THE INFLUENCE OF DISCHARGE PORTS ON ROTOR CONTACT IN SCREW COMPRESSORS

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ABSTRACT

One means of minimizing interlobe leakage in a screw compressor is to design it so that the pressure distribution across the female rotor causes it to maintain contact with the trailing flank of the male rotor. This is because the sealing length on that side of the rotor is longer than on the, more traditionally used, leading flank. The disadvantage of this is that it creates torque on the female rotor opposite in sign to that caused by the drag forces. The the net torque, which is the result of the small difference between relatively large forces, may then change its sign during the compression cycle and this is the main cause of rotor instability and mechanical noise in screw compressors. Thus, some adjustment of the pressure distribution may be required to avoid rotor flutter or rattling. A well proven mathematical model was used to calculate the torque on the female rotor and determine the best pressure distribution that would avoid this effect. The improvement resulting from this were confirmed experimentally on a prototype compressor, which had previously generated high levels of contact noise and vibration under some operating conditions. Trial and error changes were made to the discharge port size and shape until the best result was achieved. By this means, the female rotor motion was stabilized. This resulted not only in reduced noise generation but also in improved compressor performance.

Keywords: Screw compressor, Mechanical noise, Vibration

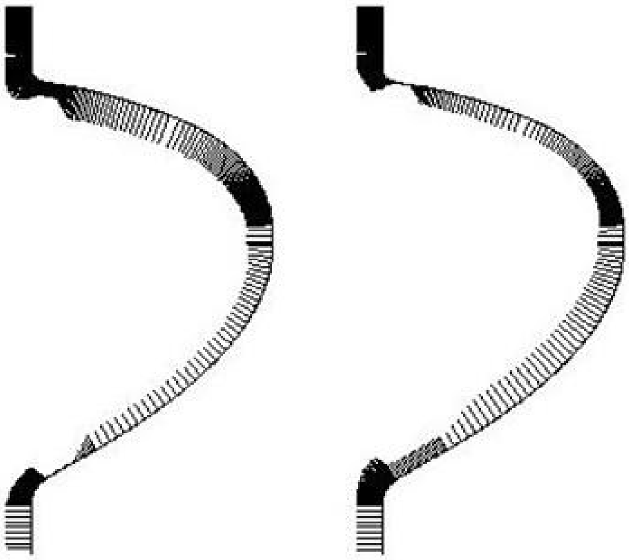
1. INTRODUCTION

The performance of screw compressors is highly affected by internal leakage. Hence, any reduction in the clearances between the rotors and the rotors and their housing always improves their efficiency. Therefore, in order to maximise screw compressor delivery rates and efficiencies, interlobe clearances are made as small as possible. Modern rotor manufacturing methods, such as grinding with simultaneous measurement, control and correction of the profile, allow profile tolerances to be maintained within ± 5 μm. This enables the clearances between the rotors to be kept below 15 μm

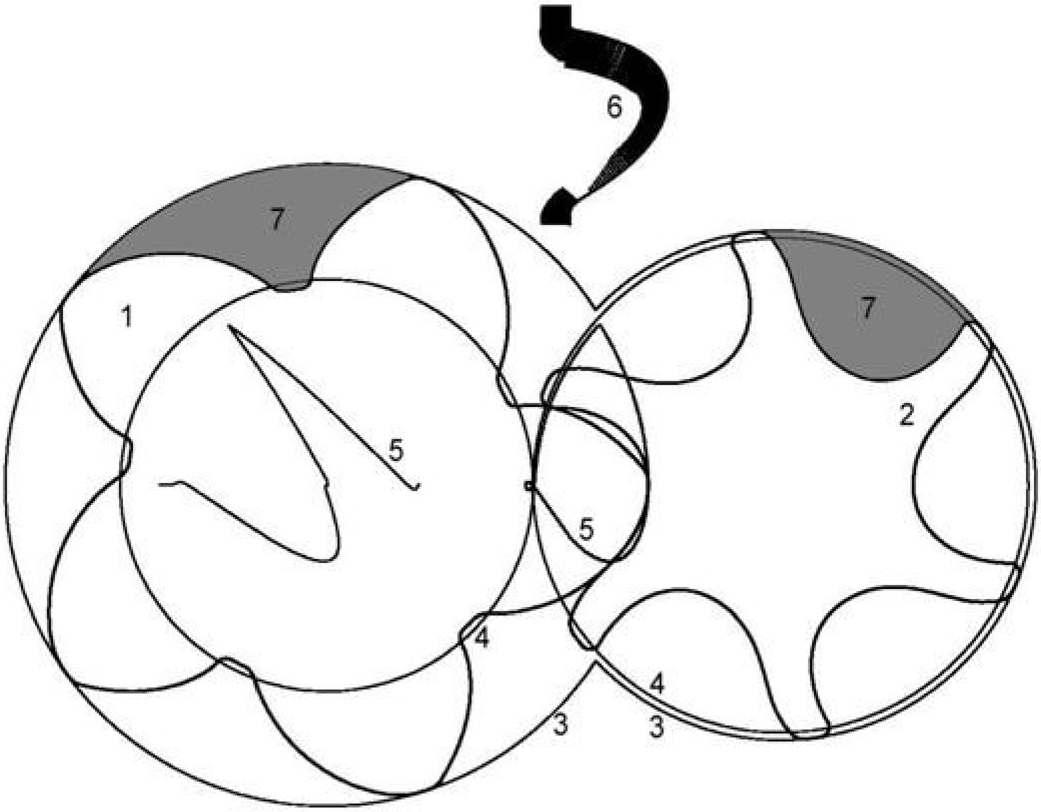
In oil flooded compressors, direct contact is made between the rotors and, in well designed machines, the clearance distribution is set so that this is first made along their contact bands, close to the rotor pitch circles. Since the relative motion between the contacting lobes in this region is almost pure rolling, the danger of their seizing, as a result of sliding contact, is thereby minimised. As shown in Fig 1, the contact band may be either on the rotor round flank a), or on the rotor flat flank b).

The traditional approach is to maintain a high, so called, positive gate rotor torque, which ensures round flank contact. More recently, it became apparent that there are significant advantages to be gained by creating a negative gate rotor torque so that contact will made be on the flat face. The reason for this can be understood by examination of the sealing line lengths, shown as item 5 in Fig 2, where it is clearly apparent that the flat flank sealing line is much longer than that of the round flank. Thus, minimising the clearance on the flat flank will reduce the interlobe leakage more than minimising the round flank clearance. Negative gate torque is achieved by making the gate rotor lobes thicker and the main rotor lobes correspondingly thinner. This has the additional effect of increasing the flow displacement area. Thereby, both the compressor flow rate and its efficiency are increased.

The torque induced by the pressure forces on the female rotor is small compared with that induced on its male counterpart and is of the same order of magnitude as that induced by other effects, such as contact friction and oil drag forces. Since negative torque on the female rotor acts in the opposite direction to the friction and drag force induced torque, the net torque on the female rotor may experience a change of sign during the complete compression cycle. This may cause the female rotor to flutter and, in extreme cases, may induce rotor rattle. This directly influences the compressor noise. *Andrews and Jones, 1990 [1]* and *Holmes, 2003 [2]* indicated that a great part of compressor noise may be attributed to the rotors. *Koai and Soedel, 1990* [3] analysed pulsations and their influence upon the discharge port and *Sangfors 1999* [5] established a procedure to calculate flow through the discharge port, while *Tantari, 2000* [7] gave a good analysis of the phenomena which influence screw compressor noise. More recently, *Mujic et al, 2005* [4] showed how the compressor discharge ports influence the noise. It follows that the design of screw compressor components must include the elimination of such effects. Neglect of this, results in reduced reliability and, usually, increased compressor noise.



a) Rotor contact on the round face, b) Contact on the flat face
Fig. 1 Rotor interlobe clearance distribution

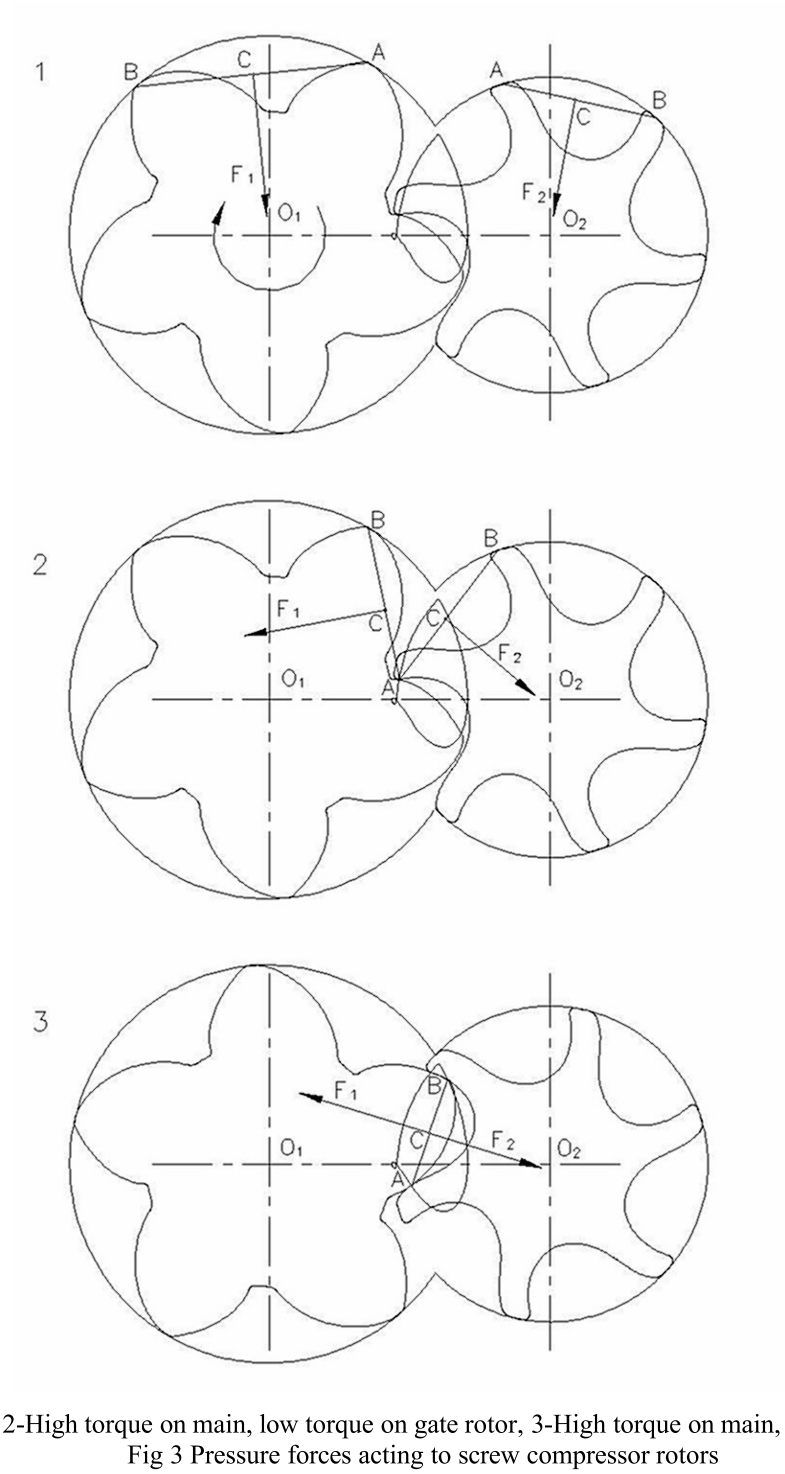


1-main, 2-gate, 3-rotor external and 4-pitch circles, 5-sealing line,
6-clearance distribution, 7-area between the rotors and housing
Fig 2 Screw rotor profile

2. CALCULATION OF TORQUE ON COMPRESSOR ROTORS

The magnitude and direction of the torque on the rotors is determined both by the rotor profile and the shape and position of the compressor discharge port. To quantify this and determine the position of rotor contact, it is necessary to calculate the pressure history on the rotors for the whole compression cycle.

The forces which cause torque at any rotor cross section are presented in Fig 3. Three rotor positions are given, where the pressure acts on the corresponding interlobes normal to line AB. Points A and B are either on the sealing line between rotors or on the rotor tips. Since they are always located on the sealing line, their positions are determined by the rotor geometry.



1-No torque, 2-High torque on main, low torque on gate rotor, 3-High torque on main, low torque on gate
Fig 3 Pressure forces acting to screw compressor rotors

In position 1, there is no contact between the rotors. Since A and B are on the circumference, the overall forces F_1 and F_2 act towards the rotor axes and thereby are purely radial. Therefore, in this position they do not create any torque.

In position 2, A is only one contact point between the rotors. Forces F_1 and F_2 are eccentric and therefore have both radial and circumferential components. The latter cause the torque. Due to the force position, the torque on the gate (female) rotor is significantly smaller than that on the main (male) rotor.

In position 3, both contact points are on the rotors and the overall and radial forces are equal for both rotors. As in position 2, they cause torque.

If x is the direction parallel to the line between rotor axes O_1 and O_2, y is perpendicular to x and the pressure is denoted by letter "p", then the torque, T, is calculated as:

$$T = p\int_A^B x dx + y dy = 0.5p\left(x_B^{\ 2} - x_A^{\ 2} + y_B^{\ 2} - y_A^{\ 2}\right)$$

This equation is integrated along the profile at all shaft rotation angle steps over one complete revolution and thereby requires the pressure history to be expressed as a function of shaft rotation. Finally, the sum is taken of all rotor interlobes, taking account of any phase and axial shift between them. Details of the calculation procedure employed are given by *Stosic at al, 2005* [6].

A production air compressor, which generated higher levels of contact noise than anticipated under some operating conditions, was taken as a case on which to examine the validity of this approach. A well proven mathematical model was used to calculate the torque on the female rotor and the effect of rotor contact on compressor behaviour, from which the resulting noise attenuation was predicted. It was found from this that the prototype discharge port design was responsible for the negative pressure torque on the female rotor being sufficient to overcome the contact friction and oil drag effects. Since the drag force is difficult to model, some allowance had to be made for errors in calculation. Nonetheless, it was shown that even a small change in the discharge port could cause the gate rotor torque to approach zero, thereby triggering rotor instability manifested by acceleration and deceleration and even loss of contact with the main rotor. This would create rattling, identified by increased rotor mechanical noise. Further analyses showed that an alternative port design would be more suitable and should eliminate or at least minimize the noise resulting from inadequate rotor contact.

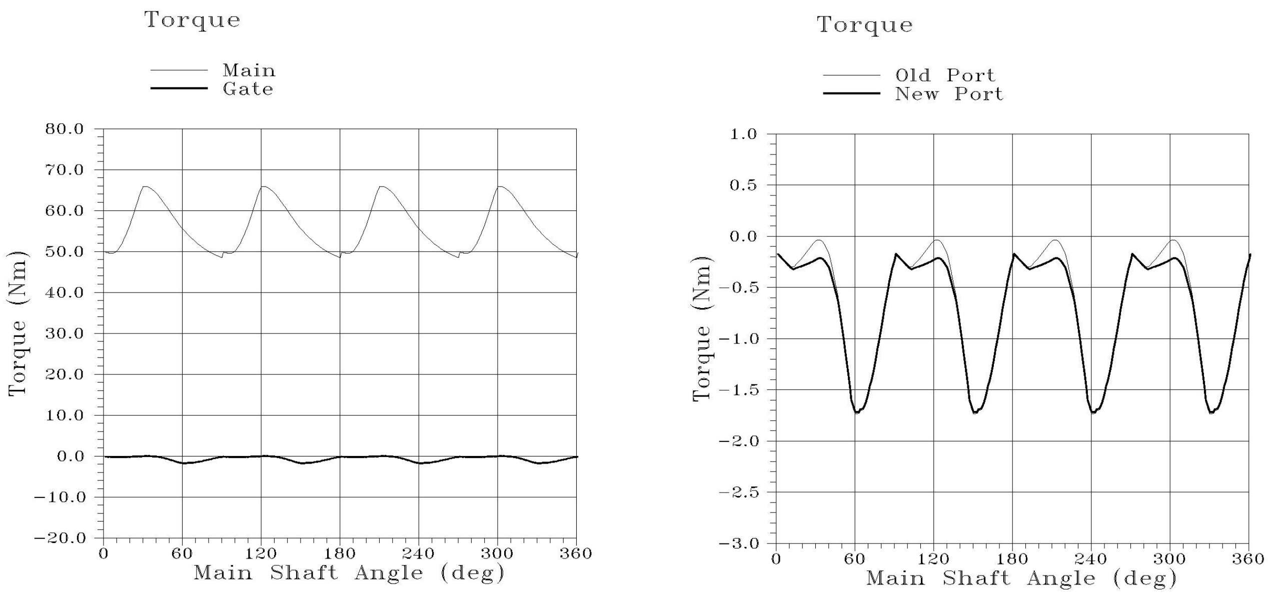


Fig 4. Torque on the main and gate rotor, left and gate rotor torque right, old port light line, new port, bold line

The results of these calculations are shown in Fig 4, for two cases, namely the original port and the finally modified port. The calculated torque on the main and gate rotors is presented in Fig 4, left, together with the torque on the gate rotor, right, for the old port, with a light line, and for the new port, with a bold line. The torque thus shown comprises the sum of the pressure force torque and drag force torque. The results presented in

Fig 4 show that the old port was prone to cause gate rotor instability, while the new port minimised this effect. Therefore, the new port was expected to ensure quieter and more stabile compressor operation.

3. EXPERIMENTAL PROGRAMME

In view of the uncertainty of the calculations, an experimental programme was then carried out on the prototype compressor to check the effects of the proposed port modifications.

The aim was to find a shape and size that simultaneously assured rotor contact at the leading flank of the male rotor while minimising the effects of gate rotor instability, without reducing the compressor efficiency,. Noise and vibration were chosen as the criteria to validate the results. Changes to the port were made progressively and until the best result was obtained.

The compressor was installed on the City University compressor test rig which is presented in Fig 5. The noise meter: SJK Scientific Ltd - Integrating Averaging Sound Level Meter HML 323 was used for noise measurements and piezo-resistive pressure sensors Endevco 8530C were used for measurements of pressure oscillations.

The noise meters were located one meter from the compressor at four positions: 1. behind the compressor discharge, 2.to the right of the compressor, 3. in front of the compressor and 4. to the left of the compressor. The pressure pulsation sensor was positioned in the compressor discharge chamber directly opposite to the discharge port. The compressor speed was operated over the 2000, 3500, 4500, 5500 and 6000 rpm range, with discharge pressures of 4, 6, 8, 10 and 12 bar. The mid point was set at 4500 rpm and 8 bar. All other compressor parameters such as suction and discharge pressures and temperatures, flow and power were measured in order to evaluate the compressor performance.

Fig 5. Screw compressor in the test rig

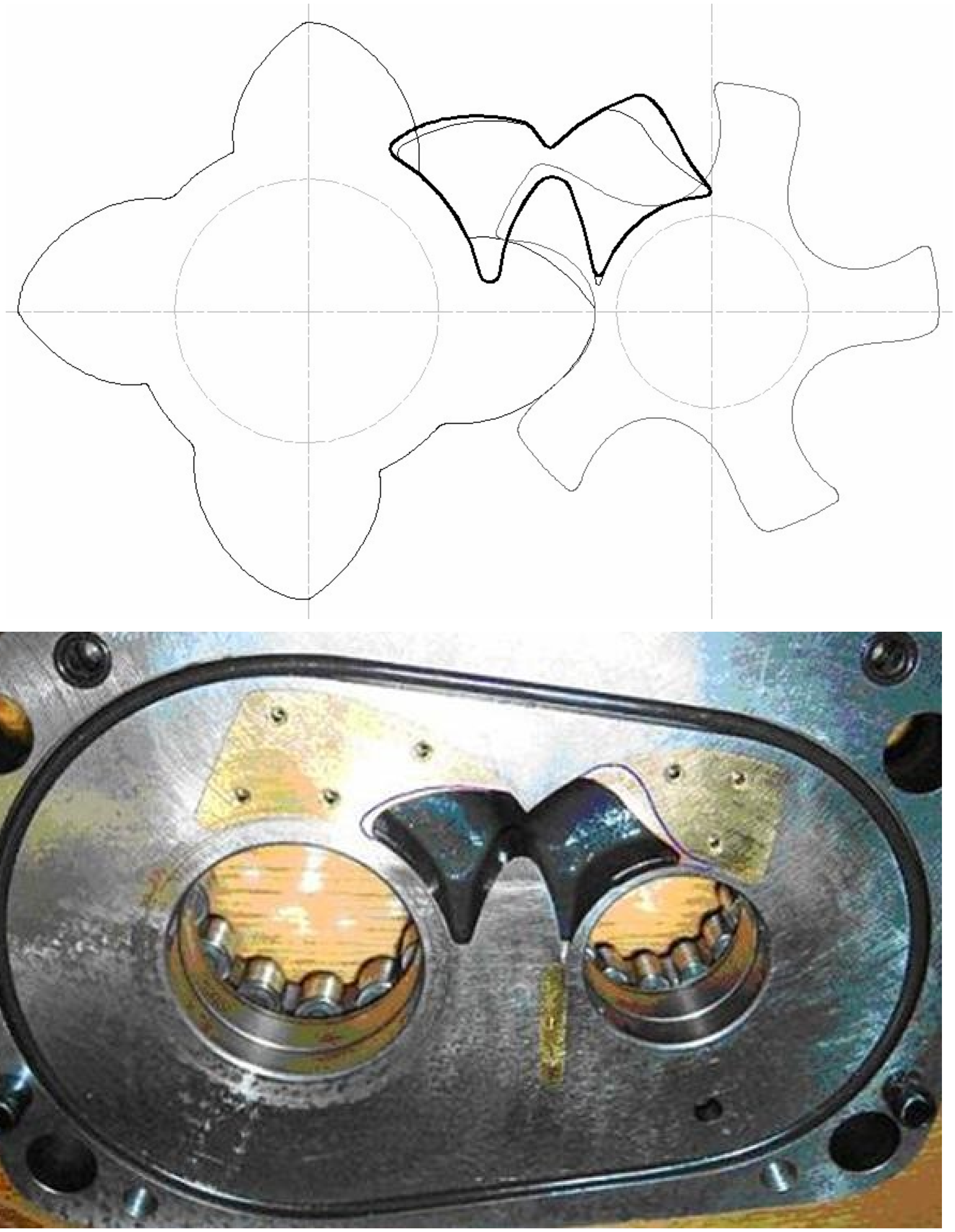
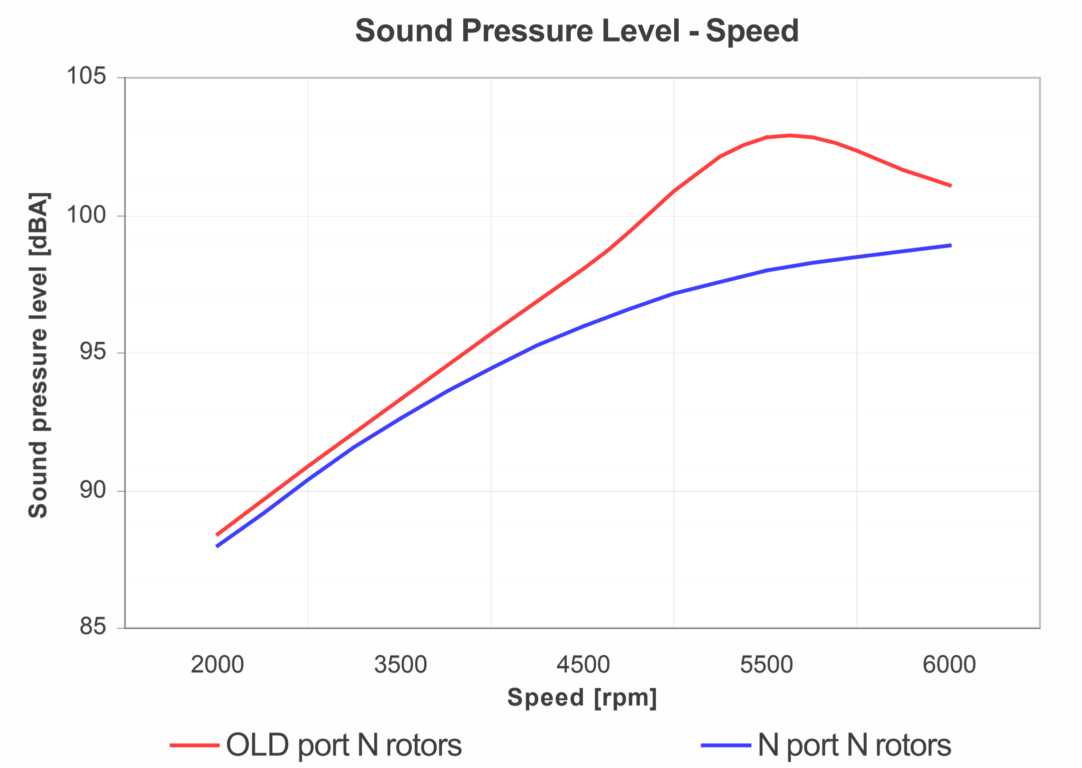


Fig 6. Discharge port sketch and photograph, original port, light line and modified port, bold line

The original and finally selected shapes of the discharge port are shown in Fig 6.

The test results conclusively confirmed that the discharge port shape, finally selected, resulted in reduced rotor contact noise. The results are presented in Table 1 and in Fig 7 which show a comparison of the sound pressure level as function of the compressor speed for the two different ports. The original compressor port caused a higher sound pressure level than the optimised modification. The difference is more visible at higher operating speeds. A comparison of the sound pressure level for the presented cases, as a function of the discharge pressure, is also presented in Table 1 and given in Fig 7. A noise reduction of up to 5.7 dB is detectable over the whole measured domain.



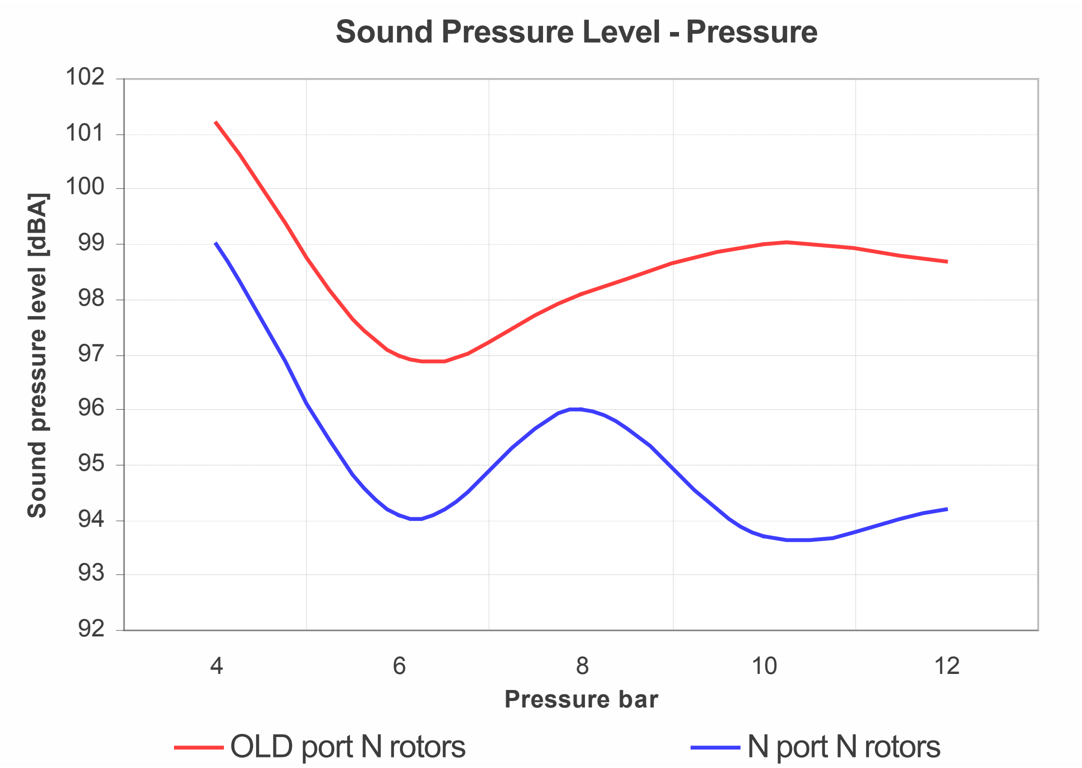


Figure 7 Sound pressure level in function of compressor speed and discharge pressure

Gas pulsations in the discharge port are presented in Fig 8 and they correlate well with the sound measurements.

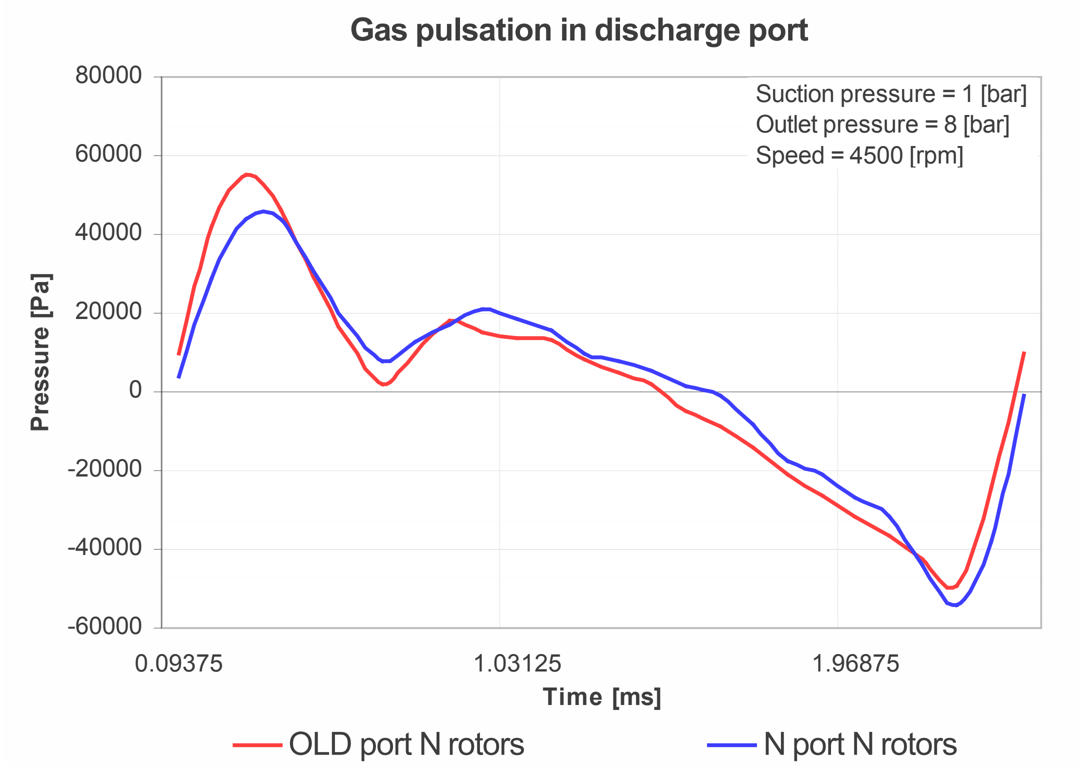


Figure 8 Gas pulsations in discharge port

Table 1 Noise measurements

Sound Pressure Level [dB]							
Date	Port	Rotors	Pressure 8 bar				
			2000 rpm	3500 rpm	4500 rpm	5500 rpm	6000 rpm
18.10.2005	OLD	N	88.4	93.3	98.1	102.8	101.1
20.10.2005	N	N	88.00	92.60	96.00	98.00	98.90

Date	Port	Rotors	Shaft Speed 4500 rpm				
			4 bar	6 bar	8 bar	10 bar	12 bar
18.10.2005	OLD	N	101.2	97	98.1	99	98.7
20.10.2005	N	N	99	94.1	96	93.7	94.2

Table 2 Compressor performance at 8 bars

Compressor performance					
Date	Port	Rotors	Q	P	Psp
			[m3/min]	[kW]	[kW/m3/min]
18.10.2005	OLD	N	5.0775	29.4978	5.7901
20.10.2005	N	N	5.3482	30.6475	5.7211

Results of the performance measurements, which are presented in Table 2 indicate that the compressor efficiency was improved by 1.5 % after the port modification.

4. CONCLUSIONS

One of the most significant sources of noise in a screw compressor is that due to rotor contact. This may be intensified if contact is made on the main rotor trailing flank in order to reduce interlobe leakage. In that case the drag forces may overwhelm the pressure forces and cause the gate rotor to flutter or even rattle. The studies presented in this paper confirm that even small modifications in the compressor discharge ports can significantly change the torque on the gate rotor and thereby substantially influence both compressor vibration and rotor noise. Experimental investigation on a prototype production compressor confirmed that, by proper port design, noise was reduced by up to 5.7 dB while, simultaneously the compressor efficiency was increased by up to 1%.

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