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Investigation of Screw Compressors for Low Pressure Ratio Applications

Y Lu[1], A Kovacevic[1], M Read[1]*, D Buckney[2], Y Endo[3]

[1] City, University of London, United Kingdom
[2] Glasgow Technical Centre, Mayekawa UK Ltd.
[3] Mayekawa MFG. Co., Ltd.

*Corresponding author: m.read@city.ac.uk

Abstract. Screw compressors have been widely investigated for many applications including air and process gas compression, and refrigeration systems. There is however a surprising lack of literature for low pressure ratio application of these machines, defined here as application requiring volume ratios in the range of 1-1.5. The aim of this paper is to characterise the loss mechanisms for oil injected screw compressors with low volume index (defined as the ratio of maximum to minimum volumes during the internal compression process), V_i. This knowledge will be applied in the future to identify appropriate optimisation of rotor profiles and porting geometry for a range of low-pressure ratio applications. The current study involves the use of chamber models to investigate the influence of wrap angles and porting. This initial analysis will be developed in future work to allow a detailed parametric study of the factors that limit the performance of low V_i oil-injected screw compressors.

1. Background

Positive displacement machines are commonly used to achieve compression of gases. There are many applications where such compressors are required to operate with a low-pressure ratio. This can either be due to the nature of the process, such as in an engine supercharger, or due to variable operating conditions, such is in gas booster systems and refrigeration systems, where low pressure ratio may be required during some operating regimes.

Roots blowers are an example of positive displacement device commonly used for low pressure ratio applications in industrial systems and engine supercharging. Significant work has been done in the past on characterising the rotor geometry and leakage characteristics of these machines [1, 2]. More recent work has focused on improving aspects such as flow pulsations [3] and using experimental and CFD methods to investigate the flow field [4]. These machines have no internal compression, and can be thought of as 'gas pumps'; pressurisation occurs due to working chambers containing low pressure fluid being exposed to the high-pressure fluid, resulting in some degree of backflow. As such, the efficiency of these machines is maximum close to a pressure ratio of 1, and drops rapidly as pressure ratio increases due to leakage at the rotor-to-rotor and rotor-to-casing clearance gaps and dynamic pressure losses during filling and discharge. Helical profiles are sometimes applied in roots blowers in order to achieve progressive engagement between the rotors and reduce vibrations and noise [5]. This leads to an increase in leakage path lengths, but the wrap angles involved are small and this does not significantly affect the operation of the machine.

Twin screw compressors use helical rotors with higher wrap angles, allowing internal compression to occur within the machine. The ability to match the working chamber and downstream pressures before discharge leads to higher efficiency at higher pressure ratio than can be achieved with a roots blower. These machines have been widely investigated for many applications including air and process gas compression, and refrigeration systems [6-8]. There is however a surprising lack of literature for low pressure ratio application of these machines, defined here as requiring a volume ratio of less than 1.5. When compared to simple positive displacement machines such as roots blowers, screw compressors have improved performance, but the increased manufacturing complexity is a drawback. Dynamic compressors are also an option which have some advantages in performance over screw compressors at design point, but have different challenges in terms of machine design, operation, and off-design performance.

This study is motivated by the identification of new industrial refrigeration applications requiring high efficiency compressor operation in this low-pressure ratio region, and hence research is necessary to understand both the fundamental loss mechanisms for screw machines in this operating regime, and the challenges of cost-effective design and optimisation.

2. Performance of a conventional screw compressor with low $\boldsymbol{V_i}$

A conventional screw compressor design has been analysed using the compressor geometry generation and thermodynamic chamber modelling features of the software package SCORG [9]. This screw compressor is designed to operate with V_i values in the range of 1.8-5, with a relatively high-pressure difference between suction and discharge conditions. This configuration does however provide a useful base case configuration for investigating the influence of various parameters on the performance of screw machines when operating with a low-pressure ratio. The modelled geometry is based on a Mayekawa 400L class UD series refrigeration compressor [10], with outlet ports adapted for low V_i operation. The compressor configuration is 4/6 lobes on main and gate rotors respectively, with a maximum operating speed of 3600rpm and a corresponding theoretical displacement of 11700m^3/h.

2.1. Low pressure ratio operating conditions

The chamber model calculations using the SCORG software package have been performed for a range of V_i values, which are achieved by varying the geometry of the high-pressure discharge port of the machine. The operating conditions considered in this analysis are for an NH_3 gas compressor with oil injection. An ideal gas model is used with the following parameters:

- P_{suc} = 3.55 bar, T_{suc} = -5°C
- P_{dis} = 4.62 bar
- Pressure ratio = 1.3
- Oil injection angle = 0°
- Oil supply temperature = 290K
- Oil density at supply temperature = 1.60 mPa.s
- Main rotor speed = 3600 rpm

The chamber model allows calculation of the working chamber pressure profile and the resulting indicated power requirement. The additional mechanical losses relating to the bearings, shaft seal and oil drag losses are then estimated based on empirical relationships between the operating speed, oil viscosity and machine geometry. The resulting net power requirement can then be calculated as shown in Figure 1, with the corresponding results for efficiency and oil-to-gas mass ratio shown in Figure 2.

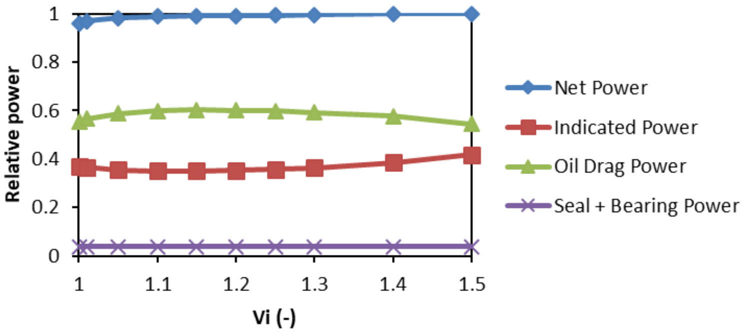
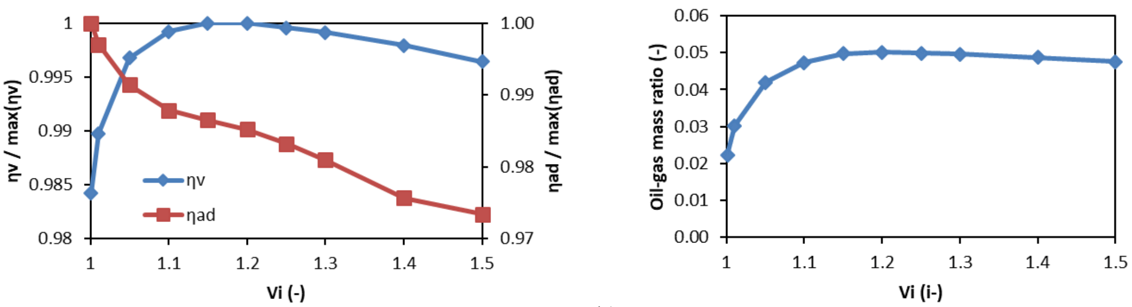
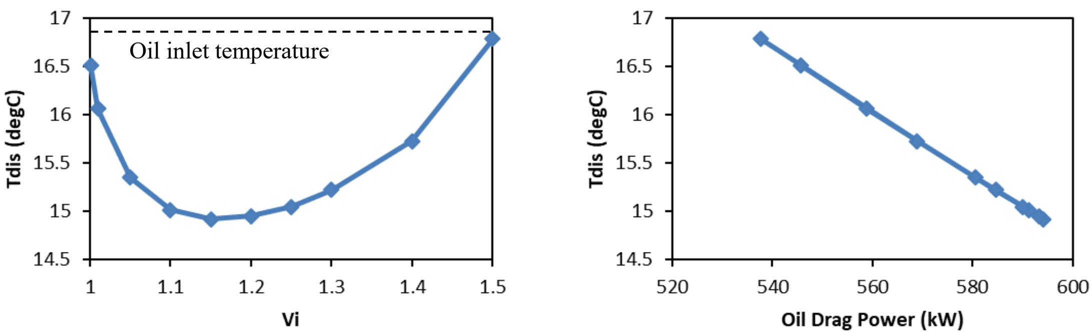


Figure 1 – Breakdown of power requirement as a function of the volume index for the base case compressor



a) b)
Figure 2 – a) Normalised volumetric and adiabatic efficiency, and b) oil-to-gas mass ratio as function of the volume index for the base-case compressor



a) b)
Figure 3 – a) Discharge temperature as function of volume index for the base-case compressor, and b) illustration of oil drag temperature dependence

This initial study of screw compressor operation and performance has indicated that oil drag is likely to be a key factor in overall power consumption for low pressure ratio applications. This is due to the low indicated power requirement relative to the machine size and operating speed. Drag losses at rotor end faces appear to be the dominant oil drag loss [11] and more detailed characterization of this mechanism is necessary to better understand the influence on specific power and adiabatic efficiency. The selection of oil is also an important factor, due to the influence of viscosity on these losses. Due to the low pressure ratio, there is very little temperature rise during the compression process, and the oil temperature at inlet and discharge are similar (Figure 3a). The effect of viscosity is shown in the relationship between discharge temperature and oil drag losses in Figure 3b. A further important factor to consider is the mass flow rate of oil. The results in Figure 2b shown that the oil-to-gas mass ratio varies between 2-5% when operating with such low discharge pressure with the oil injection supply pressurized to the same level. This is lower than would be expected for operation at higher pressure ratios. However, due to the low pressure-ratio, the gas temperature rise is relatively small and the oil is not required to achieve cooling of the gas. The small pressure difference between suction and discharge conditions also leads to relatively low leakage flow rate, reducing the need for oil to be present in clearance gaps to achieve sealing. The main requirement for oil is therefore to lubricate the rotors, and the oil-to-gas ratio can be substantially lower than in screw compressors operating at more conventional conditions with higher pressure ratios. The effects of this reduced oil mass flow rate on oil drag losses should also be considered, along with the potential for oil-free compressor operation, although this has the drawback of requiring more complex bearing and seals design.

3. Influence of geometric parameters on performance of machine with low V_i

The wrap angle of the rotors is a geometric parameter that has an influence on both the swept volume of the machine and the size of the leakage areas, particularly that of the rotor tip-to-casing clearances, during the compression cycle. The influence of this geometrical parameter on the performance of the machine has been investigated by keeping all other machine geometry fixed and varying the wrap angle from 150-300 degrees. The influence of this on three important leakage paths, the rotor tip-to-casing radial leakage areas (A_{rad}) of the main and gate rotors and the blow hole (A_{bh}) are shown in Figure 4. The general trend is that decreasing the wrap angle leads to an increased blow hole area, but decreased max radial leakage areas for both rotors. The change in the total of these three leakage areas is seen to decrease with decreasing wrap angle, suggesting leakage flow rates between working chamber may be reduced. Decreasing the wrap angle also reduces the duration of the internal compression phase so there is less time for leakage to occur. This is supported by the results of the chamber modelling, which show that the volumetric efficiency is increased by decreasing the wrap angle (Figure 5a).

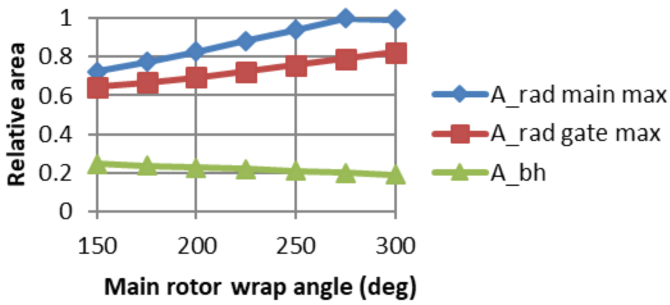


Figure 4 – Influence of rotor wrap angle on maximum radial and blow-hole leakage areas ($A_{rad,max}$ and A_{bh} respectively). Notes areas are normalised by the largest value of $A_{rad,max}$

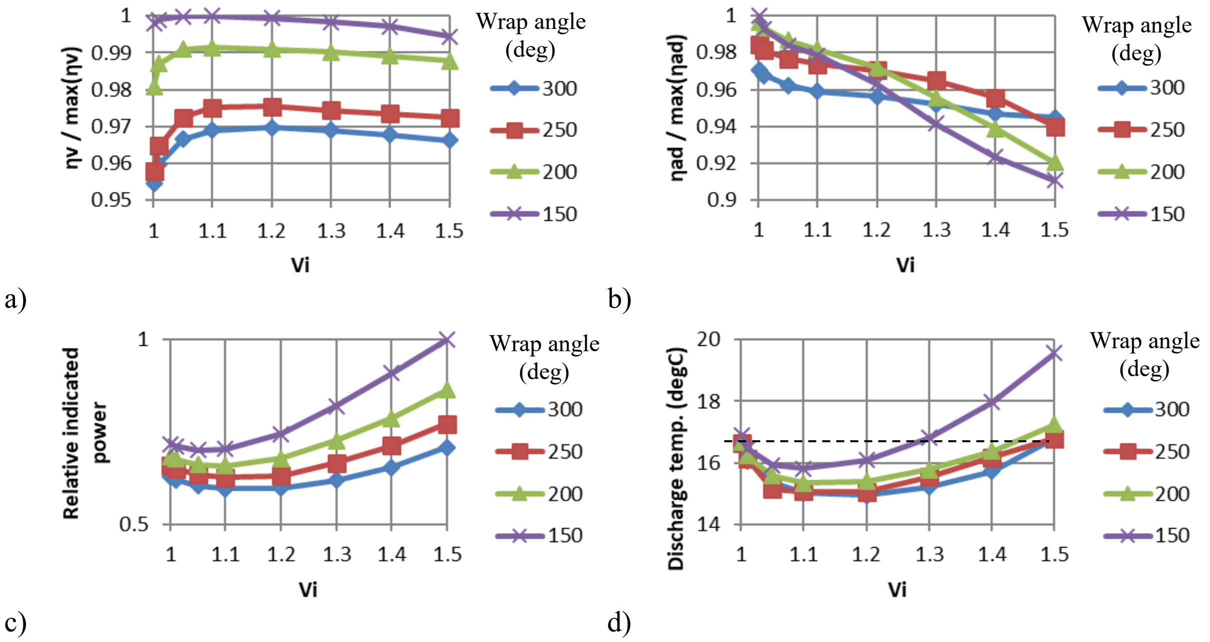


Figure 5 – Results showing normalised a) volumetric and b) adiabatic efficiency, c) normalised indicated power and d) discharge temperature for fixed screw machine geometry with a range of wrap angle values

The results for volumetric efficiency suggest that leakage is minimized with a V_i value of between 1.1-1.2 in all cases (Figure 5a). The results for adiabatic efficiency (Figure 5b) suggest that where under-compression occurs (corresponding to $V_i < 1.15$), this is also improved by reducing the wrap angle, while when over-compression occurs ($V_i > 1.15$) this is no longer the case. For the significantly over-compressed case when $V_i = 1.5$ the maximum adiabatic efficiency is seen to occur with the highest value of wrap angle (300deg). The cause of this reduction in adiabatic efficiency with increasing wrap angle can be explained by considering the indicated power, as shown in Figure 5c. Reducing the wrap angle leads to a higher maximum rate of change of volume in the working chamber, while the maximum inlet and discharge port areas remain similar; the fluid must pass through the ports over a shorter time period, resulting in higher velocities and increased pressure losses during the filling and discharge phases, increasing the indicated power required for the compression process. Figure 6 shows a comparison of the working chamber volume and port areas for cases with $V_i = 1.1$ and wrap angles of 150 and 300 degrees, while Figure 7 shows a comparison of the resulting $p - V$ diagrams for the same cases, clearly showing the significantly higher indicated power for the lower wrap angle. The higher discharge temperature achieved with lower wrap angle does tend to reduce the oil drag power, and the slightly higher maximum working chamber volume combined with the higher volumetric efficiency results in 6% higher mass flow rate for this case. The net effect is a small increase in the adiabatic efficiency when the wrap angle is decreased from 300° to 150° for machines with $V_i = 1.1$ (as seen in Figure 5b).

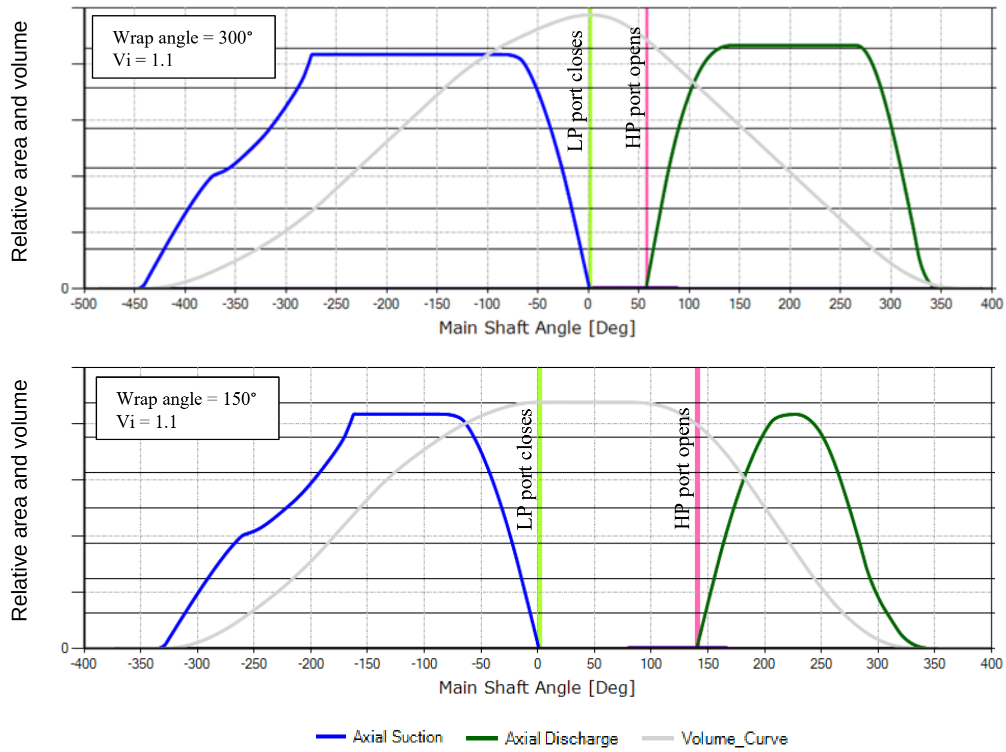


Figure 6 – Working chamber volume and port areas for cases with $V_i = 1.1$ and wrap angles of 300° and 150° respectively

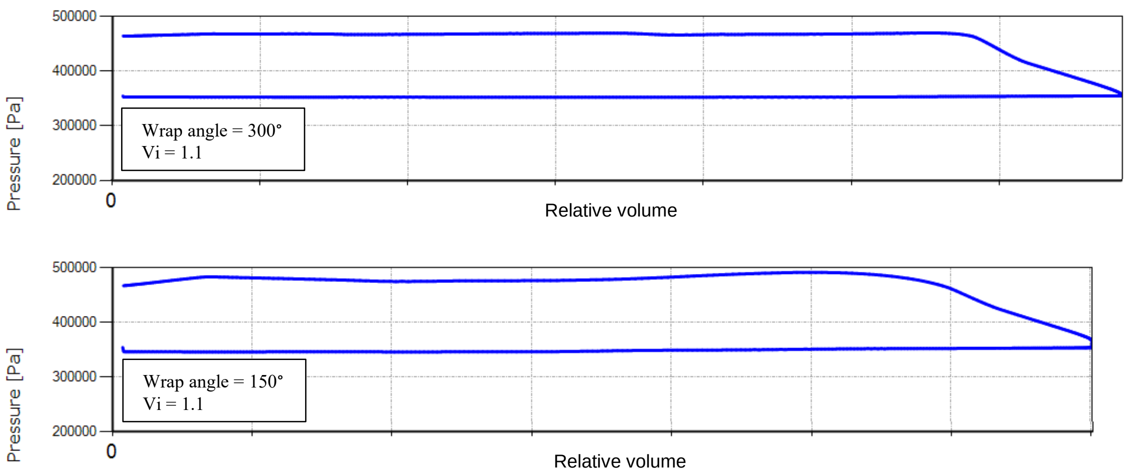


Figure 7 – Working chamber pressure vs. relative volume for cases shown in Figure 6

4. Conclusions

The main conclusions from the current study are as follows:

- Mechanical losses are an important consideration for screw machines operating at low pressure ratio due to the relatively low indicated power requirement.
- Oil drag losses appear to be a major factor in determining the adiabatic efficiency of the machine, and further consideration of the oil properties and the oil injection system is necessary.
- Oil mass flow rate should be provided in order ensure sufficient lubrication of rotors, but the sealing and cooling effects are less important for low pressure ratio operation. The optimum oil-to-gas mass ratio is therefore expected to be significantly lower than in normal screw compressor operation.
- The wrap angle of the rotors is shown to affect the volumetric efficiency and indicated power. For the case considered here, decreasing wrap angle from 300deg to 150deg increases adiabatic efficiency as long as the fluid is not over-compressed ($V_i < 1.15$)

Future work will focus on more detailed characterization of the loss mechanisms, and investigate the influence of a wider range of geometrical parameters such as rotor profile, lobe number and rotor length. It is also planned to use experimental test data to improve and validate these empirical models for the loss mechanisms at low V_i operation. The aim is then to carry out a detailed parametric study of the factors that limit the performance of low V_i oil-injected screw compressors, and identify cost-effective design solutions to improve efficiency of these machines.

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