Use of CFD to Investigate Flow Characteristics and Oil Distribution Inside an Oil-injected Screw Compressor

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Abstract. The well-developed CAE tools are used to investigate flow characteristics and oil distribution inside an oil-injected screw compressor. The flow field of gas mixed with oil is calculated by ANSYS CFX. TwinMesh generates high quality hexahedral grids for the compression chambers. There are two cases studied in this study. In the first case, the compressor models under four loading conditions are analyzed. The pressure curve is used to see if the designed built-in volume index is appropriate. In the second case, the compressor models under oil-free and oil-injected conditions are analyzed. The pressure gradients on rotor surfaces and velocity vector fields are used to describe the effects of lubrication and sealing in the sealing line gap, the tip gap, and the outlet end gap of rotors. The designer could optimize the slider and know if the oil is properly distributed in each gap.

1. INTRODUCTION

The oil-injected screw compressor is a positive displacement compressor. The gas in the compressor is compressed by compression chambers. The compression chambers are formed by the lobes of rotors and the case, and deform periodically. With oil injection, the compressor could efficiently operate under safe temperature for a long time. A chiller system with an oil-injected screw compressor would operate under different loading conditions. The power consumption of the compressor is mainly caused by the gas torque, which comes from non-uniform pressure distribution on rotor surface. The compressor should have a slider to regulate the inlet flowrate and volume index to avoid over or insufficient compression to reduce power wastage. The designer needs to consider lots of design parameters to predict performance of an oil-injected screw compressor. With the help of CAE tools, the designer could do the parametric study and optimization accurately and efficiently. The time and cost of repeated tests with prototypes could be greatly reduced.

Thanks to the well-developed CAE tools, it is now possible to get reliable results of volumetric efficiency, pressure curves, gas forces, gas torques, and oil distribution of an oil-injected screw compressor [1-3]. Rane et al. [4] introduced how to perform the CFD calculation of an oil-injected screw compressor in detail. The multiphase flow was considered with thermal and sealing effects. They showed the comparisons of performances and pressure curves between different operating conditions. They also described the oil distribution, and did the analysis of outlet temperature. Basha et al. [5] compared the calculated results between inhomogeneous and homogeneous Eulerian-Eulerian multiphase models. Based on inhomogeneous model, the calculation takes more time to get more accurate results, and the calculated oil distribution in the tip gaps is more in line with expectations than...
the one based on homogeneous model. Ding et al. [6] did the multiphase flow calculation with VOF model. They compared the results between oil-free and oil-injected conditions. Randi et al. [7] traced the oil drops with Lagrangian approach. The momentum and energy transfers between oil drops and gas were considered by two-way coupling model. The oil-drops contact walls and become oil film were described by impingement model and fluid-film (FF) model. They showed that the smaller the oil drops are, the more probabilities are for oil drops flying to actual working chamber without impinging on the walls. They also showed the distribution and thickness of oil film on walls of chambers in detail.

In this study, there are two cases studied by using ANSYS CFX and TwinMesh. In the first case, the compressor models under four loading conditions are analyzed. The pressure curve is used to see if the designed built-in volume index is appropriate. In the second case, the compressor models under oil-free and oil-injected conditions are analyzed. The pressure gradients on rotor surfaces and velocity vector fields are used to describe the effects of lubrication and sealing in the sealing line gap, the tip gap, and the outlet end gap of rotors. The designer could optimize the slider and know if the oil is properly distributed in each gap.

2. Theoretical Model and Case Study
Performance of an oil-injected screw compressor is analyzed in this study. A pair of “5x6” rotors are in the compressor. The displacement of this compressor is 260 m³/hr at 3,000 rpm. In order to reduce the physical calculation time, the simplified geometry is adapted and is shown in figure 1. The inlet channel, case, by-pass path, compression chambers, outlet channel, and oil injection channel are considered in the simplified geometry. Figure 2 shows the mesh grids and fluid domains. The dynamic mesh grids of compression chambers are generated by TwinMesh. The size of tip gap between rotor tip and the case is 0.08 mm. The minimum size of sealing line gap between rotors is 0.04 mm, and the maximum one is 0.16 mm. The size of inlet end gap between the end face of rotor and the low pressure end plate is 0.25 mm. The size of outlet end gap between the end face of rotor and the high pressure end plate is 0.05 mm. The element number of dynamic mesh is about 3,000,000. The dynamic mesh qualities are the minimum angle of 29.83, the maximum aspect ratio of 703.44, the maximum volume change of 5.028, and the minimum determinant of 0.20. Figure 3 shows three different cross sections of dynamic mesh grids of compression chambers. The static mesh grids of the inlet channel, case, by-pass path, outlet channel, and oil injection channel are generated by ANSYS Meshing. The element number of static mesh is about 1,300,000.

![Figure 1. The simplified geometry. (D1) is the inlet channel. (D2) is the case and the by-pass path. (D3) is the compression chamber. (D4) is the outlet channel. (D5) is the oil injection channel.](image1)

![Figure 2. Mesh grids and fluid domains. (D1) is the domain of inlet channel. (D2) is the domain of case and by-pass path. (D3) is the domain of compression chamber. (D4) is the domain of outlet channel. (D5) is the domain of oil injection channel.](image2)
The calculation is done by ANSYS CFX. The selections of CFD models and schemes are listed in Table 1. In order to choose the multiphase model, the particulate loading and Stokes number are calculated by equations (1) and (2) respectively, and are listed in Table 2. The inhomogeneous Eulerian–Eulerian multiphase model is recommended to be used. In this study, the homogeneous model is used for the shorter physical calculation time.

There are two cases studied in this study. The first case is the analysis of compressor models under four loading conditions. Figure 4 shows these four compressor models. The inlet and outlet gas pressure and temperature are listed in Table 3. The rotation speed of male rotor is 3,600 rpm. Oil is injected from the side of male rotor into the chamber, where is labeled 2 in figure 4. The oil-injected flowrate is 0.23 kg/s. The model under full loading condition is with the largest radial outlet path of all and without the additional by-pass path. The model under 75% loading

Table 1. The selections of CFD models and schemes

| Models and Schemes | Selection |
|--------------------|-----------|
| Turbulence model   | SST       |
| Multiphase model   | Homogeneous Eulerian–Eulerian Model |
| Advection Scheme   | High Resolution |
| Turbulence Scheme  | First Order Upwind |
| Transient Scheme   | Second Order Backward Euler |

Particulate loading \( \beta = \frac{V_{oil} \cdot \rho_{oil}}{V_{gas} \cdot \rho_{gas}} \)  
Stokes number \( t = \frac{t \cdot (U \cdot L)}{18 \cdot \mu_{gas}} \)

Table 2. The particulate loading and Stokes number.

| Parameter                | Unit   | Value   |
|--------------------------|--------|---------|
| Particulate Loading (\( \beta \)) | --     | 0.16    |
| Stokes number            | --     | 35.8    |
| Inlet volumetric flowrate of oil (\( V_{oil} \)) | m3/s   | 2.48x10^{-4} |
| Inlet volumetric flowrate of gas (\( V_{gas} \)) | m3/s   | 0.087   |
| Density of oil (\( \rho_{oil} \)) | kg/m3  | 927     |
| Density of gas (\( \rho_{gas} \)) | kg/m3  | 16.54   |
| Particle diameter (\( d_{oil} \)) | \( \mu \)m | 100     |
| Dynamic viscosity of gas (\( \mu_{gas} \)) | kg/m-s | 11.33x10^{-6} |
| Relaxation time of the particle (\( t \)) | s      | 0.0455  |
| Fluid velocity (\( U \)) | m/s    | 25.2    |
| Characteristic length (\( L \)) | m      | 0.032   |
condition is with the small radial outlet path and with the small additional by-pass path. The model under 50% loading condition is without the radial outlet path and with the medium-sized additional by-pass path. The model under 25% loading condition is without the radial outlet path and with the largest additional by-pass path of all.

The second case is the comparison of flow in each leakage gap under full loading condition, listed in table 3, between oil-free and oil-injected conditions. The inlet pressure is 0.35 MPa. The outlet pressure is 0.96 MPa. The inlet temperature is 15 °C. The outlet temperature is 58 °C. Figure 5 shows the compressor model used in the second case. The rotation speed of male rotor is 3,600 rpm. Oil is injected from both sides of male and female rotors into the chamber, where are labeled 2 in figure 5. In order to easily and clearly observe the oil distribution, the oil-injected flow rate in this case is 2.3 kg/s.

These models were solved by the 26 nodes parallel calculation with Intel Xeon E5-2699 @ 2.30 GHz and 256 GB RAM. Each calculation took about 260 hours physical time to solve 2880 time steps, which are 8 rotations of male rotor. The calculation time step size is $4.63 \times 10^{-5}$. The convergence criterion is $10^{-3} \text{MAX Residual Level}$.

### Table 3. The inlet and outlet gas pressure and temperature under four loading conditions

| Loading | Pressure [MPa] | Temperature [°C] |
|---------|---------------|-----------------|
|         | Inlet | Outlet | Inlet | Outlet |
| Full    | 0.35  | 0.96   | 15    | 58     |
| 75%     | 0.35  | 0.75   | 15    | 49     |
| 50%     | 0.35  | 0.59   | 15    | 43     |
| 25%     | 0.35  | 0.59   | 15    | 44     |

![Figure 4](image1.png)

**Figure 4.** The compressor models under four loading conditions are used in the first case. (a) is under full loading condition. The by-pass path (i) is not connected to the compression chamber (ii). (b) is under 75% loading condition. (c) is under 50% loading condition. (d) is under 25% loading condition. Under 75%, 50%, and 25% loading conditions, the by-pass path (i) is connected to the compression chamber (ii) by the additional by-pass path (iii).

![Figure 5](image2.png)

**Figure 5.** The compressor model under full loading condition is used in the second case.
3. Result and Discussion

3.1. The first Case: The performance analysis of compressor under four loading conditions

The calculated volumetric efficiencies and gas powers under four loading conditions are listed in Table 4. Figure 6 shows the calculated pressure curves. The pressure in the lobe of male rotor is probed near the front flank of male tooth, and the pressure in the lobe of female rotor is probed near the root of female tooth. The probing points are close to the outlet ends of rotors. The built-in volume index of this compressor is designed for the full loading condition. Under the full loading condition, there is a small over-compression area during the start section of outlet. Under the 75% loading condition, the compression is started at about 100 degree. Before this angle, some gas in chamber is pushed back to the inlet channel through the by-pass path. The gas pressure in chamber is a little higher than the inlet pressure. For the mismatch of volume index, the over-compression area under the 75% loading condition is larger than the one under the full loading condition. Under the 50% and 25% loading conditions, the operating conditions are the same. The volume index is about 2.2 under the 50% loading condition, and the one is about 1.8 under the 25% loading condition. Under the 50% loading condition, the largest over-compression area of all appears. In the middle and end sections of outlet, gas pressure in the chamber is lower than the back pressure. The inertial force of gas in chamber is large enough to overcome the pressure difference. Gas could continue to flow out of the chamber. Under the 25% loading condition, the over-compression area is smaller than the one under the 50% loading condition. In the end section of outlet, the gas pressure in chamber is higher than the back pressure. This shows that gas needs some driving force generated by pressure difference to flow out of the chamber.

![Figure 6](image)

**Figure 6.** The calculated pressure curves under four loading conditions. (a) is under full loading condition. (b) is under 75% loading condition. (c) is under 50% loading condition. (d) is under 25% loading condition.

| Loading | Volumetric efficiency | Gas power [kW] | Mass imbalance for mixed phase | Mass imbalance for gas phase | Mass imbalance for oil phase |
|---------|-----------------------|----------------|--------------------------------|----------------------------|-----------------------------|
| Full    | 91.6% (93.7%*)        | 36.8 (40.0*)   | 7.81%                          | 17.32%                     | -24.31%                     |
| 75%     | 73.8%                 | 22.5           | 2.93%                          | 5.48%                      | -6.29%                      |
| 50%     | 61.8%                 | 13.3           | 2.22%                          | 7.99%                      | -13.32%                     |
| 25%     | 26.2%                 | 6.7            | -0.37%                         | -5.83%                     | 11.06%                      |

*Measured volumetric efficiency and power consumption under full loading condition
3.2. The second Case

In this case, the characteristic of flow in each leakage gap and the pressure gradient on rotor surface are presented. The designer could know if the oil is properly distributed in each gap by considering the effects of lubrication and sealing together. It is ideal to achieve sufficient lubrication in contact region of rotors, and to have proper sealing effect with less oil friction in the other gaps.

3.2.1. The characteristic of flow in each leakage gap. The flows in the sealing line gap, the tip gap, and the outlet end gap of rotors under full loading condition between oil-free and oil-injected conditions are analyzed with the model shown in figure 5. Although the assumptions of Couette flow are not perfectly applied to the flow inside the compressor, the characteristic and trend of velocity variations of Couette flow are used to illustrate the flows, which are affected by oil, in the three leakage gaps [8]. Figure 7 shows velocity profiles of two Couette flows. The sliding plates are at the location of 0.1 mm. The speeds of sliding plates are the relative speed between rotors and the tip-speed of rotor. The fixed plates are at the location of 0 mm. The fluid is gas mixed with oil. The viscosity of gas is 1e-5 Pa-s, and the one of oil is 0.0258 Pa-s. The OVF is the abbreviation of oil volume fraction. The pressure gradient is 61 MPa/m, which is calculated by the pressure difference and the tip length. The pressure difference is 0.61 MPa which is the maximum pressure difference between inlet and outlet listed in table 3. The tip length is 0.01 m. Figure 7 shows that when the oil volume fraction increases, the velocity decreases or even changes direction.

![Figure 7. The velocity profiles of Couette flows.](image)

![Figure 8. The velocity vector fields present the gas velocity in the meshing region of rotors. The left end face is the outlet end of rotor. (a) is under oil-free condition. The average velocities are 75.1 and 83.6 m/s in the regions of (i) and (ii), respectively. (b) is under oil-injected condition. The average velocities are 26.6 and 33.2 m/s in the regions of (i) and (ii), respectively.](image)
The characteristic of flow in figure 7 (a) is close to the one in sealing line gap between rotors. Figure 8 (a) shows the velocity vector field in meshing region of rotors under oil-free condition. The gas flows from the high pressure zone to the low pressure zone through the sealing line gap. Figure 8 (b) shows the velocity vector field in meshing region of rotors under oil-injected condition. The flow velocity is significantly reduced in the region where oil is sufficient. The characteristic of flow in figure 7 (b) is close to the one in the tip gap and the end face gap of rotor. Figure 9 (a) shows the velocity vector field in the tip gaps under oil-free condition. The gas leaks from the chamber with higher pressure to the one with lower pressure continuously, and tends to leak along the direction normal to tip gap. Figure 9 (b) shows the velocity vector field in the tip gaps under oil-injected condition. In the region where oil is sufficient, the gas tends to flow along the direction tangent to rotor rotation. In the region where oil is rare, some gas tends to flow along the direction tangent to rotor rotation, some tends to leak along the direction normal to tip gap.

![Figure 9](image)

**Figure 9.** The velocity vector fields present the gas velocity in the tip gaps. (a) is under oil-free condition. The average velocity is about 51.6 m/s on the side of male rotor, and the one is about 52.3 m/s on the side of female rotor. (b) is under oil-injected condition. The average velocity is about 11.5 m/s on the side of male rotor, and the one is about 10.5 m/s on the side of female rotor.

Figure 10 shows the normalized velocity vector fields on the surface where is between the end face of rotor and the high pressure end plate. Near area “A”, the adjacent chambers are all in outlet process. The pressure difference is small. The velocity directions are the same as rotor rotation under oil-free and oil-injected conditions. Near area “B”, the flow condition is like the one shown in figure 7 (a). The velocity directions are the same as rotor rotation under oil-free and oil-injected conditions. Near areas “C”, “D”, and “E”, the chambers are all experiencing compression. The flow condition is like the one shown in figure 7 (b). The velocity directions are against to the rotor rotation under oil-free condition. The velocity directions are the same as rotor rotation under oil-injected condition. Near area “C” of female rotor, the velocity directions are quite the same under oil-free and oil-injected conditions. It could be rare oil and be affected by the flow near the root of female rotor. The area “F” is inside the root of rotor. Under oil-free condition, the gas flows from the high pressure region. Then, the gas flows over the shaft to the low pressure region. Under oil-injected condition, the gas circularly flows in the same direction as rotor rotation with occasional inflow and outflow. The circular flow under oil-injected condition will provide better shaft sealing than the flow under oil-free condition. The flow phenomena in area “F” will affect the pressure and temperature of gas inside the bearing box. Based on the velocity vector fields shown in figures 8, 9, and 10, the equations of mass flowrate of leakages used in thermodynamics model could be improved.
Table 1. Comparison of oil flow rate (mL/min).

| Condition   | Rotor  | A   | B   | C   | D   | E   | F   |
|-------------|--------|-----|-----|-----|-----|-----|-----|
|             | Male   | 23.2| 116.9| 56.8| 58.9| 51.6| 66.5|
|             | Female | 21.8| 155.9| 79.1| 96.9| 83.4| 80.5|
| Oil-injected| Male   | 10.3| 14.3 | 3.7 | 4.3 | 6.4 | 5.4 |
|             | Female | 7.7 | 20.4 | 11.6| 2.2 | 3.3 | 7.3 |

Figure 10. The normalized velocity vector fields clearly present the velocity direction of flow on the surface between the end face of rotor and the high pressure end plate. (a) is under oil-free condition. (b) is under oil-injected condition. The average velocities [m/s] near areas of A, B, C, D, E, and F are listed in the above table.

Figure 11. (a) is equivalent lubrication models. (b) is surface pressure distribution.

3.2.2. The pressure gradient on rotor surface. In the following discussion, the concept of hydrodynamic lubrication is used to see if rotors are properly lubricated in contact region of rotors. The area “A” shown in figure 11 (a) is the contact region between rotors. The relative motion of rotors could be imagined as a roller bearing. The area “B” shown in figure 11 (a) is the tip gap. The relative motion of rotor and case could be assumed as a journal bearing. The theory of hydrodynamic lubrication (HL) tells that changing the gap size, the relative speed, and the viscosity of fluid will affect the hydrodynamic pressure. If oil is properly filled in the contact region of rotors, elastic hydrodynamic lubrication (EHL) could be generated to ensure long-term rotors meshing [9].

Whether it is HL or EHL, the surface pressure on rotor will rise sharply and then drop rapidly along the direction of rotor rotation near the gap. The obvious positive pressure gradient is followed by negative one, as shown in figure 11 (b). If the relative speed is small or the fluid viscosity is small, the pressure on rotor surface will only be a significant pressure rise or pressure drop caused by pressure difference between upwind and downwind. Figure 12 shows the cross sections near the outlet end of
rotors, and shows the curves of pressure gradient along the direction of rotor rotation on the rotor profiles. The pressure gradient curve with positive value is outside the rotor profile, and the one with negative value is inside the rotor profile.

Figure 12 (a) shows the model under oil-free condition. The pressure gradients are generated by gas pressure difference between upwind and downwind. Figure 12 (b) shows the model under oil-injected condition. Near area “A”, rotors should actually contact with each other. The value of pressure gradient is negative followed by positive under oil-free condition. The value of pressure gradient is positive followed by negative under oil-injected condition. This shows that hydrodynamic pressure appears. Near area “F” of male rotor, the value of pressure gradient is positive under oil-free condition, and the one is positive followed by negative under oil-injected condition. The hydrodynamic pressure also appears near area “F” of male rotor, where is much more oil than near areas “C”, “D”, and “E”.

![Figure 12. Pressure gradient on the rotor surface. (a) is under oil-free condition. (b) is under oil-injected condition.](image)

4. Summary
In this study, there are two cases studied by using ANSYS CFX and TwinMesh. In the first case, the compressor models under four loading conditions are analyzed. The pressure curve is used to see if the designed built-in volume index is appropriate. In order to accurately calculate performance of compressor under different loadings, the oil-injected flowrate which is affected by the inlet and the outlet pressure must be exactly estimated or measured to be the proper boundary condition. The inhomogeneous Eulerian – Eulerian multiphase model is recommended to be used. In the second case, the compressor models under oil-free and oil-injected conditions are analyzed. The pressure gradients on rotor surfaces and velocity vector fields are used to describe the effects of lubrication and sealing in the sealing line gap, the tip gap, and the outlet end gap of rotors. The designer could optimize the slider and know if the oil is properly distributed in each gap. Furthermore, there is an opportunity to ideally perform CFD calculation of an oil-injected screw compressor model that takes into account the contact of rotors. Based on hydrodynamic lubrication, the gap size in contact region of rotors should
be less than 2 µm. Oil is ideally filled in the gap. The effect of finger-cavitation produced by hydrodynamic lubrication could also be used to analyze the rippling damage with fish scale patterns on rotor surface.

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