# Improving screw compressor performance

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## ABSTRACT

Screw compressor efficiency is improved by reducing internal leakage and this can be effected by minimising the clearance between the rotors and the casing. The effect on performance of three factors which influence the working clearance was analysed. These are: the interlobe clearance, adjusted to the rotor temperature change, rotor contact on the lobe flat side and displacement of the discharge bearing centres. Due account of the results of this study was taken in the design of a large screw compressor, which was then built and tested. The efficiencies obtained were the highest ever reported for screw compressors in the open literature. This confirmed the validity of this approach.

## **1 INTRODUCTION**

Since the performance of screw compressors is highly affected by leakage, any reduction of the clearances within them must improve their efficiency.

Modern rotor manufacturing methods, such as grinding with simultaneous measurement, control and correction of the profile, enable the profile tolerance to be maintained within  $\pm 5$  µm. This enables the clearances between the rotors to be kept below 15 µm. With such small clearances, rotor contact is very likely and hence the profile and its clearance distribution must be generated in such a manner that damage or seizure will be avoided should this occur.

Current practice to avoid rotor contact and seizure is to make clearances larger than required by manufacturing limitations. However, the clearances can be reduced by making their distribution non-uniform around the profile so that should hard rotor contact occur, it will not be in rotor areas where sliding motion between the rotors is dominant.

The design of screw compressors is an interactive process in which the performance estimated in the design process is compared with that obtained from prototype testing. The prototype is then repeatedly modified until the desired values are obtained. However, as more accurate methods of simulating the performance become available, the need for modifying the prototype can be minimised. To this end, the authors have developed a suite of subroutines for the estimation of screw compressor performance. These include facilities for the generation of new rotor profiles, the estimation of the thermodynamic processes within the compressor and hence, the performance changes that are likely to result from modifying the profiles. The accuracy of these estimates has been enhanced by extensive comparison with test results on many machines, followed by modification of the model to obtain better agreement not only with the bulk parameters, such as flow delivery and power consumption, but also with the cyclic variation of important instantaneous values, such as the pressure distribution within the compressor working chamber. More details of this are given by Hanjalic and Stosic 1997. Essentially, the computational procedure is as follows:

- (i) A pre-processor generates the lobe profiles from which the complete screw rotor pair is formed and the working chamber volume function is generated.
- (ii) The performance is estimated by the numerical solution of a set of differential equations, which account for the conservation of mass, momentum and energy and which include estimates of the thermodynamic properties of the working fluid and associated flow processes. The results include not only estimates of bulk parameters, such as the delivery flow rate and power input, but also instantaneous values of how the working fluid pressure and temperature vary within the compressor cycle.
- (iii) A post-processing program is then used to present the results in graphical and tabular form.

This procedure was used to estimate what effect the three main factors that govern the compressor working clearance have on its performance. The results thus obtained were then applied to the design of a 4/5-220 mm rotor diameter oil injected air compressor.

# 2 ROTOR MODIFICATION AND ITS EFFECT ON PERFORMANCE

In order to maximise screw compressor delivery rates and efficiencies, interlobe clearances must be made as small as possible without the likelihood of hard rotor contact between the rotors, in regions where the sliding velocity is high. Three main effects must be allowed for in the design in order to ensure this.

## 2.1 Thermal Expansion of the Rotors and Housing

Although the temperature range over which screw compressors operate is not large, the effects of thermal expansion are highly significant if the small clearances required between the rotors and between the rotors and the housing are to be maintained under working conditions. Thus, the rotor clearances obtained under manufacturing conditions must be estimated while taking account of thermal distortion that will occur when the compressor reaches its operating temperature and pressure and the calculation must allow for the unequal expansion of the rotors in different coordinate directions. An example of this is given in Fig. 1, where the left diagram shows the estimated clearance distribution when the rotors are cold, while, the centre and right diagrams show the clearances after the rotors reach their working temperatures. Additional information about the screw compressor clearance management and other means of improving efficiency may be found in Stosic, 2004.

Despite the cold clearances being adequate, under operating conditions, rotor contact may occur on the round flank, as shown in Fig 1b, or on the flat flank, as shown in Fig 1c.



a) Cold rotors, b) Rotor contact on the round face, c) 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** 

#### 2.2 Rotor contact on the lobe flat side

Oil flooded compressors have direct contact between their rotors. In well designed rotors, the clearance distribution will be set so that this is first made along their, so called, contact bands, which are positioned 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 already shown in Fig 1, the contact band may be either on the rotor round flank b), or on the rotor flat flank c).

The traditional approach is to maintain a high, so called, positive gate rotor torque, which ensures round flank contact. What is not widely appreciated is that there are significant advantages to be gained by maintaining a negative gate rotor torque to ensure that contact, when it occurs, will 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. Here it can be seen clearly 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. Also, negative gate torque is achieved by making the gate rotor lobes thicker and the main rotor lobes correspondingly thinner. The displacement is thereby increased. Thus both these effects lead to higher compressor flows and efficiencies.

# 2.3 Displacement of the bearing centres

Since there must be some clearance in the bearings, the pressure loads will tend to push the rotors apart and displace their centres from their design position with respect to the compressor housing. Thus, if the bearing centres are set to be the same as those of the rotors, the rotors will be eccentric and as a result the clearance between the rotors and housing will be smaller at the low pressure side of the rotors and larger at the high pressure side. Since leakage is caused by the pressure difference, this displacement creates the least favourable rotor position for efficient compressor operation.

Also the resulting rotor displacements from their design positions may result in contact between the rotor tips and the housing unless allowance is made for them during the design. The situation can be remedied by making the bearing centre distance smaller than that of the rotor housing and aligned to maintain a uniform clearance between the rotors and housing. To minimise the rotor interlobe clearance, the bearing centre distance must be even further reduced.

# 2.4 Optimising the rotor profile

Although rotor profiles are designed to minimise the blow-hole area, this is frequently associated with increase in the sealing line length. The optimum profile shape is therefore that which minimises the sum of both the blow-hole and sealing line leakage areas.

## 2.5 **Performance simulations of optimised rotors**

The three possible improvements described in Sections 2.1-2.3 were applied systematically to performance simulations of an oil flooded air compressor of 223mm male rotor diameter when running at a shaft speed of 2100rpm with a suction pressure of 1 bar and a discharge pressure of 9 bar. The results of these are presented in Table 1.

Q[m <sup>3</sup> /min]	$\eta_v$	P[kW]	Psp[kW/m3/min]	η	$t_{out}[^{o}C]$
15.9	0.901	103.7	6.506	0.835	86
16.1	0.911	102.9	6.383	0.850	85
16.2	0.916	102.6	6.327	0.858	84
16.0	0.906	103.7	6.444	0.843	85

## Table 1 Compressor Performance at 1500 rpm and 9 bars

The first row gives the results without any modification. The second row shows the performance obtained by introducing 'hot clearances'. The third row shows the effect of displacing the discharge bearings 50  $\mu$ m in the direction opposite to that of the line of action of the pressure radial forces and the fourth row represent the effects of inducing rotor contact on the flat flank side.

# **3** MODIFICATIONS IN SCREW COMPRESSOR DESIGN

Following the above analysis, a 220 mm diameter rotor oil flooded 4/5 air compressor was designed with a L/D ratio of 1.55, including all the proposed improvements described in Section 2. The rotors are shown in Fig 3 and its estimated performance in Figs 4 and 5.



Fig. 3 4/5 220 mm rotors

The rotors were made of steel and were manufactured by grinding, thus ensuring excellent precision and contact. The rotor L/D ratio was kept optimally low to minimise deflection, maintain favourable dynamic characteristics and achieve the required compressor compactness. The housing was designed for manufacture from normalized nodular iron for easy and precise machining, while ensuring both tightness and structural integrity.

Cylindrical roller bearings and two-point contact ball bearings for the axial loads were employed to ensure long operating life. A high quality mechanical shaft seal was provided for safe and reliable operation.

After manufacture of all the components, the compressor was assembled and tested.



Fig. 5 Estimated specific power as function of shaft speed

# 4 TESTING OF THE PROTOTYPE COMPRESSOR

A photograph of the test rig with the compressor installed in it is presented in Fig 6 and a basic piping and instrumentation diagram of it is shown in Fig 7. The compressor was driven by a 100 kW electrical motor, with its speed controlled by a frequency converter. A six band belt drive with a speed reduction ratio of 1:2.52 was used to transmit the power from the motor to the compressor drive shaft, which was mounted on two pedestal bearings to resist the side load imposed on it. After discharge from the compressor, oil was removed from the compressed air in a separator. The separated oil was then recirculated into the compressor, via a water cooled heat exchanger, and reinjected through a nozle, as a result of the pressure difference between the discharge pressure and the pressure in the compressor working chamber at the oil admission point. The separated discharge air was then passed through an orifice plate, in order to determine its flow rate and then vented to a flue below the test cell.



Fig. 6 Layout of the test rig

Instrumentation scheme of the test rig is shown in Fig 7. The measurements from which performance estimates of the compressor were derived are compressor speed and torque, compressor inlet pressure and temperature  $p_1$  and  $t_1$ , compressor discharge pressure and temperature  $p_2$  and  $t_2$ , the orifice plate inlet pressure and temperature  $p_3$  and  $t_3$  and the pressure drop across the orifice plate  $\Delta p$ . The orifice plate section of the discharge pipe was 100 mm diameter and the orifice platewas designed according to BS 1042. Additionally, the atmospheric pressure  $p_0$  and temperature  $t_0$  were measured.



Fig. 7 Instrumentation of the test rig

Apart from the laboratory atmospheric pressure, which was input manually, all measurements were obtained from electrically generated signals derived directly from the speed, torque, pressure and temperature measurements. These were logged in an Instrunet data logger, programmed to display average values at 5 second intervals. In addition, instantaneous values were displayed visually by a variety of digital electronic meters and centrally at the computer screen. The latter, were very helpful in setting up the required test conditions. The recorded data was then input to a computer as an unformatted data file for further processing. The test schedule employed followed the guidelines laid down by PNEUROP and CAGI in document PN2CPTC1.

The test data were obtained at discharge pressures of 7, 9 and 11 bars. The estimated delivery, power and specific power are compared with the experimentally derived values, as shown in Figs 8 and 9. The results show good agreement between the predicted and test values.

The specific power obtained, as a result of the optimisation procedure described, is 5-8 % lower than that of any other machine of this type, as reported to date in published literature.



Fig. 8 Comparison of the estimated and measured flow and power



Fig. 9 Comparison of the estimated and measured specific power