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Design Optimisation of a Screw Compressor with a Focus on Rotor Depth: A Computational Fluid Dynamics Approach

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ABSTRACT

The continuing demand for improving performance and reliability of twin-screw compressors has led to the need for application of the state-of-the art optimisation tools in order to minimize flow losses and improve adiabatic efficiency. In this study, the profile parameters of the standard 5/6 compressor with a typical rotor depth were optimised using a so-called response surface optimisation (RSO) method to minimise specific power of the compressor. Following that, the rotor pair with the optimised parameters was used as the basis for optimisation of the rotor depth. The rotor depth was changed by varying the addendum and dedendum which ultimately changed the outer diameter of the compressor. The demonstrator “N” profile generated by SCORGTM was used. The multi-chamber thermodynamic analysis built in software SCORGTM was used for calculation of geometric values and thermodynamics quantities. The optimisation results showed that the change in the rotor depth did not noticeably affect the volumetric efficiency. However, the adiabatic efficiency was reduced up to 5% and specific power was reduced by 4%, due to increase in the rotor depth for 14%. Finally, the performance of the compressor with the optimised profile was calculated by computational fluid dynamics (CFD) using the casing to rotor conformal mesh generated by SCORGTM and ANSYS CFX multiphase solver.

Keywords: Screw Compressor, Rotor Optimisation, RSO, Rotor Depth,

Nomenclature

z (—)	Lobe number of rotor
r_o (mm)	Outer radius of rotor
W (deg)	Wrap Angle

$R_l (-)$	Relative Length
$C (deg)$	Compression Angle
$V_i (-)$	Volume Index
$\dot{n} [rpm]$	Compressor Speed
$P_0 [bar]$	Suction Pressure
$P_d [bar]$	Compression Pressure
$p_1 [mm]$	Radius of arc AB
$p_2 [mm]$	Radius of arc HI
$u_n [deg]$	Pressure Angle in High Pressure
$v_n [deg]$	Pressure Angle in Low Pressure
$t [mm]$	Length of BC
$s [mm]$	Length of GH
$\kappa [mm]$	Normal distance of CD
$\tau [mm]$	Normal distance of DE
$d [mm]$	Length of IJ
$\gamma [^\circ]$	Radius angle of EF
$e_a [mm]$	Minor radius of FG ellipse
$A_c [mm]$	Distance between rotor axes
$O_a [deg]$	Oil Injection Angle
$P_s [kWh\ m^{-3}]$	Specific Power
$\eta_v [\%]$	Volumetric Efficiency
$\eta_a [\%]$	Adiabatic Efficiency
$\eta_t [\%]$	Total Efficiency
$P_w [kW]$	Compressor Power
$\dot{V} [m^3\ m^{-1}]$	Volumetric Flow Rate
$\dot{m} [kg\ h^{-1}]$	Mass Flow Rate
$T_{outlet} [^\circ C]$	Temperature
$r_p [mm]$	Pitch Radius
$h_a [mm]$	Addendum Height
$h_d [mm]$	Dedendum Height

Subscripts

g	Female
m	Male
oil	Injection Oil
air	Air

Introduction

Twin-screw compressors are mature machines initially conceived in the 1930s to meet the demands of variable and constant pressure applications, as well as high-volume flow rate requirements. These compressors consist of two rotors, male and female, which rotate in opposing directions, and the casing thereby compressing gas by reducing the fluid volume.

The geometry of these machines is inherently intricate, and the gas compression process within them unfolds in three distinct stages: suction, compression, and discharge. Initially, gas enters through an inlet port and fills the space between the lobes during rotation. As the rotating rotors move in opposite directions, the air is compressed by reducing its volume. The compression process concludes before the air reaches the discharge port, from which the compressed air is expelled.[1].

Industrial sectors are responsible for more than 50% of global energy consumption [2]. Surprisingly, approximately 10% of this energy usage can be traced back to compressed air systems within the industrial realm. Consequently, even modest improvements in this area can lead to significant energy savings [3]. In the literature, the primary focus for enhancing the performance of screw compressors lies in the design of both the rotor and the port. In this context, Stosic introduced a rotor profile generated using a rack profile comprised of multiple curves in a general implicit form [4].

Design and development efforts relying on Computational Fluid Dynamics (CFD) once posed formidable challenges in evaluating performance during the design phase and devising optimisation procedures for screw compressors. This was primarily due to the time-dependent moving boundary conditions tied to the rotation of the compressor rotors. However, Kovacevic, S. Rane, and their colleagues made significant contributions in this field, particularly through innovations in meshing processes for screw compressors [5], [6], [7], [8], [9].

The task of solving two-phase flow in oil-injected compressors using CFD presented further challenges. These challenges stemmed from factors such as narrow clearance gaps, a substantial density contrast between gas and oil, complex mixing of the phases and intricate geometric structures.. S. Rane, conducted an analysis of multiphase flow using Eulerian-Eulerian multiphase approach which produced favourable results [10].

Further advancements were made by N. Basha et al, who explored both the Eulerian-Eulerian multiphase approach and the homogeneous model, successfully validating their findings through experimental results [11] . They also conducted comparative studies involving different multiphase models [12].

S. Yang et al. [13] delved into simulating oil-injected screw compressors to investigate the impact of pressure pulsations on the rotors and housing inner surface. J. Suh et al. [14] developed a numerical analysis method for assessing the hydraulic performance of multiphase pumps, employing the Eulerian-Eulerian multiphase approach. Lu et al. compressor [15] proposed a novel method for compressor design, shedding light on the characteristics of non-uniform pressure within a twin-screw. We et al [16] established an oil-injected screw compressor in a test rig to determine pressure and volumetric efficiency at different frequencies.

N. Stosic [17] derived rack generated “N” rotor profile and outlined an optimisation procedure that combines an analytical algorithm for generating rotor profiles with a differential model accounting for the fluid flow and thermodynamic processes within the machine. Yu Ren Wu [18], published a paper in which they added a parameter in the “N” rotor profile for Fu-Sheng profile and employed a genetic algorithm to optimize the rotor group of a screw compressor and then validated this optimised profile using experimental data. C. Wang[19] introduced a specialized rotor profile designed to address issues related to leakage and high-temperature rise, particularly in helium twin-screw compressors. This profile was tested on two developed compressors. E. Mujic et al, [20], [21]. conducted investigations into the effects of pressure pulsations within the discharge chamber Additionally, N. Stosic [22]. Explored in detail the impact of rotor tooth depth on rotor performance.

The optimization studies in the literature are primarily focused on the optimization of parameters representing two-dimensional rotor profiles with fixed tooth depths. However, simultaneous optimization of rotor and port parameters has not been explored in previous research. This study aims to fill this gap by proposing a two-stage optimization approach. In

the first stage, a Response Surface Optimization Technique (RSO) is utilized to optimize parameters related to the rotor profile, including those representing the rotor wrap angle, tooth depth, and rotor axis distance, with the objective of minimizing specific power. Additionally, the influence of tooth depth on the objective function is investigated based on the principles outlined by Stosic et al. [22]. The second stage of optimization includes intake, discharge, and oil port parameters for the optimized rotor profile, with the goal of optimizing all parameters affecting compressor performance. The proposed design is validated through CFD-based simulations.

1. Thermodynamic Model of Screw Compressors

Optimising screw compressors solely through CFD is not feasible, given the multitude of input parameters involved. Therefore, an approach based on a thermodynamic model is necessary to calculate and assess the thermodynamic performance during the optimisation process. Zaytsev, developed a homogeneous thermodynamic model using modeling tools including Matlab Simulink, Fluidprop, and Refprop [23]. Kovacevic established a one-dimensional thermodynamic model for screw compressors, which was primarily dependent on the male rotation angle. The mass and energy balances are defined as follows [24];

$$\frac{d(\dot{m})}{d\theta} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} \quad [1]$$

$$\dot{m}_{in}h_{in} = \dot{m}_{suc}h_{suc} + \dot{m}_{l,in}h_{l,g} + \dot{m}_{oil}h_{oil} \quad [2]$$

$$\dot{m}_{out} = \dot{m}_{dis} + \dot{m}_{l,out} \quad [3]$$

Rane and Kovacevic [24], [25] designed a tool known as SCORGTM for thermodynamic calculations, employing a one-dimensional thermodynamic model. This tool also allows generation of numerical meshes for CFD calculation of the compressor. For oil injected screw compressors, the heat transfer between the working chamber and environment is relatively insignificant compared to the rate of internal energy change and often can be disregarded [24], [25].

2. Geometric Model and Parameters

The male and female rotors were generated using the rack generation method proposed by Stosic [4]. This method allowed for the simultaneous creation of both the male and female

rotors, employing the rack generation procedure. Yu Ren Wu [18] employed the same procedure and added one small curve at the tip of the male rotor which made insignificant improvements to the original “N” rotor profile. Despite this, the profile of Yu Ren Wu is explored in this paper. The rack profile consisting of nine different curves was defined parametrically by Yu Ren Wu [18] as shown in Fig. 1. The depth of the rotor profile is changed by changing addendum and dedendum of the rotors. Addendum is the radial distance between the pitch circle and the tip of the male rotor while the dedendum is the distance between the pitch circle and the inner rotor diameter of the male rotor. These design parameters were defined within specified band of values, as detailed in Table 1 and Table 2.

Design parameters in screw compressors are limited by the undercut limits [18]. Undercutting occurs when the lobe profile extends beyond the limits of conjugate profile, leading to interference during meshing. Therefore, both lower and upper boundaries are established for this particular profile to avoid the undercuts as shown in Table2. Table 1 presents the reference parameters of the compressor rotors.

Table 1. Input Reference Parameters

Parameter	Parameter value	Description
z_m/z_f	5/6	Lobe number male and female rotor
r_{om} [mm]	63.6	Male rotor outer radius
r_{og} [mm]	50.4	Female rotor outer radius
W [deg]	300	Wrap angle
R_l	1.55	Relative Length
C_s [deg]	0	Compression Start
C_e [deg]	230.2	Compression End
V_i	3.35	Volume Index
$\dot{n}$ [$r\ min -1$]	4800	Angular velocity
P_0 [bar]	1	Suction pressure
P_d [bar]	5	Compression pressure
GAPR [mm]	0.06	Radial Gap
GAPI [mm]	0.04	Interlobe Gap

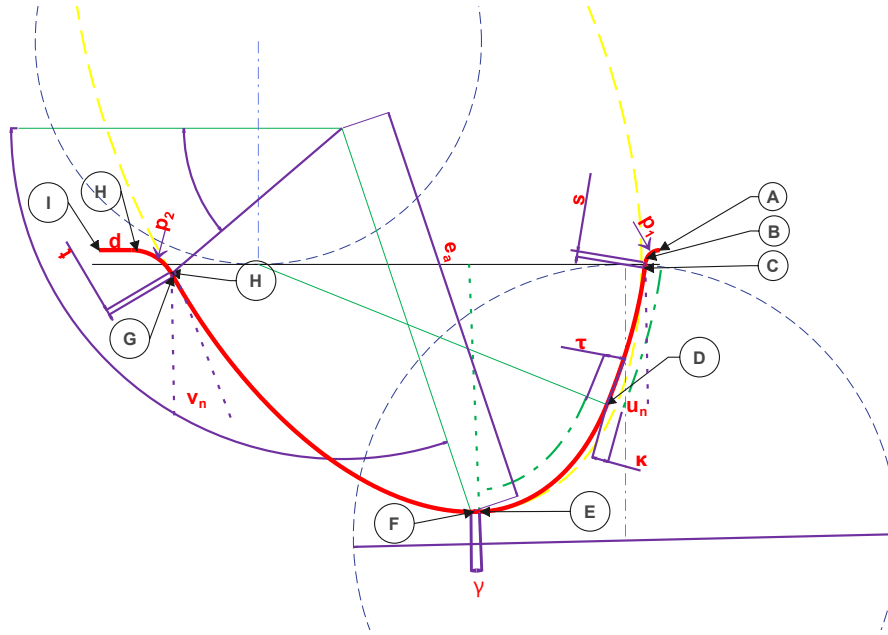


Figure 1. Parameters of Rack Profile [18]

Table 2. Lower-Upper and references values of Optimisation parameters

No	Item	Lower Bound	Reference Value	Upper Bound	Describe
1	p_1 [mm]	1.3	1.6	3.3	Radius of arc AB
2	p_2 [mm]	1.5	2.5	3.5	Radius of arc HI
3	u_n [deg]	8	10	15	Pressure angle in high pressure
4	v_n [deg]	20	25	28.5	Pressure angle in low pressure
5	t [mm]	0.4	1	2	Length of GH
6	s [mm]	0.4	1	2	Length of BC
7	κ [mm]	1	1.5	3.5	Normal distance of CD
8	τ [mm]	1.5	2	4.82	Normal distance of DE
9	d [mm]	1	1.5	2	Length of IJ
10	γ [deg]	1	3	7	Radius angle of EF
11	e_a [mm]	25	40	50	Minor radius of FG ellipse
12	A_c [mm]	88	90	91	Distance between rotor axes
13	r_{o2} [mm]	50.4	50.6	50.9	Outer radius of female rotor

Parameters which define suction, discharge, and oil ports, play a vital role in correct operation of a screw compressor. These port parameters were defined with the respect to the male rotor

rotation angle [25]. The parameter range for port optimisation conducted through RSO are presented in Table 3.

Table 3. Parameter Ranges

Parameters	Lower bound	Reference bound	Upper bound
Oil injection Angle (O_a) [deg]	45	50	65
V_i (Compression End [deg])	2.5 (200.38)	3.35 (230.22)	4.2 (247.99)

3. Design Optimisation using Response Surface Optimisation (RSO)

Response Surface Optimization (RSO) is a statistical technique used in experimental design and optimization. It is particularly valuable in situations where the goal is to optimize a response variable influenced by multiple independent variables. RSM helps researchers and scientists understand the relationship between the independent variables and the response, allowing them to identify the optimal conditions for a desired outcome.

3.1. Basic theory of RSO

The parameters under study can be influenced by numerous variables. Managing small contributions of each of the parameters individually is practically not feasible. Consequently, the selection of variables which have major impacts on performance becomes imperative. Screening calculations were conducted to identify the most influential experimental variables and their interactions,. Full or fractional two-level factorial designs are commonly employed for this purpose, primarily due to their efficiency and cost-effectiveness [26].

In this study the advanced polynomial model, containing additional terms which describe the interaction between the different design parameters was used as shown in equation [4] below where β_{ij} represents the coefficients of the interaction parameters and β_{ii} represents the coefficients of the quadratic parameter [26].

$$y = \beta_0 + \sum_{i=1}^k \beta_i x_i + \sum_{i=1}^k \beta_{ii} x_i^2 + \sum_{1 \leq i < j \leq k} \beta_{ij} x_i x_j + \varepsilon \quad [4]$$

The parametric design should ensure the implementation of at least three levels for all variables. The selection of experimental points in the central composite design used in this study determines the number of levels and blocks for the variables.

According to the Multi-Objective Genetic Algorithm (MOGA) approach used to optimize multiple objective functions, the expression for the objective point is formulated as follows [27].

$$\begin{aligned} \min_{x \in X} & (f_1(x), \dots, f_n(x)) \\ \forall i \in \{1, \dots, k\}, & f_i(x_1) < f_i(x_2) \\ \exists i \in \{1, \dots, k\}, & f_i(x_1) < f_i(x_2) \end{aligned} \quad [5]$$

3.2 Optimisation procedure

The RSO optimisation procedure is focused on identifying parameter combinations that can minimize the specific power of the screw compressor while adhering to predefined boundary conditions. In this study, the RSO optimisation technique was employed in two stages to address all geometric parameters of both the port and rotor, all of which significantly impact the overall performance. The first stage is the optimisation of rotor design parameters, and the second is optimisation of port design parameters.

The optimisation process aimed at minimizing the specific power input of the screw compressor was achieved through the successful application of the RSO. This optimisation process made effective use of MATLAB and SCORGTM, involving several sequential steps, and leveraged the Central Composite Design approach in RSO to create variations for MATLAB execution. Here are the key steps involved as shown in Fig. 2:

- A variation file was generated in .xls format to define the rotor and port parameters, which were used as input for RSO.
- In the subsequent step, a MATLAB script was developed to generate rotor profiles for each variation by importing the variation file.
- These MATLAB-generated rotor profiles were then utilized for thermodynamic calculations, with SCORGTM running in the background to perform these calculations efficiently.
- The MATLAB script read and processed the results calculated by SCORGTM, and the RSO tool evaluated candidate points based on these results.
- The optimised profile was recalculated for thermodynamic performance using SCORGTM.
- The results created by SCORGTM were read from the file path and transferred to MATLAB.

- Finally, to validate the results obtained from SCORGTM, CFX, a computational fluid dynamics tool, was proposed for conducting numerical analyses.

This comprehensive approach allowed for the effective optimisation of the screw compressors' performance and subsequent validation through numerical simulations using CFX.

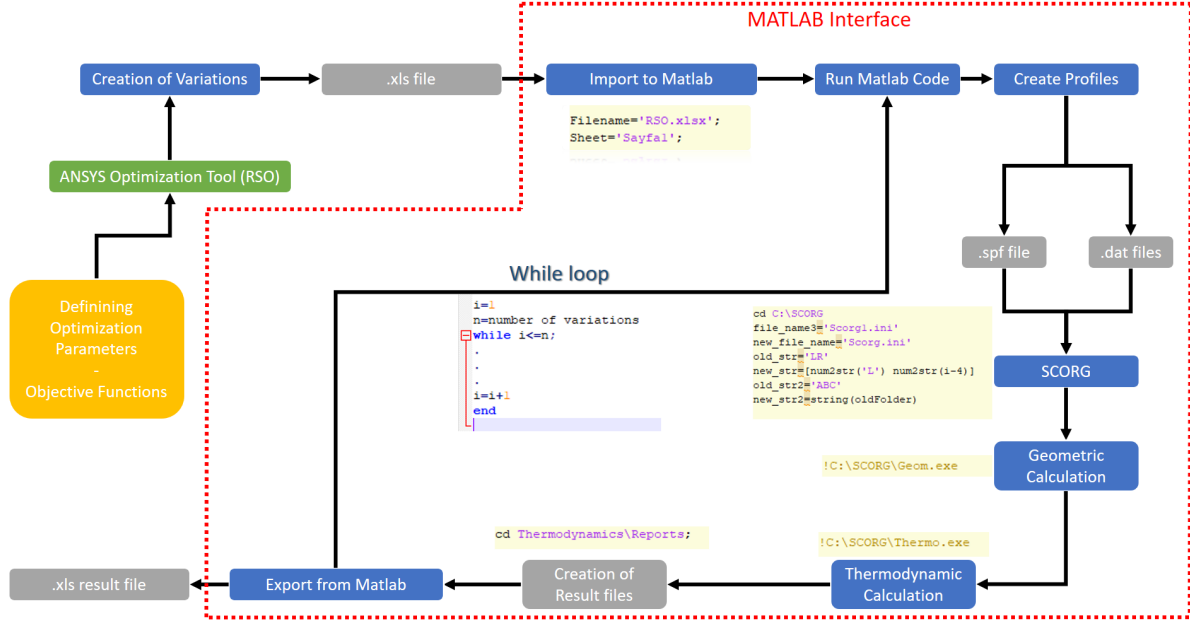


Figure 2. Flow Chart

4. CFD Approach and Boundary Conditions

Flow problems can be analysed by calculating mass, momentum, and energy balances using the finite volume method within Computational Fluid Dynamics (CFD). These balances can be expressed to account for both compressible and incompressible flow scenarios, incorporating custom source terms (S_m) as needed [28] :

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = S_m \quad [6]$$

$$\frac{\partial \rho u}{\partial t} + \text{div}(\rho u \vec{u}) = -\frac{\partial P}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + F_x \quad [7]$$

$$\frac{\partial \rho v}{\partial t} + \text{div}(\rho v \vec{u}) = -\frac{\partial P}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + F_y \quad [8]$$

$$\frac{\partial \rho w}{\partial t} + \text{div}(\rho w \vec{u}) = -\frac{\partial P}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + F_z \quad [9]$$

The CFD simulation of the screw compressor is notably complex due to the intricate 3D turbulent flow field, along with specific operational and geometric constraints as previously discussed. Initially, a computational domain was established, comprising four distinct domains: the rotor, oil injection, suction, and discharge ports, as depicted in Fig. 3. The inlet, discharge, and oil injection ports are located on the stationary casing of the compressor, while the rotor rotate in opposite directions causing the change in the working volume. Within this setup, ambient air enters the compressor through the suction port and undergoes compression due to change of the working chamber volume. The initial conditions are defined on the static state with the reference pressure at 0 barg. The pressure port is defined as a "Pressure Outlet," whereby the pressure linearly increases to an effective pressure of 4 bar within 600 time steps and remains constant at 4 bar thereafter. Thus, by selecting an outlet pressure consistent with the initial condition of the analysis, divergence is prevented. In screw compressors, oil feed is realized through the oil port where oil is pushed by the outlet pressure. The pressure at the inlet of the oil port is equal to the outlet pressure. Additionally, the rotors were modelled as rotating walls, with the male rotor operating at 4000 rpm.

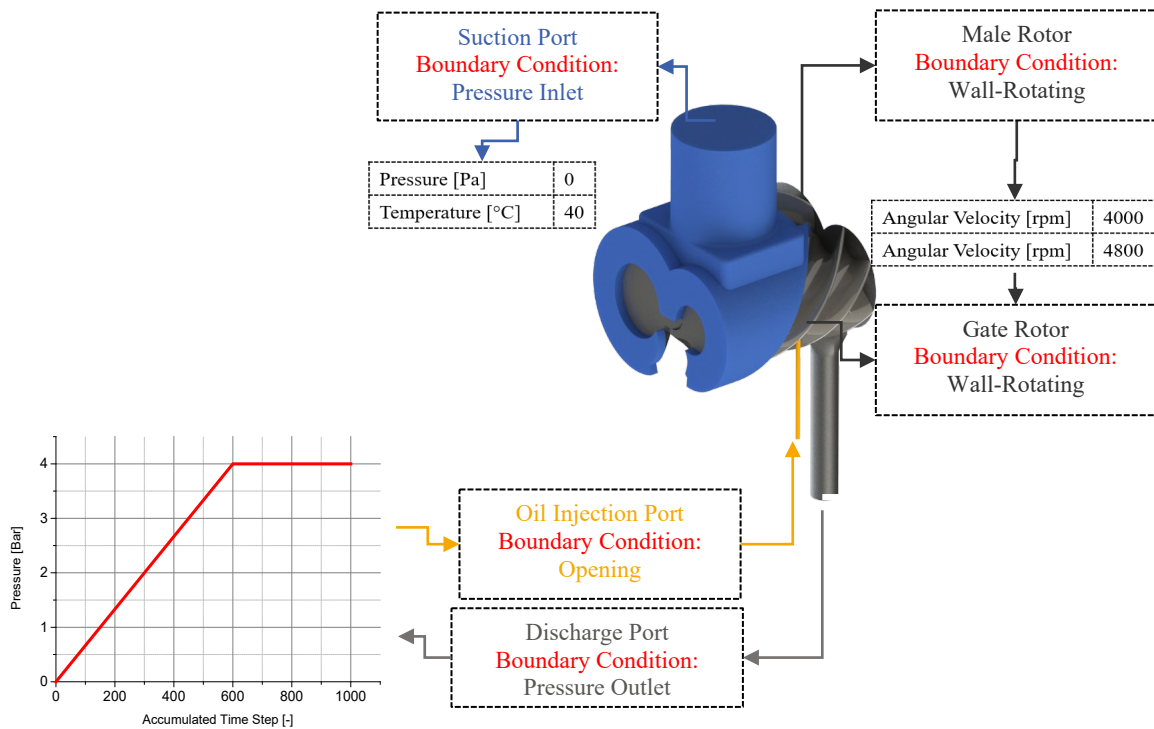


Figure 3. Fluid flow domain and boundary conditions

The rotor domain mesh was generated using SGORGTM, which is a stand-alone numerical CAD-CFD interface used to generate a numerical mesh of rotating parts of a screw machine for predefined number of time steps. The grids are imported to the CFX solver for each time step using a junction box routine . [25].

The size of the mesh shown in Figure 4 is defined by the mesh parameters as shown in Table 4.

Table 4. Specification of grid creation parameters [25]

Parameter	Parameter Value
Circumferential Divisions Female Rotor	60
Radial Divisions	6
Angular Divisions	72
Axial Divisions	0
Interlope Divisions	80

The orthogonality of the generated numerical mesh is shown in Fig. 4. Orthogonality is the essential quality of a numerical mesh Rotor geometry was created with a minimum mesh quality of at least 16 deg, as depicted in Fig. 4., satisfying the minimum mesh quality criterion of 15 deg [29]. Another important criteria for mesh quality is the density of the mesh in clearance gaps. In this region, as shown in Figure 5, the y^+ value is below 5.

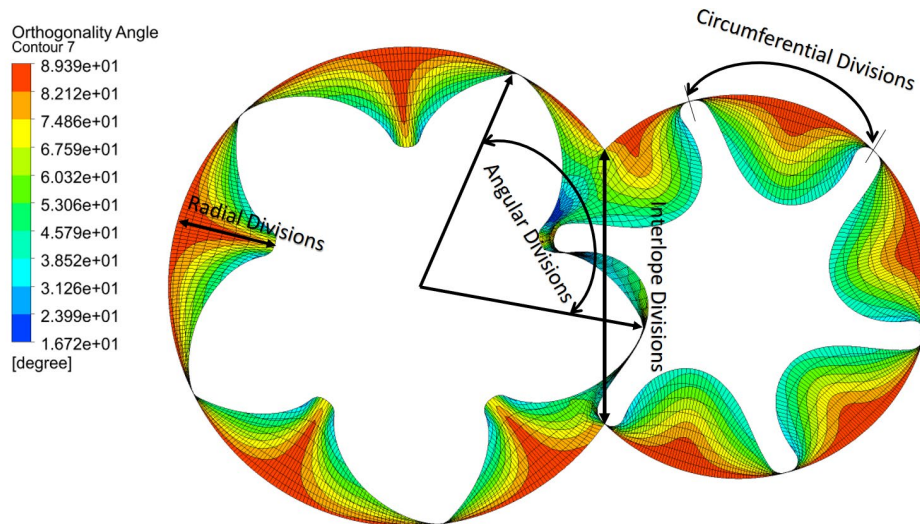


Figure 4. Mesh orthogonality angle

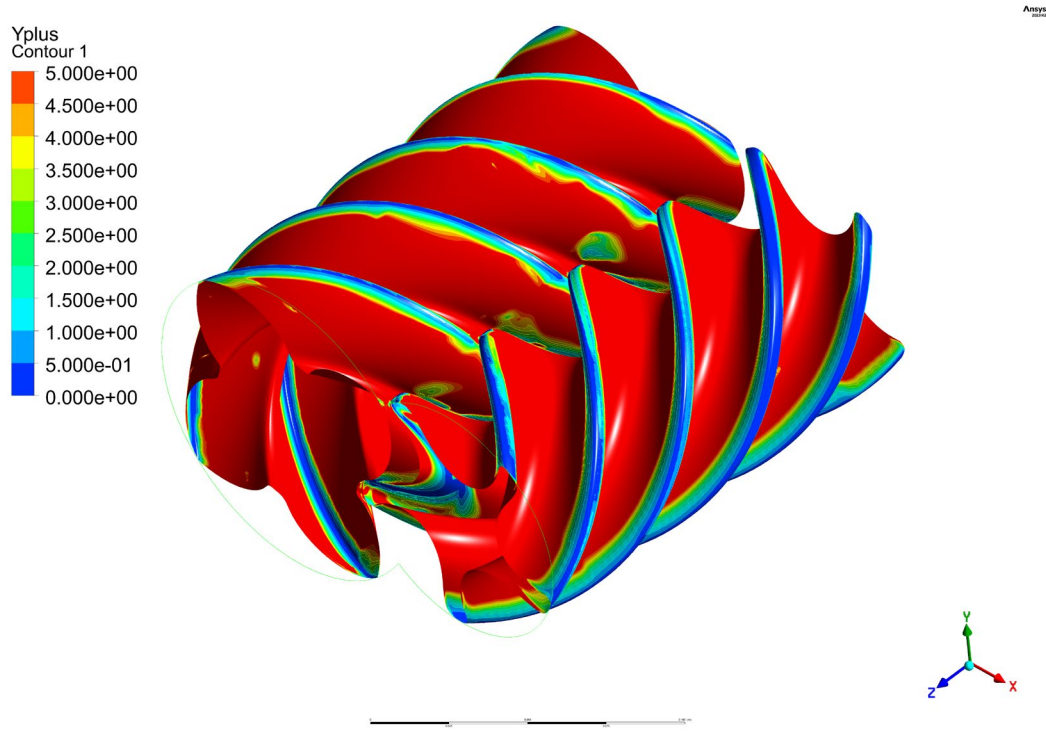


Figure 5. y^+ Contour

5. Results and Discussion

Optimising the screw compressor to minimize specific power is conducted in two stages: Rotor Optimisation and Port Optimisation. To achieve this, 284 design sets were generated using the central composite design approach. These design sets encompassed 13 parameters crucial for defining the rotor profile, forming the basis for rotor optimisation using Response Surface Optimisation (RSO).

Due to the time required for conducting CFD-based solutions, the variations generated by the MATLAB script were processed using multi-chamber thermodynamics from SGORGTM. Following these calculations, the subsequent step involved assessing the minimum and maximum performance values. This assessment considered the parameter range and depended on the variations created. The results were summarized using a response surface methodology, as detailed in Table 5.

Table 5. Calculated minimum and maximum value

Output Parameter	Calculated Min. Value	Calculated Max. Value	Max. Calculated Error
η_v [%]	0.70	0.82	0.019
η_a [%]	0.56	0.66	0.017

$\dot{V}$ [$\text{m}^3 \cdot \text{m}^{-1}$]	5.22	6.35	0.114
$\dot{m}$ [$\text{kg} \cdot \text{h}^{-1}$]	372.5	452.2	8.11
T_{outlet} [$^{\circ}\text{C}$]	60.6	69.2	0.078
P_w [kW]	29.3	33.6	0.267
P_s [kW]			
$\cdot (\text{m}^3 \cdot \text{h}^{-1})^{-1}$]	5.2	6.1	0.124
η_t [%]	0.40	0.55	0.026

Upon establishment of the response surface model, sensitivity analysis involves calculating partial derivatives with respect to each input variable. These derivatives quantify the rate of change of the response as the input variables vary. Subsequently, the evaluation of magnitude entails scrutinizing the amplitude of the partial derivatives to discern the sensitivity of the model to each input variable. The effect of the geometric parameters on specific power was shown in Fig. 6 according to sensitivity analysis used to calculate the impact of input parameters on output variable [30]. The distance between axes (A_c) and the female rotor outer diameter, affecting the tooth depth, have 51 % and 6 % sensitivities, respectively. Therefore, the most critical parameter for thermodynamic performance is the rotor depth. Other critical parameters are p_2 with 14 %, p_1 10.5 %, τ 9.8 %, and v_n 8.6 %.

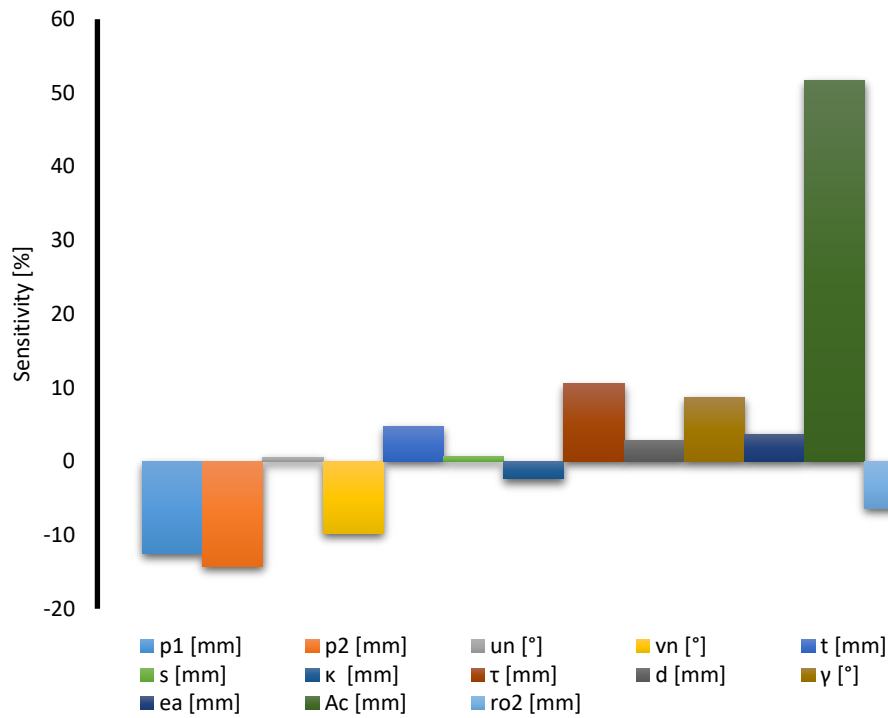


Figure 6. Sensitivity for geometric parameters.

The optimum rotor geometric parameters and the obtained thermodynamic results are shown in Table 6 and Table 7 respectively. The maximum difference of optimum point results between SCORGTM and RSO is 2.52 %, which can be considered acceptable from general engineering point of view.

Table 6. Optimum rotor parameters and thermodynamic results

Parameter	Candidate Point
p_1 [mm]	2.96
p_2 [mm]	3.47
u_n [deg]	8.4
v_n [deg]	24.78
t [mm]	0.52
s [mm]	1.82
κ [mm]	2.46
τ [mm]	1.57
d [mm]	1.43
γ [deg]	1.18
e_a [mm]	26
A_c [mm]	88.02
r_{og} [mm]	50.88

Table 7. Thermodynamic Results for Candidate Point

Output Parameter	Candidate Point	SCORG	Error [%]
P_s [kW · (m ³ · h ⁻¹) ⁻¹]	5.12	5.19	-1.35
η_v [%]	80.6	80.6	0
η_a [%]	64	65.61	2.52
$\dot{V}$ [m ³ · m ⁻¹]	6.22	6.25	0.48
$\dot{m}$ [kg · h ⁻¹]	450.51	445.74	-1.06
T_{outlet} [°C]	68.4	68.4	0

The distance between axes (A_c) and the female outer radius (r_{og}) defining the tooth depth were examined in this study, as shown in Table 8. Reducing the distance between axes (A_c) from 90 mm to 88.02 mm while increasing the female outer radius (r_{og}) from 50.4 mm to 50.9 mm resulted in a 2.89% improvement in specific power and a 10% increase in tooth depth, as shown in Fig. 7.

Table 8. Tooth depth effects

Parameter	$A_c = 90$ mm	$A_c = 88.02$ mm	Difference [%]
-----------	---------------	------------------	----------------

	$r_{om} = 50.4 \text{ mm}$	$r_{og} = 50.9 \text{ mm}$	
$r_{pm} \text{ [mm]}$	40.91	40.01	-2.20
$r_{pg} \text{ [mm]}$	49.09	48.01	-2.20
$r_{dm} \text{ [mm]}$	39.60	37.12	-6.27
$h_a + h_d \text{ [mm]}$	24	26.48	10.34
$P_s \text{ [kW} \cdot (\text{m}^3 \cdot \text{h}^{-1})^{-1}]$	5.34	5.19	2.89
$n_v \text{ [%]}$	78.5	80.6	2.68
$n_a \text{ [%]}$	63.5	65.61	3.32
$\dot{V} \text{ [m}^3 \text{ m}^{-1}]$	6.08	6.25	2.80
$\dot{m} \text{ [kg h}^{-1}]$	433.94	445.74	2.72

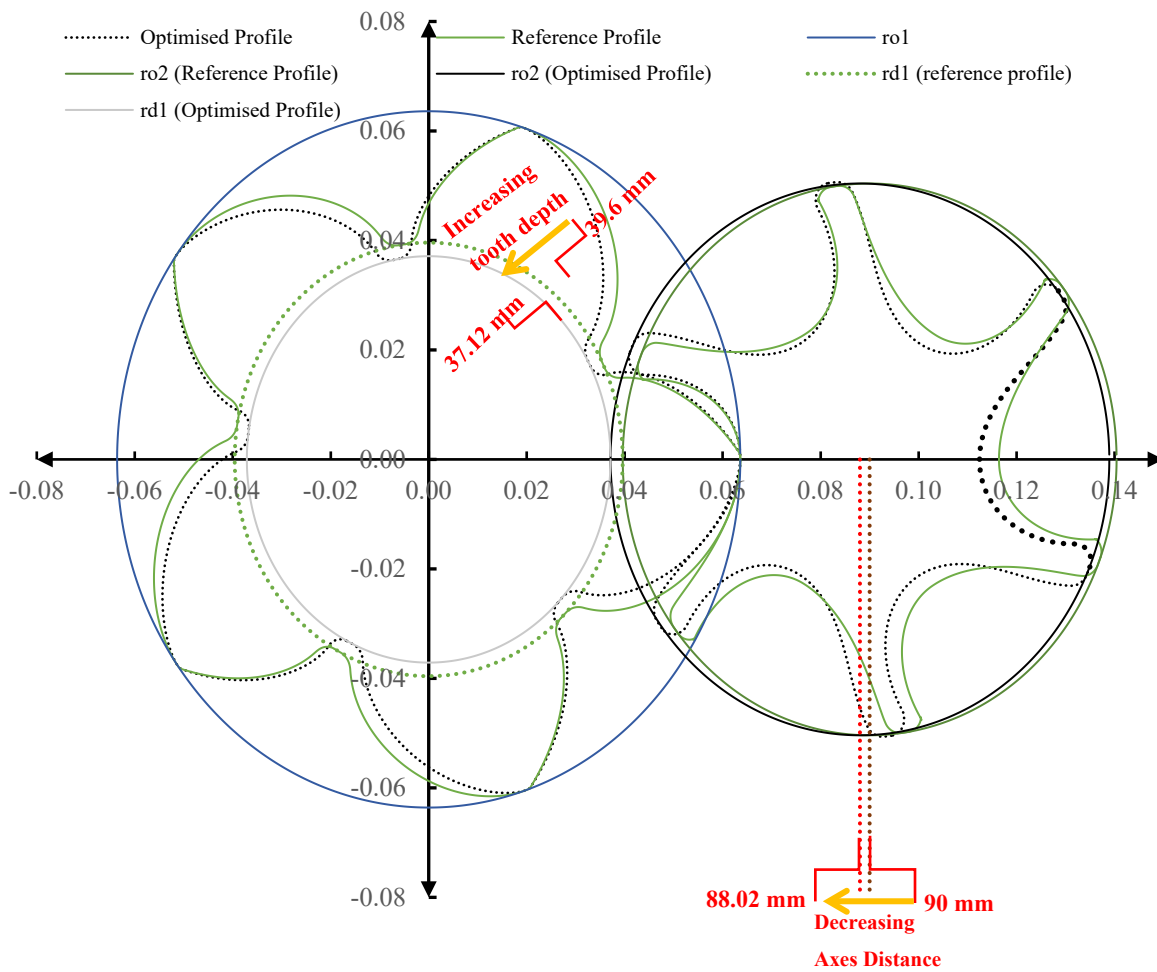


Figure 7. Rotor Depth Defining

Subsequently, rotor optimisation was carried out, followed by port optimisation, utilizing the optimised rotor profile to establish parameters of the port geometry. The impacts of these parameters on P_s and the resulting thermodynamic outcomes were assessed, as shown in Fig. 8. The maximum difference between the optimum point results of SCORGTM and CFD is only 0.19%, indicating an acceptable error level. While rotor optimisation yielded a specific power

of at $5.19 \text{ kW} \cdot (\text{m}^3 \cdot \text{h}^{-1})^{-1}$ this figure decreased slightly to $5.13 \text{ kW} \cdot (\text{m}^3 \cdot \text{h}^{-1})^{-1}$ with port optimisation.

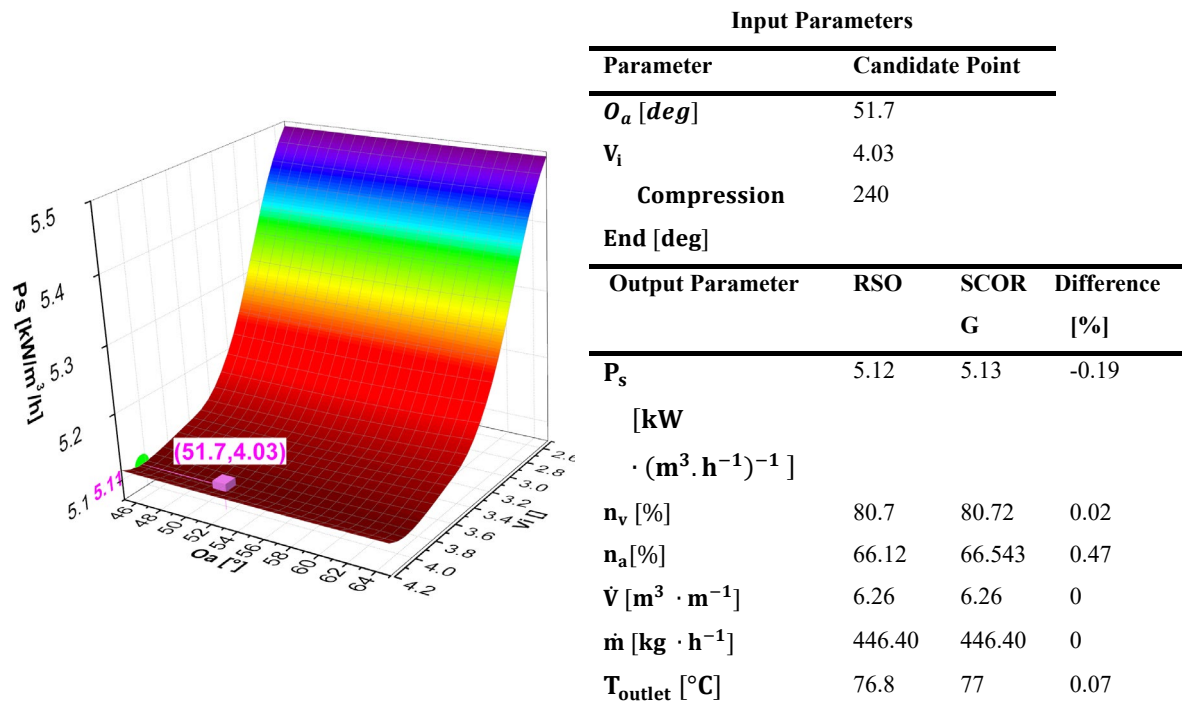


Figure 8. Effects of parameters for P_s [kW · (m³ · h⁻¹)⁻¹]

The analysis of the optimised screw compressor design, conducted through CFX, underwent evaluation via CFD post-processing. When comparing the pressure values calculated by SCORGTM with CFX results at various male rotation angles of 210 deg, 240 deg, and 260 deg, as shown in Fig. 9 and Fig. 10 during compressor operation, an acceptable maximum difference of 8% was observed, as detailed in Table 9.

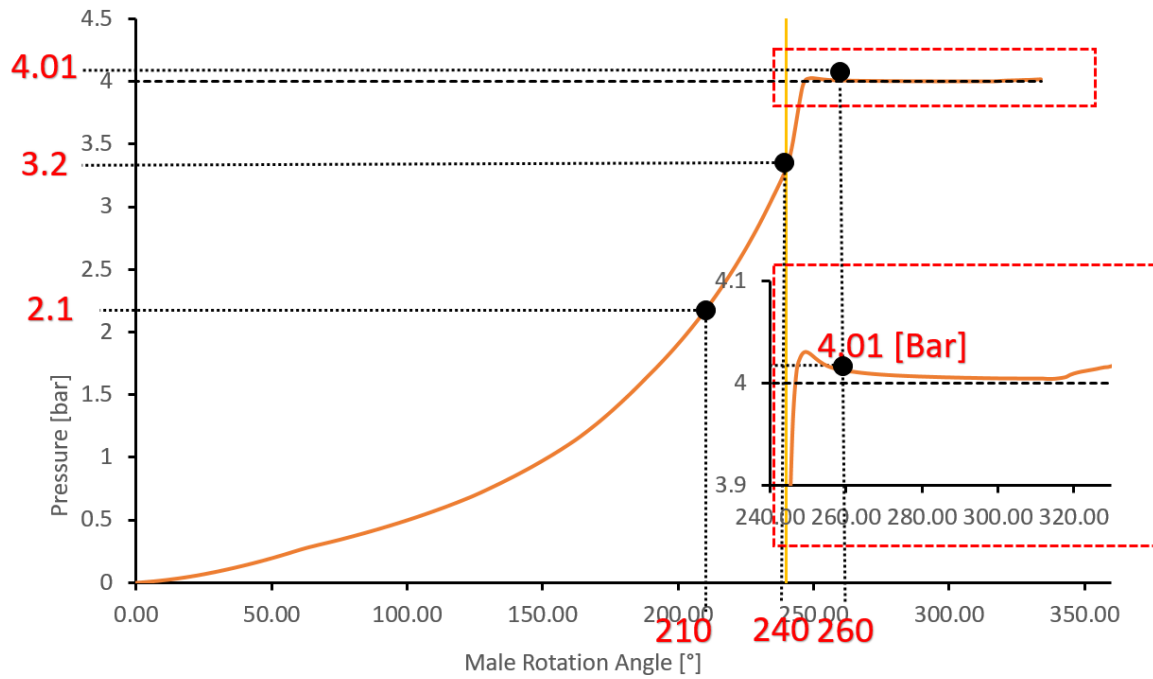
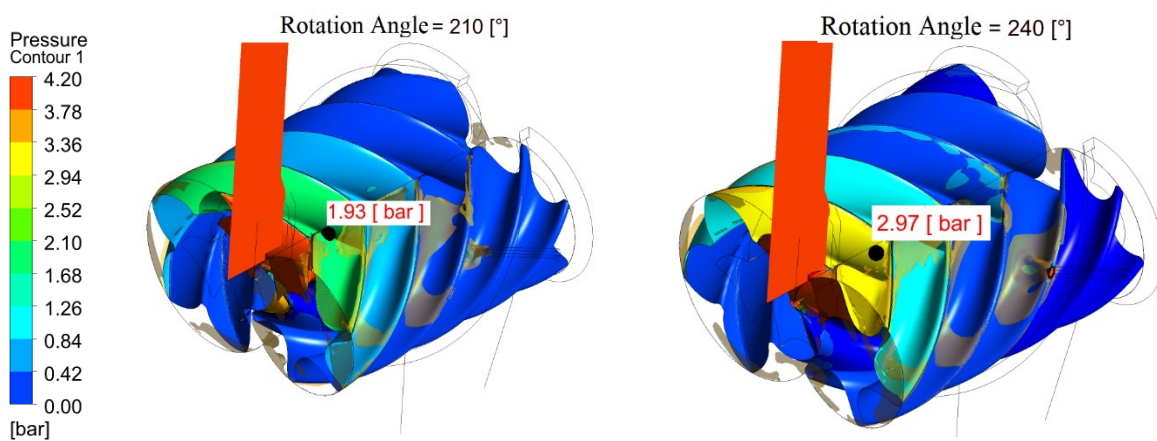


Figure 9. Pressure angle diagram of twin-screw compressors

Table 9. Comparison SCORG and CFX results

Rotation Angle	SCORG (Fig. 9)	CFX (Fig. 10)	Difference [%]
210 °	2.1 [bar]	1.93 [bar]	8 %
240 °	3.2 [bar]	2.97 [bar]	7 %
260 °	4.01 [bar]	4.05 [bar]	1 %



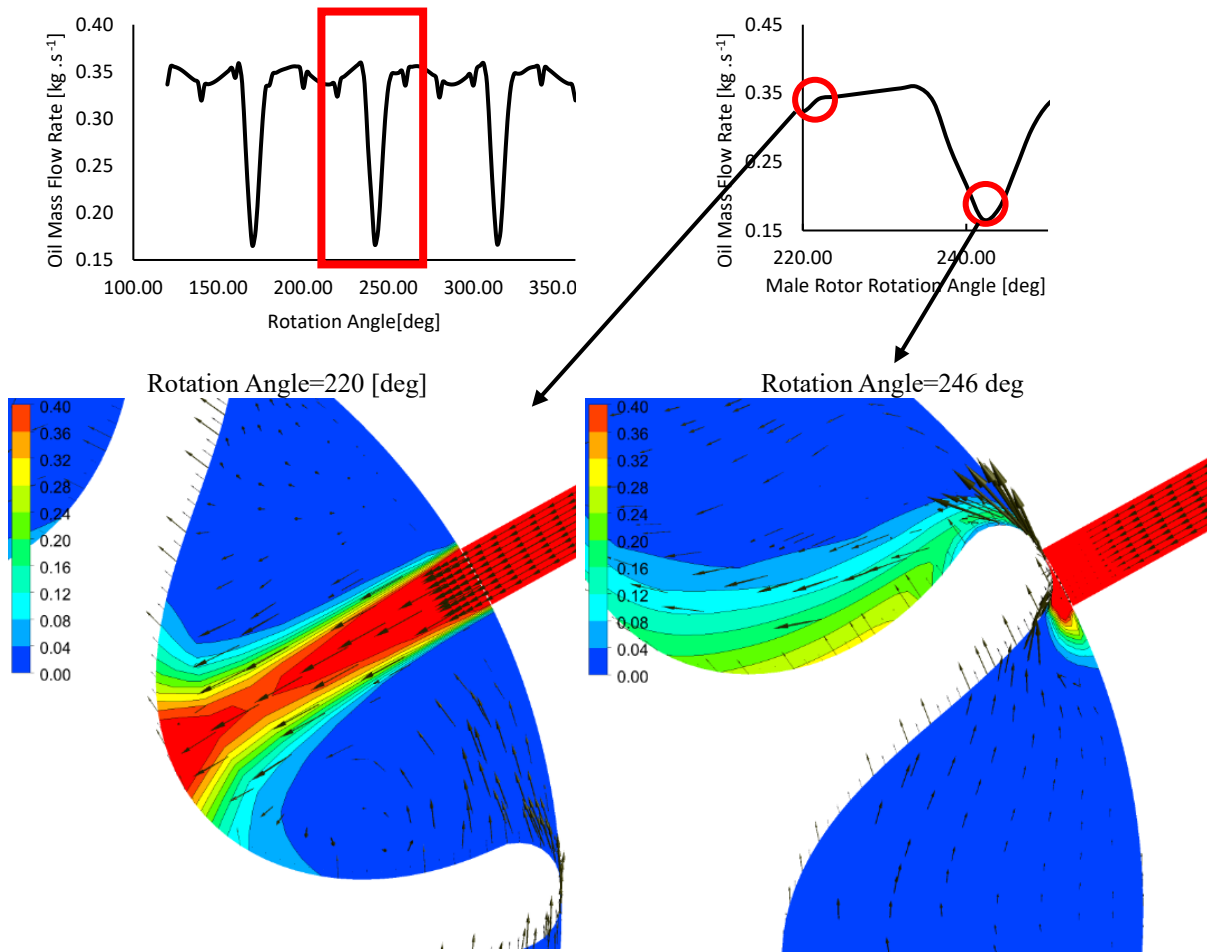


Figure 11. Oil Volume Fraction and the injection port position

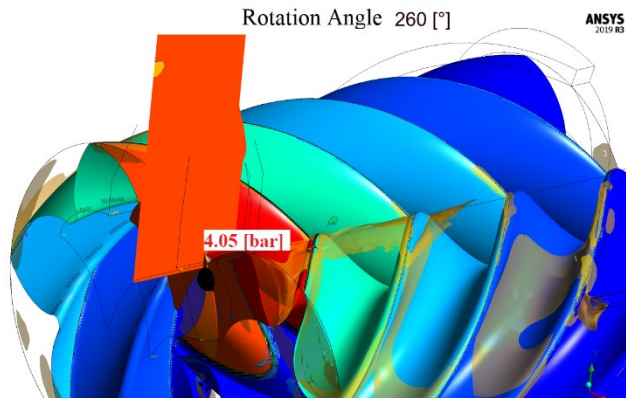


Figure 10. Pressure Contour for different rotation angles

The oil characteristic feeding the compressor, as shown in Fig. 11, is contingent upon both the position of the rotor and its profile. This profile ensures thermal control of the flow volume while also preventing condensation on the air side. Therefore, accurate determination of the oil flow rate is a crucial parameter. The average oil flow rate has been calculated as $0.26 \text{ kg} \cdot \text{s}^{-1}$ using CFX. This value indicates a 3.8% deviation from the calculated oil flow rate.

The leakage flow, a key performance parameter impacting volumetric efficiency, occurs due to pressure difference between neighbouring chambers, as illustrated in Fig. 12. Due to the leakage flows shown in Fig. 12, a loss of 19.4% in volumetric flow occurs, limiting the volumetric efficiency to 80.6%. Therefore, improving leakage flows is an important research topic for compressors.

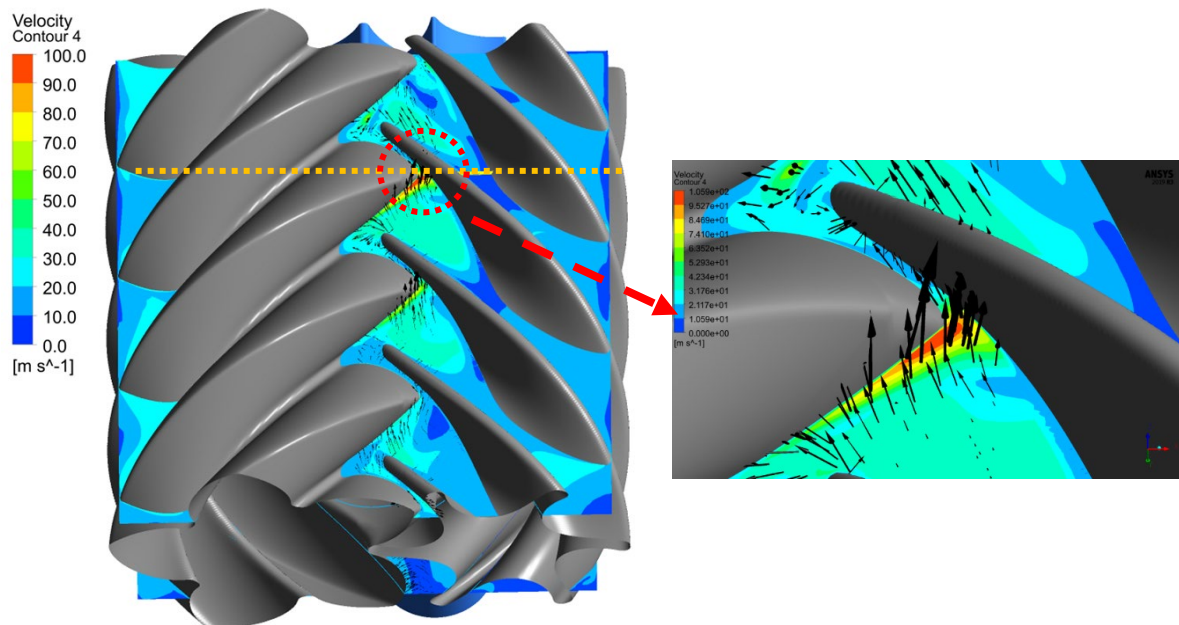


Figure 12. Leakage flows

In screw compressors, the working chamber is linked to ports, resulting in intermittent unsteady flow during the suction and discharge processes, occurring periodically [21], as shown in Fig. 13. Therefore, the compressor flow rate is computed by averaging the instantaneous flow rates over a periodic time interval.

As a result, the fluctuation between the suction and discharge ports arises from the variation in surface area dependent on rotor positions, primarily due to the significantly smaller axial and radial surface areas at the discharge port. Consequently, the mass flow rate of the discharge port varies between $0.05 \text{ kg} \cdot \text{s}^{-1}$ and $0.30 \text{ kg} \cdot \text{s}^{-1}$ for each rotor position.

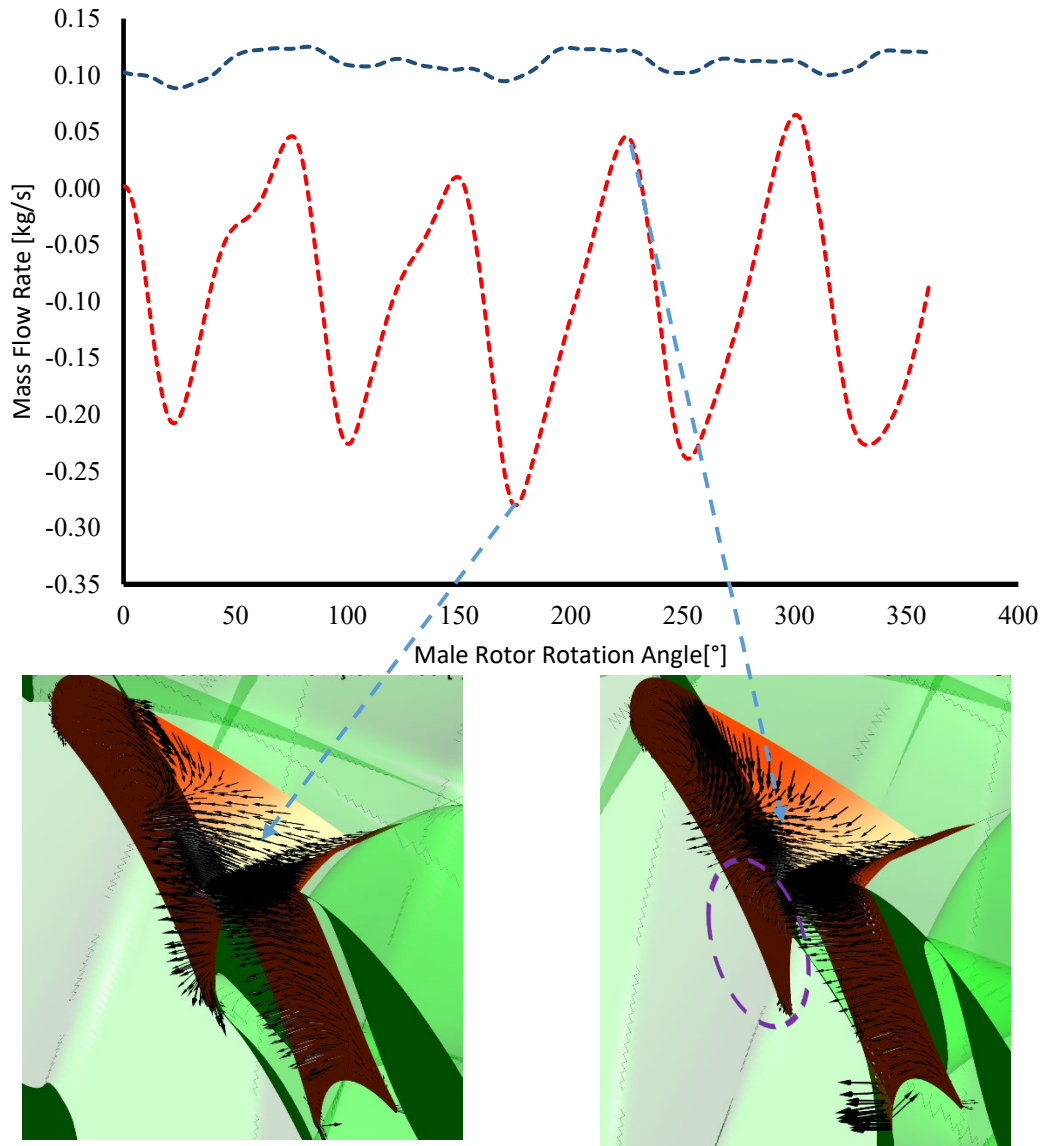


Figure 13. Mass flow rate of suction and discharge Port

When the CFD performance results obtained by CFX are compared with the SCORG thermodynamic results, the maximum difference is 3.8 %, as shown in Table 10.

Table 10. Comparison SCORG and CFX results

	SCORG	CFX	Difference [%]
P [kW]	18.8	18.2	-3.19
$\dot{m}_{air}$ [kg · h ⁻¹]	446.21	432	-3.18
$\dot{m}_{oil}$ [kg · s ⁻¹]	0.26	0.27	3.8
T _{outlet} [°C]	68.3	70.2	2.78

6. Conclusion

Optimisation procedure , encompassing both rotor and port optimisation, was implemented to achieve the most efficient air pressurization in an oil-injected twin-screw air compressor, minimizing specific power under similar operating conditions. The thermodynamic results of the optimised design were subsequently validated through CFX simulations, utilizing a VOF multiphase model to account for compressible flow. Specifically, this validation was carried out for a 5/6 lobe combination compressor operating at a discharge pressure of 5 bar and a male rotor revolution speed of 4000 rpm.

The key conclusion that can be drawn from the study are as follows:

1. Rotor and Port Optimisation: The study successfully conducted rotor and port optimisation and subsequently validated the results using ANSYS CFX for an oil-injected screw compressor. The optimisation of rotors was performed by increasing the rotor depth.
2. Rotor Optimisation Impact: Reducing the distance between axes (A_c) from 90 mm to 88.02 mm, and increasing the female outer radius (r_{og}) from 50.4 mm to 50.9 mm, resulted in a significant reduction in specific power of 2.89%. Moreover, increasing tooth depth by 10% also contributed to this improvement. These changes led to enhancements in specific power (2.89%), volumetric efficiency (2.68%), adiabatic efficiency (3.32%), and volumetric flow (2.80%).
3. Port Optimisation Impact: Port optimisation yielded a notable 1.17% improvement in specific power. The adiabatic efficiency was levered up to 5% by increasing the rotor depth for 14% while at the same time the specific power was reduced by 4% according to Reference Profile.
4. Comparison of CFD and thermodynamic Results: When comparing the performance results obtained from CFX with those from SCORG, a maximum difference of 3.8% was observed. This discrepancy falls within an acceptable margin of error with maximum limit 10%.

These findings underscore the effectiveness of rotor and port optimisation in enhancing the performance of oil-injected screw compressors, with CFX validation results validating the efficacy of this optimisation approach.

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