

The influence of slide valve on the thermodynamic efficiency and bearing forces of a refrigeration helical twin screw compressor

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ABSTRACT

Utilizing a suite of computer programs, the influence of the slide valve on compressor efficiency and bearing forces is examined. A design procedure for choosing optimal volume ratios for discharge ports and for choosing the optimal slide stop is presented. The possibility that bearing forces peak at a certain part load condition is discussed. The results suggest that the addition of a slide valve to a compressor results in a decrease in the indicated efficiency at the full load condition. For a particular minimum part load condition the volume ratio fixed by the axial discharge port should not be over high. If it is, a decrease in the indicated efficiency at the low load range will result. However, the volume ratio for the radial discharge port should be chosen so as to obtain the highest indicated efficiency at full load. The slide stop must be optimized to avoid the occurrence of the maximum bearing forces at certain part load conditions.

ZUSAMMENFASSUNG

Mit Hilfe von Rechenprogrammen wurde der Einfluß des Regelschiebers auf den Wirkungsgrad und die Lagerkräfte eines Schraubenverdichters untersucht. Eine Entwurfsmethode für die Wahl der optimalen Volumenverhältnisse von Auslaßöffnungen und des optimalen Gleithaltes wird präsentiert. Die Möglichkeit, daß die Lagerkräfte in einem bestimmten Teillastbetrieb einen Maximalwert annehmen, wird diskutiert. Die Resultate deuten an, daß der Zusatz eines Regelschiebers in einem Verdichter eine Abnahme des indizierten Wirkungsgrads im Vollastbetrieb verursacht. Für einen bestimmten minimalen Teillastbetrieb sollte das durch den Auslaß festgelegte Volumenverhältnis nicht zu hoch sein. Wenn es ist, wird die Abnahme des indizierten Wirkungsgrads im niedrigen Lastbetrieb erfolgen. Das Volumenverhältnis für den radialen Auslaß sollte jedoch so gewählt werden, daß der höchste indizierte Wirkungsgrad im Vollastbetrieb erhalten wird. Der Gleithalt muß optimiert werden, um das Auftreten von Maximallagerkräften unter bestimmten Teillastbedingungen zu vermeiden.

1 INTRODUCTION

In many applications of helical twin screw compressors to refrigeration duties, the load varies over a wide range. Capacity control is thus needed to enable the compressor to run at part load conditions. The most common method makes use of a slide valve which allows a measured quantity of the compressed cavity volume to "blow out" back to suction, as shown in Figure 1. The slide valve has the advantage of being a geometrically simple device, but it exerts an important influence on compressor efficiency /1-4/.

The addition of a slide valve mechanism to a compressor usually limits the width of the radial discharge port at full load to the width of the slide, due to space constraints. The shaded area on Figure 2 shows the area typically lost. The maximum attainable size of the axial discharge port is similarly constrained. It controls the discharge process when the compressor runs at low load capacity. These changes in the construction of the discharge ports mean that the area of discharge flow in the machine with the slide valve at full load is less than that for the equivalent machine without a slide valve. The result is an increase in the gas flow resistance through the ports, over pressure in the cavity and a decrease in the indicated efficiency of the machine. In view of the huge number of helical twin screw compressors used for refrigeration duties, it is of interest to investigate the extent of this efficiency penalty and other important aspects of behaviour such as bearing reaction forces. This paper describes the outcome of such an investigation.

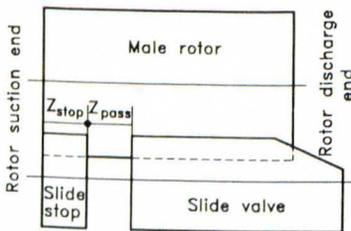


Fig. 1 Slide stop of a slide valve

Bild 1 Gleithalt eines Regelschiebers

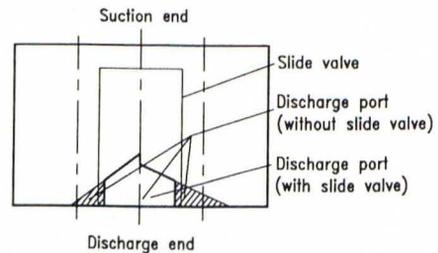


Fig. 2 Radial discharge port construction

Bild 2 Konstruktion des radialen Auslasses

As the slide-valve opens to allow bypass of gas, its far end moves towards the discharge end, resulting in a reduction of the area of the radial discharge port. Although the volumetric capacity of the compressor is reducing as the compressor is unloaded, the reduction of the radial discharge port area will increase the discharge resistance and pressure drop, and cause a decrease in the indicated efficiency. Factors which influence the compressor performance at part load condition include the compressor built in volume ratios for both

radial and axial discharge ports and the slide valve design parameters (especially Z_{stop} as defined in Figure 1). In order to provide a good performance in refrigeration systems these parameters should be optimized to suit the running conditions of the compressor.

Over pressure in the cavity introduces the possibility of increased bearing forces. Indeed, a possibility exists that the bearing forces under certain part load conditions may be larger than those at the full load condition. To ensure the safe operation of the compressor the maximum bearing forces must be determined at the design stage for various possible running conditions.

Utilizing a suite of computer programs developed by the authors, the influence of the slide valve on compressor efficiency and bearing forces is examined. A design procedure for choosing optimal volume ratios for both the radial and axial discharge ports and for choosing Z_{stop} is presented. The circumstances under which bearing forces peak at part load condition are identified and discussed. The results should be of value to the designers and operators of twin screw compressors required to run under a range of operating conditions.

2 COMPUTER PROGRAMS USED, COMPRESSOR SPECIFICATIONS AND RUNNING CONDITIONS

The following computer programs developed by the authors are used to do the calculations presented in this paper:

A Profile Generation Program: This program is a profile library which can generate any profile within the SRM A and D definition. Programs to generate other profiles are being developed. The profile generation program is the starting point of all calculations. It outputs its calculated results into data files, which are required by other programs for the calculation of geometrical characteristics, cutter blade shapes, clearances, thermodynamic behaviour and bearing forces, etc.

B Geometrical Characteristic Calculation Program: The output of the above program together with rotor geometry and port detail are the inputs to this program /5/ which calculates all the required geometrical characteristics and parameters needed by designers and for simulating the suction, compression and discharge processes (Program C). Geometrical characteristics are normally expressed as functions of the male rotor rotational angle. Those used in the working process simulation program must be expressed in this form.

C Working Process Simulation Program: This program /6/ simulates working processes of a refrigeration twin screw compressor which may run under various conditions, such as

partial loading, oil injection, liquid refrigerant injection, gaseous refrigerant superfeeding and different refrigerants etc. The program calculates each thermodynamic property in the cavity as a function of the male rotor rotational angle or of the cavity volume. The volumetric and indicated isentropic efficiencies are also calculated. The accuracy of the program has been checked against a range of measured data and found to be good /6/.

D Bearing Force Computation Program: Utilizing the pressure-volume results of the working process simulation (Program C), this program /7/ calculates the resultant gas forces and torques applied to male and female rotors and the axial and radial components of the bearing reactions. In addition, the rotor-rotor contact forces are calculated by assuming that the contact forces are constant in magnitude along the power transmission section of the profile. The lines of action of the contact forces are normal to the common tangent of the profiles at the contact point. Since the axial forces, including those applied to rotor end faces by the gas leaking across the end faces and to the rotor helical screw surfaces by both the compression gas and the rotor-rotor contact, do not act along the rotor axes, a moment exists tending to tilt the rotors which influences the bearing (radial) forces. This effect is taken into account. The results of calculated forces and torques are given as functions of male rotor angle of rotation. The accuracy of the program has been checked against measured data for an air-compressor having SRM-A profile rotors and shown to be good. The calculation techniques used in this paper for a machine having an SRM-D profile are identical to those used and experimentally verified for the SRM-A profile. The authors believe that it is reasonable to expect similar accuracy.

The specifications of the compressor used for the calculations are as follows:

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|-----------------------|-------|--------------------------|--------|
| • Rotor lobe profile: | SRM D | • Rotor diameters: | 204 mm |
| • Lobe Combination: | 4/6 | • Length/diameter ratio: | 1.65 |
| • Wrap angle (male): | 300° | • Driving rotor: | male |

The chosen running conditions:

- | | | | |
|---------------------|-------------------|----------------------------|-------|
| • Refrigerant: | R 22 | • Condensing temperature: | 25 °C |
| • Male rotor speed: | 3000 rpm | • Evaporating temperature: | -5 °C |
| • Oil injected: | 150 kg/min, 40 °C | • Superheat degrees: | 30 °C |

3 THE INFLUENCE OF SLIDE VALVE ON COMPRESSOR FULL LOAD PERFORMANCE

The influence of the slide valve on compressor full load performance is shown in Figure 3, in which the indicated efficiencies of the specified compressor with and without a slide valve

are displayed as a function of the volume ratio for the radial discharge port. The volume ratio for the axial discharge port is 5.0 for the compressor with a slide valve, while for the compressor without a slide valve it is set to be the same as that for the radial discharge port.

It can be seen that over a wide range of volume ratio for the radial discharge port, the indicated efficiency for the compressor without a slide valve is higher or much higher than that for the compressor with a slide valve. The curves peak at different volume ratios, with the highest indicated efficiency being 84.4 percent at a volume ratio of 2.1 for the compressor without a slide valve, and 83.9 percent at a volume ratio of 2.0 for the compressor with a slide valve. This means that, assuming the compressors in both cases were designed to achieve

the highest efficiency, there is still a loss of indicated efficiency of about 0.5 percent due to the use of the slide valve at the full load condition. From the Figure 3 it is apparent that if a higher volume ratio is used for the radial discharge port in the compressor with a slide valve the lost of the indicated efficiency at the full load condition will be much larger.

The main reason for this lost in the indicated efficiency is, as mentioned before, the reduction of both the radial and axial discharge port areas due to the addition of the slide valve (see Figure 2), which increases the resistance to gas flow through the discharge port, thus increasing the indicated power considerably. Figure 4 shows the variation of the Mach number of the gas flow through the ports during the discharge process. The Mach number in the compressor with a slide valve is much higher than that in the machine without a slide valve. The larger the discharge Mach number, the larger the indicated power and the lower the indicated efficiency. The waste of energy is obvious in Figure 5 in which over-pressure occurs due to the reduction of the discharge port area caused by the presence of the slide valve.

Since the reduction of the discharge port area is inevitable due to the use of a slide valve (see Figure 2), this lost of indicated efficiency is unavoidable if a compressor with a slide valve runs at full load condition. However, it is quite common for two or more twin screw compressors to be employed in one refrigeration plant. In order to reduce the total energy

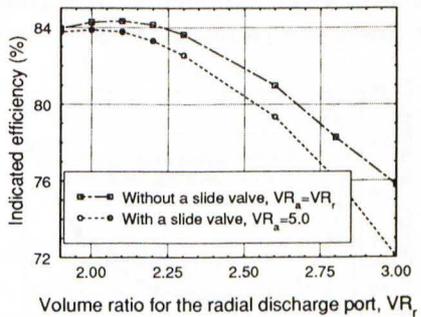


Fig. 3 Effect of slide valve on compressor full load performance

Bild 3 Auswirkung des Regelschiebers auf die Vollleistung eines Verdichters

consumption of the compressors, the authors suggest that in such a situation some of the compressors should be without slide valves. They should have the same volume ratios for the radial and axial discharge ports, and always run under the full load condition. In view of the huge applications of twin screw compressors in the refrigeration industry, the potential for energy saving of such an arrangement is significant.

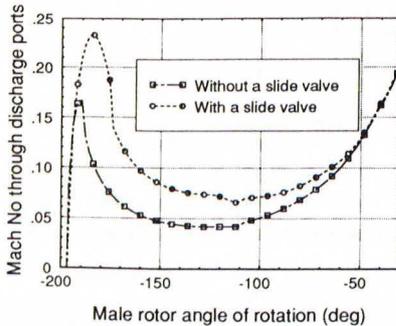


Fig. 4 Influence of slide valve on Discharge Mach No

Bild 4 Einfluß des Regelschiebers auf die Ausström-Mach-Zahl

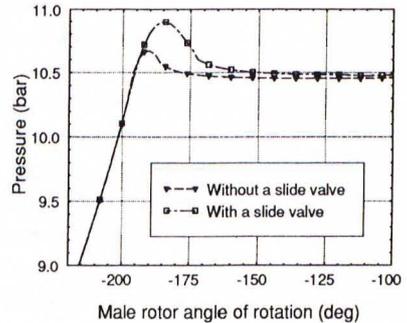


Fig. 5 Effect of slide valve on discharge pressure

Bild 5 Auswirkung des Regelschiebers auf den Auslaßdruck

4 OPTIMISATION OF THE VOLUME RATIOS FOR DISCHARGE PORTS

A typical twin screw compressor fitted with a slide valve has an axial discharge port which is smaller than the radial discharge port (at full load). It is possible to have a radial discharge port sized for a volume ratio of 2.6 combined with an axial port designed for an equivalent volume ratio of 5.0. These volume ratios for determining the discharge ports are defined as the ratio of the maximum suction volume at full load over the trapped volume just before the discharge process through the individual port, i.e. $VR_a = V_{s,max} / V_{d,a}$ and $VR_r = V_{s,max} / V_{d,r}$. It is apparent that these volume ratios for the discharge ports are different from the actual volume ratios during slide valve unloading, which are defined as the effective suction volume over the V_d . Consequently, the effective volume ratios vary with the slide valve position. The operating volume ratio of the compressor is always the lower of the two deriving from axial or radial flow. In general, the radial port sets the volume ratio at higher loads while the axial port dominates behaviour at low loads. In order to get the highest possible indicated efficiency over the entire operating range, the influence of these two volume ratios must be considered simultaneously in an optimisation procedure.

The Volume Ratio for the Axial Discharge Port VR_a

The volume ratio for the axial discharge port, VR_a , not only controls the discharge process at low load, but also sets the theoretical minimum volumetric load capacity at which the compressor can be run. This minimum volume, $V_{s,min}$, as shown in Figure 6, is in fact the trapped volume in the rotors when the axial port is about to deliver, i.e. $V_{s,min} = V_{d,a}$. Thus, the minimum volumetric load cavity

ratio, $VLCR_{min}$, can be expressed as $1/VR_a$ (i.e. $VLCR_{min} = V_{s,min} / V_{s,max} = V_{d,a} / V_{s,max} = 1 / VR_a$). The relationship between the VR_a and $VLCR_{min}$ is shown in Figure 7. For the given minimum volumetric load capacity ratio a corresponding VR_a can then be determined.

As far as the authors are aware, many helical twin screw refrigeration compressors are at present designed with a volume ratio of about 5.0 for the axial discharge port. In practice, few run at such a low load as 20 percent of the full load capacity. The use of a higher VR_a implies a smaller discharge port and thus a decrease in the indicated efficiency. Figure 8 reveals this effect of the VR_a for a minimum volumetric load capacity ratio of 0.4, for which VR_a should be larger than 2.5 according to Figure 7.

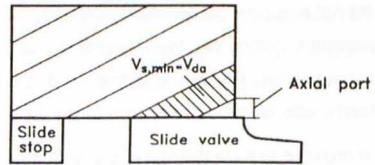


Fig. 6 Minimum load capacity of a slide valve
Bild 6 Minimallast eines Regelschiebers

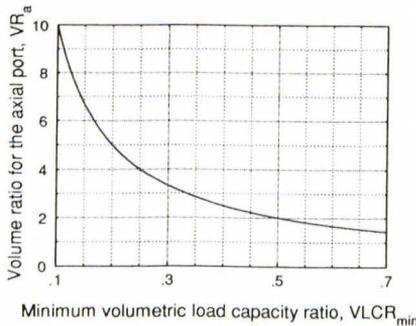


Fig. 7 Volume ratio for the axial discharge port as a function of minimum load capacity ratio

Bild 7 Volumenverhältnis für den axialen Auslaß in Abhängigkeit vom minimalen Lastverhältnis

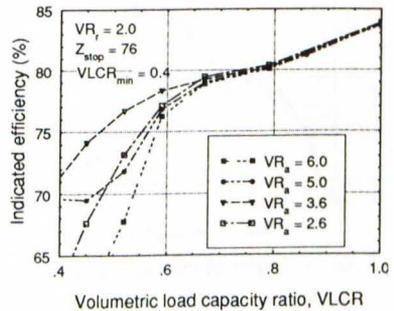


Fig. 8 Effect of the axial volume ratio on compressor part load performance

Bild 8 Auswirkung des axialen Volumenverhältnisses auf die Teillastleistung eines Verdichters

It can be seen that the uses of the high volume ratios of 5.0 and 6.0 considerably reduce the indicated efficiency within the load range of 0.4 - 0.7; while the low volume ratio of 2.6 also did not produce the best performance. This is because the high volume ratio results in over-pressure while the low volume ratio causes under-pressure, both increasing energy consumption. For the condition used, the best compressor performance over the entire load range was obtained with an intermediate axial volume ratio of 3.6.

In order to reduce the lost of the indicated efficiency caused by using a high volume ratio for the axial discharge port, the authors suggest that the VR_a should be optimised for the most common running condition of the compressor.

The Volume Ratio for the Radial Discharge Port VR_r

In general, the volume ratio for the radial discharge port, VR_r , should be chosen according to the full load condition. From Figure 3 it is known that the best VR_r equals 2.0 for the conditions specified, and, in fact, with this optimised VR_r , the compressor provides a relatively good performance over the entire operating range, provided that the slide valve is properly designed, as shown in Figure 9a.

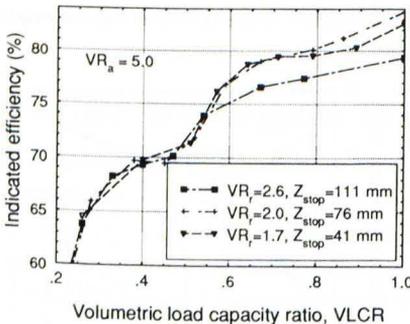


Fig. 9a Influence of the radial volume ratio on compressor part load performance

Bild 9a Einfluß des radialen Volumenverhältnisses auf die Teillastleistung eines Verdichters

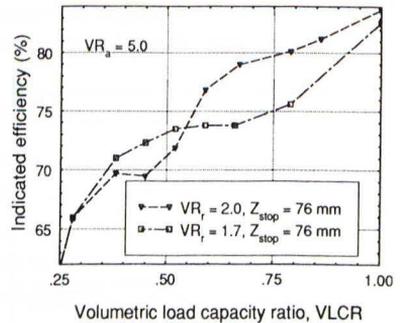


Fig. 9b Improving part load performance by changing VR_r

Bild 9b Verbesserung der Teillastleistung durch die Änderung des Volumenverhältnisses VR_r

However, this optimised VR_r usually can not provide the best performance at all the part load conditions. For instance, a smaller volume ratio of 1.7 could improve the compressor performance if the load capacity is lower than 50% for the specified condition, as shown in

Figure 9b. This is because a bigger radial discharge port (i.e. smaller VR_r) reduces the overpressure at the high load range. This suggests that VR_r may also be determined according to the compressor part load performance, provided that the machine operates unloaded for a substantial part of its duty. However, it must be noted that this approach of improving the part load performance by changing VR_r will affect the fully loaded performance and other part load conditions, as shown in Figure 9b.

5 OPTIMAL SLIDE VALVE DESIGN AND CORRESPONDING BEARING LOADS

For the chosen volume ratios for both the axial and radial discharge ports, the slide valve parameters must be optimised for the given running condition. The most important one is the slide stop, which determines the compressor part load performance to a great extent.

Figure 10 shows the effect of the slide stop on the indicated efficiency of the compressor during slide valve unloading. A small slide stop (e.g. 41 mm) tends to have a higher indicated efficiency at the high load range (0.8-1.0), while a big one tends have a better performance at the low load range (<0.5). Thus, the right choice of the slide stop depends on the running conditions of the compressor.

If a slide valve is designed in the way indicated here, there should be no severe overpressure during slide valve unloading. The bearing loads corresponding to such an optimized slide valve design are shown in Figure 11a. It can be seen that the maximum values of all the components occur at the full load condition. This is also predicted for other running conditions and length/diameter ratios by the authors' programs.

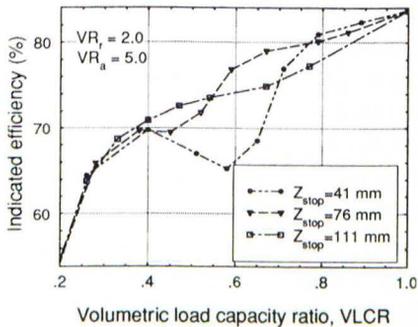


Fig. 10 Effect of slide stop on compressor part load performance

Bild 10 Auswirkung des Gleithaltes auf die Teillastleistung eines Verdichters

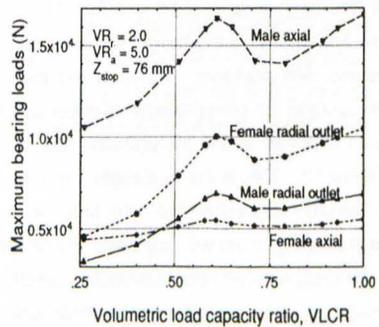


Fig. 11a Bearing loads for a optimized slide valve

Bild 11a Lagerkräfte für einen optimierten Regelschieber

However, if the machine is not run under its design condition or/and the slide stop or discharge ports are not the optimum for the given condition, severe over-pressure may occur during slide valve unloading. Accordingly, the bearing loads at certain part load slide valve settings may be equal to or even higher than those at the full load condition, as shown in Figure 11b.

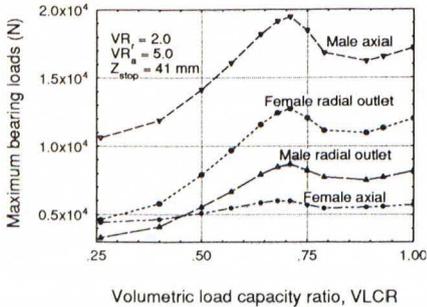


Fig. 11b Bearing loads for a improperly designed slide valve
Bild 11b Lagerkräfte für einen ungeeigneten entworfenen Regelschieber

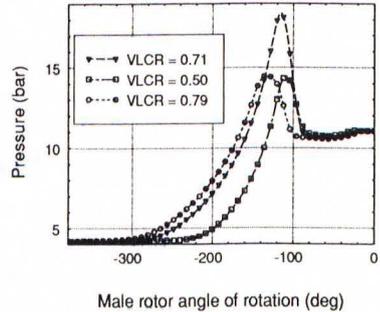


Fig. 12 Effect of load capacity ratio on discharge pressure
Bild 12 Auswirkung des Lastverhältnisses auf den Auslaßdruck

When this situation occurs, the peak values of the maximum bearing loads usually appear in the load range of 0.55 - 0.75. This is explained as follows. At the high load range (>0.75), the radial discharge port is still big enough for the gas to discharge without causing severe over-pressure. While at the low load range (<0.55), although there exists over-pressure in the compression chamber due to the reduction of the radial port area, the rotor area exposed to the compressing gas is considerably reduced, as may be deduced from Figure 12. This is the explanation of why high bearing forces appear at part load only for small slide stop values; i.e. over pressure is combined with a large cavity area. In summary, high bearing forces will only occur at part load in a compressor of a particular design (a small slide stop) running under particular conditions (about a volumetric capacity ratio of 0.65). In these circumstances, the female rotor radial bearing force at the discharge end and the male rotor axial force are both larger than their full load equivalents for the specified compressor and running condition.

6 CONCLUSIONS

- A slide valve ensures that the compressor obtains a relatively high indicated efficiency at part load condition, but reduces the compressor full load performance considerably. It is thus suggested that in the situation in which two or more twin screw compressors are used together, one or more should be without a slide valve and should always run under the full load condition with both the radial and axial discharge ports having the same volume ratios.
- The volume ratio for the axial discharge port not only controls the discharge process at low load, but also sets the theoretical minimum load capacity at which the compressor can be run. Using a high volume ratio for the axial discharge port, e.g. 5.0 for the minimum load capacity of 40 percent, may considerably decrease the indicated efficiency at the load range of interest. It is therefore suggested that the VR_a should be optimised for the most common running condition of the compressor.
- The volume ratio for the radial discharge port should be chosen according to the best full load performance. It can provide a relatively high indicated efficiency over the entire adjusting range of the slide valve. Additionally, the radial discharge volume ratio may also be used to improve compressor part load performance, but to the detriment of the full load performance.
- A suitable slide stop should be determined to achieve a relatively high indicated efficiency over the load range within which the compressor is to be run.
- If the volume ratios and slide valve are designed to optimise the indicated efficiency, the maximum bearing forces appear at the full load condition. However, a change in running conditions combined with an unwise choice of slide stop may result in the occurrence of the maximum bearing forces at the part load condition of 0.6 - 0.7. To ensure safe operation, it is suggested that in addition to the performance prediction, a force analysis should also be carried out when changing the running condition or changing the slide valve design of a twin screw refrigeration compressor.

7 NOMENCLATURE

V_s	Cavity volume at the end of the suction process Zahnlückenvolumen am Ende des Ansaugvorganges
V_d	Cavity volume at the beginning of the discharge process Zahnlückenvolumen zu Beginn des Auslaßvorganges
VR	Volume ratios for the discharge ports Volumenverhältnisse für die Auslaßöffnungen

- VLCR Volumetric load capacity ratio, i.e. the ratio of the effective suction volume V_s over the maximum suction volume at full load, $V_g/V_{s,max}$
 Volumetrisches Verhältnis, i.e. das Verhältnis des effektiven Ansaugvolumens zum Maximalansaugvolumen im Vollastbetrieb, $V_g/V_{s,max}$
- Z_{stop} Slide stop length of a slide valve (mm)
 Gleithalt Länge eines Regelschiebers (mm)

Subscripts

- a axial
 r radial

8 ACKNOWLEDGEMENTS

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