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# Manufacturing method and performance assessment for variable lead vacuum rotors

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**Abstract.** In recent years variable lead rotors have been produced, mainly for vacuum applications, involving a multiple pass manufacturing process which is necessarily time-consuming. A faster method of manufacturing such rotors uses a full profiled disc-type milling or grinding tool but involves clearance variations along the length. These effects have been assessed by computer modelling to quantify any disadvantages. The results indicate that the effects on performance are negligible and the profiled disk tool process is suitable for such components.

## 1. Introduction

For certain types of twin-screw compressor, notably those used in vacuum applications, there has been a requirement for a reducing inter-lobe spacing along the length of the rotors. This is referred to as ‘variable lead’, where lead is defined as the axial advance per 360 degrees progression around a helical flank.

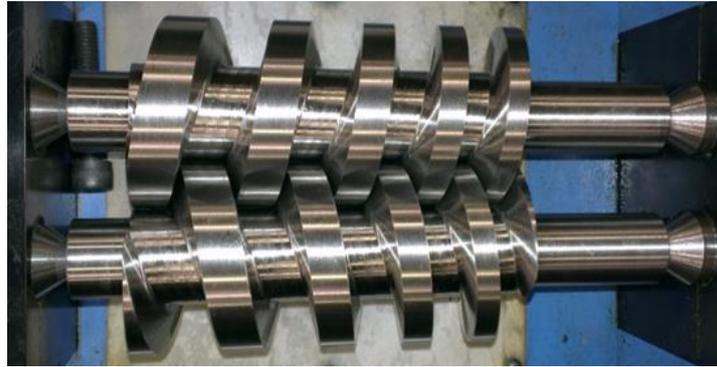
One method which has been used in production has been a rotating ‘end mill’ machine principle, where the flank is generated by multiple passes along the length of the rotating part. The successive engagement positions between the tool and the part are calculated such that the envelope of the individual facets so produced will approximate to the desired profile. The ‘end’ tools used have a relatively simple shape such as a sphere, since an attempt at a dedicated profile would be adding cost without any benefit.

This method of manufacture can require cycle times of several hours for a medium-sized rotor, and if larger production quantities are required, the cost per part may be high compared with disk-type tool which produces the entire profile in a single pass.

It has been assumed that the variable lead is an obstacle to using full-profiled wheel or disk type tool, because although the tool path can be progressively changed during a cycle, the profile which results may be correct for one end of the helix, but will be distorted at the other end, causing a mismatch between the two engaged profiles, with a consequent increase in leakage and reduction in efficiency.

It was decided to optimise the tool path to minimize the mismatch and leakage effects (Section 2), and then to investigate, by CFD modelling, how much this mismatch impacts the final performance of the compressor (Section 3).



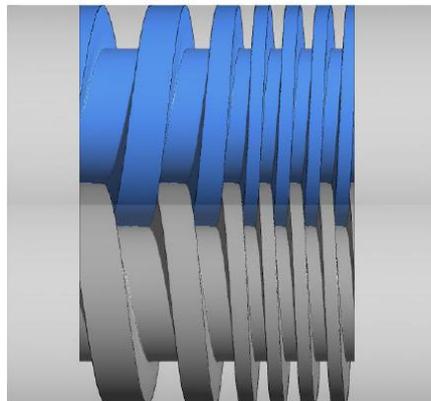


**Figure 1.** Variable lead screws made using a full-profile disc tool

## 2. Manufacturing process and simulation for a full-profile disk tool

The method of machining a rotor with a variable-lead screw was developed many years ago and is described in Davey [1]. An example is shown in Figure 1. The method uses a full-profile disk tool with adjustments of some or all of the following: head angle, approach angle, work rotation and infeed. The rotors engaged and rotated smoothly.

Assuming that the clearances at the discharge end are more important when determining leakage, it is reasonable to keep these as initially designed, and in this case the deviations will increase towards the inlet/suction end, which will therefore have slightly greater clearances and might be expected to give more leakage. In order to estimate the commercial viability of using a fast manufacturing process using disk-type milling, it is necessary to estimate the ‘leakage cost’ of the compromise profile clearances which result.



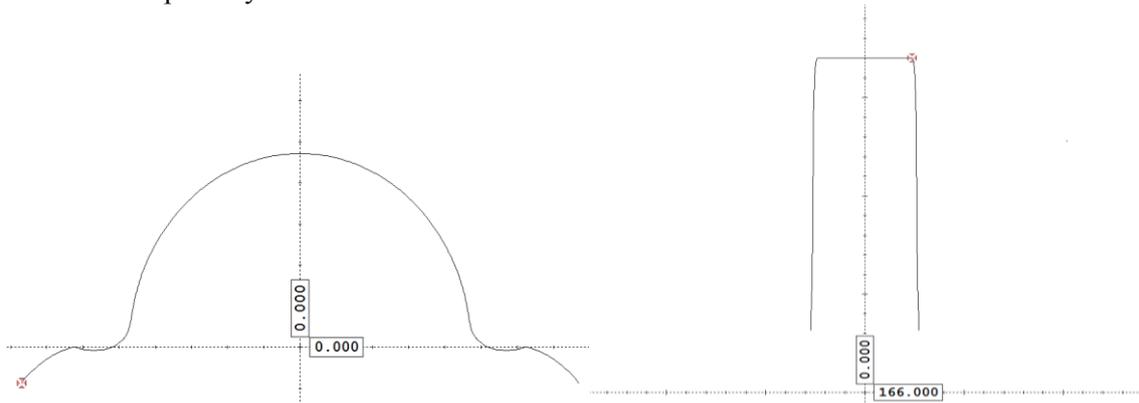
**Figure 2.** Combined constant and variable lead model, discharge end to the right

Figure 2 shows the rotor design used in this study of the effect of small clearance mismatch on performance. It has two segments, with constant lead at the discharge end, and variable lead at the suction end. The main body dimensions were:

Outside diameter	156 mm
Short lead	21 mm
Longest lead	72 mm
Total length	190 mm
Inter-lobe clearance	0.2 mm

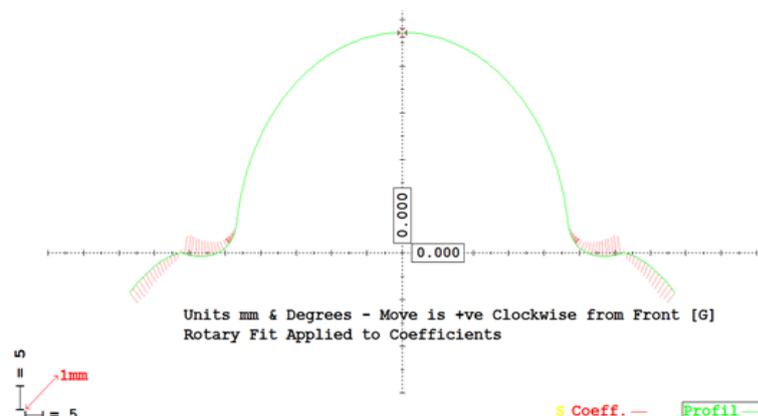
The transverse profile of the rotor, and the tool used to produce the constant lead segment are shown in Figure 3. This tool produces correct clearances over the constant lead segment only.

The same tool was used to make the variable lead segment, i.e. there was no modification of the tool profile. There is no doubt that a more favourable profile would be achieved if movements of head angle, approach angle, infeed and work rotation are applied. In this case, to reduce the complexity imposed on the process, the profile of the variable lead segment was simulated without infeed or approach angle movement. Also, it will be obvious that where variable lead is required, opposite flanks must be manufactured separately.



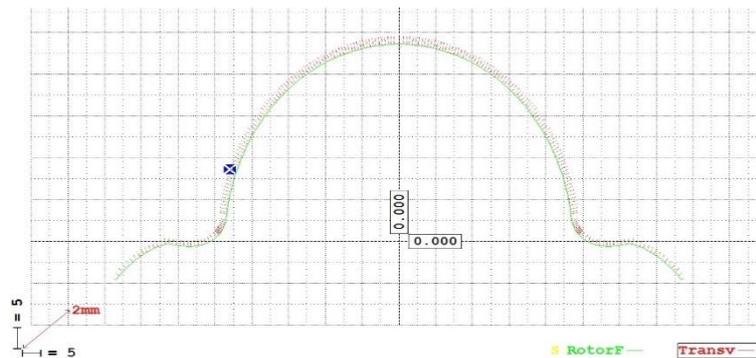
**Figure 3.** Transverse section of rotor (excluding O.D.) and tool

The variable lead segment was divided into five sub-segments, and the profile of each sub-segment was calculated for input to a CAD model. The intention was to keep the minimum inter-lobe clearance close to its nominal value. An iterative process was used to find the best tool paths. The program calculated the rotor profile with estimated tool path adjustments, next the clearance was calculated using a pair of rotors with this actual profile, then feedback was used to give further adjustments to the profile calculation to optimise the clearances. The clearances at the suction end were optimised by adjusting the tool head angle and work rotation, using an iterative procedure. The effect of changing head angle is shown in Figure 4.

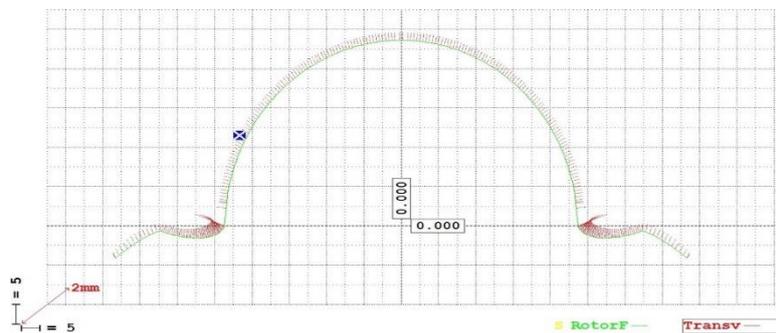


**Figure 4.** The effect of changing tool head angle

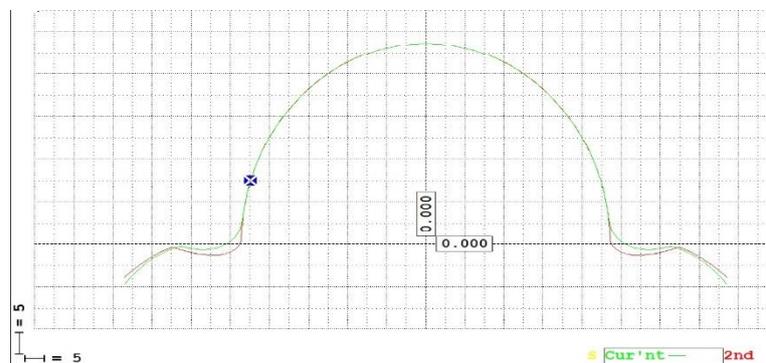
Figure 5 shows the target clearances as designed for the full length of the rotors. Figure 6 shows the calculated clearance distribution at the suction end using the tool profile which is correct for the discharge end. It can be seen the clearance is no longer uniform. The inter-lobe clearance has increased significantly at the intake end. Figure 7 shows the two suction end profiles superimposed, green being the target, and red being actual.



**Figure 5.** Designed clearance distribution over full length



**Figure 6.** Calculated suction end clearance distribution using same tool profile as for discharge end



**Figure 7.** Suction end profiles superimposed, green=target, red=actual

Two CAD models were created, one with uniform clearances over the whole length based on the reference profiles, and the other using the simulated profiles with non-uniform clearance. In both cases the reference profile was used over the uniform lead section

### 3. CFD analysis for comparison of rotor performance

Various modelling approaches are available for the design and analysis of thermodynamic and flow processes inside the working chamber of twin screw machines such as compressors and expanders. For a long time these models have been formulated based on lumped parameters and solution of equations of conservation of mass and internal energy of the working fluid. Although very fast, such quasi-one dimensional models need fine tuning of important geometrical parameters such as leakage gaps, port areas and the chamber volume variation with time. In recent times, a more accurate representation of the real 3D rotor geometry has been made possible by the use of computational fluid dynamics. Although currently employed CFD models are enormously slower in comparison to lumped parameter models, they have high resolution and capture the flow process in its entirety, so were felt to be more reliable. (The model is based on conservation of mass, momentum and energy equations applied within continuous fields with negligible molecular interactions, and absolute vacuums cannot be solved by these methods, so this is a limitation. It will be valuable to compare the two methods in a future study.)

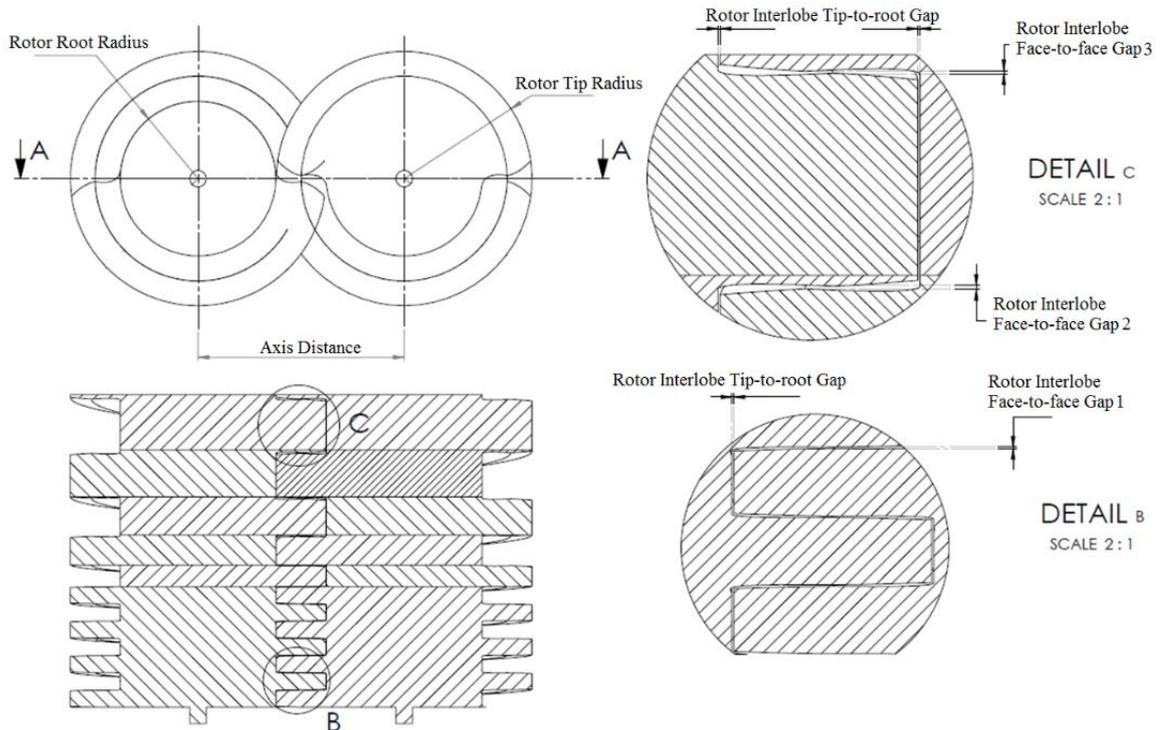
The major challenge in CFD modelling for twin-screw rotors is the treatment of highly deforming domains that need a mesh to represent the geometry. Studies such as Kovačević et al [2], Vande Voorde et al [5], and Rane et al [3] have used customised grid generation tools in order to address this challenge of deforming domains for CFD analysis of twin screw compressors, hook and claw type pumps, oil injected twin screw compressors etc. In the application of these models to the analysis of twin screw vacuum pumps with variable lead rotors, it is possible to create such a customised model for grid generation and solve the flow. Rane [4] has presented a full development procedure for variable twin-screw geometry rotor grid generation and an application to dry air, low-pressure compressor performance has been evaluated.

The challenge required for the presented study, i.e. the analysis of variable lead rotors from a manufacturing and performance perspective, was of a different nature. The variable lead rotors to be manufactured using the proposed method had small deviations in rotor to rotor clearances and the impact on pump performance was required to be estimated at the desired operating conditions. Analysis of such a variable lead screw rotor with variation of clearance has not been reported in available literature. Existing CFD modelling techniques have used uniform clearances in the representative mesh for calculations. There are reports which demonstrate the change in clearance and performance impact but these clearance changes are uniform along the rotor length [4].

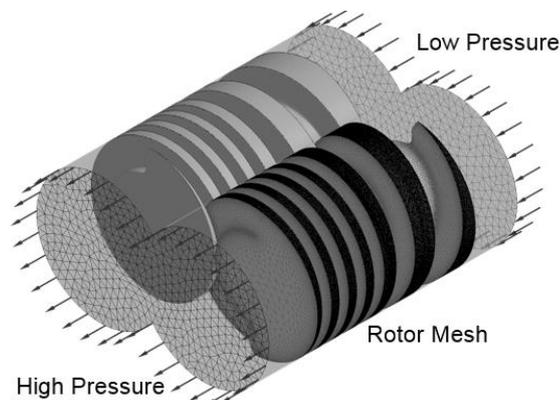
Figure 8 shows the construction of the rotor geometry with variable lead and variable clearance. Five rotor length segments were identified and cross section profiles were varied in each of these lengths to achieve the lead and clearance variation. For the CFD model shown in Figure 9 the fluid volume with leakage gaps is required. This was extracted from the CAD model by subtracting the rotor volume from the bore volume. This operation captured the rotor-to-rotor and rotor-to-bore leakage gaps. The entire fluid volume was then meshed using ANSYS meshing with a refined tetrahedral grid. The total number of nodes in the mesh was 1,324,250. Another geometry was constructed with the same lead variation but with ideal clearances i.e. without the manufacturing deviation. The objective of the CFD analysis was now to compare the performance of the two rotor geometries under same operating conditions. The performance has been compared using the parameters of leakage flow, pressure distribution, torque and thrust forces.

Important assumptions of the model were: the rotors are non-rotating to simplify the model, the casing and rotor surfaces are smooth walls i.e. no roughness, the gas is air with ideal gas properties for density variation, the high-pressure side gas temperature was set at 300K which is the leakage gas temperature, the flow equations used in the CFD solver for continuum are assumed to be valid even under very low density conditions, the rotors do not contact (friction between the rotors which is a big source of heat and mechanical torque has been ignored), and no thermal deformation occurs. The leakage

flow solution was calculated for both the rotor geometries at a set of decreasing low pressure levels which went from 75000 Pa down to 50Pa.



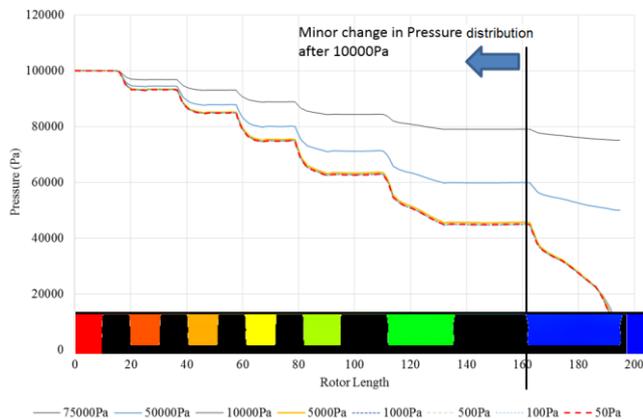
**Figure 8.** CAD model of variable lead variable clearance rotor geometry



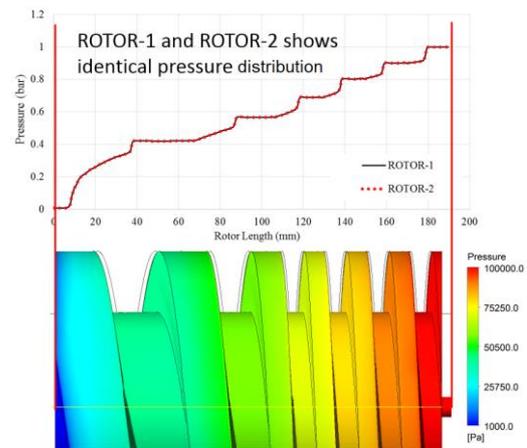
**Figure 9.** CFD model grid and boundaries of the fluid geometry

### 3.1. Pressure Distribution

Figure 10 shows the pressure variation in rotor pair 1, having uniform clearances for the full set of suction pressure conditions. It is seen that the distribution of pressure follows a stepped variation corresponding to the rotor lobes. As the suction pressure is lowered from 75000Pa to about 10000Pa, a noticeable pressure drop is observed. But beyond 10,000Pa and lower pressure there is very insignificant drop in pressure.



**Figure 10.** Comparison of pressure distribution along the length of rotor pair 1 for a set of suction pressure conditions



**Figure 11.** Comparison of pressure distribution on rotor pairs 1 and 2 at 1000Pa suction pressure condition

Figure 11 shows a comparison of rotor pair 1 with uniform clearance, and rotor pair 2 with variable clearance caused by the manufacturing deviation. It can be seen that both rotors produced identical pressure distribution.

3.2. Leakage flow

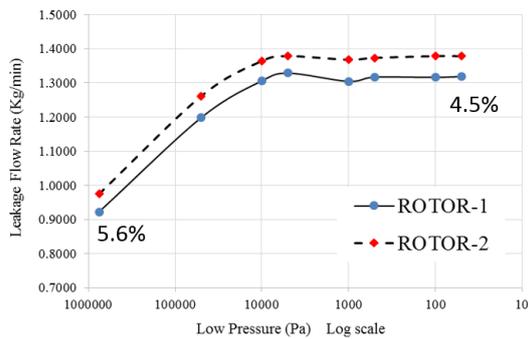
Figure 12 shows the comparison of leakage flow through the rotor pairs 1 and 2 under a set of reducing suction pressure conditions. In both rotor sets it is seen that initially when the suction pressure drops, the leakage flow increases. But after about 10,000 Pa at the suction end the increase in leakage flow ceases. This trend can be attributed to the effect of choking. It happens in the leakage gaps between the first and second rotor chambers which have large pressure distribution. This flow field trend also corresponds to the observation in pressure distribution results. With non-uniform clearance in rotor pair 2, leakage flow increases by 4.5 – 5.6 % over the entire operating suction pressure range.

Although there is an increase in leakage flow observed with the variable clearance rotor pair 2, volumetric efficiency analysis presented in Table 1 helps in evaluating the magnitude of the impact. Here the efficiency of ideal rotor pair 1 is given a set of probable values from the best case of 95% to a worst case of 70%. As seen the decrease in volumetric efficiency of rotor pair 2 produced by the proposed manufacturing method is very small and even in the worst case only about 1.5%.

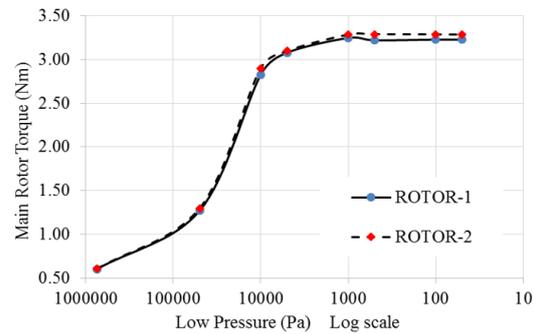
(The angular position of the rotor affects the port positions, and this will affect the axial pressure and hence the torque, but due to the large wrap angle, the leakage is unchanged.)

**Table 1.** Volumetric efficiency analysis for the two rotors

Rotor pair 1 Efficiency	Inefficiency	Effect of 5% higher leakage	Rotor pair 2 Efficiency	Decrease in Efficiency of Rotor pair 2
x%	(1-x)%	$y=1.05*(1-x)\%$	$z=(1-y)\%$	(x-z)%
95	5	5.25	94.75	0.25
90	10	10.5	89.5	0.5
85	15	15.75	84.25	0.75
70	30	31.5	68.5	1.5



**Figure 12.** Comparison of leakage flows



**Figure 13.** Comparison of torques

### 3.3. Rotor Torque

Figure 13 presents the comparison of torque on the two rotor types for different reducing suction pressure conditions. The thrust on the rotors is calculated from the integration of the pressure solution. Hence the trend in thrust variation also closely follows the trend in leakage flow. In both the rotors as the suction pressure starts dropping the rotor thrust starts to increase. After about 10,000 Pa suction pressure and further drop, the thrust magnitude ceases to increase. Unlike the observation in leakage flow, the thrust and also axial force variation (not presented here) remained identical between rotor pairs 1 and 2, signifying that indicated power will be of similar order in both the rotors.

In summary, the CFD model was able to predict the leakage flow, pressure distribution, the axial thrust and pressure torque acting on the rotors under different suction pressure conditions. With rotating effects in real operation, flow conditions will have differences and the presented results should be used only for a relative comparison. A reduced volumetric efficiency due to increase in leakage caused by manufacturing deviation is likely, but as presented in Table 1 the relative impact is very low.

## 4. Conclusions

It is shown that with careful calculation and iterative design of tool paths it is possible to use a profiled tool designed for a uniform lead rotor to manufacture a rotor (or rotor segment) having non-uniform lead. The clearances between such rotor pairs deviate slightly from the design clearances, and the effect on the performance of the two rotor types has been compared using CFD. The predicted differences are very small, suggesting that the disadvantages of using a full-profile disc-type tool process for variable lead rotor production are negligible, whilst the economic advantages may be significant, especially where volume production is required. The CFD modelling inevitably used some assumptions and simplifications, and experimental verification would now be valuable.

The above work relates to multi-wrap vacuum screw machines with timing gears, and no conclusions may be drawn for oil-injected screw compressors with traditional profiles and variable lead.

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