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Mass Flow Rate Optimization of a Reciprocating Inverter Compressors Oil Pump with CFD Simulations

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ABSTRACT

The lubricating oil used in the reciprocating compressor is located in the lower shell and with the operation of the crankshaft, oil climbs to helical channels on the crank shaft by the centrifugal forces. Oil is distributed from crank outlet and this ensures that all parts in the compressor are lubricated. The lubricating oil, which reaches the crankshaft bearings and other parts of the compressor prevents wear on the bearings and it also cools other components. Oil pump system includes both helical and centrifugal pump characteristics.

The aim of this study is to determine the oil pump design that gives the maximum oil flow rate at the crank outlet. A parametric numerical fluid dynamics analyses are performed to evaluate the effect of crank-screw pump geometric parameters, i.e., the number of screw-helix, screw form and depth between crank and screw. Results of this study are used in deciding the amount of oil to be used. In addition, the vortex effects in oil reservoir and its effects on sound power level of the compressor are investigated.

1. INTRODUCTION

Hermetic reciprocating compressor, is a key element for household refrigerator system, consumes electrical energy and converts it to the energy stored in the evaporated refrigerant fluid. Although there are many different types of compressors, the hermetic reciprocating compressor is the most widely used type of compressor in domestic refrigerators in the competitive environment as a result of globalization.

One of the important system in compressor is lubricating system. Since the cooling capacity is directly related to the crankshaft speed, the design of the oil management system which provides required amount of the lubricant to the journal bearings and other moving parts are crucial to warrant expected lifetime and performance of the compressor. (Ozsipahi et al., 2019). The lubricating oil used in the reciprocating compressor is located in the lower shell and with the operation of the crankshaft, oil climbs to helical channels on the crank shaft by the centrifugal forces. Oil is distributed from crank outlet and this ensures that all parts in the compressor are lubricated.

After developing inverter control technology, variable speed compressor is of great interest in air conditioning and refrigeration system because it offers better capacity regulation than the conventional on/off control. The modeling of this kind of compressor plays a very important role in the simulation of air conditioning and refrigeration system

(Chen et al., 2005). With this development, lubrication system design become a crucial part of the compressor for different speed ranges.
In this study, oil pump optimization is investigated to give the maximum oil flow rate at the crank outlet especially in minimum working speeds. In addition, by the aim of diminishing the amount to be oil used in compressor, maximum oil flow rate is studied for maximum working speeds. A parametric numerical fluid dynamics analyses are performed to evaluate the effect of crank-screw pump geometric parameters, i.e., the number of screw-helix, screw form and depth between crank and screw. Moreover, the vortex effects in oil reservoir and its effects on sound power level of the compressor are investigated.

2. CFD MODELING

CFD analysis is designed for the examination of four different screw parameters with a view to mass flow rate and pressure difference optimizations. These parameters are number of screw helix, depth between crank and screw, length of screw and diameter of screw.

Table 1: Screw parameters and prepared different samples

Parameters	Sample-a	Sample-b	Sample-c	Sample-d	Sample-e
Number of screw helix	4	4	3	3	3
Depth between crank and screw	short	long	long	short	short
Length of screw	short	short	short	long	long
Diameter of screw	large	large	large	small	large

2.1 Analysis Steps

Oiling crank-screw geometry consists of three parts. These parts are crank, screw and attachment parts. In compressor inner volume, oil volume fluid is set at the lower case and gaseous is set at the upper side. Then, fluid volume of oil supply system is created. When the fluid volume is generated, the sharp edges joining the narrow sides of the geometry and the skewed surfaces that would not affect the flow are matched to the CFD analysis. The fluid volume scheme is showed in Figure 1.

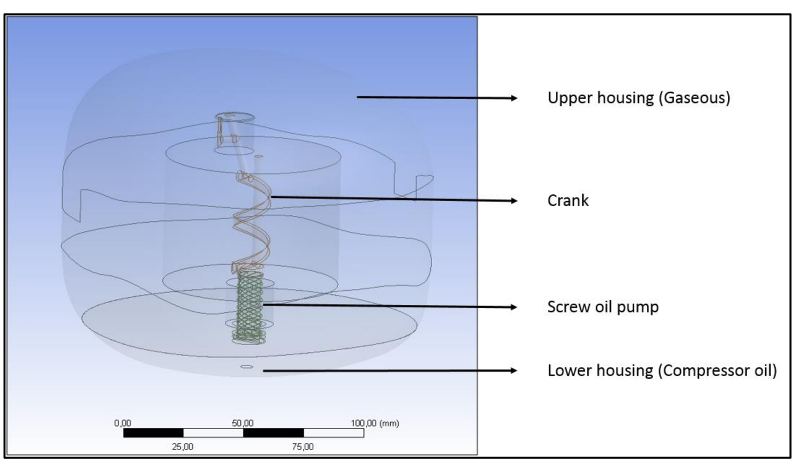

Figure 1: Fluid volume of compressor oil system

After geometrical arrangements, solution network is build, the physical component is divided into a small number of small elements. Trying many mesh configurations, 3 million elements is decided to provide convergence criteria. The solution network consists of tetrahedral elements and the skewness value is 0.85.

Next, boundary conditions is set in commercial software code, ANSYS Fluent. In this case, model includes two phase flows, so volume of fluid approach (VOF) is executed. The VOF method is an applicable numerical technique for interfaces in two-phase flows (Ansys, Inc., 2013). The method is based on tracking the volume fraction of each fluid in the domain and solving the volume fraction equation (1).

$$C(t) = \frac{1}{V}\int_V \chi(x,t)dx \tag{1}$$

The volume fraction C represents the volume of the cell which is occupied by the reference phase; the range is defined $0 \le C \le 1$ (Bnà et al., 2014)

The transport equations of the material properties are regulated by the presence of the component phases in each control volume (Ansys, Inc., 2013). In a two-phase system, volume fraction of the second of these is being tracked, for instance, density in each cell is given by

$$\rho = \alpha_2\rho_2 + (1-\alpha_2)\rho_1 \tag{2}$$

Other properties like viscosity are computed in this way.

In solution methods, Fractional step method is chosen. The momentum equations are decoupled from the continuity equation using a mathematical technique called operator-splitting or approximate factorization and the order of splitting error can be controlled with this formalism. By reason of that, velocity-coupling scheme in a non-iterative time-advancement (NITA) algorithm can be used with Fractional step method (Ansys, Inc., 2013).

Table 2: Model, material properties, boundary conditions and convergence criteria

<table>
<tr><td>Multiphase Model</td><td colspan="2">Volume of Fluid (VOF)</td></tr>
<tr><td>Viscous Model</td><td colspan="2">Laminar</td></tr>
<tr><td rowspan="2">Material Properties</td><td>Fluid 1
Density (kg/m3)
Viscosity (kg/m-s)</td><td>Air
1.225
1.7894e-05</td></tr>
<tr><td>Fluid 2
Density (kg/m3)
Viscosity (kg/m-s)</td><td>Compressor oil (40 °C)
890
0.0089</td></tr>
<tr><td rowspan="2">Boundary Conditions</td><td>Crank</td><td>Frame Motion (Rotational velocity: 800 rpm and 4500 rpm for each case)</td></tr>
<tr><td>Screw wall</td><td>Moving Wall (Rotational velocity: 800 rpm and 4500 rpm for each case)</td></tr>
<tr><td>Solution Methods</td><td>Fractional Step Method</td><td>Non-Iterative Time Advancement</td></tr>
</table>

Convergence criteria	Continuity	10e-4
	Velocity	10e-3
	Volume fraction	10e-4

Assumptions

- Air is used for the gaseous phase.
- Physical properties of the fluids are constant.

2.2 CFD Results

Crank-screw oil pump system provide oil supply on piston-connecting rod dynamic mechanism, crankcase and bearings. Climbing oil time and oil mass flow rate can be measured from crank-oulet with transient CFD analyses. These analyses conduct with 800 and 4500 rpm crank rotation speed respectively. Numerical calculations of oil mass flow rate from crank outlet versus time is seen in Figure 2,3,4,5 and 6.

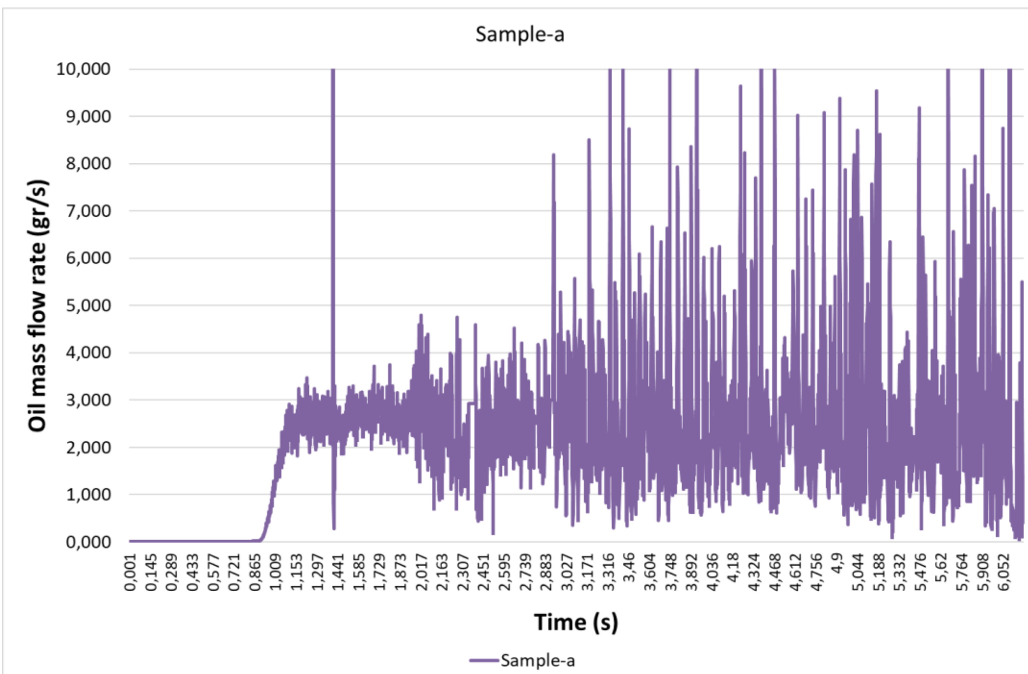

Figure 2: Oil mass flow rate of Sample-a @4500 rpm

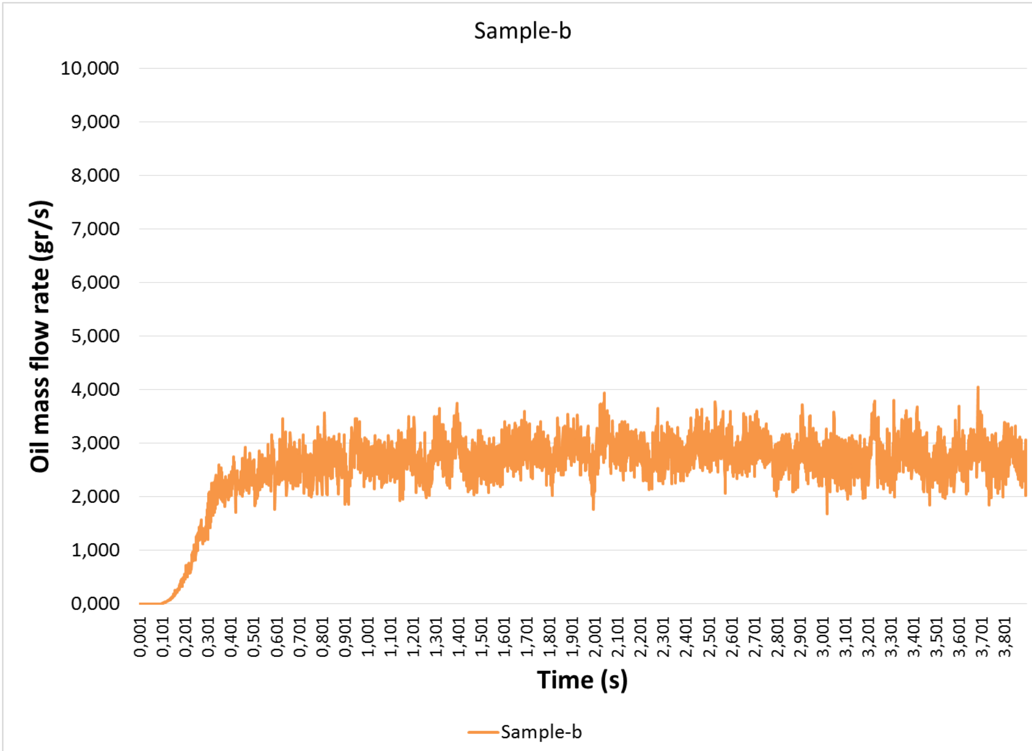

Figure 3: Oil mass flow rate of Sample-b @4500 rpm

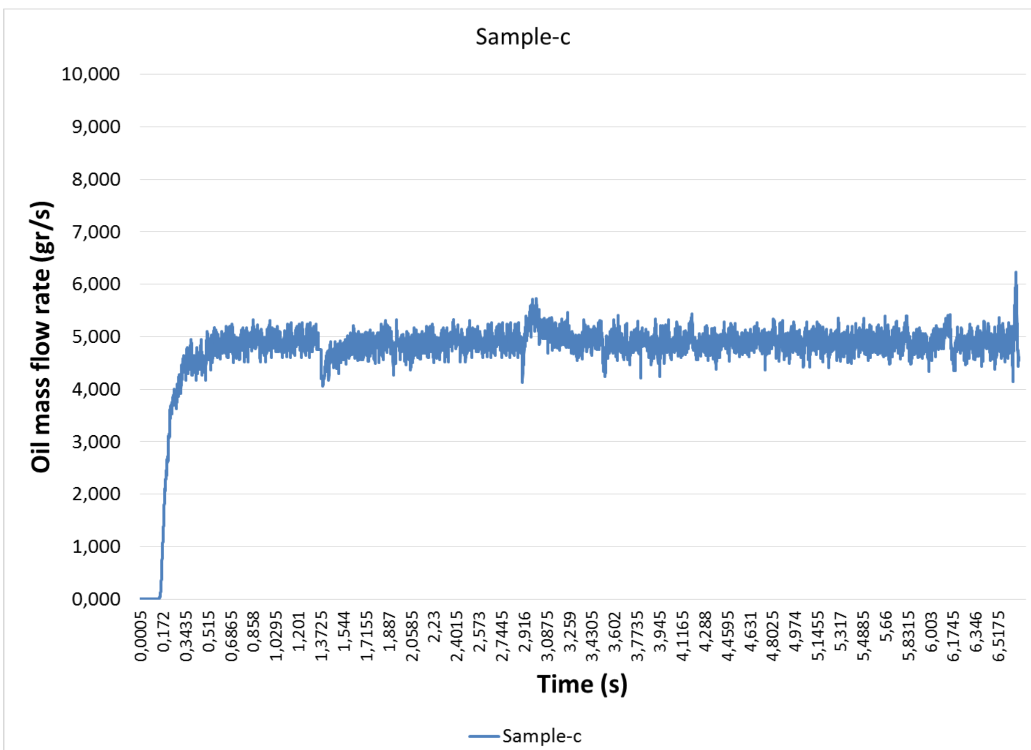

Figure 4: Oil mass flow rate of Sample-c @4500 rpm

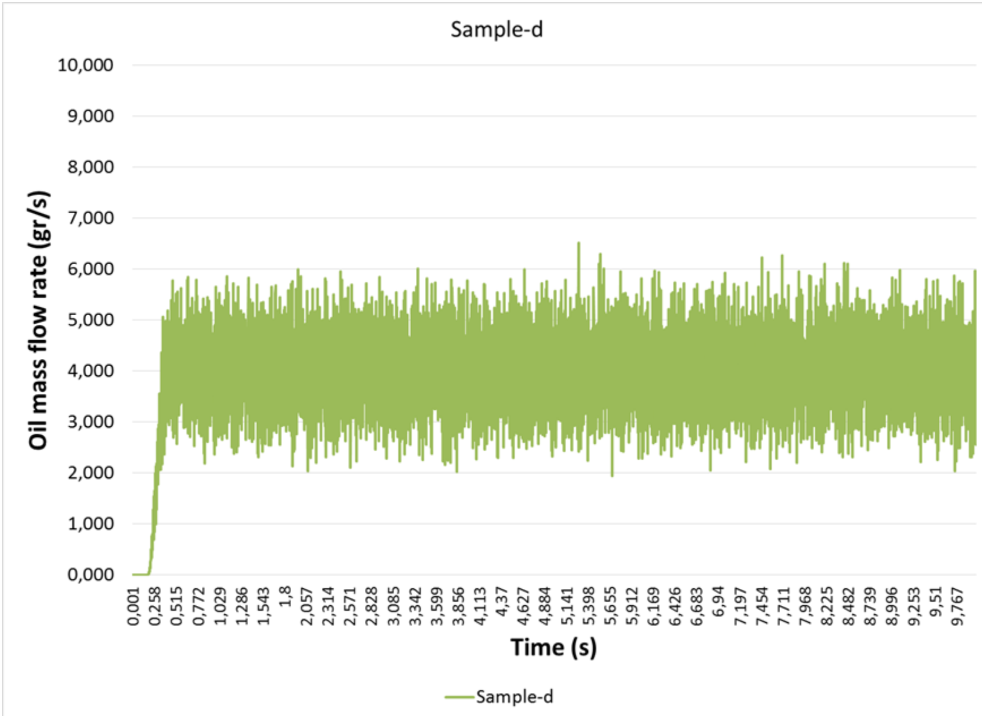

Figure 5: Oil mass flow rate of Sample-d @4500 rpm

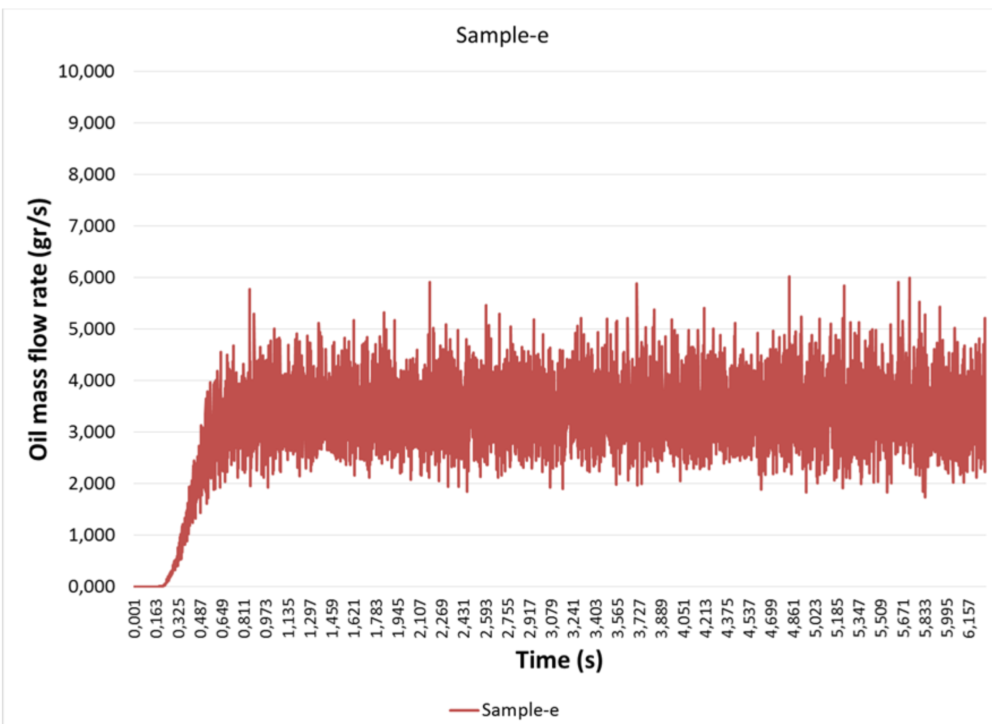

Figure 6: Oil mass flow rate of Sample-e @4500 rpm

After seeing the oil mass flow rate versus climbing time, average oil mass flow rate is calculated for regime conditions in each model.

Table 3: Average oil mass flow rate for each model @800 rpm and @4500 rpm

Model	Sample-a	Sample-b	Sample-c	Sample-d	Sample-e
Oil mass flow rate (gr/s) (Crank rotation speed: 800 rpm)	0.59	0.71	0.79	0.67	0.72
Oil mass flow rate (gr/s) (Crank rotation speed: 4500 rpm)	2.63	2.79	4.81	4.02	3.42

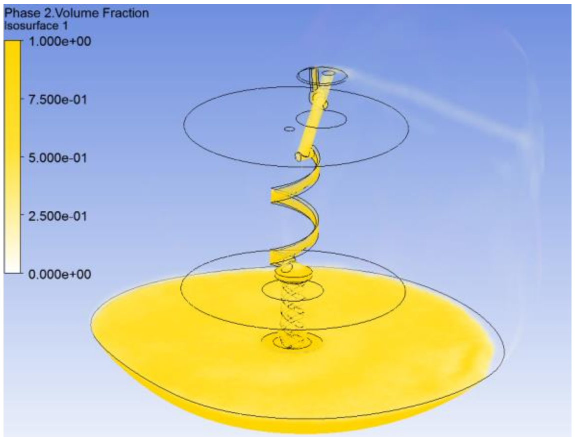

Figure 7: CFD Post Process**:** Volume fraction rendering of compressor oil

Pressure changes in flow come out through the helical channels of the crank-screw oil pump system. It is predicted that structural and acoustic problems can appear as a result of high-pressure differences and high-pressure forces acting on the screw.

Figure 8 shows pressure changes in fluid along the crank-screw oil pump system as Sample-d.

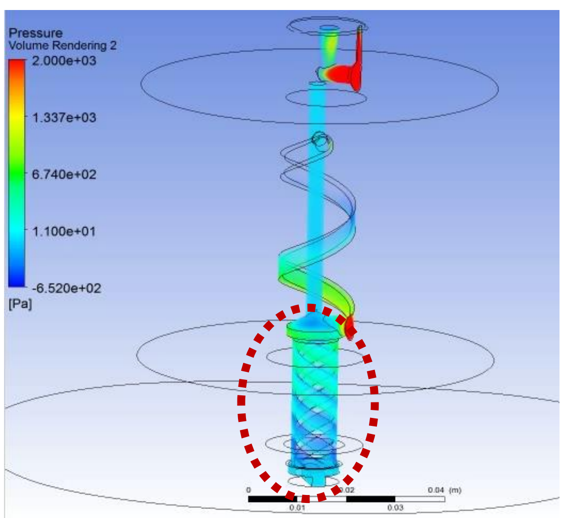

Figure 8: CFD Post Process**:** Volume fraction rendering of compressor oil

Table 4: Pressure changes in passing screw to crank side (Marked area from Figure 8) for all samples

Model	Sample-a	Sample-b	Sample-c	Sample-d	Sample-e
Crank rotation speed (rpm)	4500	4500	4500	4500	4500
Pressure difference in passing screw to crank (Pa)	≥2000	1300-1400	670-1300	100-670	670-1300

These results show that minimum pressure changes is detected in Sample-d and maximum pressure values are observed in Sample-a with the same conditions.

3. EXPERIMENTAL STUDIES

For experimental tests, hermetic reciprocating compressor (165 kcal/h @3000 rpm in ASHRAE) is used. One of the critical tests to decide new oil-pump design is trying to see oil climbing in crank outlet. In experiment, the amount of lubricating oil is controlled and increased gradually. In each condition, compressor starts with 800 rpm. If oil climbs through the channel and lubrication is seen, then it is noted.

For current design, minimum operating speed is 1300 rpm and the amount of lubricating oil is 180 cc. On the other hand, for sample-d model, minimum operating speed is 800 rpm and the amount of lubricating oil is 60 cc. New oil pump model has advantage of working on lower operating speed and using lower the amount of lubricating oil (Figure 9).

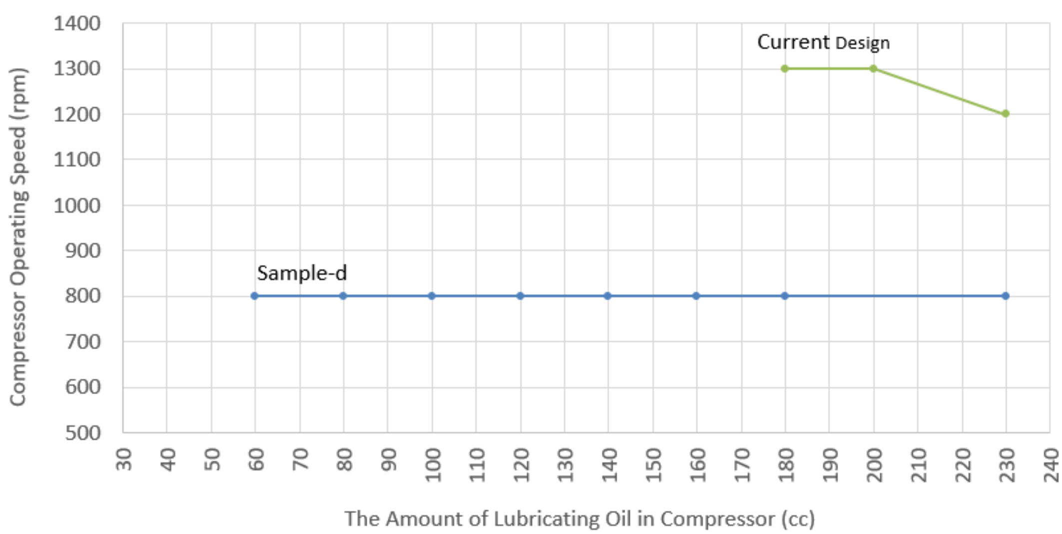

Figure 9: Operating speed versus the amount of lubricating oil for current design and Sample-d

The second critical test to decide new oil-pump design is sound pressure level measurements. Sound pressure level measurements made on the hemispherical surface in a semi-anechoic room in according to ISO 3741 and ISO 3745 standards in 1/3 octave frequency bands in the frequency range of 100 Hz to 10 kHz.

Controls are carried out by taking background sound measurements with 10 microphones at regular intervals.

Figure 10: Sound pressure level measurement room

Figure 11,12 and 13 show that sound pressure level results of current design and sample-d model. In this experiment, operating speeds are selected as 1300, 3000 and 4500 rpm.

According to Figure 11, total SPL of sample-d is 1,6 dBA lower than current design. Sample-d has better performance in especially 500-800 Hz, 1250-3150 Hz, 5000 Hz and 8000-10000 Hz.

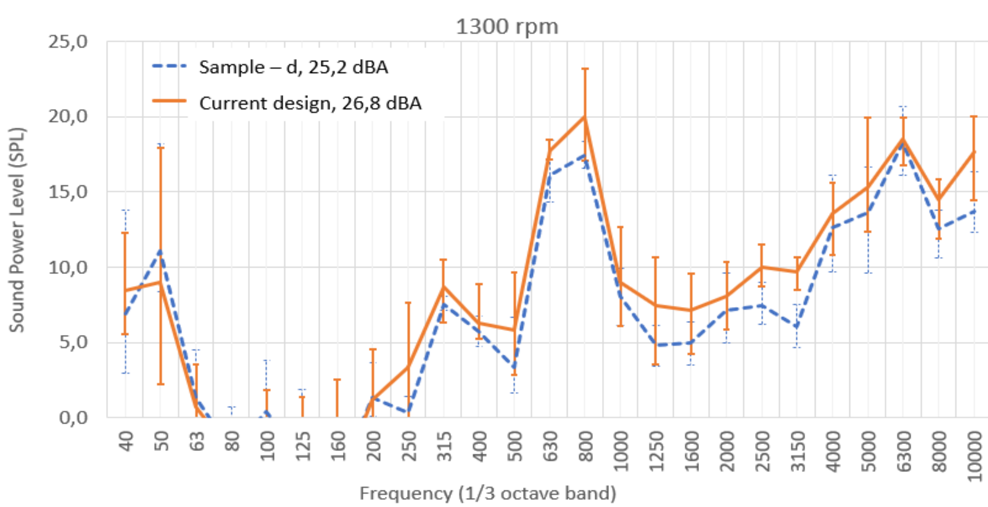

Figure 11: Sound power level in 1300 rpm for current design and sample-d

According to Figure 12, total SPL of sample-d is 2,4 dBA lower than current design. Sample-d has better performance in especially 1600-5000 Hz.

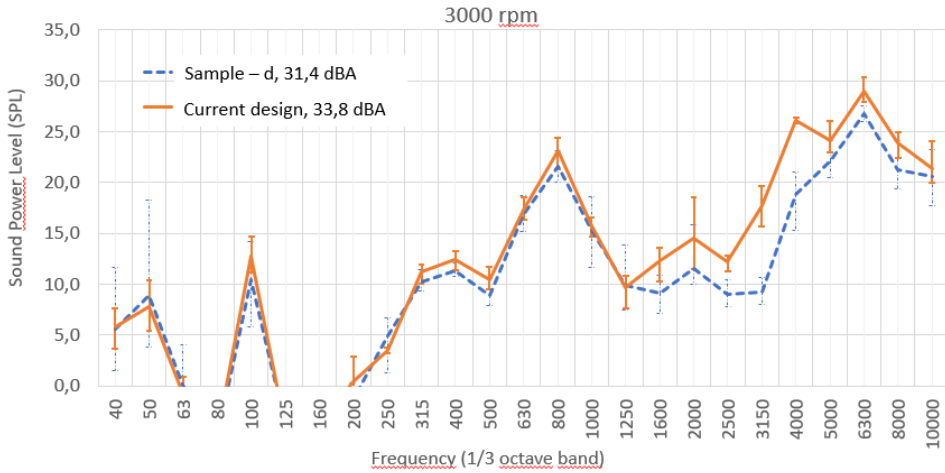

Figure 12: Sound power level in 3000 rpm for current design and sample-d

According to Figure 13, total SPL of sample-d is 0,4 dBA upper than current design. Sample-d has worse performance especially in 2000 Hz and 6300 Hz.

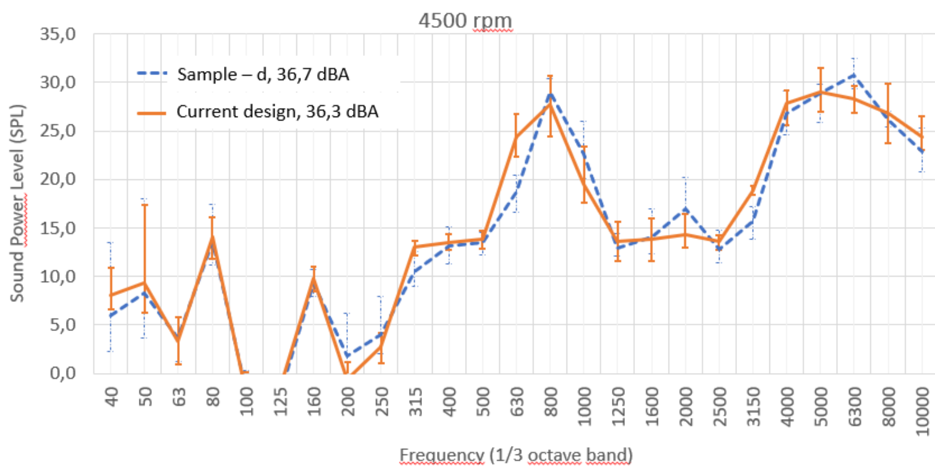

Figure 13: Sound power level in 4500 rpm for current design and sample-d

4. CONCLUSIONS

The design of an adequate lubrication system is a major task in the compressor development process because of the need to fulfill all the requirements. In addition, the lubrication of the moving parts and the hydrodynamic bearings, the oil also transports heat from the hot compressor parts to the shell, acts as sealing between piston and cylinder and protects against corrosion. The oil supply system in hermetic reciprocating compressors is usually integrated in the compressor crankshaft and it uses the energy of the rotating crankshaft to pump the oil (Posch et al., 2018).

In this study, oil pump design parameters are investigated and optimized with CFD Modeling. A parametric numerical fluid dynamics analyses are performed to evaluate the effect of crank-screw pump geometric parameters, i.e., the number of screw-helix, screw form and depth between crank and screw. Five different models are developed and volume of fluid approach (VOF) is executed. According to oil mass flow rate in crank outlet results, sample-c has maximum flow rate for 800 rpm and 4500 rpm, but it has worse pressure fluctuation in 4500 rpm which can leads to wear on oil pump system. So, sample-d model is selected as best option in accordance with results.

In experimental studies, the amount of oil to be used is determined trying to see oil climbing in crank outlet. Sample – d model has advantage of working on lower operating speed and using lower the amount of lubricating oil. Moreover, sound pressure level measurements are tested for current design and sample – d. For different operating speeds, sample – d has better performance in some frequency ranges.

To conclude, CFD modeling of oil pump system presented here is a tool that can be recommended for use in the design phase of reciprocating compressors in order to rapidly obtain simulation results.

NOMENCLATURE

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
CFD	Computational Fluid Dynamics
dBA	Decibel (filter of A)
ISO	International Organization for Standardization
VOF	Volume of Fluid
SPL	Sound Power Level

Subscript

rpm	Revolutions per minute
kcal/h	kilocalorie per minute
cc	cubic centimeters

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