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Numerical investigation on flow characteristics and working performance in oil-injected sliding vane rotary compressors

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Abstract:	Oil is injected into Sliding Vane Rotary Compressors (SVRCs) for cooling, sealing and lubrication purposes. This paper presents numerical investigations on the oil-gas two-phase flow characteristics of the SVRC followed by studying the effects of the oil injection parameters on the compressor performance. The analytical grid generation methodology was employed to discretise the rotating and deforming rotor fluid domain. The Computational Fluid Dynamics (CFD) model of the SVRC was developed and validated with experimental test data. The flow topology was analyzed to illustrate the flow and oil distribution features within the compressor. Moreover, the influence of the oil injection parameters (oil to gas ratio, oil injection angle, oil injection temperature and rotational speed) on the compressor performance were explored. The results show that the oil accumulates at the leading sides of the blades and then flows into the blade tip region, which helps to reduce the internal leakage. In the reference case, the compressor achieves the volumetric efficiency of 95.1%, the adiabatic efficiency of 60.3%, the exhaust temperature of 337.4 K and the specific power of 9.86 kW/(m ³ ·min ⁻¹). Furthermore, increasing oil to gas ratio leads to better cooling and sealing performance, but also results in higher specific power. Oil to gas mass ratio has higher sensitivity than oil injection temperature or angle. About 8° shift in oil injection angle results in 4.2% decrease in specific power. The lubricating oil capacity should be increased accordingly to the increasing rotational speed.
Response to Reviewers:	RESPONSE TO REVIEWERS NUMERICAL INVESTIGATION ON FLOW CHARACTERISTICS AND WORKING PERFORMANCE IN OIL-INJECTED SLIDING VANE ROTARY COMPRESSORS International Journal of Refrigeration JIJR-D-24-01153 The Authors would like to thank all the Reviewers for their comments, which have served to improve the paper quality. The original Reviewers' comments have been reported in red and italic fonts. The Authors' answers are reported in black. In the revised manuscript, the modified text is highlighted in yellow. RESPONSE TO REVIEWER #1 CFD analysis of oil injected sliding vane rotary compressors for flow characteristics and working performance is observed in the manuscript. It is appreciated the effort taken by the authors for writing good manuscript.

Numerical investigation on flow characteristics and working performance in oil-injected sliding vane rotary compressors

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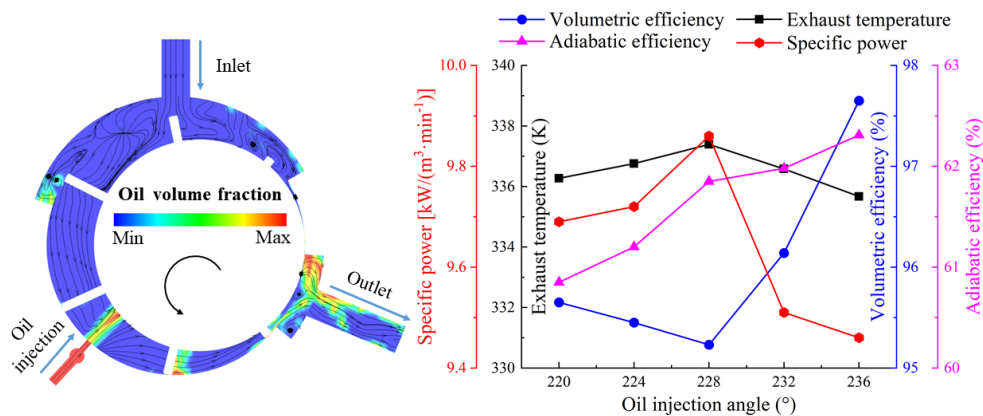
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Highlights

- CFD model of an oil-injected vane compressor was validated with experimental data.
- Custom deforming mesh generation using Matlab routines was applied in the model.
- Effects of the injection parameters on the compressor performance are explored.
- About 8° shift in oil injection angle resulted in 4.2% decrease in specific power.
- Oil to gas ratio has higher sensitivity than oil injection angle or temperature.

Graphical abstract



Abstract

Oil is injected into Sliding Vane Rotary Compressors (SVRCs) for cooling, sealing and lubrication purposes. This paper presents numerical investigations on the oil-gas two-phase flow characteristics of the SVRC followed by studying the effects of the oil injection parameters on the compressor performance. The analytical grid generation methodology was employed to discretise the rotating and deforming rotor fluid domain. The Computational Fluid Dynamics (CFD) model of the SVRC was developed and validated with experimental test data. The flow topology was analyzed to illustrate the flow and oil distribution features within the compressor. Moreover, the influence of the oil injection parameters (oil to gas ratio, oil injection angle, oil injection temperature and rotational speed) on the compressor performance were explored. The results show that the oil accumulates at the leading sides of the blades and then flows into the blade tip region, which helps to reduce the internal leakage. In the reference case, the compressor achieves the volumetric efficiency of 95.1%, the adiabatic efficiency of 60.3%, the exhaust temperature of 337.4 K and the specific power of $9.86 \text{ kW}/(\text{m}^3 \cdot \text{min}^{-1})$. Furthermore, increasing oil to gas ratio leads to better cooling and sealing performance, but also results in higher specific power. Oil to gas mass ratio has higher sensitivity than oil injection temperature or angle. About 8° shift in oil injection angle results in 4.2% decrease in specific power. The lubricating oil capacity should be increased accordingly to the increasing rotational speed.

Keywords

Sliding vane rotary compressor; Oil injection; CFD; Deforming grid; Volumetric efficiency; Specific power

41 Nomenclature

Symbols		W	Power [W]
α	Volume fraction [-]	Z	Axial length [mm]
γ	Specific heat ratio [-]	Subscripts	
η	Efficiency [-]	ad	Adiabatic
μ	Dynamic viscosity [Pa·s]	axi	Axial
ρ	Density [kg/m ³]	bla	Blade
τ	Torque [Nm]	$comp$	Compressor
λ	Thermal conductivity [W/(m·k)]	g	Gas
c_p	Specific heat capacity [J/(kg·K)]	gap	Tangential clearance
C	Clearance [μm]	in	Inlet
e	Eccentricity distance [mm]	inj	Injection
F	Body force [-]	l	Liquid
g	Gas phase [-]	out	Outlet
G	Gravity term [-]	spe	Specific
h	Specific enthalpy [J/mol]	th	Theoretical
l	Liquid phase [-]	tip	Blade tip
m	Mass flow rate [kg/s]	vol	Volumetric
n	Rotational speed [RPM]	Acronyms	
p	Pressure [bar]	CFD	Computational Fluid Dynamics
Q	Volume flow rate [m ³ /min]	FVM	Finite Volume Method
R	Radius [mm]	SVRC	Sliding Vane Rotary Compressor
t	Time [s]	UDF	User Defined Function
T	Temperature [K]	UDND	User Defined Nodal Displacement
Th	Thickness [mm]	VOF	Volume of Fluid
v	Velocity [m/s]		

1. Introduction

According to National Aeronautics and Space Administration, the global average temperature has risen by over 1.18°C compared to the late 19th century (NASA, 2021). This increase has led to significant climate and environmental issues, including extreme weather events, glacier melting and rising sea levels (Yuan et al., 2022). Therefore, addressing carbon dioxide emissions has become an urgent necessity. In the industrial sector, compressed air is one of the main energy consumers, which accounts for around 10% of the electricity consumed in the European Union and China, while in the US, Malaysia and South Africa accounts for about 9% of total energy consumption (Cabello Eras et al., 2020). And this share may exceed 20% if residential and commercial usage (air conditioning, ventilation, pneumatic heating, air pumps, etc.) are included (Vittorini and Cipollone, 2016b). Compressed air systems nowadays are inefficient and characterized by high carbon dioxide emissions (Çağman et al., 2022), presenting high potential for energy savings. Rotary positive displacement compressors, such as screw and sliding vane types, are the most widely used compressor technology for pressures ranging from 7-12 bar and flow rates less than 1000 m³/min (Bianchi et al., 2015). Among them, Sliding Vane Rotary Compressors (SVRCs) have some intrinsic features which state an unforeseen potential due to the advantages of compact design, low vibration, long lifespan, and stable operation (Shouman et al., 2017). Indeed, an energy consumption analysis on positive displacement machines showed that SVRCs were more efficiently than screw compressors if on/off load conditions were taken into consideration (Roberto and Diego, 2014). In addition to being applied for compressed air (Murgia et al., 2017; Valenti et al., 2021), SVRCs are also extensively used for refrigeration (Aw and Ooi, 2021; Liu et al., 2023), automotive air conditioning (Wang et al., 2020) and heat pump (Hosseinnia et al., 2024; Olympios et al., 2024). Therefore, improving the performance and its efficiency contributes to significantly reduce energy consumption and carbon dioxide emissions.

To improve the performance of rotary positive displacement compressors, an oil injection technique has been introduced to transform the current adiabatic compression process towards an isothermal one, which demonstrated the cooling capability of the oil during the compression stage (Deichmann et al., 2023; Li et al., 2023; Zhang et al., 2019). In addition to cooling purpose, oil injection can also achieve lubrication and sealing effects (Wu et al., 2023). Research on improving compressor performance through oil injection technology has mainly focused on screw compressors. He et al. (2018) conducted experimental tests on twin-screw compressors and found that increasing the oil injection flow rate improved both the adiabatic and volumetric efficiencies, and also decreased

the exhaust temperature and specific power. While [Wu et al. \(2022\)](#) performed an experimental investigation on the effects of the oil injection amount on the screw compressor performance, indicating that an oil mass flow rate of 2.28 m³/h achieved the highest efficiency. [Basha et al. \(2018\)](#) studied the performance of an oil-injected screw compressor under various discharge pressures and rotational speeds, and found that the oil injection flow rates had a greater impact on the power consumption than the air flow rate, revealing the oil injection quantity as a key parameter for screw compressors. They also suggested that the screw compressor with two port configurations reduced the maximum gas temperature by 30~35 °C compared to that with one port configuration ([Basha et al., 2021](#)). [Peng and Yan \(2023\)](#) theoretically analyzed the effects of the injection speed, temperature difference, droplet diameter and initial velocity on the heat transfer between the droplets and the working fluid based on a spatial kinematic model, finding that the droplet diameter significantly impacted the gas-liquid heat transfer.

The application of the oil injection technology on the SVRC was a breakthrough in this scientific field. [Kruse \(1980\)](#) introduced oil injection to decrease the blade tip friction, which decreased the driving torque for rotational speeds up to 1500 RPM. Compared to oil-free compressors ([Raykovskiy et al., 2024](#); [Raykovskiy et al., 2023](#)), oil-injected ones achieve high efficiency due to the cooling, lubricating and sealing effects ([Valenti et al., 2013](#); [Valenti et al., 2014](#)). The existing research on oil-injected SVRCs mainly encompasses both theoretical analysis ([Wang et al., 2021a](#); [Wang et al., 2021b](#); [Wen et al., 2024](#)) and experimental approaches ([Hu et al., 2022](#); [Murgia et al., 2015](#); [Park and Park, 2024](#)). Although theoretical studies can reduce computational costs, they cannot capture the complex shapes of real equipment. Experimental studies provide more intuitive and accurate results but face limitations due to the nature of experiments, such as high costs, long duration and low resolution, which prevent comprehensive understanding of all factors. In oil-injected SVRCs, [Bianchi and Cipollone \(2015\)](#) studied the oil film thickness in the blade tip using both theoretical and experimental methods and revealed the influences of the compressor structure on the compressor efficiency. A Lagrangian theoretical model of oil injection technology was then used to evaluate the cooling effects of lubricants due to the high surface-to-volume ratio of oil droplets and predicted a reduction in indicated power ([Bianchi et al., 2015](#)). The model validation on a medium-sized SVRC equipped with a range of pressure-swirl atomizers showed that using an oil injection pressure of 7 bar reduced the energy consumption by nearly 20%.

Limited Computational Fluid Dynamics (CFD) investigations are available on vane machines due to the challenge for representing the moving and deforming domain enclosed between rotor, stator and blades ([Prasad,](#)

2004). Most of existing research relied on commercial meshing software for domain discretization. Lobsinger et al. (2020) employed TwinMesh™ to generate the computational mesh for a vane oil pump in automatic transmission systems of vehicles and performed the CFD simulations with ANSYS CFX. Murthy et al. (2024) developed special routines called User Defined Function (UDFs) which was written in C language to read node coordinate stored in data files and update the computational domain at each step of the simulation. Gu et al. (2021b) used a program including macros and UDFs to control the dynamic grids as well as the motions of different components. The initialization function was utilized to save the initial calculation angle and read the outer contour for user-defined shape. To avoid negative volume in the computational mesh, Rak et al. (2018) modified the geometry by reducing blade thickness and increasing tip clearance. Without using commercial software, Bianchi et al. (2017) developed a novel analytical grid generation method based on the User Defined Nodal Displacement (UDND) approach for vane machines, which employed algebraic techniques to perform transfinite interpolation, post-orthogonalization and smoothing for internal node distribution. Furthermore, Ye et al. (2021) proposed a fully analytical grid generation method suitable for a broader range of rotating blade machinery structures, including multiple working chambers, non-circular stators, offset blades and asymmetric tip profiles.

Most of existing CFD research on vane machines mainly focused on single-phase flow conditions without consideration of lubricating oil (Błasiak et al., 2023; Gu et al., 2021a; Gu et al., 2021b; Montenegro et al., 2014), which may lead to inaccurate results. Gas-liquid two-phase flow in oil-injected SVRCs still poses a great challenge due to the complex geometry of the rotors, the very thin clearance gaps, high-density ratios between the gas and the oil and the smearing of the gas-oil interface when the compressible gas and the incompressible liquid exist in the same domain. Consequently, CFD research on the oil-gas two-phase flow in the oil-injected SVRCs is quite limited. Bianchi et al. (2016) carried out multiphase simulations on an oil-injected SVRC, which allowed to track a recirculation region in the chamber induced by the blade tip leakage flow. The convective heat transfer with air displayed the cooling effects of the oil. The rise in the oil temperature reduced its dynamic viscosity, leading to weaken the hydrodynamic potential which prevent dry contact between stator and blade tips.

In terms of two-phase numerical model, the models used for oil-injected screw compressors can provide reference for the oil-injected SVRCs. Rane et al. (2016) carried out flow analysis in an oil-injected screw compressor based on a Eulerian – Eulerian model, which helped to visualize the internal oil distribution. Another study examined a standard oil-injected screw air compressor, which compared the Euler-Euler approach with the

homogeneous model and achieved strong agreement with experimental data (Basha et al., 2018). The Euler-Lagrange method could be an alternative, but it is not practical for oil-injected compressors due to the absence of initial data on oil droplet sizes and the oil volume fraction exceeds the 12% limit set by CFD vendors (Peng and Yan, 2023). Basha et al. (2019) compared three commonly used numerical models for oil-injected compressors: the Euler model, the Volume of Fluid (VOF) model and the Mixture model. The VOF model predicted power and flow rates closely matching experimental data and was 25% faster than the full Eulerian-Eulerian model. Therefore, the VOF model was selected for gas-liquid two-phase flow simulations on oil-injected SVRCs in this paper.

To date, the existing research on oil-injected SVRCs is not able to reveal the oil-gas two-phase flow topology and how the oil injection parameters affect the compressor performance. Therefore, in this work, a numerical simulation model of SVRCs was established based on the analytical grid generation methodology proposed earlier by the authors. After validating the model with the experimental data in the literature, the evolution characteristics of the pressure, temperature and velocity fields as well as the oil distribution were revealed. Finally, the effects of oil injection parameters (oil to gas ratio, oil injection angle, oil injection temperature and rotational speed) on the compressor performance (volumetric efficiency, adiabatic efficiency, exhaust temperature and specific power) were identified.

2. Numerical methodology

2.1. Geometrical model

The SVRC is a rotary positive displacement machine consists of a stationary stator, a rotating rotor and several blades. The number of the blades is equal to 7 in this work. Fig. 1 illustrates the schematic of a seven-blade SVRC with port positions. The stator and rotor are installed with an eccentric distance (e). The rotor has seven slots machined at equal angular intervals along the rotor side surface to host the blades. The blades slide along the slots towards the stator inner surface because of centrifugal force. The variable working chamber enclosed by the rotor, stator and blade is thus periodically formed as the rotor rotates along the centre, which performs the intake, compression and exhaust processes of the working medium.

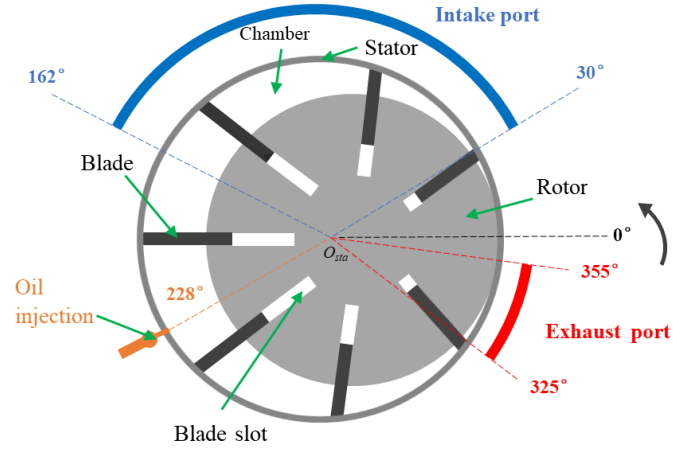


Fig. 1. Schematic of the SVRC section with port positions.

The reference geometric parameters of the SVRC are displayed in Fig. 2 and listed in Table 1. Among them, the clearance gap between the blade tip and the stator is assumed to be constant at 50 μm based on previous research which studied vane machines of similar dimensions (Lobsinger et al., 2020; Zhang and Xu, 2014). The tangential clearance gap between the rotor and the stator, which can be determined according to the assembly size, is also fixed at 50 μm in this work. The axial length of the entire SVRC is 40 mm. The leakage occurs through the blade tip clearance and the tangential clearance, leading to a reduction in volumetric performance.

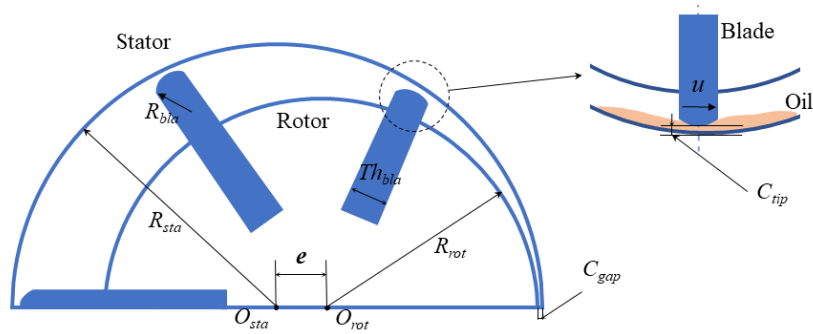


Fig. 2. Geometrical parameters of the SVRC.

Table 1 Geometrical parameters of SVRC.

Quantity	Value	Unit
Number of blades	7	-
Stator radius/ R_{sta}	68.0	mm
Rotor radius/ R_{rot}	55.5	mm
Rotor axial length/ Z_{axi}	40.0	mm
Blade thickness/ Th_{bla}	4.7	mm
Blade tip radius/ R_{bla}	9.5	mm
Blade tip clearance between stator and blade/ C_{tip}	50	μm
Tangential clearance between stator and rotor/ C_{gap}	50	μm

2.2. Computation domain and mesh model

A crucial step for CFD simulations of SVRCs is to develop discretization algorithms of the rotating and deforming rotor fluid domain. An analytical grid generation methodology for rotary vane machinery developed in an earlier work by the authors was employed in this study. The detailed grid generation approach is reported in (Ye et al., 2021) and here briefly recalled. Fig. 3 displays the workflow of this analytical grid generation algorithm. MATLAB routines are used for this custom deforming mesh generation procedure. As illustrated in Fig. 3, all parameters for boundary generation, boundary discretization and node distribution of the rotor fluid domain are input for the first step. The inner and outer boundaries of the rotor profile are generated and discretized to form the rotor domain. It should be noted that a small geometrical simplification is introduced to convert the blade side wall into a short segment on the rotor surface when the blade slides inside the rotor, which does not influence the tip leakage flow as the leakage gap remains unchanged. After that, the mapping processes are performed between nodes from inner boundary to outer boundary. All mapping lines are then equidistantly divided into radial divisions respectively for interior node distribution, resulting in an “O” structure for the rotor mesh. This mesh topology structure contributes to improve accuracies in the non-matching mesh connection between the blade tip region and the core region. At the initial position, the two-dimensional mesh is generated and then assembled into the corresponding three-dimensional mesh which is read by the ANSYS FLUENT solver. A grid database including the three-dimensional node coordinates of the rotor mesh at all time steps is generated through the proposed analytical grid generation procedure. To interface the FLUENT solver with customized grids, a UDF is developed to read node coordinates saved in the grid database and subsequently update the mesh at every time step. Finally, the port domain is connected to the rotor domain by a nonconformal interface.

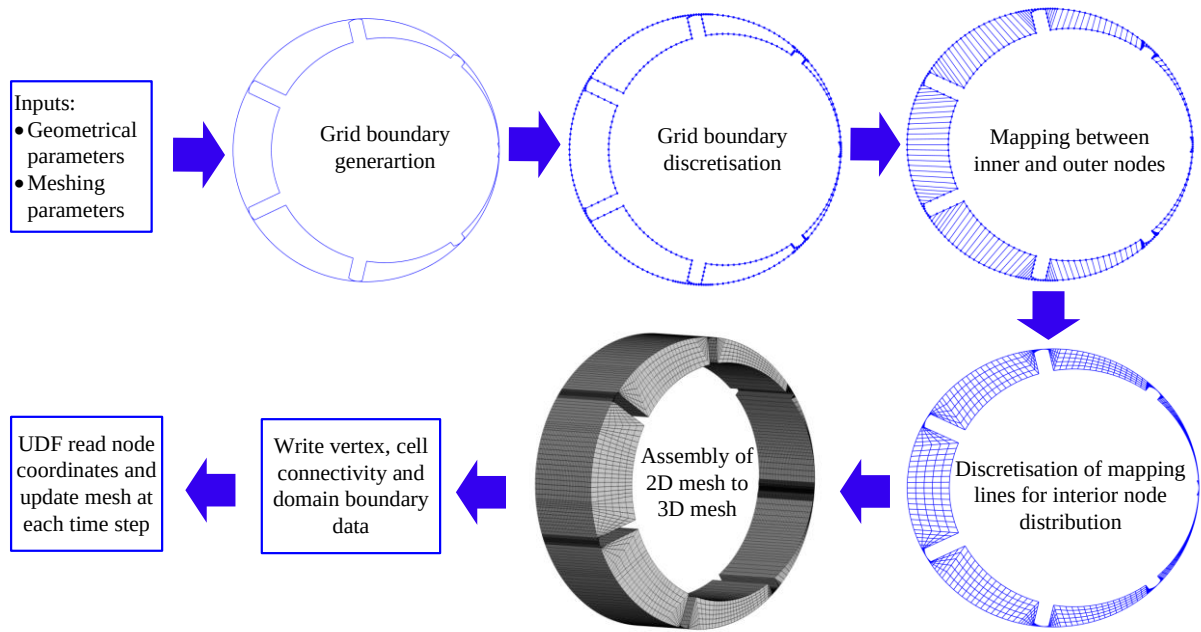


Fig. 3. Analytical grid generation workflow for the rotor fluid domain.

The computational domain consists of the core region and three ports (the intake port, the exhaust port and the oil injection port). The core region mesh is generated using the aforementioned analytical grid generation method, while the port sections are discretized by the meshing tool available in the ANSYS software suite. The computational mesh of the rotor and port fluid domains is showed in Fig. 4. It should be noted that the inflation layers are not take into consideration in this work since the mainstream flow rather than the boundary layer falls into our interest. In addition, neglecting the inflation layers helps to reduce the mesh count and consequently decrease the computation time. The O-Grid topology in the rotors means that there are same number of radial cells in the core as well as the leakage gaps. The blade side walls are parallel surfaces to form a convex connection with the blade tips, and this convexity increases the difficulty of mesh generating due to the fact that the aspect ratio between the tip gap and the core varies by many orders of magnitude, which affects the stability of the solver.

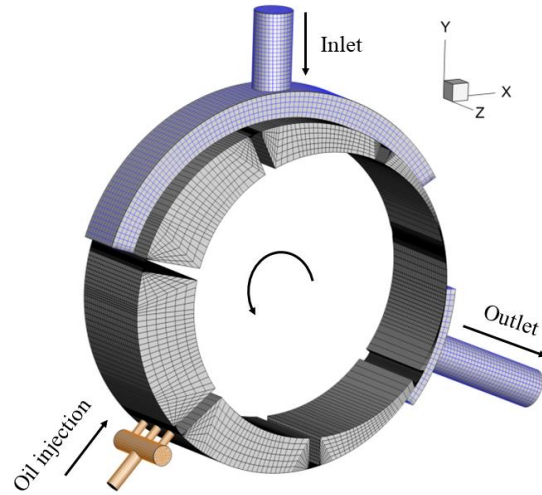


Fig. 4. Computational mesh of rotor and port fluid domains.

2.3. Mathematical calculation model

2.3.1. Assumptions

The CFD simulations on the SVRC involve complex physical processes such as turbulence and multiphase flow. To ensure data accuracy while reducing model complexity and computational costs, a set of assumptions were employed to improve computational efficiency. In terms of the geometry, the clearances between the blade tip and stator inner surface remain unchanged along a full vane revolution. This is a trade-off between the ideal geometrical tangency and actual working conditions, while the blade tip slides along a fluid layer preventing dry contact with the stator (Bianchi et al., 2017). The constant tip clearance assumption avoids the solving of the instantaneous minimum liquid film thickness in the blade tip which would require either simplified hydrodynamic methods or the complex fluid-structure calculations.

Assumptions were also introduced into the physical properties of air and oil to simplify the calculation process. Both oil and air are Newtonian immiscible fluids, leading to no mass transfer between them. Air is regarded as an ideal gas since the pressure values are within only a few bars. The lubrication oil is treated as a continuous incompressible fluid and has constant thermophysical properties in terms of density, specific heat capacity, dynamic viscosity and thermal conductivity. This simplification allowed authors to discard the resulting complexity of calculating the mentioned temperature-dependent properties. Indeed, the temperature at the intake and outlet are around 25 °C and 65 °C respectively. The comparison of the main thermophysical properties of oil at 25 °C and 65 °C is presented in Table 2, from which it can be concluded that the density, specific heat capacity, dynamic viscosity and thermal conductivity do not change significantly with the temperature from 25 °C to 65 °C.

Therefore, the mentioned thermophysical properties at the temperature of 45 °C are used for simulations in this study. Also, there is no heat transfer between the working fluids and the compressor body.

Table 2 Variation of oil thermophysical properties with temperature.

Variables	25°C	45°C	65°C
Density ρ [kg/m ³]	962	950	938
Specific heat c_p [J/(kg·K)]	2160	2250	2395
Viscosity μ [Pa·s]	0.0097	0.0090	0.0078
Thermal conductivity λ [W/(m·K)]	0.152	0.145	0.134

2.3.2. Multiphase Model

The oil undergoes topological changes after being injected into the SVRC. It accelerates within the clearance gaps transitioning from continuous flow to dispersed flow. For complex cases like oil-injected SVRCs, it is not feasible to calculate these topological changes using a single macroscale model. The VOF model is used for gas-liquid two-phase numerical simulations in this work. This model assumes that both air and oil are Newtonian immiscible fluids. The immiscible assumption is applicable here because the oil and air are usually immiscible in typical oil-injected air compressors. The physical properties of the system, such as density, is controlled by the volume fraction:

$$\rho = \alpha_l \rho_l + \alpha_g \rho_g \quad (1)$$

where ρ , ρ_l and ρ_g are densities of mixed phase, liquid phase and gas phase respectively, α_l and α_g are volume fraction of liquid phase and gas phase respectively.

The volume distribution of gas and liquid phase in the flow field can be determined by the constraint:

$$\alpha_g + \alpha_l = 1 \quad (2)$$

These flow variables are then employed in a single set of governing equations for continuity, momentum and energy as follows (ANSYS, 2017):

Continuity:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \quad (3)$$

where t is time, $\vec{v}$ is velocity.

Momentum:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot [\mu(\nabla \vec{v} + \nabla \vec{v}^T)] + \rho \vec{G} + \vec{F} \quad (4)$$

where μ is viscosity, G is gravity term. F is body force. The terms on left-hand side of Eq. (4) describe the transient and convection terms, while the pressure gradient, stress-strain, gravity and external body forces are located on the right-hand side.

Energy:

$$\frac{\partial}{\partial t} \sum_{k=1}^n (\alpha_l \rho_l h_l) + \nabla \cdot [\alpha_l (\rho_l \vec{v}_l h_l - \delta \nabla T)] = 0 \quad (5)$$

$$\frac{\partial}{\partial t} \sum_{k=1}^n (\alpha_g \rho_g h_g) + \nabla \cdot [\alpha_g (\rho_g \vec{v}_g h_g - \delta \nabla T)] = 0 \quad (6)$$

where h_l and h_g are enthalpy of liquid phase and gas phase respectively, v_l and v_g are velocities of liquid phase and gas phase respectively, T is temperature.

2.4. Simulation setup

The ANSYS FLUENT solver was used for CFD simulations of the SVRC. The solver settings are detailly displayed in Table 3. Among them, the SST $k-\omega$ turbulence model and VOF multiphase model are employed in this work. As for the convergence standard, the residual value for energy is 10^{-6} , while the ones for other variables is 10^{-3} . The angular step size is set as 0.25° per time step, thus a full revolution cycle is represented by 1440 angular grid positions. With the rotational speed of 1000 RPM, the time step size is 4.167×10^{-5} s in the solver. Calculation of four revolutions were carried out, leading to a periodic solution in 5760 time steps. The working fluids are air and lubricating oil whose properties are showed in Table 4. The reference operational conditions of the SVRC are listed in Table 5 and used as boundary conditions for the rest of this study. Among them, a constant temperature and pressure boundary condition is set for the intake port, while a constant pressure boundary condition is for the exhaust port. As for the oil injection, at a specified angle position of the rotor in the compression process, oil is injected into the compressor cells through three calibrated orifices displaced along the axial direction of the compressor and fed by a common rail. The operating conditions that were considered as benchmark for the numerical results were 1000 RPM as revolution speed, 6.6 bar as oil injection pressure, 338.2 K as oil injection temperature and 0.14 kg/s as oil injection mass flow rate. And the selection of the oil injection pressure is the same with our previous experimental work (Bianchi et al., 2016).

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Table 3 Detailed settings for ANSYS FLUENT solver.

Criteria	Specification	Setting or value
Solver type	Pressure-based	—
Turbulence model	SST $k-\omega$	—
Multiphase model	Volume of Fluid	Volume Fraction Parameters Formulation: Explicit Volume Fraction Cutoff: 1×10^{-6} Courant Number: 0.25 Interface modelling type: Sharp
Solution methods	Pressure-velocity coupling	Coupled
	Control volume gradients	Green-gauss Node Based
	Spatial discretization	Pressure: PRESTO! Density: First order upwind Momentum: First order upwind Volume fraction: First order upwind Turbulent kinetic energy: First order upwind Specific dissipation rate: First order upwind
	Transient formulation	First order implicit
	Explicit relaxation factors	0.1
	Under-relaxation factors	0.1
	Convergence criteria	Energy: 1×10^{-6} Others: 1×10^{-3}
	Angular step size	0.25°
Solution controls	Number of time steps	5760
	Max iterations per time step	500

Table 4 Physical and thermal properties of air and oil phase.

Property	Air	Oil
Fluid type	Ideal gas	Constant ρ 950 kg/m ³
Specific heat c_p [J/(kg·K)]	1004.4	2250
Viscosity ν [Pa·s]	1.7894×10^{-5}	0.009
Thermal conductivity λ [W/(m·K)]	2.61×10^{-2}	0.145

Table 5 Operational parameters (reference state).

Quantity	Value	Unit
Rotational speed/ n	1000	RPM
Intake temperature/ T_{in}	298.6	K
Intake pressure/ p_{in}	1.01	Bar
Exhaust pressure/ p_{out}	8.7	Bar
Oil injection temperature/ T_{inj}	338.2	K
Oil injection pressure/ p_{inj}	6.6	Bar
Oil injection mass flow rate/ m_{inj}	0.14	kg/s

A grid independence verification was carried out on the reference case of the SVRC. Three different sizes of the rotor domain mesh were considered for comparison, as shown in Table 6. The relative difference of the air mass flow rates, exhaust temperature and shaft power between the medium mesh and the fine mesh were 0.31%, 0.15% and 0.28% respectively. Therefore, taking both simulation accuracy and computational time into account, a rotor domain mesh of about 150,000 cells was employed for all CFD simulation cases. All simulations were carried out with reference to the same port meshes whose cell count was equal to 151,232. With regards to computational resources, all CFD cases ran on a PC with AMD Ryzen 9 7950X CPU at 4.50 GHz. Fourteen cores of the CPU were used while the RAM usage was 12.0 GB. The parallel simulations solved in double precision required about 40 h to accomplish four full revolutions of the SVRC.

Table 6 Results of mesh independence test.

Item	Coarse mesh	Medium mesh	Fine mesh
Cell count of rotor domain	109689	151232	182592
Air mass flow rate (kg/s)	7.2265×10^{-3}	7.2150×10^{-3}	7.1923×10^{-3}
Relative deviation of air mass flow rate (%)	-0.16	—	+0.31
Exhaust temperature (K)	338.42	337.39	336.89
Relative deviation of exhaust temperature (%)	-0.30	—	+0.15
Shaft power (kW)	3.57	3.55	3.54
Relative deviation of shaft power (%)	-0.56	—	+0.28

2.5. Method validation

To validate the CFD simulation model of the SVRC, the numerical results have been compared with the corresponding experimental data (Bianchi et al., 2015), as showed in Fig. 5 and Table 7. Three piezo-resistive

pressure transducers were employed to measure the cell pressure in the experiment, while in the present simulations a series of monitoring points were set to obtain the internal pressure value. Fig. 5 displays the comparison of the cell pressure trace against the cell volume between in the simulation and the experiment. Within the testing range, the maximum error between simulated and experimental pressure values was within 5.0%. The air mass flow rate, exhaust temperature, shaft power and indicated power were also obtained in simulations for comparison with the experimental data, as illustrated in Table 7. The errors between simulated and experimental values of the air mass flow rate, exhaust temperature, shaft power and indicated power were 4.2%, 2.7%, 3.8% and 6.3% respectively. The errors between the simulated results and experimental data could be the absence of the axial leakage gap in the model. Therefore, this outcome allows to retain the quantitative validation of the numerical CFD model developed in this study as satisfactory.

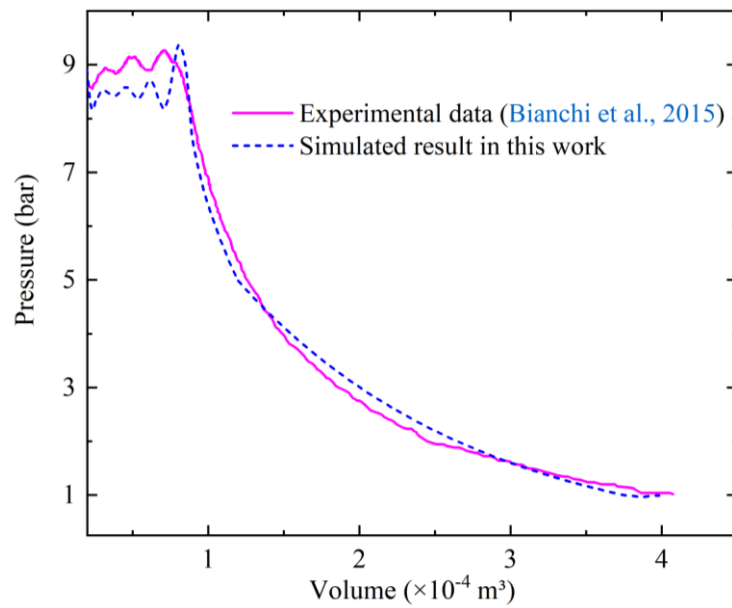


Fig. 5. Comparison of pressure trace between in the experimental data and in the simulated result.

Table 7 Comparison of experimental and simulation results.

Items	Experimental data (Bianchi et al., 2015)	Simulated result in this work	Relative error
Air mass flow rate (kg/s)	0.069	0.072	4.2%
Exhaust temperature (K)	361.85	352.46	2.7%
Shaft power (kW)	23.10	24.00	3.8%
Indicated power (kW)	20.9	19.6	6.3%

3. Flow analysis and discussion of the results

The flow topology including pressure, velocity and temperature fields as well as the oil distribution are visualized in this section. The evolution properties of the instantaneous mass flow rates at the intake and exhaust ports, the rotor torque and the cell pressure are then analyzed. Also, the compressor performance was evaluated by volumetric efficiency, adiabatic efficiency, exhaust temperature and specific power.

3.1. Internal flow characteristics

3.1.1. Pressure field

The seven-blade configuration remains unchanged for all simulation cases, leading to an angular spacing of approximately 51.4° per cycle. Fig. 6 illustrates the pressure contours at mid-length cross section over one cycle. The pressure in the three ports remains uniform, while the pressure in the core presents a large gradient. The air is fed to the compressor at a pressure of 1.01 bar through the intake port ranging from 30° to 160° . After that, the air is continuously compressed with the rotor revolution, leading to a gradual rise in the cell pressure. The oil injection port is positioned at 228° whose intake pressure remains constant at 6.6 bar. As the rotor rotates, the pressure at the injection port fluctuates because the pressure of the chamber connected to the oil injection port changes with time. When a blade slides along the injection port, the pressure at the port rises sharply and even exceeds the injection pressure. This is due to the small blade tip clearance, leading to intense compression of the fluid at the injection port. After the blade passes through the injection port, the oil flows smoothly into the chamber and the pressure at the injection port gradually returns to normal values. Before the chamber is connected to the exhaust port, the pressure reaches its maximum value slightly above 8.7 bar. This indicates that there is a certain over-compression of the working medium. During the exhaust stage from 325° to 355° , the chamber pressure tends to be consistent with the exhaust pressure. The pressure distribution at the tangential clearance clearly shows the effective sealing of the high-pressure exhaust port from the intake port.

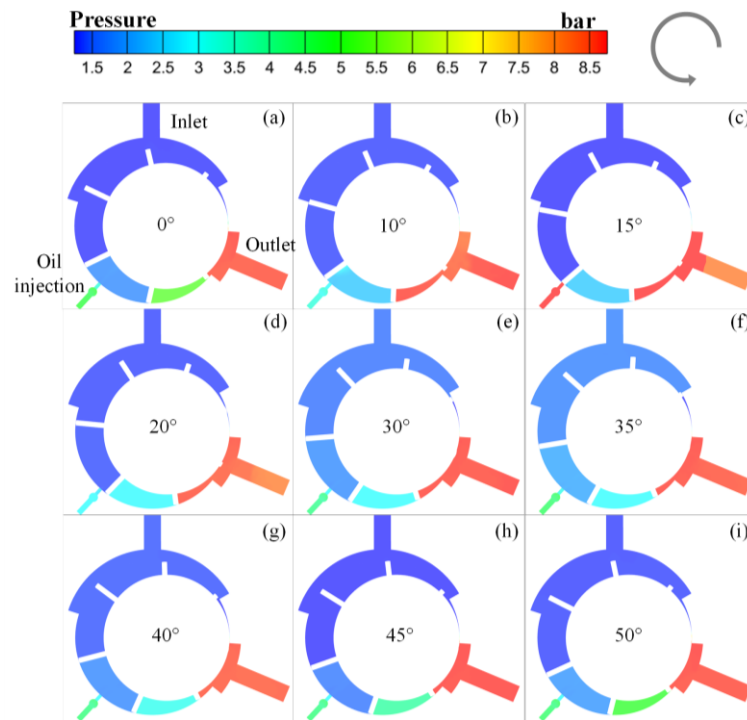


Fig. 6. Variation of pressure field with rotational angle.

3.1.2. Oil distribution

[Fig. 7](#) shows the variation in oil phase distribution over one cycle. Lubricant oil radially flows into the chamber at 228° and mixes with air for cooling. The oil spreads along the leading side of the blade and tends to accumulate on the walls of the rotor grids, i.e., on the stator inner surface, rotor outer surface, blade side surface due to the high viscosity of the oil. This local accumulation of the oil hinders the mixing between oil and air, and thus affect the cooling of the oil to the air. After the oil accumulates at the leading edge of the blade with the rotor revolution, it flows into the gaps between the blade tip and stator, which helps to prevent the air leakage between adjacent chambers and consequently ensure reliable internal sealing performance. Additionally, the oil accumulation at the clearance gaps helps to enhance the lubrication between relative moving surfaces, i.e., between the blade tip and the stator inner wall. As the blade passes through the injection port, the narrow blade tip clearance limits the entry of the oil, leading to a decrease of the oil flow rate, as shown in the bottom left of the in [Fig. 7\(b\)](#). The mixing between the oil and the air tends to be more uniform at the exhaust stage, leading to a good cooling performance. Finally, most of the oil is discharged from the exhaust port, while the remaining small part leaks into the next cycle through the tangential clearance gap.

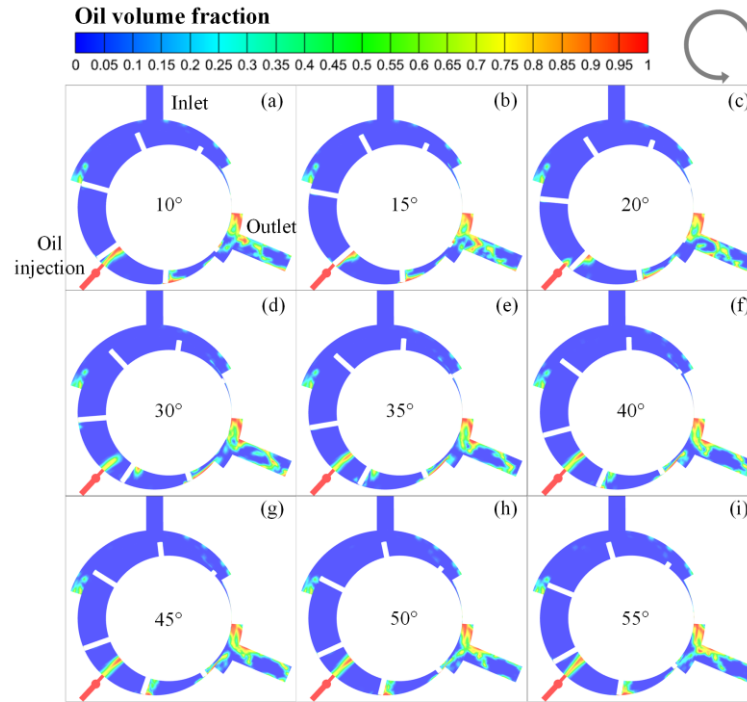


Fig. 7. Variation of oil volume fraction with rotational angle.

The distribution of the oil phase in the blade tip clearance significantly affects local leakage and lubrication. [Fig.8](#) shows the oil distribution in the blade tip clearance varying with rotational angles. A low oil phase volume fraction near the blade tip is present at 30°, which is due to the fact that the oil-gas mixture leaks from the exhaust to the intake through the tangential clearance gap. At 60°, 90°, 120° and 150°, as the blade passes through the intake region and does not form a blade tip gap, almost no oil accumulation is observed in the blade tip. While at 180° and 210°, the formation of the blade tip clearance results in oil accumulation due to the viscosity difference between the oil and the air, leading to effective sealing. After the oil is injected into the chamber at 228°, it spreads into the inner surface of the rotor and accumulates here, leading to lower oil volume fraction in the blade tip at 240°, as shown in [Fig. 8\(h\)](#). As the blade rotates, oil accumulates at the blade leading edge and flows towards the blade tip, resulting in significant oil accumulation in the blade tip gap at angles of 270° and 300°. As a result, the oil is filled with the blade tip clearance and a good sealing performance is achieved. The blade tip clearance disappears at 330° in the exhaust stage, thus the oil volume fraction decreases near the blade tip. At 360°, a significant oil phase distribution is observed in the tangential clearance region, which helps to reduce the tangential leakage of the air.

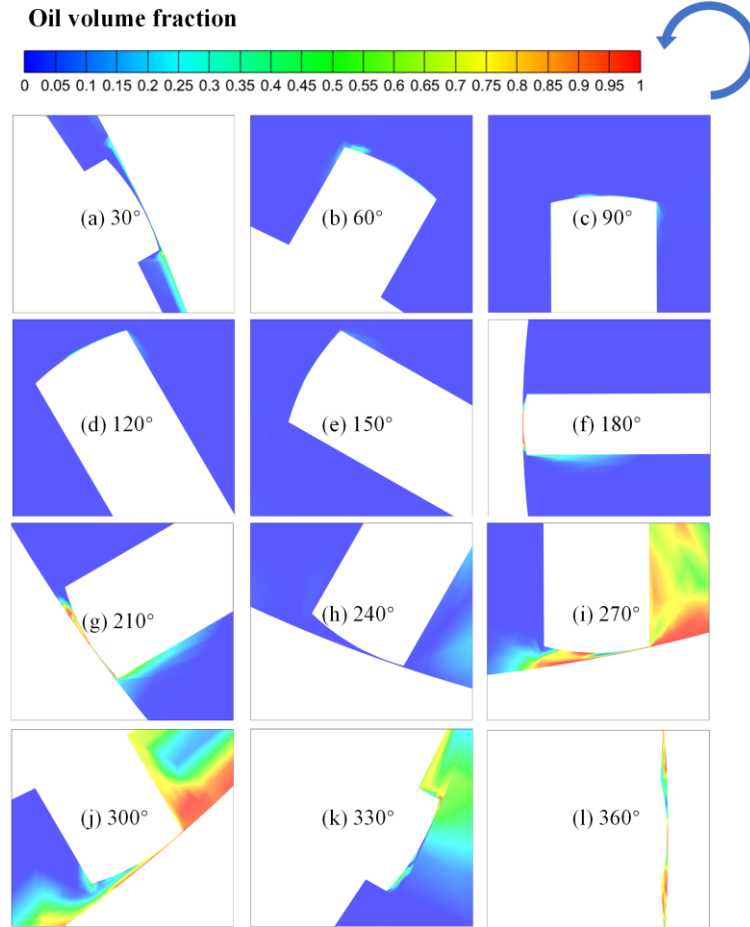


Fig. 8. Oil phase distribution in the blade tip region.

3.1.3. Temperature field

[Fig. 9](#) illustrates the temperature variation within a cycle. During the intake process, the air at a temperature of 298.6 K flows into the compressor chamber through the intake port. The volume of the chamber decreases as the compression proceeds, converting mechanical energy into the internal energy of the air and thus raising the temperature of the air. At 228°, the oil flows into the chamber through the injection port for the cooling of the air, leading to a decrease of the temperature. It can be seen from [Fig. 9](#) that the temperature distribution of the chamber is not uniform during the compression stage. The temperature in the leading side of the blade is lower than that in the core due to the accumulation of oil in the leading side of the blade. The chamber volume reaches its minimum at the end of the compression stage, leading to the appearance of the high temperature region. The maximum temperature in the chamber is even above 500 K, as showed in [Figs. 9\(f\) and 9\(i\)](#). After that, the oil and air mix evenly at the exhaust port, leading to more uniform temperature filed. High-temperature air at the exhaust port leaks through the tangential clearance into the intake port, which results in a slight increase in the temperature at

the intake port for the next cycle.

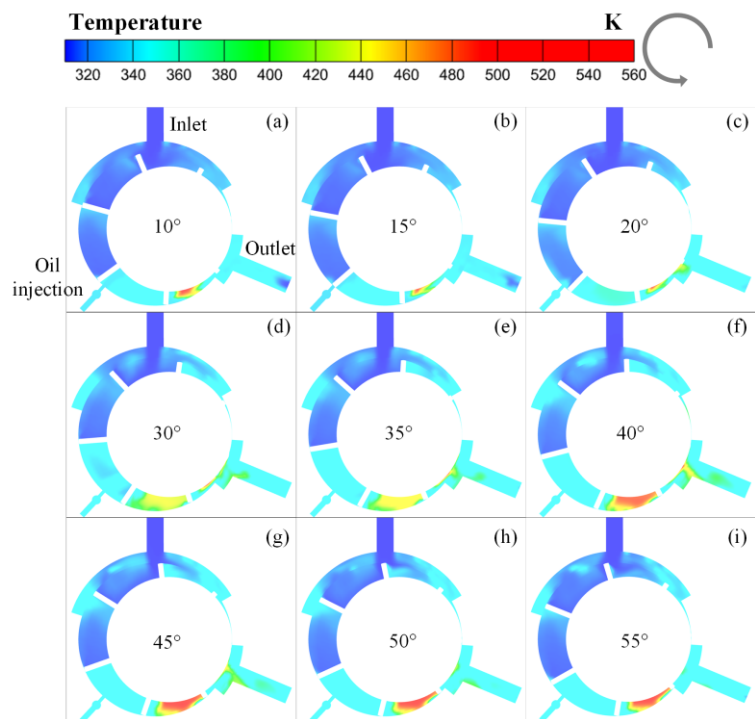


Fig. 9. Variation of temperature field with rotational angle.

3.1.4. Velocity field

[Fig. 10](#) displays the velocity contour with superimposed streamlines. As the fluid flows into or out of the chamber, the sudden change in the cross-sectional area leads to unsteady flow characteristics and the formation of multiple complex vortex structures. While in the straight pipe section of the intake and exhaust ports, the flow field is relatively uniform. During the compression stage, the blade tip clearance induces a backflow driven by a reverse pressure gradient which is opposite to the counterclockwise rotational speed. The leakage flow in the blade tip causes a recirculation region in the downstream chamber, as shown at the bottom left in [Fig. 10\(d\)](#). It can be seen from [Figs. 10\(e\)-\(g\)](#) that the recirculation region evolves from one to two influenced by the oil injection flow. This phenomenon contributes to enhance the heat transfer between the compressing air and the oil that, after the radial injection, spreads along the leading side of the blade and mixes with air.

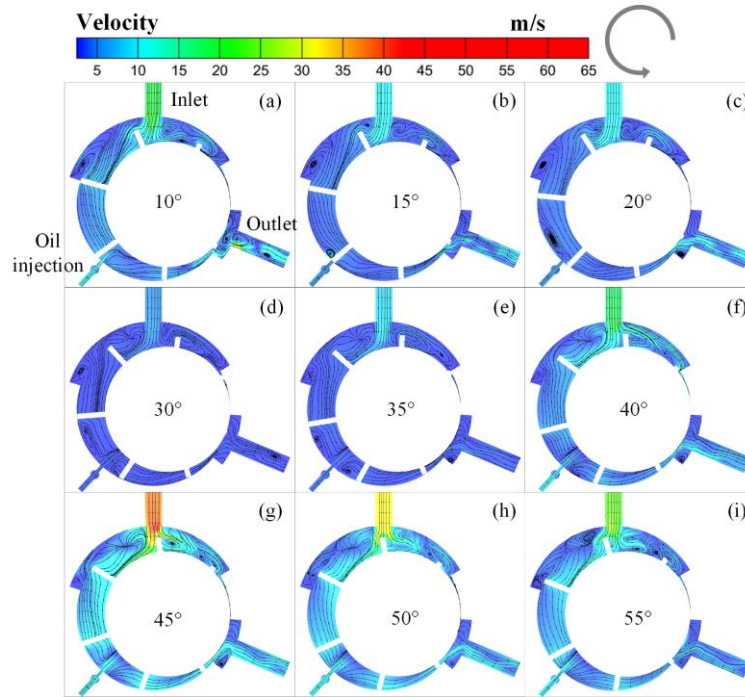


Fig. 10. Variation of velocity with rotational angle.

3.2. Compressor performance analysis

Fig. 11 shows the angular evolution process of the instantaneous mass flow rate, the rotor torque and the average cell pressure of the SVRC. As showed in Fig. 11(a), the mass flow rates at the intake port remain stable at positive values, while the ones at the exhaust port fluctuates significantly throughout the cycle. The mass flow rate at the exhaust port increases sharply from negative to positive values before reaching its maximum at 10°, indicating the occurrence of backflow. It subsequently falls back to negative values, followed by a slight oscillation, at which point the backflow disappears. The amplitude of the rotor torque increases due to the backflow phenomenon and then falls back to near the initial value after the backflow disappears, as illustrated in Fig. 11(b). As displayed in Fig. 11(c), the average pressure of the chamber remains stable throughout the intake stage and then increases exponentially during the compression stage. At the end of the compression stage, the average cell pressure reaches a maximum of approximately 9.5 bar which is higher than the set pressure at the exhaust port, indicating the presence of over-compression in the chamber. When the chamber connects to the exhaust port, the cell pressure value decreases slightly to the exhaust pressure. Some fluctuations can be then noticed due to the small geometrical simplification of the blade side walls according to the analytical grid generation method (Ye et al., 2021).

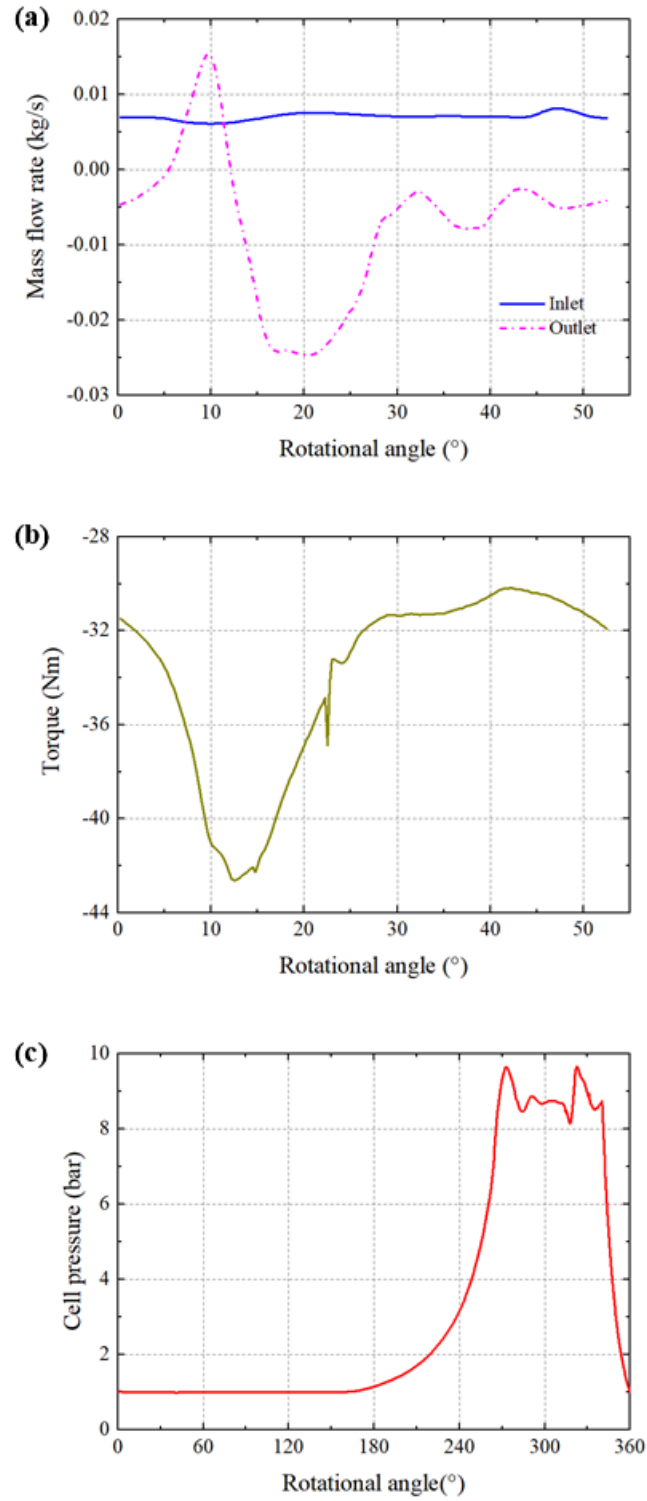


Fig. 11. Angular evolution of (a) inlet and outlet mass flow rates, (b) rotor torque, (c) average cell pressure.

In this work, the volumetric efficiency, the adiabatic efficiency, the exhaust temperature and the specific power are utilized to evaluate the performance of the SVRC. The volumetric efficiency is defined as the ratio of the measured mass flow rate to the theoretical one of the compressor, where the measured mass flow rate refers to

the simulated result here. The adiabatic efficiency is defined as the ratio of the theoretical work based on the ideal adiabatic compression process (hereinafter referred to as adiabatic work) to the actual work consumed by the compressor, where the actual work refers to the simulated result here. And the specific power is defined as the power consumption per unit volumetric flow rate, where the power consumption refers to the simulated result here. According the above definitions, Eqs. (7)-(11) are used to obtain the volumetric efficiency, the adiabatic efficiency and the specific power. According to the simulated results of the reference case, the SVRC achieves the volumetric efficiency of 95.1%, the adiabatic efficiency of 60.3%, the exhaust temperature of 337.4 K and the specific power of 9.86 kW/(m³·min⁻¹).

$$\eta_{vol} = \frac{Q_{vol}}{Q_{th}} \quad (7)$$

$$\eta_{ad} = \frac{W_{ad}}{W_{comp}} \quad (8)$$

$$W_{ad} = \frac{\gamma-1}{\gamma} p_{in} \cdot 60 \cdot Q_v \left[\left(\frac{p_{out}}{p_{in}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (9)$$

$$W_{comp} = \frac{\tau \times n}{60} \quad (10)$$

$$P_{spe} = \frac{W_{comp}}{Q_{vol}} \quad (11)$$

where η_{vol} is volumetric efficiency, η_{ad} is adiabatic efficiency, W_{ad} is adiabatic work, W_{comp} is actual work, P_{spe} is specific power, γ is specific heat ratio of air, p_i is intake pressure, p_o is exhaust pressure, Q_{vol} is volumetric flow rate, τ is torque, n is rotational speed, Q_{th} is theoretical volumetric flow rate.

3.3. Comparison with market compressor

The specific power and adiabatic efficiency are two representative indexes to evaluate the compressor performance. A comparison of specific power varying with compression ratio between in existing technology (Vittorini et al., 2015) and in this work is shown in Fig. 12. It can be seen from Fig. 12 that both air cooled premium compressors in existing technology and the oil-injected compressors in this work wholly increases with the compression ratio. However, the specific power obtained in this work is higher than that in market compressors. One reason is that oil jet form instead of oil spray form is used for air cooling in this work, leading to higher power consumption. Another reason is the differences in cooling methods. Specifically, air cooled approach belongs to external cooling type, while oil-injected method is an internal cooling type. As a whole, the variation trend of the specific power as a function of the compression ratio is consistent, which

further validate the reliability of the present study.

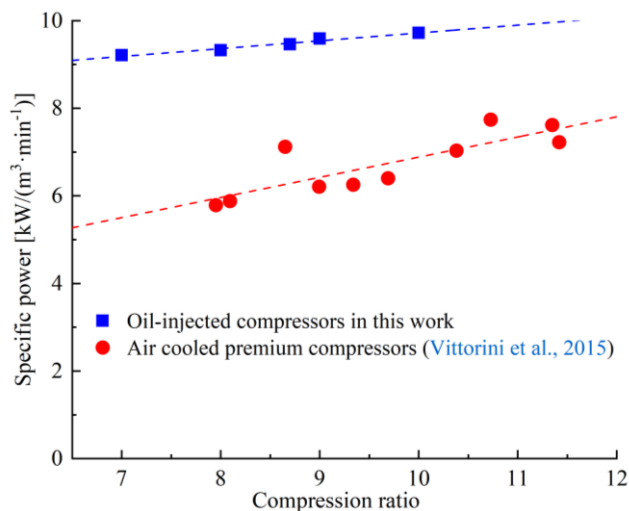


Fig. 12. Specific power varying with compression ratio.

The isentropic efficiency of market compressors is reported as a function of air flow rate (log scale) (Vittorini and Cipollone, 2016a), as shown Fig. 13. This isentropic efficiency refers to the average value of class SVRCs measured with delivery pressure of 9 bar. It can be observed from Fig. 13 that the isentropic efficiency of market compressors increases with air flow rate, and a fitted line is obtained. Although the air flow rate of the reference case in this work is smaller than that of market compressors reported in Ref. (Vittorini and Cipollone, 2016a), the isentropic efficiency obtained in this work aligns well with the fitted line. This indicates that the efficiency predicted in the present study is consistent with real data of market compressors.

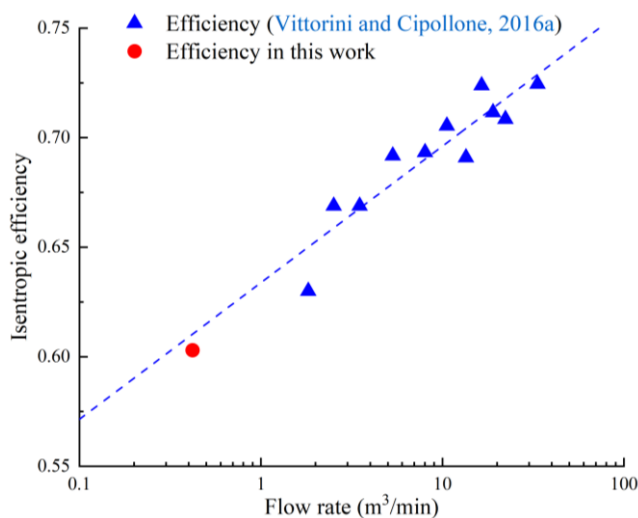


Fig. 13. Isentropic efficiency varying with air flow rate.

4. Effects of oil injection parameters on compressor performance

The oil injected into the chamber of the SVRC significantly affects its performance. In this section, the effects of the oil to gas mass ratio, oil injection angle, oil injection temperature and rotational speed on the volumetric efficiency, adiabatic efficiency, exhaust temperature and specific power are investigated.

4.1. Effects of oil to gas ratio

Five simulation cases with different oil to gas mass ratios (5, 10, 15, 20 and 25) have been carried out to identify the influences on the performance of the SVRC. Fig. 14 illustrates the effects of the ratio on the volumetric efficiency, the adiabatic efficiency, the exhaust temperature and the specific power. It can be seen from the blue line that the volumetric efficiency markedly increases from 90.1% to 97.2% with the ratio from 5 to 25. This is because higher ratio results in a greater amount of oil in the blade tip clearance during the compression stage, leading to lower leakage between adjacent chambers. As shown in the pink line, the adiabatic efficiency increases with the ratio from 5 to 15 and then decreases with the ratio from 15 to 25. This is because higher ratio increases the adiabatic work due to the improved sealing effect and the actual work due to the more power required to handle the additional oil at the same time. Due to the effects of the ratio on both the adiabatic work and the actual work, the adiabatic efficiency reaches its peak around 61.9% at the ratio of 15. It is expected that increasing the ratio allows to enhance the heat transfer between the air and the oil, leading to lower exhaust temperature. Actually, the exhaust temperature decreases with the ratio from 5 to 20 and then shows a slight increasing trend with the ratio from 20 to 25, as indicated by the black line. The suitable ratios (e.g., 15 or 20) allow effective cooling for the air since the specific heat capacity of oil is much higher than that of air, while the larger ratio (25) may be detrimental to the mixing between the oil and the air. It can be seen from the red line that the specific power defined by the ratio of the adiabatic work to the actual work decreases from 10.06 to 9.59 kW/(m³·min⁻¹) with the ratio from 5 to 10 and then increases from 9.59 to 9.94 kW/(m³·min⁻¹) with the ratio from 10 to 25. Higher ratio helps to improve sealing effect and thus increases the air volumetric capacity, while more oil increases the actual work consumption. Due to the influences of the ratio on both the air volumetric capacity and the actual work, the specific work reaches its bottom about 9.59 kW/(m³·min⁻¹) at the ratio of 10. Comprehensively considering the effects of the oil to gas

ratio on the compressor performance, the choice of 10 is appropriate for the ratio of the present SVRC.

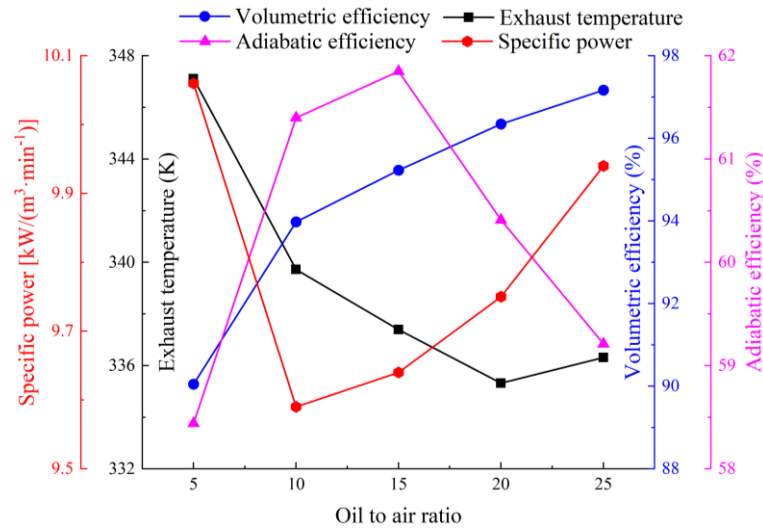


Fig. 14. Influences of oil to gas ratio on compressor performance.

4.2. Effects of oil injection angle

The oil injection angle is a vital parameter that influence the compressor performance. Fig. 15 displays the effects of the injection angle on the volumetric efficiency, the adiabatic efficiency, the exhaust temperature and the specific power. It can be seen from the blue line that the volumetric efficiency slightly decreases with the injection angle from 220° to 228° and then sharply increases with the injection angle from 228° to 236°, reaching its bottom (95.2%) at the angle of 228°. With smaller injection angles, the oil is injected into the chamber earlier at the compression stage, which helps to extend the time for the oil sealing in the blade tip and thus improve the volumetric efficiency. While with larger injection angles, more oil is distributed in the blade tip clearance at the later compression stage where serious blade tip leakage occurs due to the high pressure difference between adjacent chambers. Consequently, increasing the injection angle from 228° to 236° leads to a notable improvement in the volumetric efficiency from 95.2% to 97.7%. About 8° shift in oil injection angle (from 228° to 236°) resulted in 2.5% gain in volumetric efficiency. The adiabatic efficiency reported by the pink line slightly increases with the angle from 220° to 236°. This is due to the fact that delaying the angle reduces the residence time of the oil in the compression stage, which helps to decrease actual power consumption and thus increase adiabatic efficiency. The exhaust temperature reported by the black line remains almost unchanged as the angle increases from 220° to 236°. This is because changing the injection angle has little effect on the state of the oil which is still effectively mixed

with the air to achieve good cooling performance. As can be seen from the red line, the specific power increases from 9.69 to 9.86 kW/(m³·min⁻¹) with the injection angle from 220° to 228° and then decreases from 9.86 to 9.46 kW/(m³·min⁻¹) with the injection angle from 228° to 236°. The variation trend of specific power is just opposite to that of volumetric efficiency. About 8° shift in oil injection angle (from 228° to 236°) leads to 4.2% decrease in specific power. Comprehensively considering the effects of the oil injection angle, a delayed angle benefits overall compressor performance. Within the range of the calculations in this paper, an angle of 236° is recommended.

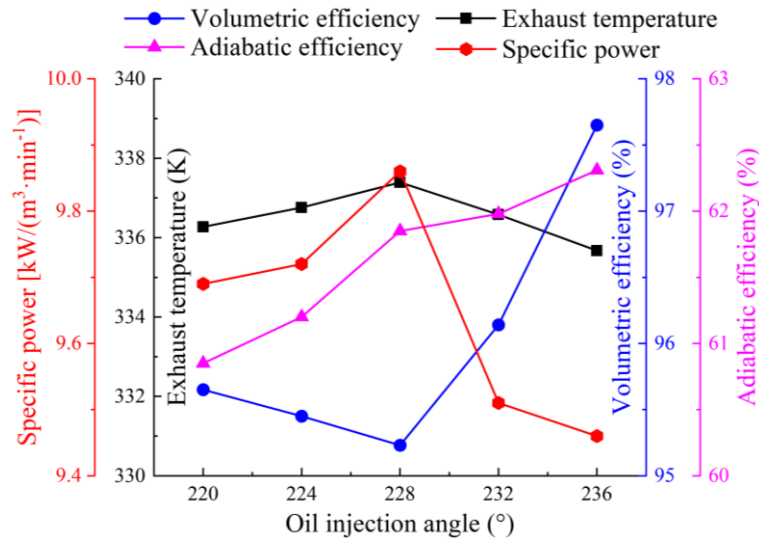


Fig. 15. Influences of oil injection angle on compressor performance.

4.3. Effects of oil injection temperature

Fig. 16 describes the influences of the oil injection temperature on the adiabatic efficiency, the volumetric efficiency, the exhaust temperature and the specific power. Both the volumetric efficiency (reported in the blue line) and the adiabatic efficiency (reported in the pink line) decrease with the oil injection temperature, while the exhaust temperature (reported by black line) and the shaft power (reported in the red line) increases with the oil injection temperature. Higher oil injection temperature decreases the temperature difference between the oil and the air, which reduces the cooling ability of the oil to the air and consequently increase the exhaust temperature. The rise of temperature in the chamber results in higher pressure values as well as high pressure difference. The reduction of the volumetric efficiency is attributed to the increasing leakage in the blade tip clearance due to higher pressure difference. High oil temperature decreases the adiabatic work due to the weakened sealing effect, leading to a reduction of the adiabatic efficiency. And higher oil temperature decreases the air volumetric capacity, resulting

in an increase of the specific power. From the above results, reducing oil temperature helps to significantly improve the compressor performance. However, the oil temperature cannot be reduced without limit since oil is recycled in compressed air systems.

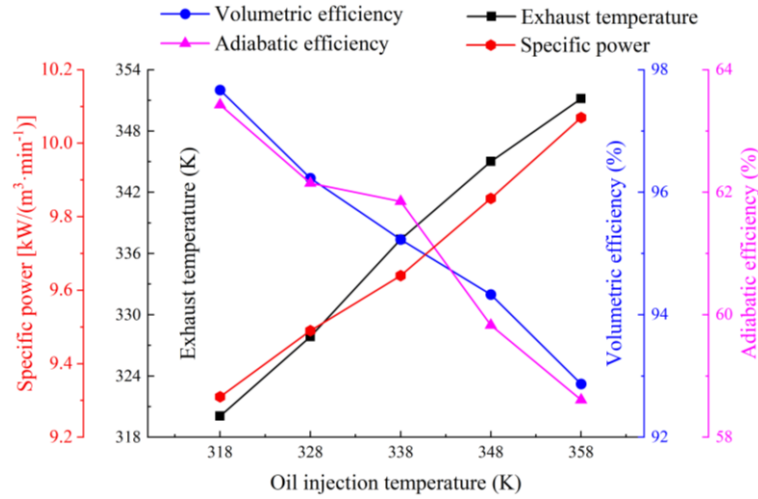


Fig. 16. Influences of oil temperature on compressor performance.

4.4. Effects of rotational speed

Fig. 17 demonstrates the impact of rotational speed on the volumetric efficiency, adiabatic efficiency, exhaust temperature and specific power. It can be seen from the blue line that the volumetric efficiency continuously decreases from 95.2% to 89.4% with the rotational speed from 1000 to 3000 RPM. This is because the supply of lubricating oil becomes more insufficient with the increase of the rotational speed, resulting in serious leakage in the blade tip during the compression stage. Higher leakage at higher rotational speed results in lower adiabatic work. Thus, the adiabatic efficiency (reported by the pink line) declines with the rotational speed. In terms of the exhaust temperature (reported by the black line), as the rotational speed increases from 1000 to 2000 RPM, the decreasing residence time for the gas cooling leads to an increase in the exhaust temperature from 337.4 to 340.5 K; while as the rotational speed increases from 2000 to 3000 RPM, the increasing leakage may result in incomplete compression of part of gas, leading to a slight decrease in the exhaust temperature from 340.5 to 338.8 K. The specific power (reported by the red line) increases from 9.46 to 15.89 kW/(m³·min⁻¹) with the rotational speed from 1000 to 3000 RPM. This is because the shaft power almost linearly increases with the rotational speed, while the air volumetric capacity slowly increases with the rotational speed. Based on the above analysis, the lubricating

oil supply is insufficient under the high rotational speed condition, which leads to the reduction of compressor performance. Therefore, the mass flow rate of circulating lubricating oil should be increased accordingly to the increasing rotational speed.

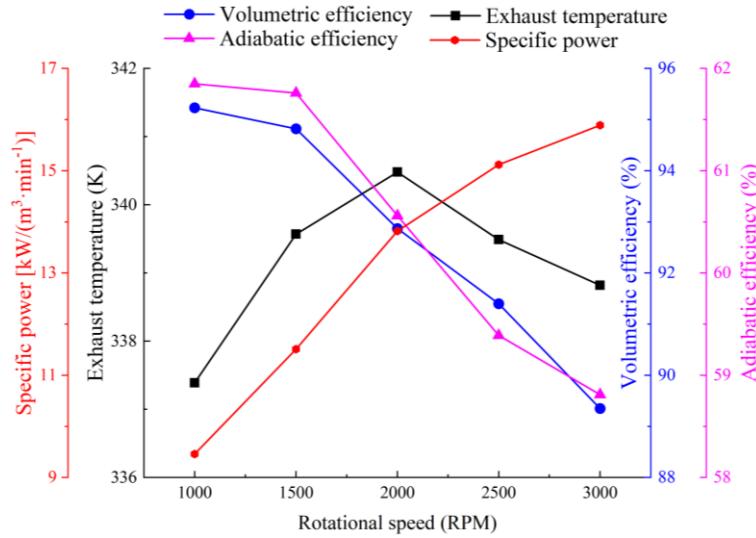


Fig. 17. Influences of rotational speed on compressor performance.

5. Conclusions

This study carried out CFD investigations on an oil-injected sliding vane rotary air compressor. The oil-gas two-phase flow characteristics and the effects of the oil injection parameters on the compressor performance have been obtained. The oil injected into the chamber accumulates at the blade leading edge and then flows into the blade tip clearance, which helps to reduce the blade tip leakage and enhance lubrication between moving surfaces. The mixing between the oil and the air tends to be more uniform as the compression process proceeds, leading to the cooling effect of the oil to the air. The mass flow rate at the intake port is uniform, while the one at the exhaust port presents backflow which leads to torque fluctuations. The cell pressure trace shows a slight over compression phenomenon at the end of the compression stage and the minor pressure fluctuations at the exhaust stage. For the reference case, the compressor achieves the volumetric efficiency of 95.1%, the adiabatic efficiency of 60.3%, the exhaust temperature of 337.4 K and the specific power of 9.86 kW/(m³·min⁻¹).

Increasing oil to gas ratio from 5 to 25 helps to improve volumetric efficiency and reduce exhaust temperature, while the adiabatic efficiency reaches its peak at the ratio of 15, and the specific power reaches its bottom at the ratio of 10. The ratio of 10 is selected for the present work. An increase in the oil injection angle from 220° to 236°

helps to improve the adiabatic efficiency and has little effect on exhaust temperature. About 8° shift in oil injection angle (from 228° to 236°) resulted in 2.5% gain in volumetric efficiency and 4.2% decrease in specific power. From the results, an injection angle of 236° is recommended to improve compressor performance. Both the volumetric and adiabatic efficiencies decrease with the oil injection temperature from 318.2 to 358.2 K, while the exhaust temperature and the specific power increases with the oil injection temperature. The oil temperature should be kept as low as possible within the allowable operating conditions of compressed air systems. As a whole, the oil to gas mass ratio has higher sensitivity than the oil injection angle or temperature. Increasing the rotational speed from 1000 to 3000 RPM leads to a decrease in both the volumetric and adiabatic efficiencies and an increase in the specific power, and the exhaust temperature reaches its peak at the rotational speed of 2000 RPM. The lubricating oil capacity should be increased accordingly to the increasing rotational speed.

CrediT author statement

Yuande Dai: Methodology, Software, Formal analysis, Writing-original draft, Visualisation; **Huiyang Zhu:** Methodology, Software, Formal analysis, Writing-original draft, Visualisation; **Giuseppe Bianchi:** Formal analysis, Writing-review and editing, Supervision; **Sham Rane:** Methodology, Software, Formal analysis, Writing-review and editing, Supervision; **Fanghua Ye:** Conceptualisation, Methodology, Software, Formal analysis, Writing-original draft, Writing-review and editing, Supervision, Funding acquisition, Project administration.

Declaration of interest statement

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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