Experimental Investigations on Rotary Vane Compressors

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A loss analysis of a rotary vane compressor is taken by measurements and shows the influences of mechanical friction, oil injection and non-adapted pressure ratio.

1. INTRODUCTION

Rotary vane compressors like other types of rotating compressors as compared with reciprocating machines have the advantage of a possible total mass balance, leading to an oscillation free machine of high possible speed. Further as compared with other rotary compressors, simple configurations of the compressor elements are favourable for a machine of low production costs. This type of compressor is often used for air and gas compression purposes, whereas in refrigeration its application is limited mainly for booster purposes in low temperature systems and newly for automotive air conditioning compressors.

The reason for this limited application is the main disadvantage of this type of compressor, namely the relative high needed power input as compared with other types of compressors, which increases remarkably with pressure difference and speed. High pressure difference causes internal leakage and high frictional forces on the sides of the vanes, high speed increases vane tip friction by high centrifugal forces.

Especially when compressing refrigerants, the high pressure differences of normal working fluids as given already by moderate temperature differences between condensing and evaporating temperature show an unfavourable leakage and side force effect. The reason for the leakage in rotary vane compressors is the poor internal sealing mainly caused by the necessary axial gaps on both sides of the rotor. As in other rotary compressors without special sealing elements, like for instance screw compressors, a good sealing can be gained by injection of oil into the machine. This, although reducing the internal gas leakage and compression temperatures can cause again a higher power consumption by the influence of too much oil in the compressor. Further alternating temperatures of cooling and heat rejection lead to different external compression ratios of refrigerant compressors, which in case of valveless machines do not correspond with the internal one as given by the built-in volume ratio, causing so additional power losses when running under non-design conditions.

Today's production methods are more favourable as in the past to minimize especially the leakage effects, so that an increasing application of rotary vane compressors for stationary and automotive refrigeration purposes can be stated. On the other hand today's energy requirements need a thorough optimization of the overall efficiency and its influences for rotary vane compressors in order for them to compete successfully also in this respect with other compressor types.

So for improving the efficiency of rotary vane compressors an analysis of the losses is favourable which shall be illustrated by measurements made on a rotary vane compressor.

2. PURPOSE OF INVESTIGATION

A purpose of the investigation was to find experimentally the reasons for a too high power input of a rotary vane compressor especially under high speeds and loads. The compressor was a testing machine for the investigation of developing problems. It was designed for normal application conditions and had been run under testing conditions up to double the speed and double the load in order to get significant informations about the various mentioned influences on needed power input by testing the compressor under extreme running conditions.

Especially the various influences
- of the amount of injected oil
- of the barrel ring friction
- of the vane tip friction and
- of internal over- or undercompression should be studied.

3. TEST EQUIPMENT

The tested rotary vane compressor was a single flow machine with an eccentric rotor of 158 mm diameter running in a cylinder of 176.5 mm diameter. The rotor was equipped with 16 vanes, each 5 mm thick and produced of a phenolite material. The length of the
rotor and the vanes was 220 mm. In the tests the compressor was run with air. It was equipped with measuring devices for shaft speed, barrel ring speed and compression end pressure in the cells as well as for suction and discharge pressures in the lines. Since the compressor could also be run as an oilflooded machine, an oil separator and accumulator were arranged behind the compressor, as well as an oil cooler between accumulator and oil pump. The normal oil pump was separated from the compressor and driven by an electrical motor in order to vary the oil injection for studying the influence of oil quantities. Further for spare lubrication with very low quantities of oil a separate oil pump with 6 piston elements was installed.

The compressor itself was driven by an electrical DC-motor, fed by a Leonard generator, in order to vary the compressor speed between 500 and 3000 rpm. The torque of the DC-motor could be measured directly, as well as the speed, so that the shaft power of the compressor could be determined easily. With powers up to 50 kW speeds up to 3000 rpm and end pressures up to 8 bar could be achieved with this installation. The end pressures \( p_e \) were measured in the oil separator behind the compressor as well as the suction pressure before the machine. Further the discharge pressure \( p_d \) in the discharge chamber immediately after the opening port of the rotor cells was measured in the machine. For ensuring stable running conditions various temperatures were measured by thermocouples.

4. TEST RESULTS

In the following chapter the test results together with special testing conditions to get these results are presented.

4.1 Influence of Oil Injection

The injected oil into the machine for lubrication, sealing and internal cooling can have different effects, positive and negative on the needed power input. In order to study these effects, three different test series were made.

4.1.1 Minimum Oil Quantities for Spare Lubrication Under Idle Conditions

In order to exclude the influence of the gas compression in the machine the suction port was sealed hermetically. A minimum oil quantity was injected by the piston element pump into the machine in order to ensure only the minimum lubricating conditions. Three tests were run under those idle conditions with three different quantities of injected oil, namely of 0.4, 0.5 and 0.7 kg/h.

Fig. 1 shows in the lower part the input torque \( T \) of the rotary vane compressor under those idle conditions with minimum oil injection in relationship to the speed. It can be seen, that the friction decreases as the oil flow increases indicating certain boundary friction conditions. These conditions were also indicated by a noise of high frequency generated in the machine and vanishing at an oil flow of 0.7 kg/h and more. The torque relationship to the speed shows a torque increasing firstly up to 1000 and secondly above 2000 rpm.

It is possible, that in the first mentioned region the centrifugal force of the vanes is so low that the barrel rings have a certain slip to the rotor speed and their friction is low. Above 2000 rpm the influence of speed on the friction of the barrel rings is increasing remarkably.

4.1.2 Normal Oil Injection for Abundant Lubrication, Sealing and Cooling Under Idle Conditions

In these three test runs the quantities of the injected oil were increased as compared with those for spare lubrication by a factor up to 1000. The test results in the same Fig. 1 shown by the three second lower curves indicate also boundary friction conditions at speeds below 1500 rpm where the friction decreases again as the oil flow increases. At higher speeds above 1500 rpm a reversed tendency can be seen indicating more fluid friction. The torque in relationship to the speed increases from a lower speed on if the injected oil quantity is higher. The overall result of the tests under idle conditions is that by increasing the quantity of injected oil into the machine by a factor of approximately up to 1000 the mechanical friction is roughly up to doubled.

4.1.3 Normal Oil Injection for Abundant Lubrication, Sealing and Cooling Under Load Conditions

Both upper curves of Fig. 1 show the additional
influence of gas compression in the machine and a further increase of oil injection. Because of a higher oil pressure in the oil accumulator the injected oil quantities were increased by the help of this pressure under the same pumping conditions of the oil pump as before. Whereas the curve for an end pressure of 3.3 bar was achieved with an open suction port the results for 3.0 bar were measured by a fault with a closed suction throttle.

It can be seen that at lower speeds the torque of both lines is approximately equal whereas at higher speeds and throttle effects a tendency like at the idle curves can be seen where the influence of hydrodynamic friction increases the torque in the same manner only at a higher level, caused by the still remaining gas compression effect. Under non throttled conditions the influence of the compressed gas also at very high speeds results in a remarkable steeper increase of the torque with relationship to the speed.

4.2 Influence of Circumferential Friction

As from the results in Fig. 1 could be seen the mechanical friction of the machine under idle conditions depends remarkably on speed and oil quantity. Part of this friction is the circumferential friction of the barrel rings as already mentioned. The question is if this friction occurs only at the outside of both rings or if a great deal of this friction also occurs between vane tips and rings. The answer can be given if the speed of the barrel rings in respect to that of the rotor itself is known.

4.2.1 Rotation of Barrel Rings

In order to investigate the question of the rotational speed of the barrel rings which support the vanes it was measured in the following way. The pressure equalizing drilled holes in the middle of the rings were used to count them by inductive pick up during ring rotation. Fig. 2a shows the arrangement of the inductive pick up installed in adapters sealed against the cooling chamber of the machine.

Two results of the speed measurements at the barrel rings are shown in Fig. 3 for two different running conditions.

![Ring speed vs Rotor speed graph](image)

**Fig. 3:** Barrel ring speed versus rotor speed under idle and load conditions

4.2.1.1 Normal Oil Quantity, Idle Conditions

The unbroken line shows a certain slip at lower speeds and a higher barrel ring speed at higher rotor speeds. The slip effect seems to be caused by the lower centrifugal forces of the vanes on the barrel rings. The higher ring speed as compared with the rotor speed is a result of the higher circumferential mean velocity by their high normal forces and static friction of the vanes causing the rings to rotate according to this circumferential velocity. Other results under idle conditions not shown here indicate that at speeds below 600 rpm the ring has the tendency not to rotate because of low centrifugal force and low static friction of the vanes.

4.2.1.2 Normal Oil Quantity, Load Conditions

All results with pressures between 3 and 8 bar show approximately the same tendency like the dotted line in Fig. 2, i.e. that the barrel rings rotate nearly with the same speed as the rotor itself. A very small tendency not shown here indicates that a certain slip occurs when the end pressure increases.

4.2.2 Vane Tip Friction

The results concerning the torque shown in Fig. 1 were taken with rotating barrel rings. The circumferential friction enclosed in those results is mainly a friction between barrel rings and housing not between vane and rings, according to the results of ring speed measurements.
Modern compressors often are designed without barrel rings because of low production costs. The question is if vane tip friction in the housing directly produces higher friction than the rotating barrel rings. In order to investigate this question two test series with rotating and fixed barrel rings were made. For this purpose the barrel rings were fixed according to Fig. 2b. The clearance between a pin of the fixing bolt and the barrel ring hole was chosen so that the barrel ring could adjust itself in its housing clearance axially and radially without the possibility of rotating. The results of the torque measurements without and with this equipment are shown in Fig. 4 in relationship to speed and load in their whole possible ranges.

\[ T = f(n, p) \]

\[ p \text{ [bar]} \]

\[ n \text{ [1/min]} \]

It can be seen that at a speed above the indicated line between 1000 rpm at low and 2000 rpm at high loads fixed barrel rings produce higher vane tip friction as barrel ring friction. The reason for this behaviour are the increasing centrifugal forces of the vanes by which they are pressed against their fixed counterparts causing with this high normal force a high dynamic friction. At lower speeds on the left side of the indicated line the rotating barrel rings themselves here produce higher friction than the vanes on fixed rings. Their moderate centrifugal force and friction in this region at a smaller diameter than the ring outside diameter induces here lower friction losses. It can be seen that for higher rotational speeds rotary vane compressors should have barrel rings if possible. If the production costs are too high for this device a very careful optimization of the vane tip friction should be made. Fig. 5 shows an optimization of the vane tip force achieved by a computer programme. It can be seen that the variation of the vane eccentricity has a much larger effect on vane tip friction than the variation of the tip radius.

4.3 Influence of Compression Ratio

The power input of a rotary vane compressor is influenced in the case of a valveless machine by the external compression ratio as for instance for refrigerating compressors given by evaporating and condensing temperatures if it does not coincide with the internal one as given by the built-in volume ratio. The effect of this influence on power input and isentropic efficiency can be remarkable if internal and external pressure ratio disagree obviously. The internal pressure ratio can be calculated approximately from the built-in volume ratio and the isentropic exponent of the working fluid, if the polytropic exponent is not known from tests. The latter is influenced by leakage effects, external and internal cooling of the working fluid. A remarkable internal cooling occurs by oil injection into the cells of a rotary vane compressor influencing so together with the leakage effects the deviation between isentropic and polytropic exponents. The influence of the oil itself in the cells by taking a certain portion of the cell volume can be neglected in respect to the pressure ratio. At the mean oil injection quantity of about 1200 kg/h for the test compressor the oil volume is only about 1% of the volume of the cells at the end of compression. So for a built-in volume of 2.5 the compression ratio with oil is 3.8 instead of 3.7 without oil and therefore not very important in respect to the power input. Therefore the deviations of the external pressure ratio from the internal one as caused either by alternating external running conditions or by not well enough adjusted internal pressure ratio because of unknown polytropic exponent or a fault of the layout may be the reason of an unnecessary high power input expressed in a too low isentropic or adiabatic efficiency.

1) The calculations for another internal research project were made by Dipl.-Ing. H. Lindemann
4.3.1 Adiabatic Efficiency

From the test results in Fig. 5 gained with rotating barrel rings the adiabatic or isentropic efficiency of the compressor was calculated. Fig. 6 shows the adiabatic efficiency in relationship to speed and end pressure which is suction pressure of 1 bar also represents the pressure ratio of the machine including oil separator. It can be seen that the built-in pressure ratio of the machine must be in the region of 3 where the highest efficiencies can be stated.

![Graph showing adiabatic efficiencies versus speed and end pressures.](image)

Fig. 6: Adiabatic efficiencies versus speed and end pressures.

In order to demonstrate the influence of different internal and external pressure ratios the compressor was tested up to external ratios of 8 where the greatest disagreement and the lowest efficiencies can be stated. In order to analyse more thoroughly the effect of different internal and external pressures the compression end pressures in the cells were measured in respect to the discharge pressure. For this purpose a piezo pressure pick-up was placed according to Fig. 7 so in the cylinder wall that at first after passing of a vane the built-up pressure could be measured at the end of compression in a still closed cell. This pressure $p_{c}$ is built up partly by the internal gas compression in the cell and partly by internal leakage flow from the high to the low pressure cells. Secondly, a pressure measurement before total opening of the cell to the discharge chamber can be made when a small triangular opening gap allows a soft pressure equalization between cell and discharge chamber. In case of a higher discharge pressure $p_{d}$ than the compression end pressure $p_{c}$ the cell pressure is raised by back-flow through this gap into the cell up to a certain value $p_{c_{max}}$ before a total opening of the cell occurs.

![Diagram showing installation of pressure pick-up.](image)

Fig. 7: Installation of pressure pick-up.

Fig. 8 shows the measured pressures in the cell $p_{c}$ and $p_{c_{max}}$ in comparison to the discharge pressure $p_{d}$ which is higher than the end pressure $p_{e}$ because of the discharge line and oil separator losses.

![Graph showing generated pressures versus speed.](image)

Fig. 8: Generated pressures versus speed.
Therefore in relationship to the speed the discharge pressure $P_d$ shows a parabolic behaviour in respect to the end pressure $P_e$. Because obviously the designed internal pressure ratio is lower than the external one, at lower speeds a remarkable backflow from the high pressure side into the closed or just opening cell occurs increasing here the pressures $P_c$ and $P_{c_{\text{max}}}$, This effect is more remarkable at lower than at very high speeds causing so the decrease of the adiabatic efficiency at lower speeds and high end pressures.

From these results it can be stated that first the disagreement between internal and external pressure ratio may cause a great loss in efficiency as well as in the case of a rotary vane compressor possible remarkable great back flow through the leakages. So at rotary vane compressors leakage inspite of oil sealing by oil injection can cause high power losses. In order to demonstrate the relative size of the investigated losses under the pressure ratio of 8 the energy flow diagram is presented in Fig. 9.

![Fig. 9: Energy flow diagram for the most unfavourable running conditions](image)

It shows about 28% losses caused by oil squeezing effects, 16% caused by the mechanical friction under idle condition and spare lubrication and 42% caused by the influence of compression effects consisting of as well load effects on friction as also additional losses by the mentioned undercompression. The difference between the total adiabatic and the adiabatic power is caused by the pressure loss in the discharge line and the oil separator which is contemplated as a necessary part of the whole compressor system.

A similar energy flow diagram for a pressure ratio of 3 where internal and external ratio correspond nearly shows oil losses of about 38%, friction losses of 22% and compression losses of only about 5%. The great amount of the losses caused by the injected oil and the mechanical friction indicates that besides of the compression losses more easily to be avoided by a good adaption of internal and external pressure ratio a very thoroughly made investigation on rotary vane compressors is necessary as well as theoretically as also experimentally in order to minimize the losses and raise the efficiency to a value by which a good competition to other compressor types is possible. For a careful experimental investigation not only the pressure measurement at the end of compression is important but also the determination of a whole $p,v$-diagram. For this purpose a certain number of pressure pick-ups along the part of the cylinder wall where compression occurs is necessary. The pressure signals of those pick-ups have to be correlated to each other and to the alternating volume of the cell in order to get a complete $p,v$-diagram. For experimental purposes this is a difficult evaluation of the pressure volume diagram which needs a lot of time and computing efforts. A better way seems to be to install pressure pick-ups in the rotor which can travel with it and measure the pressure in the cells when their volume is alternating. This solution has been chosen now at our investigations on a small automotive air conditioner refrigerant compressor (Fig. 10). 2)

![Fig. 10: Installation of pressure pick-ups in the rotor of an automotive a/c compressor](image)

One pressure pick-up immediately behind and another before the corresponding vane ensure that the total suction compression, discharge and reexpansions phases are picked up by the two pressure transducers despite if a gap sealing between cylinder and rotor is located timewise between them both. The first results have shown that rotational speeds up to 7000 rpm have no influence by possible centrifugal acceleration on the pressure signal and a good pressure signal can be measured during running.

2) The experimental work on this internal research project was made by Dipl.-Ing. H. Kaiser
SUMMARY

The experimental investigation on a rotary vane compressor has shown that remarkable power losses can occur by not adjusted compression ratios, by oil injection and mechanical friction. To decrease this mechanical friction especially at higher speeds barrel rings are more favourable than a direct running of the vanes in the housing. The losses caused by oil effects are in such an order that the amount of injected oil should be minimized as possible where to such an amount that the losses are minimized too. For an as high as possible efficiency it is necessary to adapt internal and external pressure ratio in the case of an valveless machine as carefully as possible or to use discharge valves.