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A Comparison of SRM and Globoid Type Screw Compressors at Part Load

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INTRODUCTION

This paper is a sequel to "A COMPARISON OF SRM AND GLOBOID TYPE SCREW COMPRESSORS AT FULL LOAD".

The capacity control systems used for SRM compressors and Globoid compressors will be described at first.

These two types of compressors both provide stable operation throughout the entire operating range, and they both provide unloaded starting. Moreover the built-in volume ratios on both types of compressors vary when the load is reduced. Using different port designs, this variation can be adapted to a certain extent to the external operating conditions. This will be explained later.

CONTROL SYSTEM FOR SRM COMPRESSORS

The control system for SRM compressors has been described in references (5), (10) and (12).

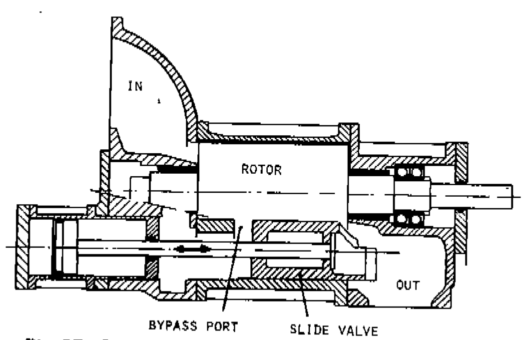

Fig. 27 Capacity control system for SRM compressors

Below and between the rotors, there is a slide valve which moves axially. This slide valve is designed as an integral part of the rotor casing. In principle, the effective length of the rotors is changed by slide valve movements. When the slide valve moves towards the outlet side in connection with unloading, a bypass port is formed at the inlet end of the slide valve. Excess gas proceeds via this bypass port back to the inlet side. The bypass port is positioned in such a way that no compression work is carried out on the excess gas.

The SRM compressor outlet port has a radial part and an axial part. The radial/movable part is formed by the outlet side of the slide valve, while the axial/fixed part is located in the outlet end plane of the rotor casing.

When the slide valve changes position, the size of the outlet port is changed, thus varying the built-in volume ratio v_i.

The control cycle will be explained with reference to Fig. 28.

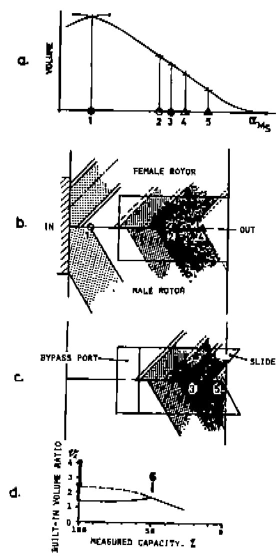

Fig. 28 Working principle for SRM capacity control system

Fig. 28a shows the compression part of the volume curve. Figures 28b and 28c show the part of the rotor casing that incorporates the slide valve. Fig. 28d shows theoretical variations in v_i at different capacities. The built-in volume ratio v_i is defined as follows

$$v_i = \frac{V_i}{V_o}$$ 8

where V_i = theoretical interlobe space volume just closed at the by pass port

V_o = theoretical interlobe space volume just about to open to the outlet port

The definition of v_i provides an inlet volume that is smaller than the actual suction volume. This will be shown later. However, this definition is essential for theoretical comparison purposes.

It is evident from Figures 28a and 28b that at full load $v_i = V_1/V_4$

On the basis of the above definition, one can imagine the slide valve opening infinitesimally. This would mean that the theoretic inlet volume would correspond to point 2 in Fig. 28. Point 2 is thus the position where the interlobe space volume has just closed at the bypass port, but V_o has not changed since the slide valve has moved through only an infinitesimal distance. Theoretically v_i has thus changed discontinuously to $v_i = V_2/V_4$.

Points 3 and 5 in Fig. 28 represent intermediate positions of the slide valve.

As mentioned previously, the outlet port consists of two parts, one of which is radial/movable while the other is axial/ fixed. By designing the axial/fixed part smaller than the radial part at the full-load point, v_i can be changed at part load, thus changing the part load characteristics to some extent. Point 6 in the illustration corresponds to the position where the two ports are of equal size. Prior to point 6 in the cycle, the area of the outlet port was reduced continuously by the slide valve. After point 6, the port area is determined by the fixed part, and v_i thus drops continuously. When $v_i = 1$ the interlobe space volume closes at the by pass port and opens to the outlet port simultaneously. The compressor is now theoretically completely unloaded. When the slide valve moves further, direct connection is established between the compressor outlet and inlet.

The actual compressor capacity will not be the same as the theoretical capacity during the part load cycle. This is due primarily to the pressure drop in the bypass port. The area of this port is small at the beginning of the control cycle. During the part load cycle, the compressor capacity will vary from 100% down very close to the completely unloaded state. Fig. 28d shows, on the basis of the aforesaid definitions, the v_i cycle as a function of the actual compressor capacity.

CONTROL SYSTEM FOR GLOBOID COMPRESSORS

Different manufacturers of globoid compressors have selected different control systems. One system uses two slide valves and operates on the same principles as the SRM compressor system. This system is described in references (8) and (13). The control system that will be analyzed here is illustrated in Fig. 29.

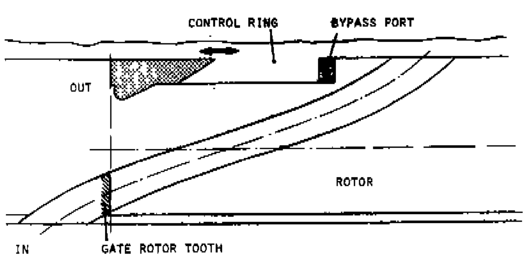

Fig. 29 Capacity control system for Globoid compressors

This illustration shows a diagrammatic development of the rotor (one groove) and the rotor casing. In the outlet plane, there is a rotatable ring which varies the area of the bypass port and also varies the area of the outlet port. This control ring can be turned by means of an external gear motor. Continuous control is obtained here, just as on the SRM compressor.

Since the outlet port has both a fixed and movable part, the built-in volume ratio v_i can be varied in the same way as in an SRM compressor. However, the system is somewhat more limited here since the entire port area is radial while in the SRM type the movable part is radial and the fixed part is axial.

Moreover, the system is limited by the fact that the part of the control ring which determines the area of the outlet port cannot be turned further than to the casing partition, i.e. the gate rotor plane. This limits the percentage of unloading. Control ring movements are particularly limited at high values of v_i .

Fig. 30 illustrates the theoretical part load cycle defined in the same way as for the SRM compressor. Fig. 30a shows the compression part of the volume curve. Figures 30b and 30c show the control ring at the full-load position and part-load position respectively. Fig. 30d shows the variation of v_i as a function of the actual capacity. v_i at full load is defined as $v_i = V_1/V_4$.

Particularly in Fig. 30a, it can be seen that a certain reduction of volume is obtained at point 1 before the gate rotor seals the groove. When the bypass port opening is infinitesimal, the interlobe space volume drops to point 2 and the v_i of the discontinuity gap thus becomes $v_{i2} = V_2/V_4$. Point 3 represents an intermediate position at which $v_{i3} = V_3/V_5$. Point 6 represents the v_i that is obtained at the outlet port position shown by a broken line in Fig. 30c, i.e. when the outlet side of the control ring has reached the fixed port. Point 7 represents the maximum unloading position. At this position the ring is turned so far that the limiting edge of the movable outlet port has reached the gate rotor plane shown by the broken line in Fig. 30c.

The variation in v_i thus obtained at different capacities is shown in Fig. 30d.

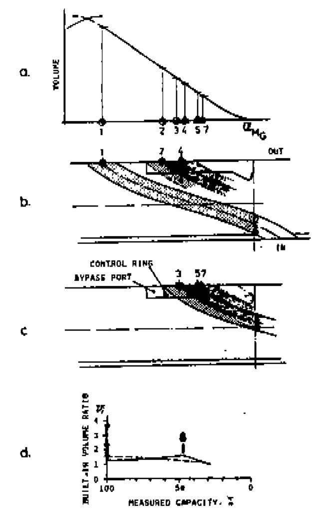

Fig. 30 Working principle for Globoid capacity control system

It can be seen in Fig. 30b that compression in the groove space is obtained before the groove opens to the bypass port. This is more marked at a high value of v_i with the selected control system due to the fact that the bypass port inlet edge is closer to the outlet port.

EFFECT OF VARIATION OF v_i ON EFFICIENCY

As indicated by part 1 of this paper (Fig. 13), the v_i is a dominant parameter, although it can be controlled to some extent at full load for these types of rotary compressors.

Normally, in a refrigeration plant, one obtains a descending condensing temperature, a descending discharge pressure at the compressor and, in certain cases, a rising evaporation temperature (rising suction pressure) when plant capacity is being reduced. When using centrifugal compressors, this phenomenon is necessary to obtain stable operation at low loads.

The above set of conditions means that the pressure ratio at which the compressor is operating diminishes gradually with the load. This in turn, means that a certain reduction in v_i is desirable during the part load cycle.

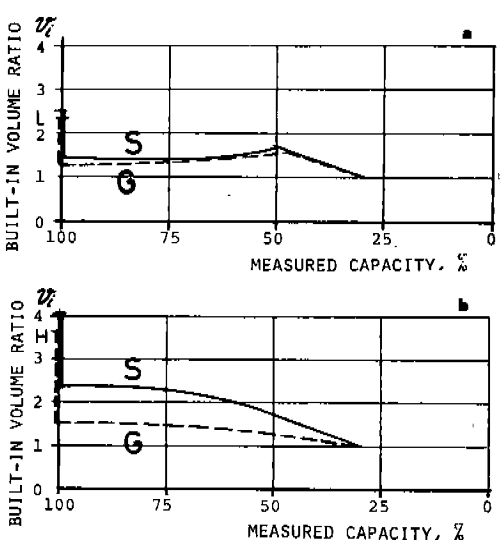

Fig. 31 Built-in volume ratio vs measured capacity for low v_i (a) and high v_i (b) at part load with ammonia

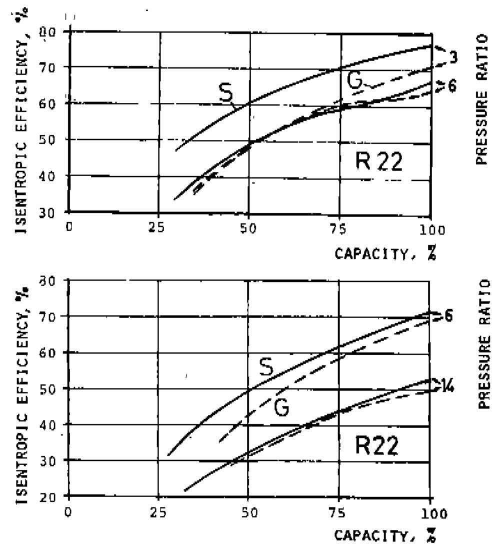

Fig. 32 R22 compressor efficiencies at part load for the SRM and the Globoid compressors

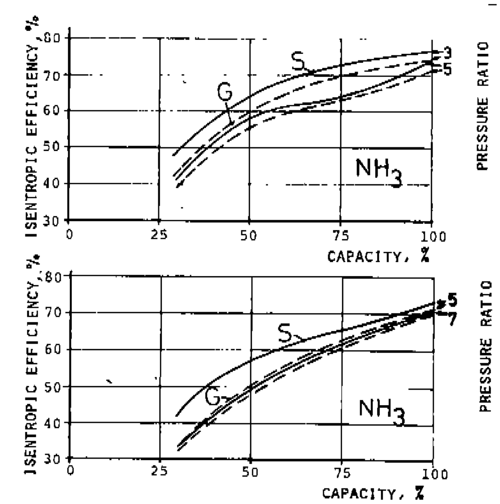

Fig. 33 Ammonia compressor efficiencies at part load for the SRM and the Globoid compressors

The v_i value becomes lower at part load as indicated in Figs. 28d, 30d and 31. Fig. 31 compares the v_i variation during the part load cycle in the two types of compressors. In the two situations compared here, v_i decreases less for the SRM compressor, but it should be noted that this in no way provides an accurate picture of part load efficiency. When the compressor carries a part load, a completely new and complicated geometry is encountered (different wrap angles, different volume curves, etc.). Consequently the theoretical analysis of the v_i cycle only provides a schematic picture of the possibilities of changing the part load characteristics.

In the full-load case, v_i is a dominant parameter. At part load, losses other than the v_i loss are more dominant. Such losses include leakage losses in the rotor mesh, past the rotor tips and in the outlet end plane as well as other losses such as pressure drops, frictional and ventilating losses.

COMPARISON OF TEST DATA

As mentioned previously, the two types of compressors were run in the same test rig using R22 and also ammonia. They were run under the same operating conditions with two different values of v_i — one low and one high. The compressors were driven by a calibrated DC dynamometer motor that was also used to measure input torque.

The rate of gas flow was measured with nozzles on the suction side of the compressor in accordance with ISO 917 - 1974 and ISO R 541. During the tests, the pressures at the inlet and outlet were kept constant for the individual slide valve positions and control ring positions. The isentropic efficiency η_{is} was selected as a basis for comparison. Isentropic efficiency is defined in "A Comparison of SRM and Globoid Type Screw Compressors at Full Load".

Fig. 32 shows η_{is} as a function of the measured capacity. Fig. 32a shows a comparison at a low value of v_i (low v_i and high v_i are defined at the full-load point). Fig. 32b shows the comparison at high v_i. The tests were run at normal operating conditions for these values of v_i. The illustration shows that the percentage of difference at the full-load points increases somewhat at part load. This increase is particularly marked at $P_1/P_2 = 3$ which corresponds to the operating conditions encountered in air conditioning applications. Probably the Globoid compressor is hampered here by port areas that are too small, since this was plainly indicated by the full-load analysis.

Fig. 33 shows the corresponding test results for NH_3. Here too, the better efficiency of the SRM compressor at the full-load point was retained, and the percentages were better throughout the entire part load cycle.

SUMMARY

These two papers have presented a study of two similar types of compressors belonging to the same family, i.e. positive displacement, oil-injected compressors. They were both screw compressors, one of the SRM type and the other of the Globoid type. The comparison was made between two compressors having the same swept volume. The compressor geometry and losses were analyzed in the theoretical part of the comparison. This analysis showed that SRM compressors have certain advantages which should result in better efficiency. Two compressors of normal production quality were run using both R22 and ammonia in the same test rigs. These compressors were tested throughout the entire operating range at full load and at part load with different values of v_i etc. The results of the test runs verify the results of the theoretical comparison which indicated that SRM compressors have better efficiency throughout the full operating range.

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