Grid Generation for Screw Compressors with Variable Geometry Rotors

*A. Kovacevic, S. Rane and N. Stosic

City University London, Northampton Square, EC1V 0HB London, United Kingdom
tel: +44 20 7040 8780, fax: +44 20 7040 8566

*Corresponding author: a.kovacevic@city.ac.uk

Abstract

An algebraic grid generation algorithm presented in this paper is applicable for Computational Fluid Dynamics (CFD) calculation of twin screw compressors with variable rotor geometry. It is based on the method previously developed by authors for standard uniform pitch rotors with constant cross-section profile. The new algorithm allows the rotors with variable pitch and/or variable rotor profiles to be analysed. Performance predictions are obtained by commercial CFD solver ANSYS CFX for three variants of an oil-free 3/5 screw compressor, namely uniform helical rotors, variable pitch rotors and variable profile rotors. The variable pitch and variable profile rotors achieve steeper internal pressure rise, lower leakage rate due to reduced sealing line length in high pressure domains and a larger discharge area for the same pressure ratio. This work will allow analysis of the existing and novel compressor configurations.

Keywords: Algebraic Grid Generation, Computational Fluid Dynamics, Twin Screw Compressor, Variable Pitch Rotors, Variable Profile Rotors.

Introduction

Twin screw compressors contain helically lobed rotors with profile optimized for an efficient compression process. These usually have a constant pitch. The pitch of screw compressor rotors is defined as an axial distance between tips of consecutive lobes. Figure 1 shows CAD model of a twin screw compressor consisting of the compression chamber formed by the helical rotors and casing. Such domain includes the main chamber, leakage paths, and rotor ports. Rotors with variable pitch were patented by Gardner in 1969. These are still rarely used due to lack of efficient and economical manufacturing techniques for their production.
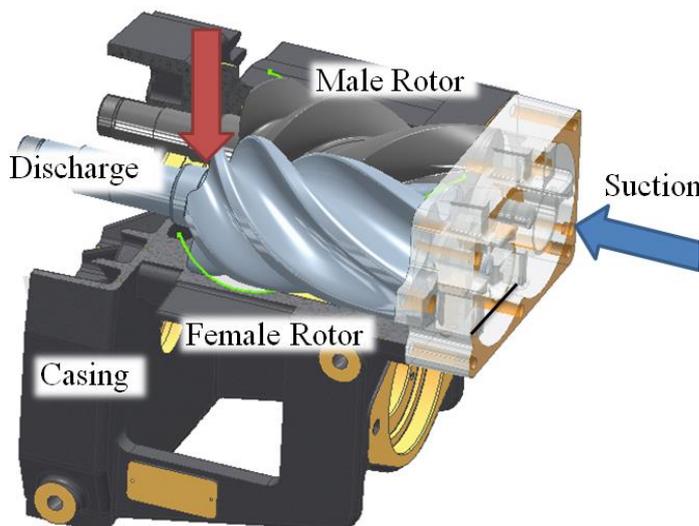

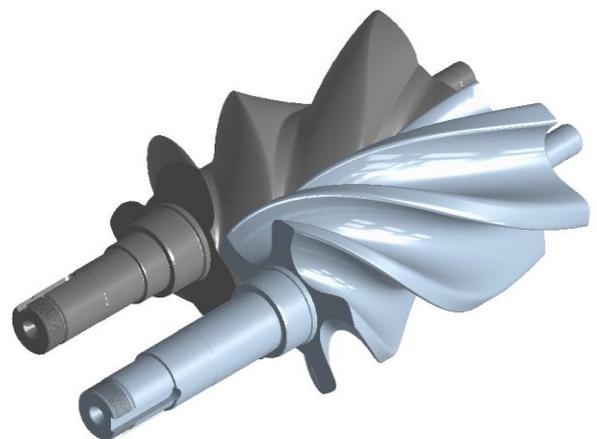**Figure 1. Sectioned view of a Twin Screw Compressor**

Figure 2. Variable Pitch Screw Rotors

Figure 2 shows configuration of twin screw rotors with variable pitch. It has been shown in the literature that for the same rotor lengths, diameter, wrap angles and lobe profiles, variable pitch rotors can provide higher pressure ratio and larger discharge port opening areas with reduced throttling losses than the equivalent rotors of a constant pitch (Gardner, 1969). Even higher compression ratios could be achieved with rotors of variable diameter.

Twin screw machines are today often analysed by use of Computational Fluid Dynamics (CFD). The prerequisite for successful CFD is appropriate space discretisation by a suitable numerical grid. Three main mathematical methods are used to generate such grids, namely algebraic methods, differential methods and variational methods, as described in (Eiseman et. al., 1994; Liseikin, 1999; Samareh and Smith, 1992; Shih et. al., 1991; Soni, 1992 and Thompson, 1999).

Kovacevic et. al. (2000, 2002, 2005 and 2007) have successfully used algebraic grid generation method to produce numerical mesh for twin screw machines with constant pitch rotors. This has been implemented in the custom made program called Screw Compressor Rotor Grid Generator – SCORG[©]. Figure 3 shows an example of the grid in one cross-section produced by SCORG[©].
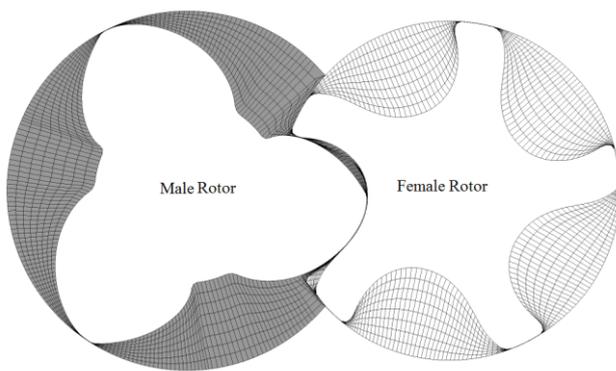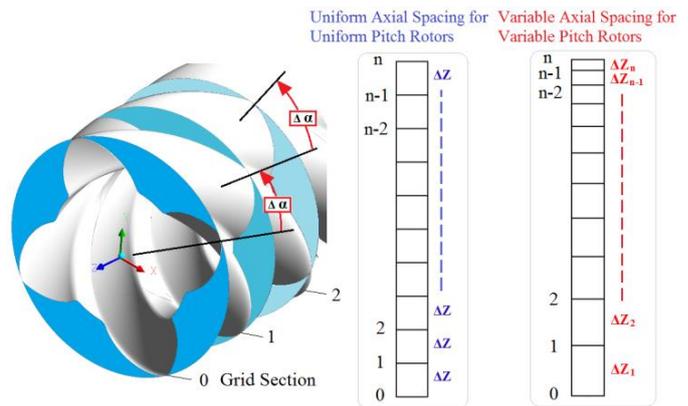

Figure 3. Numerical grid in a cross section

Figure 4. Cross sectional spacing for uniform and variable pitch rotors

Grid Generation

Rotors with uniform pitch and constant rotor profile have a fixed relationship between the lead and the rotor rotation angle. This allows for the numerical grids generated in one cross section at different angular positions in time to conveniently be used for spatial definition of rotors along the rotor axis.

The challenge in generating a numerical grid for variable lead rotors is in establishing the relationship of the axial distance between consecutive cross sections and the angular rotation of the rotor. The uniform rotation does not correspond to the uniform axial spacing of cross sections in variable pitch rotors. Figure 4 shows the difference in grid spacing for the constant and variable pitch rotors. The existing procedure used in SCORG[©] has been reformulated and adapted for the variable pitch and variable profile screw rotors as presented in this paper.

The rotor pitch is usually constant. However it can also change linearly, non-linearly or in steps. The former is often used for screw vacuum pumps while the latter is applied in some car superchargers. The change in the pitch can be expressed through following equations:

Constant Pitch

$$p = constant \tag{1}$$

Linear Pitch

$$p = \left(\frac{p_e - p_s}{L}\right) z + p_s \tag{2}$$

Quadratic Pitch

$$p = \left(\frac{p_e - p_s}{L^2}\right) z^2 + p_s \tag{3}$$

Authors developed two approaches for grid generation for machines with variable pitch rotors, as described in detail in previous publications (Rane et. al., 2013).
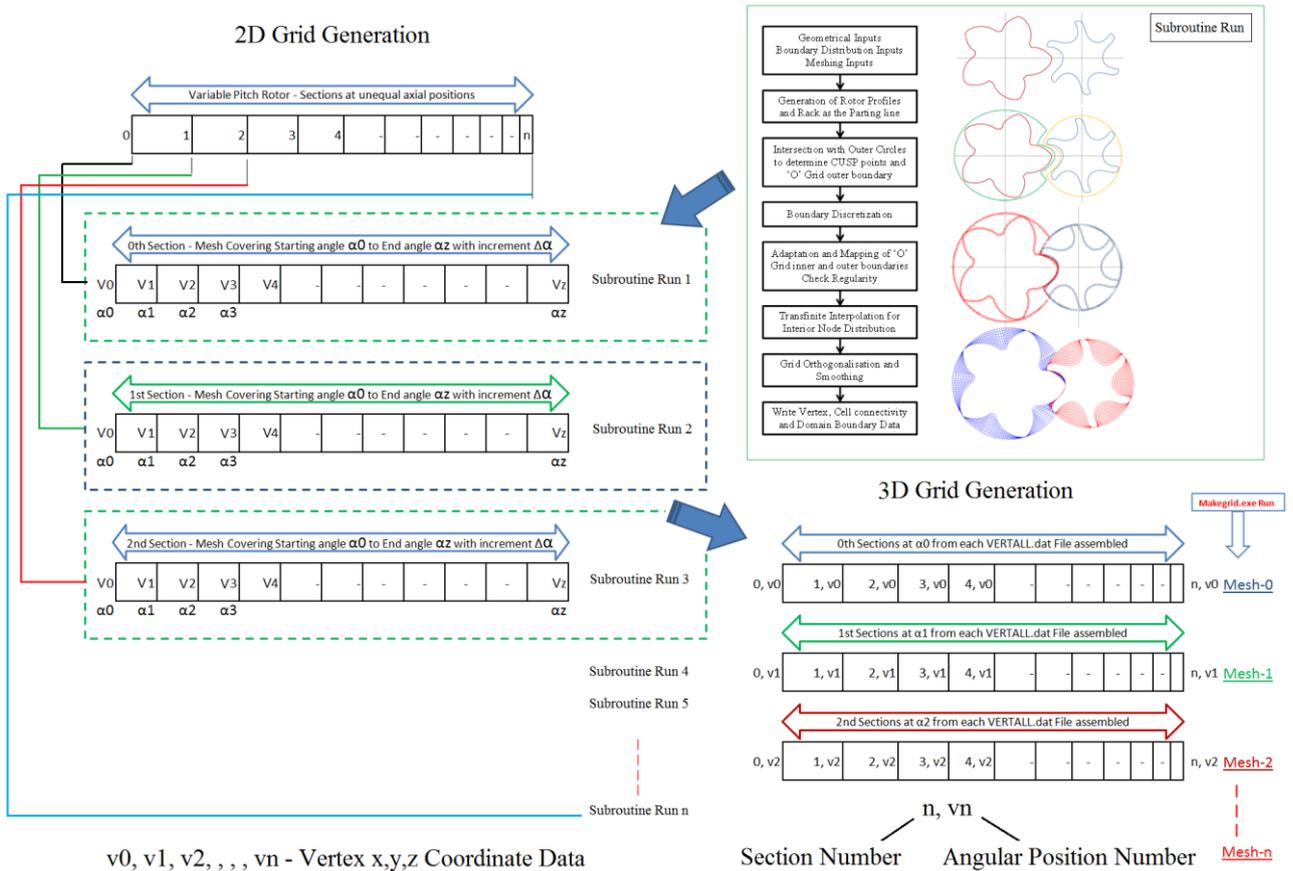

Figure 5. Variable Pitch and Variable Profile Grid Generation

The first approach uses relations like (1) to (3) to define corresponding axial distance at different axial positions of a cross section ΔZ_n which will relate to a uniform rotation of a rotor $\Delta\alpha$. It does not allow change of a profile along the rotor axis. The second approach shown in Figure 5 is more general. In it, it is assumed that screw compressor rotors form a conjugate pair in each cross section. Therefore each section can be calculated independently using its specific rotor profile. The procedure is described in detail in Rane et. al. (2013). Once 2D grids are generated for all time steps in each cross section, these could be assembled to form a set of rotor flow domain grids for every time step. An example of the constant rotor profile with variable pitch rotors is shown in Figure 6 while the grid for variable rotor geometry and uniform pitch is shown in Figure 7.
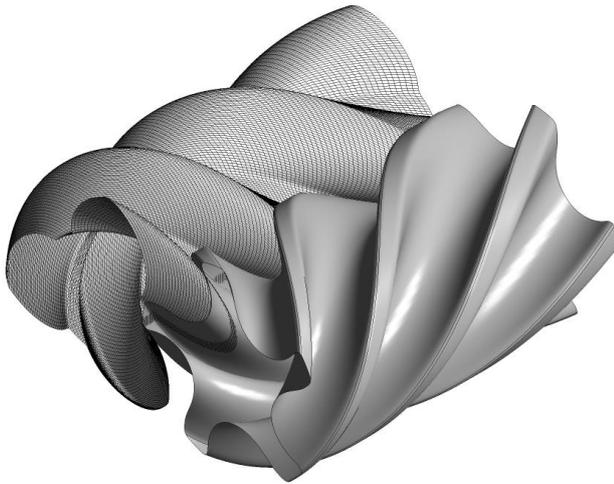
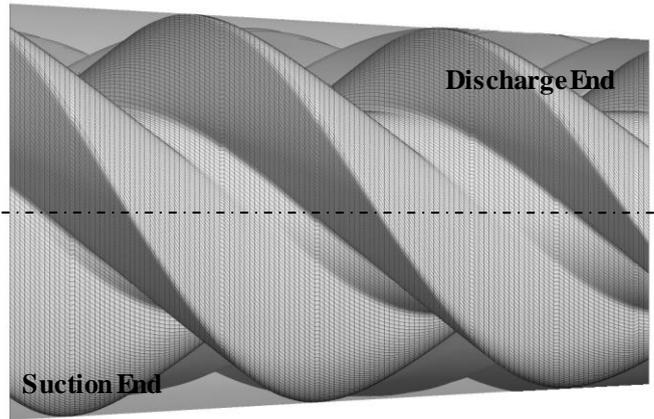| Figure 6. Numerical grid for variable pitch screw rotors | Figure 7. Numerical grid for uniform pitch, variable profile rotors |

CFD Analysis

Numerical analysis was carried out by use of ANSYS CFX solver with objectives to validate the grid generation approach for variable rotor profile and to study performance of compressors with various configurations. The male rotor has 3 lobes with 127.45mm outer diameter, L/D ratio 1.6 and wrap angle Φ_w of 285°. The female rotor is with 5 lobes and the rotor centre distance is 93mm. Both rotors are rack generated 'N' profiles. The compressor speed is 8000 rpm with pressure ratio of 2.0 and 3.0. Three test cases were calculated.

Case 1. Uniform pitch and uniform profile rotors with built in volume index V_i of 1.8.

Case 2. Variable pitch with uniform profile rotors and built in volume index $V_i > 1.8$. The Suction side pitch was 130mm and Discharge side pitch was 40mm. The wrap angle of Φ_w 285° was maintained.

Case 3. Variable profile and uniform pitch rotors. Rotor profile in successive sections is generated using rack generation procedure by variation of addendum on the defining racks (Stosic et. al., 2005). Addendum on the suction end of the rotors was 33mm while on the discharge side it was reduced to 21mm. The addendum on the uniform profile rotors had constant value 28.848mm. Due to the variation of addendum the outer diameter of the male rotor is changing while the inner diameter remains constant and vice versa for the female rotor as shown in Figure 7. The displacement of these rotors is smaller than for the uniform profile rotors.

For the purpose of comparison, same suction and discharge ports were used for all three cases. Each rotor configuration was analysed on three different grid sizes. Table 1 shows the number of computational nodes in each case. Stationary compressor ports were meshed by use of the commercial grid generator. Figure 8 shows the compressor domain decomposed into three main regions; the suction port, the discharge port and the rotor domains.

Table 1. Size of refined grids, number of nodes

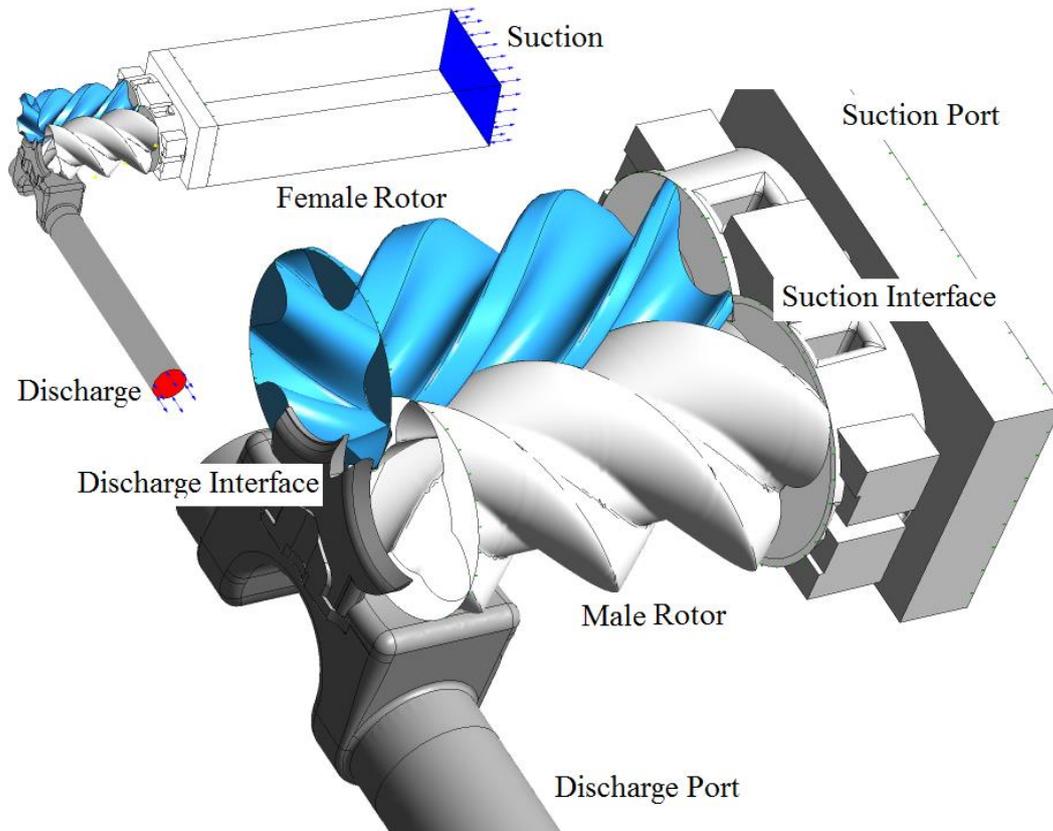| Case	Uniform	Variable Pitch	Variable Profile
Coarse	691174	648190	675918
Medium	838378	915184	889794
Fine	1214418	1344944	1297434

Figure 8. Compressor model with variable profile rotors

The working fluid is air following an ideal gas law. The suction pressure in all cases was 1.0 bar. The discharge pressures were 2.0 bar and 3.0 bar. The target convergence criterion is 1.0×10^{-3}. During solution, r.m.s residuals were between 1.0×10^{-3} and 5.0×10^{-3} for the momentum equation, while for continuity and energy equations the convergence criteria was always achieved. The calculations were run sufficiently long to allow a cyclic repetition of flow and pressure at boundaries. Each case was calculated with both Laminar and SST k-Omega Turbulence model.

Results and Discussion

Pressure field

Figure 9 shows the pressure field on the variable pitch rotors – Case 2 with discharge pressure of 2.0 bar. The red colour indicates higher pressure while the blue colour is the lower pressure. The indicator diagrams for all three cases calculated on fine grid with SST turbulence model are shown in Figure 10. The increase of internal pressure is steeper for the variable pitch and variable profile rotors. The vertical dashed lines show the angle of rotation of the male rotor at which the working chamber is exposed to suction and discharge port. Since the same discharge port is used for all three cases, the variable geometry rotors, Case 2 and Case 3 achieve full internal compression for the case of 3 bar discharge pressure while in that case constant geometry rotors under-compress making them more inefficient for this higher pressure. In the case of the 2bar discharge pressure, the constant geometry rotors achieve full internal compression while variable geometry rotors over-compress. This will make them less efficient in case of lower discharge pressures.

Depending on the geometry, variable profile rotors may have smaller displacement than the uniform rotors. Therefore these are better suited for high pressure applications.
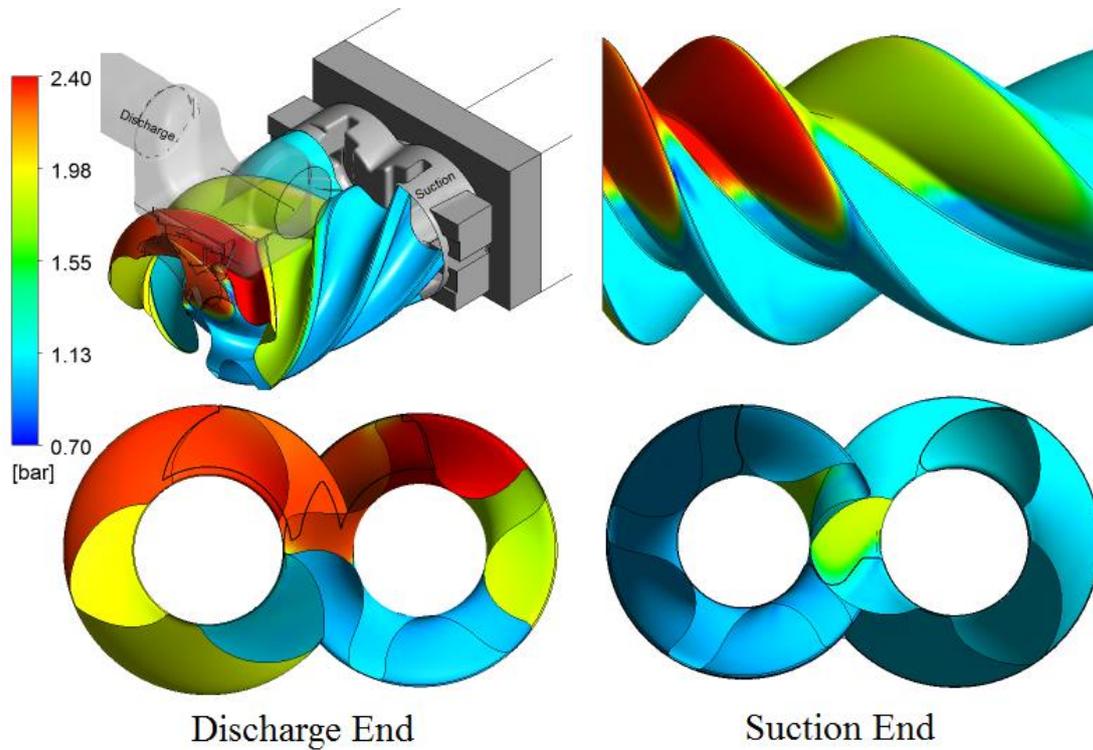
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Figure 9. Pressure variations in the variable pitch rotors with discharge pressure 2.0 bar
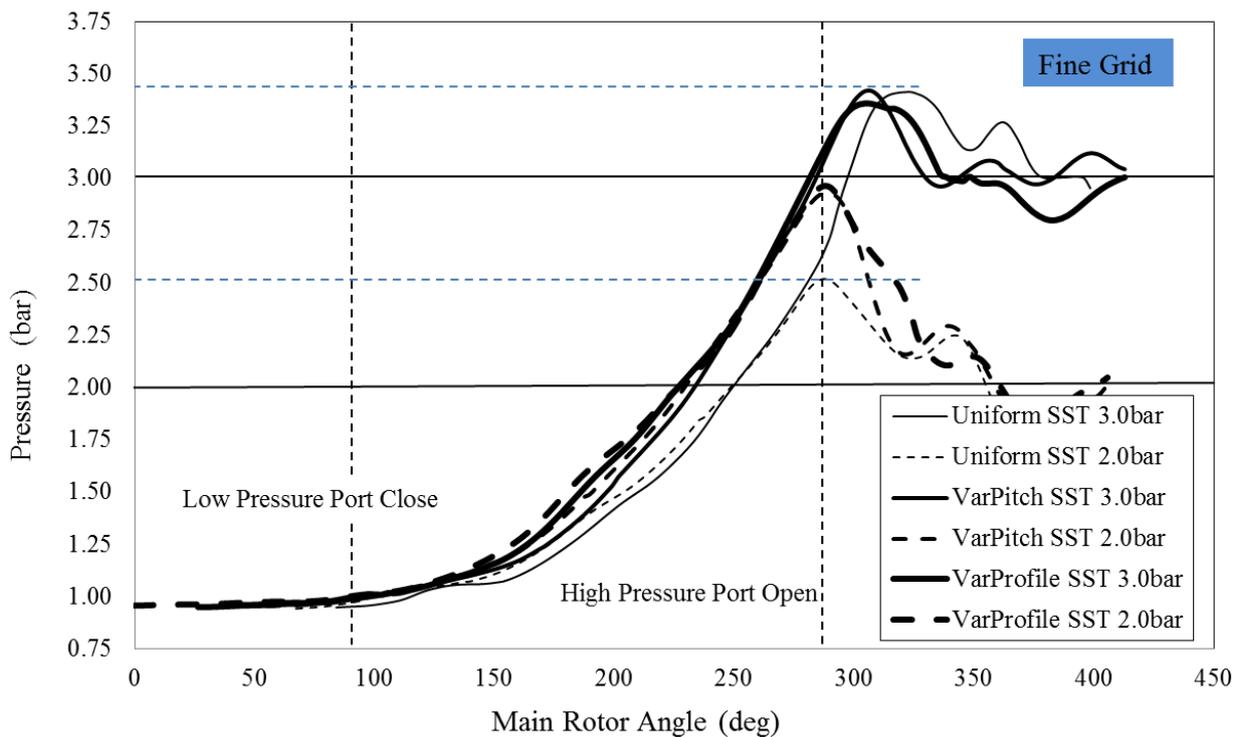

Figure 10. Indicator diagram

Influence of Grid Refinement

Figure 11 shows the effect of grid refinement on the prediction of integral quantities such as the mass flow rate, indicated power and specific power. The cases presented in the figure are with 2 bar discharge pressure. The increase in mass flow rate is observed with grid refinement in all cases. The

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difference between each consecutive refined grids is around 10%. This indicates that the rotor geometry is captured better with finer grids, which results in better representation of leakage losses which are progressively reduced with grid refinement. Similarly, indicated power increases for 2-5% each time the grid is refined.
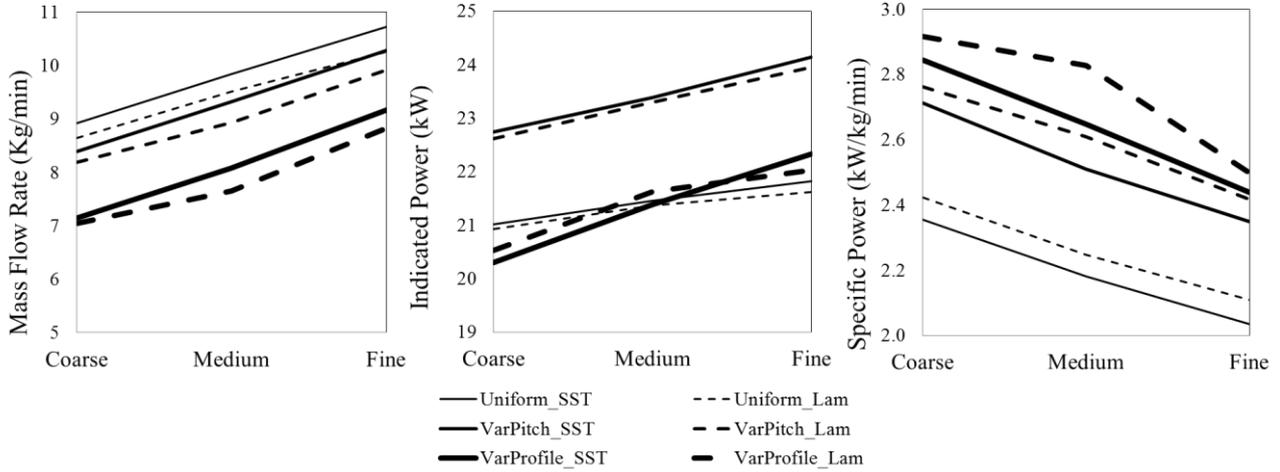

Figure 11. Effect of grid refinement on integral parameters

Influence of Turbulence Model

As shown in Figure 11, the higher mass flow rates are achieved when the turbulence model is applied. Indicated power is also increases as the consequence of increased mass flow. However, specific power is reduced for turbulent cases and is closer to the expected values. This suggests that the appropriate turbulence model must be applied for calculation of screw compressor performance. For variable profile rotors, the results obtained on the finest grid are contradictory which may be consequence of the grid generation method used in this case. This needs to be investigated further.

Sealing Line Length

The interlobe sealing line is the line of closest proximity between two rotors. The leakage of gas through the interlobe gap is proportional to the length of the sealing line and the size of the clearance normal to the profile (Fleming and Tang, 1994; Hanjalic and Stosic, 1997 and Stosic et. al., 2005). Contours of pressure distribution on the rotor can be established from numerical calculations. The line which divides the high and the low pressures between the rotors represents the sealing line. The comparison of the sealing line lengths for three calculated cases is shown in Figure 12 and in Table 2.

Table 2. Comparison of Interlobe Sealing Line Length [mm]

Interlobe No	Uniform	Variable Pitch	Difference	Var. Prof	Difference
1	145.8	175.9	-30.1	158.4	-12.6
2	170.3	164.0	+6.4	162.2	+08.1
3 (part)	069.1	056.8	+12.2	068.4	+00.6
Total	385.2	396.6	-11.5	389.1	-03.9

The sealing line for the uniform pitch rotors has the same length for each interlobe space along the rotor. At the suction end on the variable pitch rotors the sealing line is 30mm longer than for the uniform case. It decreases towards the discharge end of rotors by 12mm. This helps to reduce leakage as the largest pressure difference across the sealing line is at the discharge end. In the case

with variable profile the suction end is longer by 12mm and discharge end has very small difference compared with the uniform profile.
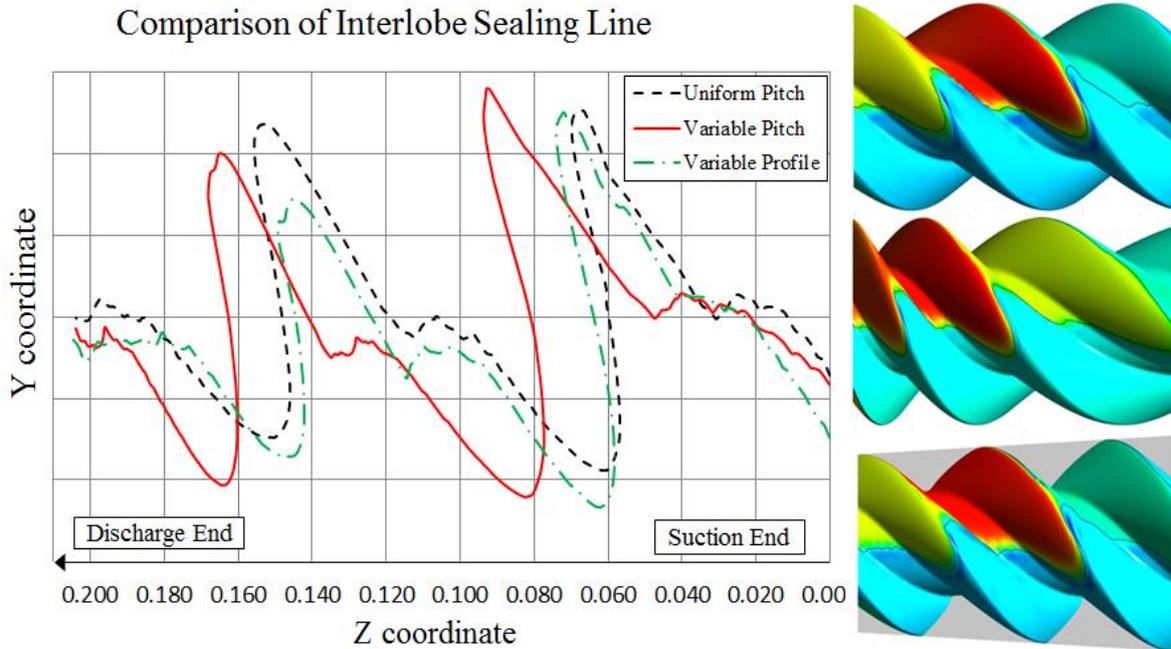

Figure 12. Comparison of interlobe sealing line lengths

Blow-hole area

Blow-hole area is the leakage area formed between the male and female rotors and the casing at the rotor cusp as shown in Figure 13. The blow-hole area on the high pressure side is important for leakage due to the pressure difference between adjacent compression chambers. A smaller blow-hole area is desired for improved performance of the machine. Table 3 shows the blow-hole areas measured on the medium size grid for different calculated cases at different positions in the rotor. The blow-hole area for the uniform rotors remains constant along the length of the rotors. In the variable pitch rotors, the suction side blow-hole area is larger than in the uniform rotors but the discharge side blow-hole area is smaller than for the uniform rotors. In the variable profile rotors, the suction side blow-hole area is nearly the same as that of the uniform rotors but the discharge blow-hole area is smaller than for the uniform rotors. Proportionally, the reduction of blow-hole area towards the discharge side rotors is most pronounced in the case of variable pitch rotors.

Table 3. Comparison of Blow-hole area [mm^2]

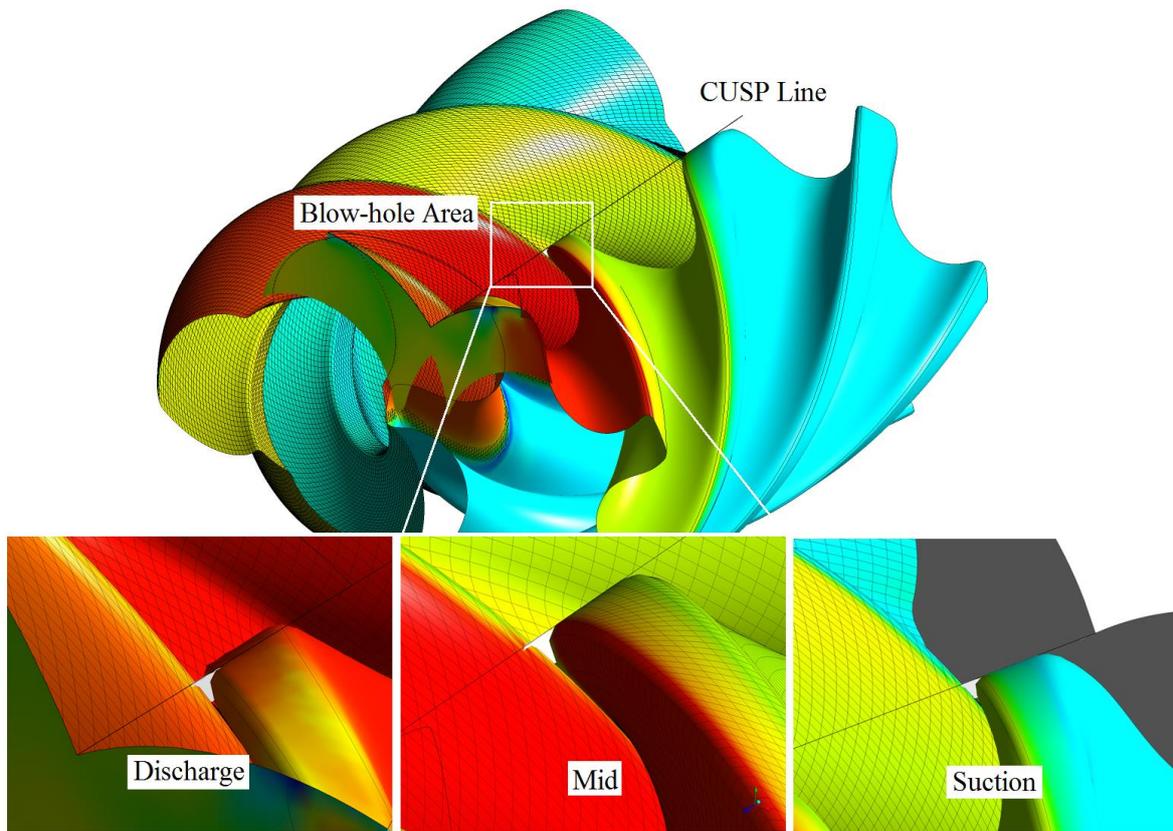| Position	Uniform	Var. Pitch	Diff %	Var. Profile	Diff %
Suction	9.817	12.49	-27.2	9.83	-0.15
Mid	9.908	9.263	6.51	9.35	5.64
Discharge	9.701	6.562	32.3	9.19	5.29

Figure 13. Comparison of Blow-hole area

Overall performance

The influence of the discharge pressure on the compressor performance for three analysed cases is shown in Figure 14. Uniform rotors have a highest flow rate and lowest specific power for both pressures. However the difference in the specific power between uniform and variable geometry rotors is much reduced at higher pressures. This further confirms that the variable geometry rotors are more suitable for high pressure applications.
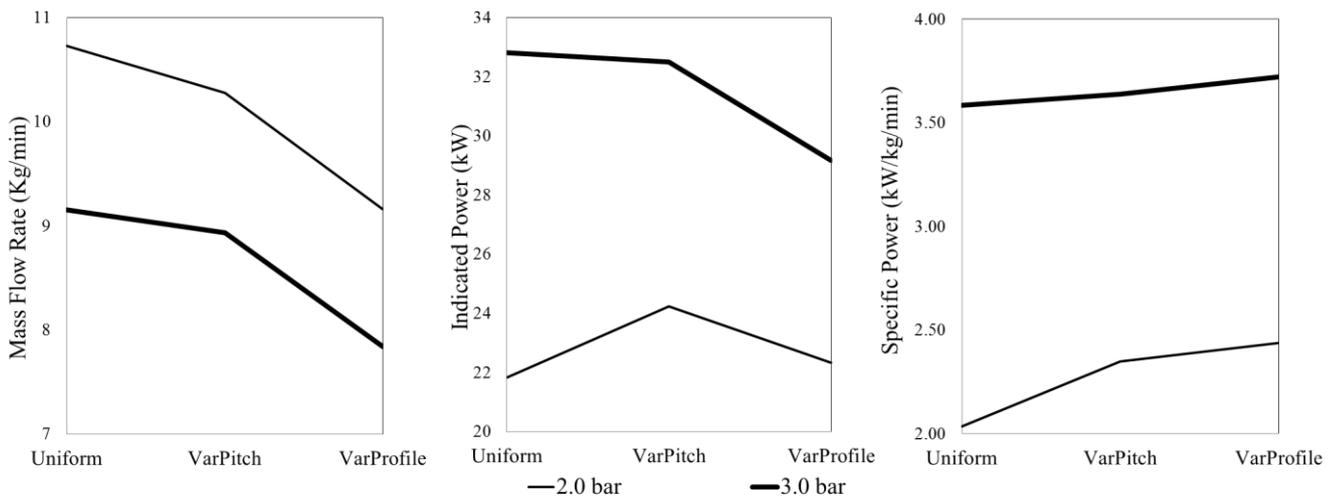

Figure 14. Influence of discharge pressure on overall compressor performance

Conclusions

The procedure for grid generation of twin screw compressors with variable pitch and variable cross section profile rotors was implemented successfully. Examples of grids and CFD results with variable lead and variable profile have been presented.

By varying the rotor lead continuously from the suction to discharge or by varying the rotor profile to reduce the cross section area towards the discharge, it is possible to improve compression characteristics and achieve a steeper internal pressure rise. The variable lead rotors additionally reduce the sealing line length and blow-hole area in the high pressure difference zone.

The grid generation enhancements allow use of CFD for novel compressor performance analysis.

Nomenclature

L – Rotor Length

D – Male Rotor Outer Diameter

Φ_w – Male Rotor Wrap Angle

α – Male rotor rotation angle

Δα – Increment in Male rotor angle

Z – Axial distance along the rotors

ΔZ – Increment in Axial distance

p_s – Starting Pitch

p_e – Ending Pitch

z_1 – Number of lobes on the Male rotor

z_2 – Number of lobes on the Female rotor

V_i – Built in Volume Index

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