Experimental validation of a geometry model for twin screw machines

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ABSTRACT

Aspects of screw compressor geometry are often idealised in mathematical models used for design and optimisation of twin screw machines which can lead to errors in performance predictions. A model that accurately represents geometrical elements such as the volume and flow area history, variable interlobe clearances and geometry of actual compressor port shape has been previously presented by the authors. In this paper the geometry values from that model were used with an existing thermodynamic model to predict the compressor performance with consideration of different rotor profiles, actual discharge port and thermal effects on clearances for an oil free compressor. Results were verified using experimental test data.

NOMENCLATURE

A	rotor centre distance / area	F	interpolation factor
G	clearance gap	h	enthalpy
m	mass	p	pressure
$\dot{Q}$	heat flow rate	R	rotor outer radius
T	temperature	U	internal energy
V	volume	Δ	delta, small change
μ	coefficient of thermal expansion	θ	cycle angle
ω	rotational speed	Vi	volume index

Subscripts:

A	axial	a	ambient
c	casing	g	gas
I	interlobe	i	flow into system
o	flow out of system	R	radial
r	rotor	x	component in x-axis

1 INTRODUCTION

Twin screw compressors are extensively used in refrigeration, gas processing and energy industries. Within these industries, compressor applications can greatly vary in terms of flow-rates, pressures, temperatures and working fluids. For each specific application, performance can be improved by use of a uniquely optimised rotor profile (1). This is possible with modern rotor profiling methods that deliver general improvements in efficiency and displacement whilst providing a high degree of versatility in the shape of the profile (2). Stosic (1) emphasises the need to simultaneously consider all design parameters such as the type of bearings and the quantity of oil injection during this rotor optimisation. Such a complex multi-variable design process is possible with the use of numerical computer simulations. Quasi one-dimensional numerical chamber models are now well established (3), (4), (5) for predicting the performance of twin screw compressors. Their ability to predict overall performance with accuracy and speed makes them indispensible and the continuing tool of choice for the preliminary design and optimisation of positive displacement machines. The thermodynamic model presented in (3) calculates the thermodynamic properties of the working fluid at successive angular positions in the compression cycle by considering conservation of energy and continuity within a control volume. It is assumed that the fluid properties are homogeneous over the control volume and kinetic energy is neglected on the basis that it is insignificant compared to the internal energy of the fluid (3).

$$\omega\left(\frac{dU}{d\theta}\right) = \dot{m}_i h_i - \dot{m}_o h_o + \dot{Q} - \omega p \frac{dV}{d\theta} \tag{1}$$

$$\omega\left(\frac{dm}{d\theta}\right) = \dot{m}_i - \dot{m}_o \tag{2}$$

Knowing the rate of mass flow in and out of the compressor chamber is critical for solution of the internal energy (equation 1) and the net mass (equation 2) within the control volume. Mass flows account for suction and discharge processes, leakage flows, recirculation and fluid injection. Estimating each of these flows relies on accurate geometry representation and suitable flow models.

Precise calculation of actual port area history is required to fully evaluate flow fluctuations in ports and give useful insight into particular aspects of compressor design. Mujic (6) developed procedure to accurately represent arbitrary port shape area for quasi-one dimensional and integrated models. He then used the actual discharge port to assess the impact of the port shape on pressure pulsations and noise generation. Chen (7) modelled the part load performance of a refrigeration cycle by including accurate area curves for the slide valve recirculation and radial discharge ports into an established model (8).

Leakage paths occur due to radial (rotor tip to casing bore), axial (rotor end face to casing end face) and interlobe (rotor to rotor) clearances. The area of each leakage path is proportional to the length of its sealing line which is calculated as a function of the compression cycle. Assuming a constant, uniform clearance gap the area of the leakage path is known. In reality it is common for the interlobe sealing line to have a non-uniform clearance distribution which can have a notable effect on the overall leakage area and the resulting thermodynamic performance (9). Accurate representation of the change in clearance gaps due to thermal and other effects is therefore critical for reliable performance predictions.

More often than not, thermodynamic models rely on some empirical feedback to compensate errors due to unknowns such as complex effect of operating deflections or reduction of the leakage flow areas due to presence of oil. For example, Hsieh (10) presented a procedure to determine empirical constants in a 'training process' which optimises multiple constants using a range of test cases. Hsieh (10) and Wu (8) presented models which once calibrated, or trained, have good agreement between simulated and measured results. Caution should be exercised when using such models, which rely on empirical constants or coefficients, to predict the performance of compressors different than the machine used to derive the coefficients. Even two equivalent compressors from the same manufacturing batch can perform differently on test due to the influence of manufacturing and assembly tolerances which are significant relative to the small clearances in screw machines. The widely utilised model developed by Hanjalic and Stosic (3) uses standard, verified flow equations and coefficients. Since operational clearances are unique to each compressor and the operating parameters (11), accurate absolute performance prediction can be achieved only if the input of the operational clearances is accurately defined. These are normally based on nominal design clearances corrected on the basis of measured correction factors or thermal deformations accounted for by complex 3D FEM or CFD calculations (12).

In this paper, the procedure for calculation of geometry characteristics (13), provided the input to the widely used thermodynamic model for performance prediction of screw compressors of Stosic et all (2). This is used to assess the impact of specific geometry features on the overall thermodynamic performance of the oil free compressor. Performance predictions are validated using a range of test data obtained by measurements on in-house test facilities.

2 GEOMETRY MODEL

2.1 Summary of the geometry model

The development of this geometry model and its validation based on comparison with 3D CAD model is presented by authors in (13). The main objective was to accurately calculate required geometric characteristics of a screw compressor in advance of thermodynamic calculations. These values are represented as a function of the main rotor angle, θ and comprise of the set of volume and flow area curves stored in a single matrix. The flow areas include axial and radial inlet and outlet ports; radial, axial, interlobe and blow hole leakage flow areas; economiser, liquid injection, oil injection, recirculation ports etc. The aim of the model is to accept rotor profile of any type and from any source either calculated or measured, allowing flexibility and independent rotor comparisons and to support the use of arbitrary ports. The model is based on the direct numerical integration of the actual chamber volume and flow areas.

2.2 Different profile types

Two distinctly different rotor profiles have been evaluated using this geometry model presented in (12). The objective is to assess how accurate the thermodynamic simulation is for different profile shapes with considerably different blow-hole and sealing line characteristics. The performance was calculated based on the coordinates of SRM symmetric profile and the retrofitted N-profile which has improved displacement and efficiency. The verification was performed by comparing calculated performance with measurements of a Howden oil free compressor with original symmetric circular profile and retrofitted "N" profile, as shown in Figure 1.

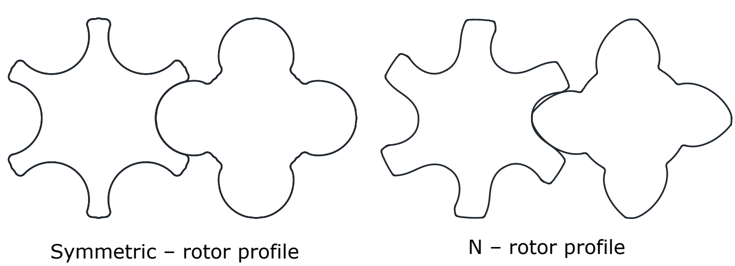


Figure 1 - Profiles of rotors used in this research

2.3 Representation of actual ports

For maximum model accuracy and validity it is necessary to represent real geometry of suction ports, discharge ports, part load re-circulation passages, liquid injection ports and economizer ports. Commonly, theoretical ports are generated from the rotor profiles based on the required volume index. The geometry model utilised for this research calculates port areas using actual point coordinates from drawings or measurements which may differ from the theoretically ideal port shape. In this case the actual volume index is derived from the real port geometry.

The left hand side of Figure 2 shows the actual port geometry of the compressor under investigation. The retrofit N-rotor profile is overlaid to show the mismatch in the rotor and port shape. By assessing area A_2, it is clear that the shape of the port does not closely follow the shape of the rotors which can potentially cause difference in the actual timing of port closing and opening and consequently different Vi compared to the circular profile. Also, due to the constraints in casing design, the minimum radius on the actual port does not coincide with the rotor root causing that the maximum theoretical port area cannot be achieved.

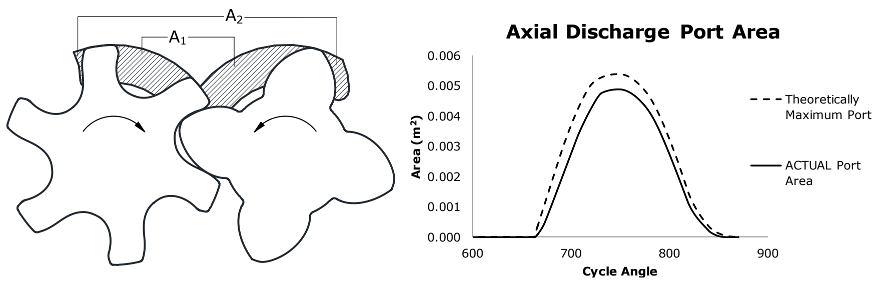


Figure 2 - Actual axial discharge port and resulting actual area curve

The right side of Figure 2 shows the history of the theoretical and actual axial discharge port flow areas. The dashed line represents the theoretically maximum port area curve that could be achieved for the current N-rotor profile with set volume index. The continuous line represents actual port area history with this combination of rotor profile and port shape. The opening timing for both ports is within 1 degree of male rotor rotation but due to the greater width on the retrofitted gate rotor lobe there is a slight delay of opening at the female side. The actual port area increases more gradually and doesn't achieve the same magnitude.

2.4 Operational clearances

The thermodynamic model calculates port and leakage flows based on flow equations which utilise a flow coefficient and a friction coefficient (3). These coefficients are well established and are not the subject of this investigation which instead focuses on the area used for flow calculation. The clearance gaps, used to calculate leakage area, are parameters that have a significant bearing on the performance predictions. The following procedure represents an attempt to approximate the operational effects of the rotor and casing temperature deflections using empirical factors that would apply for this compressor.

In oil free compressors, the clearance gaps change predominantly due to thermal expansion of the rotors and casing (11). The local change in the radial gap, ΔG_R, depends on the local relative expansion between the rotors and the casing. In practice the actual rotor and casing temperatures are difficult to determine and are not uniform. Thermal displacements are typically calculated using finite element analysis, for example in (12). This is not used in the current, generic modelling approach. Instead, the following method which assumes an '*effective temperature increase'*, ΔT_{rc} was applied (Equation 3). This represents the uniform increase in rotor temperature from suction to discharge that is equivalent to the average net effect of rotor (subscript 'r') and casing (subscript 'c') expansion. It is calculated here as a function of the operational gas discharge temperature and the suction temperature:

$$\Delta T_{rc} = F_{rc}(T_{2g} - T_a) \tag{3}$$

F_{rc} is an interpolation factor which is specific for this compressor and derived experimentally. This procedure allows the gas discharge temperature, T_{g2}, to be used for estimating changes in the radial clearance gap due to thermal deformation at specific duties as per Equation 4.

$$\Delta G_R = \mu R \Delta T_{rc} \tag{4}$$

Similarly, equations 5 and 6 apply to the change of the interlobe gap size, ΔG_I. In this case the effective temperature *increase*, ΔT_{rr}, describes the uniform rotor temperature increase that is equivalent to the average net effect of rotor expansion and any change in the rotor centre distance due to casing expansion. This uses a second interpolation factor, F_{rr}:

$$\Delta T_{rr} = F_{rr}(T_{2g} - T_a) \tag{5}$$

In equation 6 the calculated value ΔG_{Ix} is the component of the change in the interlobe gap that occurs in the x-direction along the line which connects rotor centres, A:

$$\Delta G_{Ix} = \mu A \Delta T_{rr} \tag{6}$$

ΔG_{Ix} is used to calculate the local change in the interlobe gap normal to the rotor surface. Figure 3 shows the resulting operational clearance distribution of the interlobe gap normal to the rotor surface due to a change ΔG_{Ix}. The original clearance distribution on the left is uniform and the new distribution on the right is predominantly reduced at the rotor roots:

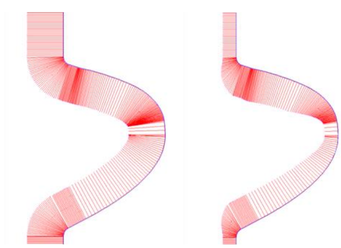


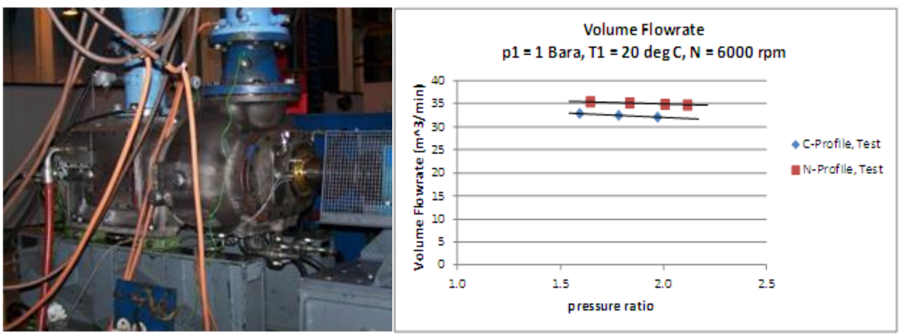
Figure 3 - Clearance distribution change due to rotor thermal expansion.

The factors F_{rc} and F_{rr} must be established empirically in the first instance if they are to represent a reasonable approximation of thermal effects on operational clearances. Once the factors are known and verified, the simulation procedure is as follows: The interlobe and radial leakage areas are initially calculated using the 'cold clearances' specified in the compressor design. These initial area curves are used to estimate the gas discharge temperature which is then fed back to the geometry program which re-calculates the leakages areas using the 'operational clearances' estimated using equations 4 and 6. The procedure is then iterated as necessary.

3 MODEL VERIFICATION

3.1 Compressor test details

Test results used in this paper are from a Howden HS/204/165/26 oil free compressor – Figure 4a. The rotors have a 4/6 lobe combination and equal diameters of 204mm. The ratio of the length to the diameter is 1.65, the built in volume ratio is 1.98 and the wrap angle of the main rotor is 300 degrees. The main rotor is directly driven from the suction end where timing gears are fitted to drive the gate rotor. This design includes a number of mechanical seals to prevent oil entering the compression chamber and also features a cooling water jacket on the compressor casing. All testing was performed with atmospheric air at suction. Tests were performed over a range of pressure ratios and speeds. Test parameters were monitored and recorded continuously using a data logging system. Averaged steady state test points were later extracted from the real time test data. Raw results were corrected to p1 = 1 Bara and T1 = 20 degrees C and to the targeted speed. The flow measured on test for each of the profile types is shown in Figure 4**Error! Reference source not found.**b. All other compressor design parameters, including the design clearances have been held constant.



a) test rig b) test results

Figure 4 Oil free compressor HS204/165/260

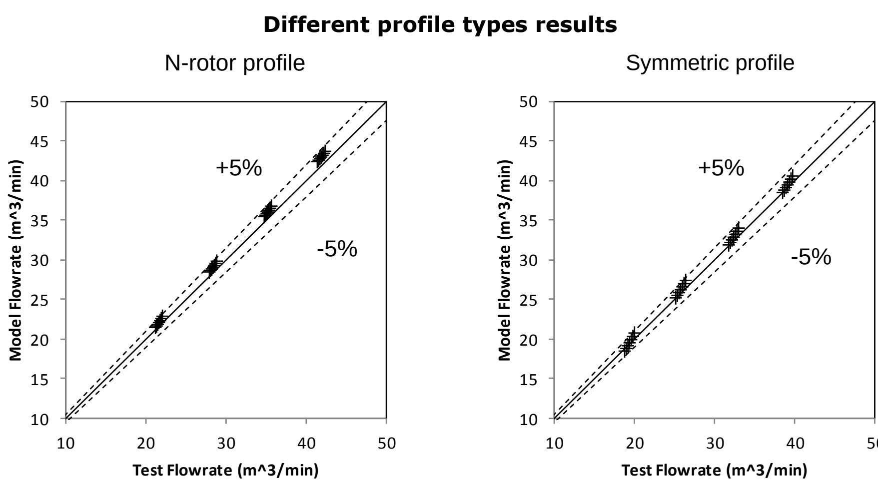


Figure 5 - Modelled flow against measured flow with different profile types

Figure 5 shows predicted volume flow against measured volume flow for speeds of 4000, 5000, 6000 and 7000 rpm at a range of pressure ratios from 1.6 to 2.2. The clearances used for modelling were adjusted to match for the N-rotor profile volume flow rate at 6000 rpm. In the case the same clearance input parameters were then used for all test points and both profile types. The accuracy of the predictions, shown in Figure 5, is similar between profiles that have significantly different geometry characteristics.

3.2 Actual Port Geometry results

Figure 6 shows the history of modelled internal pressure with the angle of rotation for theoretical and real ports which are presented in Figure 2. The slower opening of the actual discharge port has resulted in a slight lag in the time taken for the internal pressure to stabilise to the discharge pressure but the peak pressure is unaffected. The effect of this extended period of over-compression on the performance is shown in Table 1 over a range of pressure ratios. The main effect on performance is a small increase of power less than 1% at a pressure ratio of 1.6. As the pressure ratio is increased (reducing the over-compression) this power increase is reduced.

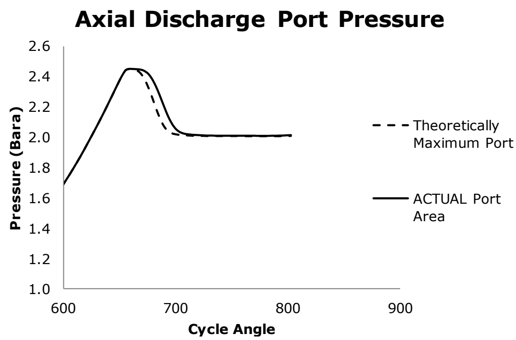


Figure 6 - Pressure history during the discharge process

Table 1 - Change in results with actual port area

	Change in modelled result with actual port		
Pressure ratio	Power	Flow	Temperature
1.6	0.88%	-0.28%	-0.23%
1.8	0.81%	-0.17%	-0.30%
2.0	0.76%	-0.12%	-0.31%
2.2	0.17%	-0.15%	0.56%

The ability to perform relative comparisons of different ports in the model offers useful insight when considering compressor design and optimisation. In this case the results suggest that investing resources to re-designing and manufacturing an updated casing will offer limited benefits.

3.3 Sensitivity to operational clearance corrections

Using the test results for the N-rotor profile a number of simulations were run to investigate the suitability of the equations 3 to 6 which are used to adjust the leakage area through the radial and interlobe clearance gaps to approximate the thermal distortion during operation. The model was initially run with the actual design clearances to provide a baseline; this was achieved by setting both factors, F_{rc} and F_{rr}, to 0. The baseline results are shown by the dashed line in Figure 7 and Figure 8. With cold clearances the model under-predicts flow and over-predicts temperature which suggests that the modelled net leakage flow-rate is too high.

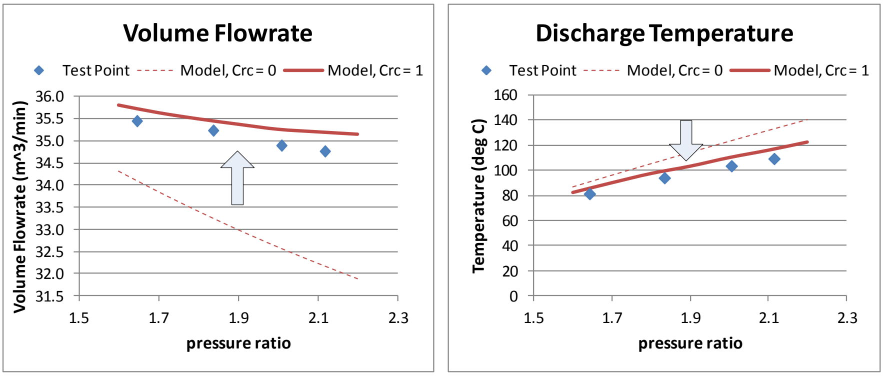


Figure 7 - Effect of modifying rotor to casing clearance gap

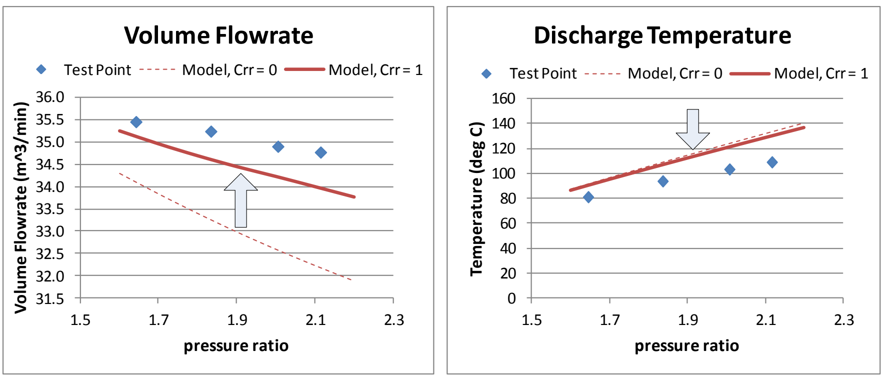


Figure 8 - Effect of modifying rotor to rotor clearance gap

In Figure 7 only the rotor to casing radial clearance gap was reduced by arbitrarily setting the factor F_{rc} to a value of 1. Modification of the radial operational gap alone is capable of bringing the modelled flow very close to the flow measured on test for all pressure ratios. The gradient of the modelled discharge temperature against the pressure ratio is improved when the radial gap is reduced however the modelled discharge temperature is still slightly higher than the measured values. In Figure 8 only the rotor to rotor interlobe clearance gap was reduced by arbitrarily setting the factor F_{rr} to a value of 1. This clearance reduction again shifts the predicted flow closer to the test flow however the effect on the discharge temperature is small in this case.

3.4 Results with operational clearance corrections

For the oil free compressor investigated in this research which has jacket cooling, the rotor to casing thermal reduction factor was set to F_{rc} = 0.8. This value closely reproduces the reduction in the radial gap measured on test for this type of compressor (14). The rotor to rotor thermal reduction factor was then modified to target the flow measured on test resulting in a value F_{rr} = 0.2.

The results from the computer model are compared against the test results at 6000rpm in Table 2. The flow calculation is consistently good across all pressure ratios with a maximum error of 1.1%. The temperature prediction was reasonably good across all the pressure ratios at this speed with error ranging from -1.2% to 5.1%. As this overall compressor power is dependent on assumed mechanical losses the main parameters of interest for verification of the current compressor model with updated geometry calculation are the delivered flow and the discharge temperature.

Table 2 - Difference between model and test results

4	Difference between model and test		
Pressure ratio	Power	Flow	Temperature
1.6	-3.57%	0.64%	-1.19%
1.8	0.89%	0.56%	2.68%
2.0	3.35%	0.76%	4.41%
2.2	4.52%	1.07%	5.07%

Figure 9 shows predicted volume flow against measured volume flow over pressure ratios of 1.6 to 2.2 bar and speeds of 4000rpm to 7000rpm. The maximum error for the modelled flow was 2.4% at a pressure ratio of 2.2 and a speed of 4000rpm.

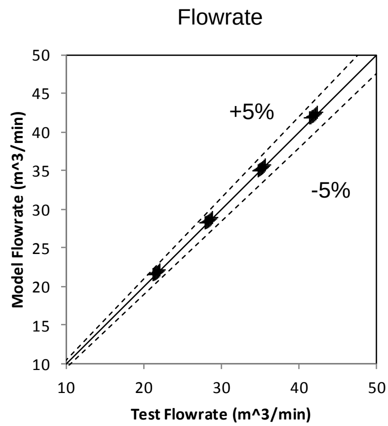


Figure 9 –"N" rotor profile –Comparison of modeled and test flow rate

5 CONCLUSIONS AND FURTHER WORK

The geometry model which calculates volume and flow area history was validated against measurements on oil free compressor with two different rotor profiles. The predicted performance is in good agreement over the full range of test results. This provides further verification of the established compressor model (3) and validates the application of the updated geometry calculation procedures. In section **Error! Reference source not found.** the model shows a consistent level of accuracy for different rotor profiles.

The results in section 3.2 suggest that in the case of this particular "N" rotor retrofit the benefit of investing resources to re-design and manufacture a casing port modification is limited. The updated model is a useful tool when strategically prioritising development work.

In section 3.4 the model accounts for operational effects and provides improved correlation with test data over the full range of test pressures and speeds. There is some debate at this stage as to how applicable the empirical correction factors derived from this test are for different compressor builds that will have different assembly clearances.

Further verification and model development are underway – specifically to assess the use of operational clearance adjustments with oil injected compressors in which the leakage flow area and component temperatures are significantly influenced by the presence of oil. Additional cases with elevated suction pressures and different fluids would also be beneficial. Measuring PV diagrams on test will allow more detailed assessment of the results. In future the developed geometry program will be used to add more novel modelling capabilities such as compressors with variable lead rotors.

6 ACKNOWLEDGEMENTS

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