Numerical Investigation of the Minimum Gap Impact on the Screw Compressor Efficiency

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Abstract. The screw compressor is a positive displacement machine, with two rotors - driver and driven. In this paper, the influence of minimum gap, between the rotors of an oil injection screw compressor, on the efficiency was investigated. Two cases were analysed, in which the minimum gap ranged from 40 to 60μm, respectively 20 to 40μm. The geometrical model, the physical properties and the initial values represent the input data for the used thermodynamic model. Starting from the 3D CAD model, the computational domain for the stator part was defined, which is composed of three subdomains: suction side, oil inlet and pressure side. The grids for the stator were generated with Ansys Meshing, whereas the meshes for the rotor subdomains were made using the TwinMesh software. In the end, smoothing functions were used to increase the mesh quality. To simulate the flow through the screw compressor, the commercial software of CFD ANSYS CFX was used, employing the SST turbulence model. In the analysis the rotational speed was 6000rpm, with a suction pressure of 2bar and a discharge of 10bar. Using the "opening" type limit conditions with the required pressure, both the air flow variation and the oil flow variation were determined.

1. Introduction
Screw compressors are often used in the petrochemical industry, for pneumatic gear drivers and other industrial applications. Based on their design, screw compressors can be divided into two main categories: dry screw compressors – with no oil between the screws and oil-flooded screw compressors. The thermodynamic behaviour of such rotating machines can be associated with that of reciprocating compressors, but screw compressors are considered less susceptible to failure compared to reciprocating compressors [1]. Therefore, they are largely spread for lower compression operations and refrigeration applications. When compared to centrifugal compressors, the main advantage would be that screw compressors do not present a surge line, since the gas volume flow at the inlet does not suffer large fluctuations with pressure ratios variations [2]. State-of-the-art regarding screw compressors integrates thermodynamic evaluations using specific relationships for analytical assessment of their performances [3], noise levels investigation and control methods [4], as well as development of new designs for silencers, that would help mitigate vibrations and therefore noise in screw compressors [5].

In order to improve the efficiency of oil injected screw compressors studies have been conducted regarding the oil level injected in the working chamber and how it affects overall performances. Milligan et al. [6] study the lubrication oil system of oil-injected screw compressors and analyse various measurement technologies for oil level, concluding that industrial markets that exploit screw
compressors prefer technologies with fewer moving parts and simpler designs, as they are low maintenance, rather than complex configurations that provide complicated technologies for oil level measurements. In this case, the industrial users prefer lower maintenance costs in time, rather than slightly improved efficiency. In paper [7] the influence of oil injection parameters on the compressor’s performances is investigated and a lubrication circuit is developed for optimum oil injection parameters that would increase efficiency of the machinery. By optimizing the oil injection system energy savings can go up to 7%, as concluded by Peng et al. [8].

Another way to increase the efficiency of screw compressors is represented by the CFD analysis of its geometry and the optimal minimum gap between rotors, since this parameter directly affects the overall quality of the compressor, as stated by Sauls [9]. Holmes [10] presents in detail the measurement process for the minimum gap, using the Holroyd pair measuring system, describing how it works in order to generate data regarding gaps occurring between difficult surfaces, as are the two rotors of the studied machinery. In paper [11], two designs with different interlobe gaps were investigated in order to evaluate their compatibility with operation regimes that present high discharge temperatures. For the design with an increased gap, the overall performances were reduced, but rotor-to-rotor contact was avoided at the root of the driver rotor (male rotor).

In this research paper, the efficiency evolution of an oil injection screw compressor was investigated for two minimum gap cases, namely 20 to 40μm and 40 to 60μm, by means of computational fluid dynamics numerical analysis, employing the software ANSYS CFX. The purpose of this research is to analyse how overall performances of the screw compressors can be improved by modifying the value for minimum gap between rotors.

2. Methodology

As stated before, the screw compressor has two rotors, one driver (the male rotor) and the other driven (the female rotor). The flow in the screw compressors is continuous but pulsatory, the compression starts by displacing the volume formed between the interlobs and the housing. During the rotation, the suction port opens, and the working fluid enters in the compressor. When the volume is highest and the passage is closed, the volume begins to decrease, and the compression process begins. The discharge phase appears when the discharge port is opened. The thermodynamic model for this study was developed to include and integrate the geometry, proprieties of the physical model and the initial values for pressure and temperature. In figure 1, the implementation process is illustrated.

![Implementation process diagram](image)

Figure 1. Implementation process diagram.
The computational domain for the CFD analysis is divided into five subdomains: three subdomains for the stator part – suction, oil inlet and pressure side, two subdomains for the rotor part – male and female rotor. The computational subdomains are presented in figure 2.

Figure 2. Computational subdomains for the screw compressor.

Regarding the grid, for the stator subdomains the grids were generated using the ANSYS Meshing software, where the grid methods employed were based on the element maximum size, for both surface and volume. For the rotor subdomains, the TwinMesh software was utilized. The XYZ coordinates of the rotors’ points are used as input data for this software. After the points are imported the curves for the rotor, as well as for the shroud, are defined, in order to generate the interfaces between the two rotors. In the end, for quality improvement of the grids, smoothing functions are applied. In figure 3, the mesh quality for the rotor domain is presented, based on the minimum angle criterion, whereas in figure 4 the grids for the stator subdomains are illustrated.

Figure 3. Rotor domain grid quality.  
Figure 4. Stator domain mesh.

The CFD investigation was realized using a solver based on the SIMPLE method and the k-ω SST turbulence model. The rotational speed of the male rotor was 6000 RPM and the suction pressure 2bar, whereas on the discharge pressure was of 10bar.
3. Results

In this section, the results for the studied cases will be presented and compared. The purpose of this investigation is to evaluate the influence of the minimum gap on the overall performances of a screw compressor. For this, the performances were studied for both a minimum gap of 40 to 60μm (Case 1) and one of 20 to 40μm (Case 2). The analysed screw compressor presents a suction pressure of 2bar and a discharge pressure of 10bar at a rotational speed of 6000rpm. In figures 5 and 6 the mass flow variation and oil mass flow variation at the inlet are presented for both cases.

![Figure 5](image1.png)  
**Figure 5.** Mass flow variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm (Case 2).

![Figure 6](image2.png)  
**Figure 6.** Oil mass flow variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm (Case 2).

In order to determine the mass flow and oil mass flow variations (figures 5 and 6) the “opening” type boundary conditions were used for both inlet, oil inlet, as well as outlet. In order to evaluate the
stability of the finite volume method used, the maximal velocity of the working fluid for the screw compressor was monitored during the unsteady analysis. Figure 7 illustrates the maximal velocity variation for the studied cases.

![Figure 7. Maximal velocity variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm.](image)

Regarding the performances of the screw compressor, the total power was also observed for both cases. The mean value for total power was approximately 897kW for the first case (40 to 60μm) and 879kW for the second case (20 to 40μm). The variation of the total power is shown for the two cases in figure 8.

![Figure 8. Total power variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm.](image)

Total torque variation, as well as torque variation for each rotor for both cases are represented in figure 9.
Figure 9. Torque variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm.

Absolute pressure variation is illustrated in figure 10. In both cases, after approximately 300 time-steps, the pressure outlet of 10 bar was assessed. On the same chart are presented the variations for maximal and minimal pressure, as well as the constant pressure inlet of 2 bar. It can be observed that for the second case the maximal pressure value is higher than for the first case, indicating that for a smaller gap pressure forces are higher. As a result, the total power in the second case is with 18kW smaller than the total power for the first case. This decrease can be associated with the increased pressure forces that occur for the second case.

Figure 10. Pressure variation: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm.

The absolute pressure on the rotors’ surface is shown in figure 11. For a smaller gap (case 2) the pressure between the rotors’ lobes is higher and this can affect the overall performances of the screw compressor through a lower total power, as discussed earlier. However, for the first case, with a bigger
gap, the pressure is higher on the rotors’ surfaces when compared to the second case. Fortunately, the impact on total power in this case is not as strong as for the smaller gap case.

![Figure 11](image)

**Figure 11.** Absolute pressure distribution: (a) Case 1 - 40 to 60μm; (b) Case 2 - 20 to 40μm.

### 4. Conclusions

In this paper the influence of the minimum gap on the overall performances of a screw compressor was numerically evaluated by means of CFD analysis. Using ANSYS CFX the variation of the mass flow was determined at inlet and outlet, as well as the variation of oil mass flow inlet. The evolution in time was assessed for maximal velocity, absolute pressure, power, torque, mass flow and oil mass flow for two cases: one with a minimum gap varying from 40 to 60μm and one with a minimum gap ranging from 20 to 40μm. The total power in the first case surpassed the one of the second case by 18kW. By analysing absolute pressure variation, it was observed that for a minimum gap of 20 to 40μm the compressor was subjected to higher pressure forces that may have caused a decrease in total power. The absolute pressure distribution on the surfaces of the two rotors shows that for the second case (with a minimum gap of 20 to 40μm) high pressure values occur at discharge between the rotors’ lobes. On the other hand, for the first case (with a minimum gap of 40 to 60μm) the pressure on the rotors’ surface is slightly higher. In conclusion, in order to obtain optimal performances in accordance with the application of the screw compressor designed, the minimum gap between the rotors should be chosen carefully, keeping in mind that it should be small enough as to reduce pressure forces on the rotors’ surface, but big enough in order not to generate high pressure forces between the rotors’ lobes.

### 5. References

[1] Broerma E B, Manthey T, Wennemar J and Hollingsworth J 2019, *Compression Machinery for Oil and Gas: Chapter 6 – Screw Compressors*, Gulf Professional Publishing, Elsevier Inc. pp. 593 – 603

[2] Mannan S 2012, *Lees’ Loss Prevention in the Process Industries – 4th Edition* (Oxford, UK: Butterworth- Heinemann, Elsevier Inc).

[3] Forsthoffer M S 2017, *Forsthoffer’s More Best Practices for Rotating Equipment: Chapter 3 – Compressors* (Oxford, UK: Butterworth – Heinemann, Elsevier Inc.) pp. 73 – 185

[4] He Z, Han Y, Chen W, Zhou M and Xing Z 2020, Noise control of a two-stage screw refrigeration compressor, *Applied Acoustics* 167

[5] van Lier L, Korst H and Smeulers J 2014, Design of new silencers for screw compressor, *Fluid Machinery Congress 6-7 October 2014* 109 – 121

[6] Milligan W J and Muir D I 2013, Oil level measurement in oil-injected screw compressor packages used in the petroleum, petrochemical refrigeration and fuel gas markets, *8th International Conference on Compressors and their Systems* 77 – 85
[7] Dhayanandh K K, Rameshkumar K, Sumesh A and Lakshmanan N 2020, Influence of oil injection parameters on the performance of diesel powered screw air compressor for water well application, Measurement 152

[8] Peng X, Xing Z, Zhang X, Cui T and Shu P 2000, Experimental Study of Oil Injection and Its Effect on Performance of Twin Screw Compressors, International Compressor Engineering Conference – School of Mechanical Engineering paper no. 1491 1003-1010

[9] Sauls J and Branch S 2013, Use of computational fluid dynamics to develop improved one-dimensional thermodynamic analyses of refrigerant screw compressors, 8th International Conference on Compressors and Their Systems 591–600

[10] Holmes C S 2004, Inspection of Screw Compressor Rotors for the Prediction of Performance, Reliability, and Noise, International Compressor Engineering Conference – School of Mechanical Engineering Paper no. 1692

[11] Buckney D, Kovacevic A and Stosic N 2016, Design and evaluation of rotor clearances for oil-injected screw compressors, Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering 231 26 – 37

Acknowledgments

This work was carried out within “Increasing Excellence in Research - Development of RRDI for Gas turbine - COMOTI” Project CREaROR ”, supported by the Romanian Minister of Research and Innovation, project number Ctr.3PFE/2018.