of air can be calculated as follows:

$$m = C_d \cdot A \cdot \sqrt{2\rho \cdot \Delta P} \cdot \Delta t \qquad (2)$$

The mass of air inside the main chamber changes according to the following equation:

$$m_f = m_i + m_{fr} - m_{rr} - m_b \qquad (3)$$

Where,

mf: final air mass;

mi: initial air mass;

mfr: front chamber air mass;

mrr: rear chamber air mass;

mb: bottom leak air mass.

The mass of air from the leading chamber is calculated by substituting the area A with the product of the length of the front vane sticking out of the rotor and the play:

$$m_{fr} = C_d \cdot h_{fr} \cdot play \cdot \sqrt{2\rho \cdot (P_{fr} - P_c)} \cdot \Delta t \qquad (4)$$

Where,

hfr: height of front vane sticking out of rotor;

play: play between vane end face and stator;

Pfr : front chamber air pressure;

Pc: current chamber air pressure.

The mass of air from the trailing chamber is calculated by substituting the area A with the product of the length of rear vane sticking out of the rotor and the play:

$$m_{rr} = C_d \cdot h_{rr} \cdot play \cdot \sqrt{2\rho \cdot (P_c - P_{rr})} \cdot \Delta t \qquad (5)$$

Where,

hrr: height of rear vane sticking out of rotor;

Prr: rear chamber air pressure;

The mass of air from the chamber lost through the bottom part is calculated by substituting the area A with the product of the length of the arc on the rotor corresponding to the size of a chamber and the play:

$$m_b = C_d \cdot L_{arc} \cdot play \cdot \sqrt{2\rho \cdot (P_c - P_{atm})} \cdot \Delta t \quad (6)$$

Where,

Larc: length of arc on rotor between two vanes;

Patm: atmospheric air pressure;

Pc: current chamber air pressure.

The algorithm uses some simplifications in order to calculate the losses. The discharge coefficient Cd is given a constant value of 0.65 for all equations. This value was chosen from a set of well-known values and it corresponds to the discharge coefficient for an orifice with straight edges in a plate. Another simplification is that the density in the equations is the same as that of the chamber that is being analyzed. An assumption is made that the air that leaks out of the chamber through its bottom part goes to atmospheric pressure since it should in the end find its way to the intake side of the compressor.

Since the equation for the flow of a fluid through a straight tube of constant cross-sectional area assumes that the fluid has no viscosity, all the calculations are based on dry air. The algorithm doesn't take into consideration any effects of lubricating oil. This means that the calculated losses are probably larger than in reality, where oil is mixed with the air from a sump or from injection ports.

Choosing an appropriate time increment is important, since the heights of the vanes that appear in equations (4) and (5) change. They are considered relatively constant for a small change in position and consequently in time.

When the chamber reaches the end of the compression phase, the actual mass of air still found in the chamber can be compared with the ideal mass when no losses are taken into account. The ratio between the real and the ideal masses of air at discharge shows the pneumatic efficiency of the system.

4. PROPRIETARY NUMERICAL SIMULATION SOFTWARE

Proprietary software for the numerical simulation of the pneumatic losses of a vane compressor was developed using MATLAB. It takes general geometrical and functional input regarding a specific compressor and calculates the mass of air lost from the compressor's chambers. It calculates the air leakage of the compressor.

The geometrical input data are:

- the radius of the stator;
- the radius of the rotor;
- the number of vanes;
- the thickness of the vanes;
- the eccentricity of the rotor to the stator;
- the length of the rotor;
- the play between the lateral parts of the rotor and the stator.

The functional input data are:

- the adiabatic constant of air;
- the molar mass of air;
- the ambient (atmospheric) pressure;
- the ambient temperature.

The type of simulation is transient and selecting the right time step is very important. If the time step is too large the calculations become inaccurate and if it's too low the simulation time is increased without a significant improvement in accuracy.

In order to decide on a good value for the time step, it can be chosen so that it corresponds to a low angle of rotation of the compressor rotor. If the angle is denoted by α [°] and the speed by n [min-1], then the following equation can be used to determine its value:

$$\Delta t = \frac{\alpha}{6 \cdot n} \quad (7)$$

5. SIMULATION EXAMPLE USING THE PROPRIETARY SOFTWARE

The simulation software was used to calculate the pneumatic losses in a vane compressor that was designed specifically for testing, figure 1.

The compressor's geometric parameters are:

- the radius of the stator: 50 mm;
- the radius of the rotor: 40.9 mm;
- the number of vanes: 7;
- the thickness of the vanes: 4 mm;
- the eccentricity of the rotor to the stator: 8.5 mm;
- the length of the rotor: 54.2 mm;

- the play between the rotor's end faces and the stator: 0.03 mm (on each side).

The functional input data are:

- the ambient (atmospheric) pressure rounded to 100000 Pa;
- the ambient temperature: 293.15 K (20°C).

The time step was chosen to correspond to a very small change in the height of the vanes that are sticking out of the rotor, namely 1.11e-4 seconds, corresponding to a 1° rotation.

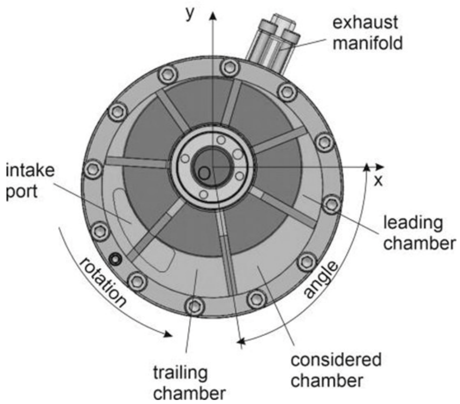

Fig. 2. Angle used in simulation

The results of the simulation are presented in table 1. The angle in the table is the angle between the trailing vane of the considered chamber and axis Ox as shown in figure 2. An angle of -115.7° corresponds to the end of the intake phase and an angle of 64.3° to the end of the exhaust phase. These values are not represented in Table 1 since they correspond to the atmospheric pressure, respectively to the exhaust manifold pressure. The last column is the percentage of air mass that remains in the chamber compared to the initial air mass. Figure 3 shows a graphic of the percentage of air remaining vs. the ideal 100% if no air would be lost.

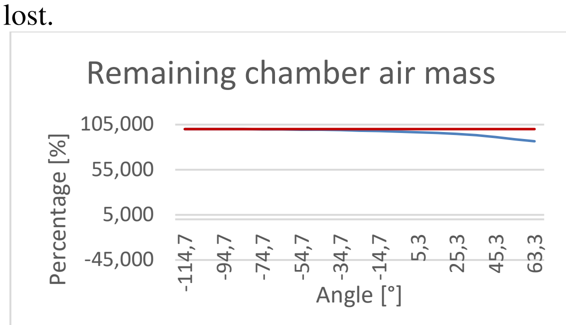

Fig. 3. Remaining chamber air mass

6 CONCLUSIONS

The paper presents an algorithm for determining the amount of leakage from a rotary vane compressor by determining the leakage from a single chamber. Air is considered to be gained from the chamber in front that has a higher pressure and lost to the chamber behind and the bottom of the considered chamber due to the lower pressures of these areas.

The efficiency of the presented compressor is calculated to be 86.4% as a ratio between the mass of air that exits the exhaust port and the mass of air that goes through the intake.

This is a good efficiency since the model doesn't account for the lubricating oil that is usually present in the air and serves to better seal due to its viscosity.

The leakage analysis helps fine-tune the tolerances of a vane compressor in practice and build resilience into the product by making it longer lasting and more energy efficient, both of which are necessary for better sustainability of businesses and resources.

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Table 1

Simulation results

Angle [°]	m_{fr} [°]	m_{rr} [°]	m_b [°]	m_f [°]	Air remaining [%]
-114.7	0.000000127	0.000000128	0.000000281	0.045268878	100.000
-113.7	0.000000163	0.000000166	0.000000428	0.045268447	99.999
-112.7	0.000000192	0.000000197	0.000000574	0.045267868	99.998
-111.7	0.000000216	0.000000224	0.000000719	0.045267141	99.996
-110.7	0.000000238	0.000000249	0.000000865	0.045266265	99.994
-109.7	0.000000257	0.000000272	0.000001010	0.045265240	99.992
-108.7	0.000000275	0.000000293	0.000001156	0.045264067	99.989
-107.7	0.000000292	0.000000313	0.000001301	0.045262743	99.986
-106.7	0.000000307	0.000000333	0.000001447	0.045261270	99.983
-105.7	0.000000321	0.000000351	0.000001594	0.045259646	99.980
-104.7	0.000000334	0.000000369	0.000001740	0.045257871	99.976
…					
-94.7	0.000000435	0.000000526	0.000003231	0.045231670	99.918
-84.7	0.000000498	0.000000660	0.000004792	0.045189556	99.825
-74.7	0.000000537	0.000000785	0.000006462	0.045130477	99.694
-64.7	0.000000559	0.000000903	0.000008287	0.045052977	99.523
-54.7	0.000000567	0.000001016	0.000010327	0.044955085	99.307
-44.7	0.000000561	0.000001126	0.000012657	0.044834157	99.040
-34.7	0.000000544	0.000001234	0.000015383	0.044686641	98.714
-24.7	0.000000515	0.000001341	0.000018651	0.044507720	98.319
-14.7	0.000000476	0.000001448	0.000022674	0.044290769	97.839
-4.7	0.000000428	0.000001556	0.000027769	0.044026493	97.256
5.3	0.000000372	0.000001665	0.000034425	0.043701554	96.538
15.3	0.000000314	0.000001775	0.000043401	0.043296393	95.643
25.3	0.000000269	0.000001872	0.000055850	0.042781939	94.506
35.3	0.000000270	0.000001913	0.000073279	0.042115895	93.035
45.3	0.000000371	0.000001772	0.000096307	0.041244951	91.111
…					
55.3	0.000000538	0.000001201	0.000119368	0.040139101	88.668
56.3	0.000000543	0.000001111	0.000121069	0.040017464	88.400
57.3	0.000000542	0.000001016	0.000122569	0.039894421	88.128
58.3	0.000000532	0.000000914	0.000123848	0.039770192	87.853
59.3	0.000000513	0.000000806	0.000124885	0.039645014	87.577
60.3	0.000000480	0.000000690	0.000125665	0.039519140	87.299
61.3	0.000000429	0.000000563	0.000126172	0.039392834	87.020
62.3	0.000000350	0.000000420	0.000126399	0.039266365	86.740
63.3	0.000000212	0.000000232	0.000126339	0.039140005	86.461

Calcule numerice ale pierderilor pneumatice ale unui compressor cu palete

Compresoarele pneumatice se bazează pe o etanșare adecvată pentru a atinge presiunea necesară și pentru a funcționa eficient. Majoritatea compresoarelor cu palete se bazează pe o peliculă subțire de ulei hidraulic care se formează între palete, stator, rotor și capace. Dacă jocul dintre diferitele componente este prea mare, o cantitate semnificativă de aer scapa și compresorul nu atinge o presiune suficientă la ieșire. Dacă este prea mic, costurile de prelucrare cresc și, de asemenea, este posibil ca paletele să se blocheze și să deterioreze compresorul. Lucrarea calculează cantitatea de aer pierdută prin scurgerile interne într-o supapă de compresor pe baza jocurilor dintre componente. Este propus un algoritm care folosește ecuația pentru curgerea unui fluid printr-un tub drept cu secțiune transversală constantă și este dezvoltat un soft propriu pentru o simulare numerică tranzitorie.

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