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Numerical Investigation of Oil Flow in a Hermetic Compressor

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Abstract— *The aim of this study is to numerically investigate the effects of various parameters on the lubricant (oil)-coolant two phase flow in the lubrication system of hermetic compressors commonly used on household refrigerators. Lubrication oil is pumped from the sump through an asymmetrically opened hole on the bottom of the crankshaft (suction side or inlet) by its rotational motion and climbs as an oil film on the internal surface of the helical channel carved on the crankshaft surface. This oil film is directed to crankshaft upper exit discharging into the coolant refrigerant and it is used to lubricate the moving components of the compressor including the cylinder piston. The oil forms an immiscible mixture with coolant, thus a two phase flow model using Volume of Fluid (VOF) method is used. Specifically, the mass flow-rate of oil is determined as a function of the rotational speed, oil viscosity and the submersion depth of the crankshaft in the oil-sump. With increasing rotational speed and submersion depth, the mass flow-rate through the crankshaft upper exit also increases. With increasing oil viscosity the mass flow-rate through the crankshaft upper exit decreases due to the increased friction.*

Keywords—Hermetic compressor; Computational Fluid Dynamics (CFD); Volume of Fluid (VoF); Two-phase flow

I. Introduction

The hermetic reciprocating compressor can be considered as the most important and complex component in a household refrigeration unit. As it heavily affects the overall efficiency of the refrigeration process, the design and optimization of various compressor parts are of paramount importance. Lubrication system in a hermetic reciprocating compressor is vital for a reliable and efficient operation and includes important parameters to influence the efficiency of the compressor. Cooling of various compressor parts, sealing, reducing the noise level are among the tasks of the lubrication in a compressor [1].

Recently, lubrication of oil flow in a reciprocating compressor using a two-phase model is investigated numerically by means of CFD. Volume of Fluid (VoF) methodology is mostly applied to model a two-phase flow where immiscible oil and coolant (mostly air) with or without free surface effects is considered. Most important output of the simulations are the prediction of the steady-state oil flow rate and this has been widely reported in literature [1-4]. An oil pumping system is modeled both numerically and analytically [3]. Numerical results after reaching steady-state condition are compared to analytically obtained results. Parametric analysis including the effect of the inlet diameter with the shaft eccentricity and the submersion depth are given in [3]. It is shown that the flow rate increases with the inlet diameter and shaft eccentricity.

Other desired information is the time required for the oil to reach bearings once the crankshaft starts rotating. Lubrication system of a hermetic reciprocating compressor is visualized by a high speed camera and oil climbing time is compared to CFD results in [4]. Furthermore, effect of viscosity and rotational speed on the flow rate is investigated in [4]. It is shown numerically that with increasing rotational speed the mass flow rate also increases. On the other hand, the mass flow rate drops when the viscosity increases. Another study shows similar results found in [4] such as a drop in the flow rate with increasing viscosity and a raise with increasing rotational speed [5]. In this study, the effects of the rotational speed, oil viscosity and submersion depth of the crankshaft in the oil-sump are investigated numerically. While our results show similar findings with the previous literature at the steady state, we also report time-accurate transitional behavior of oil climbing through the crankshaft. In addition, by reporting the first time the velocity field in the crankshaft inlet, we investigate and propose modifications for the maximization of the oil flow rate. The variations of the flow rate through the crankshaft exit with time have been plotted for different rotational speeds, viscosities and submersion depth in the oil sump.

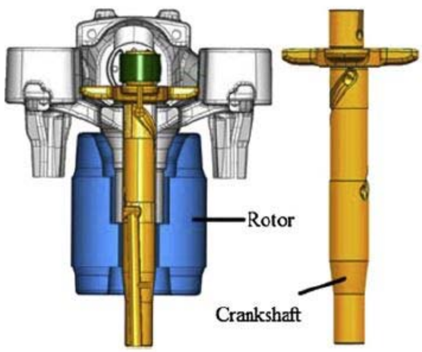

Fig.1. Detailed appearance of a hermetic reciprocating compressor [4]

II. COMPUTATIONAL MODELLING AND GOVERNING EQUATIONS

A. Governing Equations

The computational model used is VoF. We use Fluent software and for details and implementation, we refer to Fluent manual [6]. Nevertheless, we note the tracking of the interface(s) between the phases is accomplished by the solution of a continuity equation for the volume fraction of one (or more) of the phases given in (1). For the liquid phase (oil) denoted by *l*, this equation has following form [6]:

$$\frac{\partial}{\partial t}(\alpha_l \rho_l) + \nabla(\alpha_l \rho_l \vec{v}_l) = 0 \qquad (1)$$

In this study, mass transfer between the phases and any source term for both phases are not present. Liquid has been treated as the secondary phase in our computational study.

As we are interested in an isothermal flow, only momentum equations are taken into account. Hence the velocity field is obtained from the numerical solution of the momentum equation given in (2) for both phases [6]:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla(\rho \vec{v}\vec{v}) = -\nabla p + \nabla[\mu(\nabla \vec{v} + \nabla \vec{v}^T)] + \rho \vec{g} + \vec{F} \qquad (2)$$

B. The Computational Domain and Boundary Conditions

The computational domain shown in Fig.2, consists of approximately 240,000 polyhedral cells where mesh is refined near walls. Lubrication system is divided into two parts: the crankshaft prescribed with frame-motion and the stationary oil sump. The accelerated start-up of the electrical motor is neglected and it is assumed that the angular velocity is constant due to infinite start-up acceleration. Top of the oil sump is assumed to be a free-surface and at the crankshaft exit pressure outlet boundary condition is used.

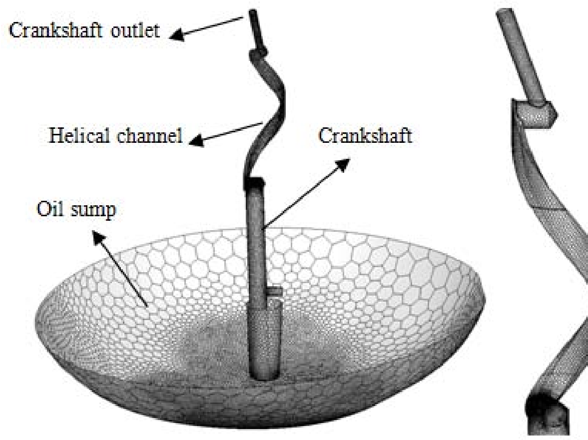

Fig. 2. Computational domain for the lubrication system

PISO (Pressure-Implicit with Splitting Operators) algorithm [7], which is a preferred pressure-based scheme for the transient flows, has been used for the pressure-velocity correction. Time step selected is 5 x 10^{-4} s and solution is obtained for a minimum time of 5 s. Steady-state solution is mostly reached after about 2-3 s.

The Reynolds number based on oil mass flow through the crankshaft inlet is calculated in (3):

$$\mathrm{Re} = \frac{4\dot{m}}{\pi \mu D} \cong 14 \qquad (3)$$

Based on the Reynolds number, the flow is assumed to be laminar thus the computations have been modeled as laminar flow.

The computations have been applied for following conditions:

- constant rotational speeds (rpm): 1000, 2100, 3000, 4000 and 5000; for oil viscosity 5 cSt and submersion depth 1.42 (normalized by the crankshaft diameter).
- oil viscosities (cSt): 3, 5, 10 and 15; for constant rotational speed (3000 rpm) and submersion depth 1.42 (normalized by the crankshaft diameter).
- submersion depths (-): 0.71, 1.07 and 1.42, (normalized by crankshaft diameter); for constant rotational speed (3000 rpm) and for oil viscosity 5 cSt.

All simulation are performed via parallel computations on 6-core Pentium processors using a finite-volume based incompressible transient flow solver (FLUENT).

III. RESULTS

Fig. 3 shows the instantaneous oil climbing through the crankshaft starting from the rest (t = 0) at n = 3000 rpm. As we neglect the motor acceleration time, the climbing starts right after the motion of the crankshaft. Beginning at 0.5 s, increasing fluctuations in the oil-air free surface is observable. The steady state solution is obtained for about $t > 3$ s where oil mass flow rate does not change with time.

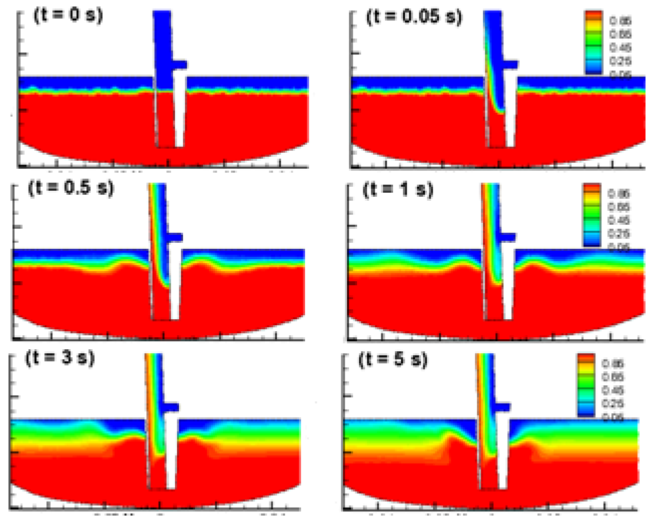

Fig.3. Instantaneous oil climbing through the crankshaft at n = 3000 rpm. Oil phase is illustrated in red.

Fig. 4a and 4b show instantaneous vector fields at the time levels t = 0.5s and 5s respectively. They are extracted as

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vertical slices from the computational domain and the colors indicate the velocity magnitudes in m/s.

Due to the centrifugal forces caused by the rotational speed of the eccentrically created flow path inside the crankshaft, oil is sucked from the crankshaft's inlet. The highest velocity magnitude is observed on the outer edges of the crankshaft because of the highest centrifugal forces acting on the fluid at those regions.

Fig.4c shows the velocity vectors (colors denote the velocity magnitude) in a horizontally sliced plane near the crankshaft's inlet. It is clear that on the left side of the inner edge we observe oil flow out of the crankshaft to the sump. This reduces the total oil flow to the helical channel so some modification is needed to reduce this back flow.

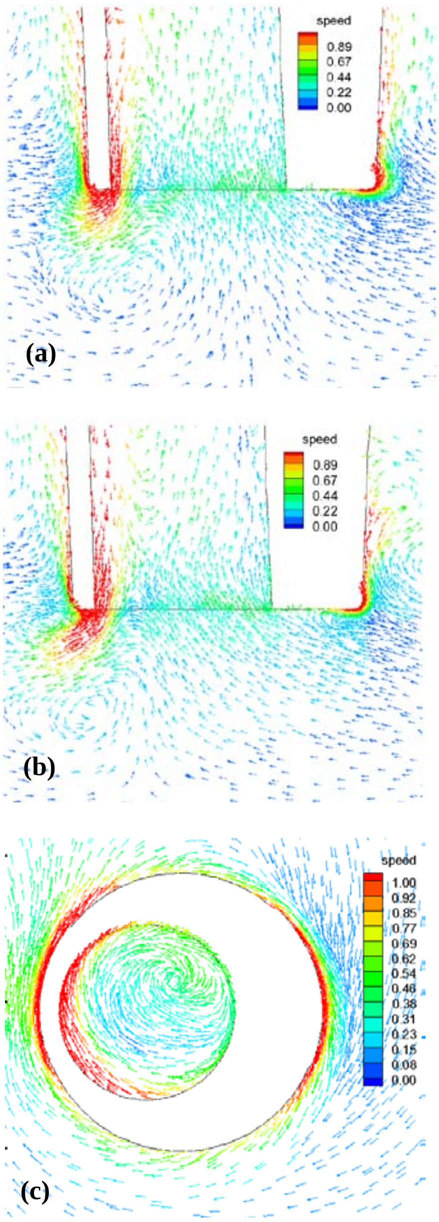

Fig.4. Closed-up view of the flow at the inlet of the crankshaft: (a) vertical plane at $t = 0.5$ s, (b) vertical plane at $t = 5$ s, (c) horizontal plane at $t = 5$ s

A. *Effect of Rotational Speed*

To investigate the effect of the rotation; simulations with constant rotational speeds of 1000, 2100, 3000, 4000 and 5000 rpm have been performed for oil viscosity 5 cSt and submersion depth of 1.42 (normalized by the crankshaft diameter). The rotational speed is prescribed as frame motion on the walls of the crankshaft. We have carried out the computations at least 5 seconds to reach steady-state conditions for the mass flow rate at the crankshaft exit. Fig. 5 illustrates variation of the mass flow rate at the crankshaft exit for various rotational speeds. The values of the mass flow rates reported in the study are normalized by the nominal 3000 rpm-value. As expected, with increasing rotational speed, a raise in the mass flow rate at the crankshaft exit is observed. Higher rotational speeds cause a peak in the mass flow rate around 0.8 seconds which is not occurring at the lowest rotational speed.

In Fig. 6, the mass flow rates at the crankshaft both at inlet and exit are plotted for 3000 rpm. Due to the transitional regime at the start up and there exists air in the most part of the crankshaft hole and helical channel at t = 0, it takes some time for the accumulation of the oil film in the crankshaft and helical channel. There is initially a big deviation between the inlet and exit mass flow rates. This deviation is the oil stored both on the helical channel and crankshaft resulting in thickening of the oil film layer with time. After about 1.4 s, the mass flow continuity at the inlet and exit is preserved.

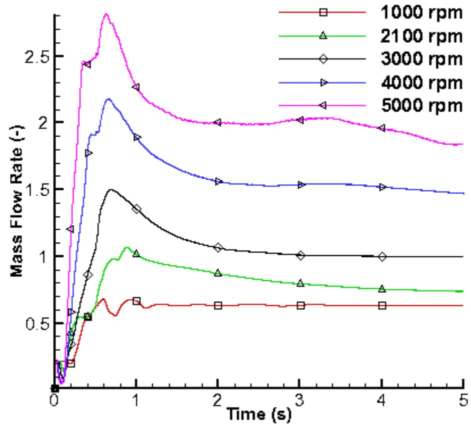

Fig.5. Variation of the mass flow rate at the crankshaft exit at various rotational speeds

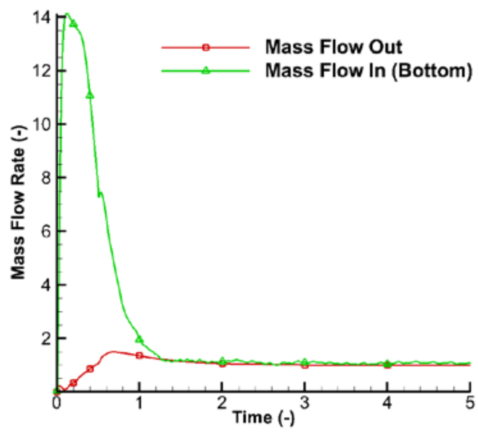

Fig.6. Variation of the mass flow rate at the crankshaft inlet and exit at 3000 rpm.

B. Effect of Oil Viscosity

To investigate the effect of the oil viscosity; viscosity values of 3, 5, 10 and 15 cSt have been used for constant rotational speed (3000 rpm) and normalized submersion depth of 1.42. Since we assumed isothermal flow, the viscosity values selected are treated constants in the simulations. We note that the viscosity values considered are of common industrial oils at about 40^{o}C. As Fig.7 shows, the mass flow rate at the crankshaft exit increases when the viscosity decreases. With a r aise in the temperature, viscous effects weaken leading to less frictional forces. As a result, oil with a lower viscosity can climb easier to the crankshaft exit resulting in the more oil flow rate.

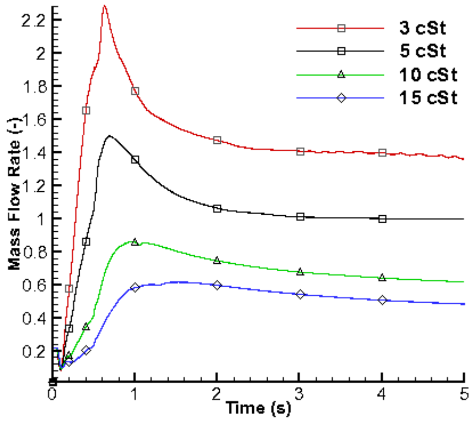

Fig.7. Variation of the mass flow rate at the crankshaft exit for various viscosities.

C. Effect of Submersion Depth

To investigate the effect of the submersion depth *h*; first submersion depths are normalized by the crankshaft diameter, *d*. Thus, simulations for the normalized submersion depths (*h*/*d*) 0.71, 1.07 and 1.42 have been carried out for constant rotational speed (3000 rpm) and oil viscosity 5 cSt. As shown in former studies [3 and 5], the mass flow rate at the crankshaft exit increases when the submersion depth in the oil sump increases. A higher flow rate can be obtained with a higher value of the submersion depth, which increases hydrostatic head acting on the lubrication system. Fig. 8 s hows the variation of the oil flow rate and the normalized submersion depth.

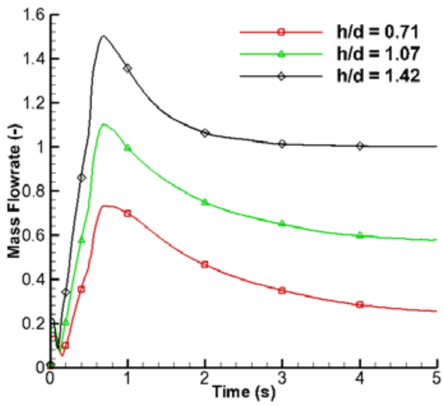

Fig. 8. Variation of the mass flowrate at the crankshaft exit for submersion depth (normalized by the crankshaft diameter).

IV. CONCLUSIONS

In this study, a simplified lubrication system (two-phase oil flow through crankshaft and helical channel) in a household hermetic reciprocating compressor is investigated. The effect of basic crankshaft parameters (rotational speed and submersion depth) and the oil viscosity on the mass flow rate at the crankshaft exit are reported. The transient variations of the mass flow rate at the crankshaft exit with time and the steady-state observations are shown for the selected parameters. It is shown that with increasing rotational speed and submersion depth, the oil flow rate increases. On the other hand, oil flow rate decreases when the oil viscosity increases.
The complex velocity field in the crankshaft inlet is reported in detail. The geometric modification of the crankshaft inlet is required so as to maximize of the oil flow rate.
This study demonstrates that governing parameters such as the rotational speed, oil viscosity and submersion depth must be chosen wisely to control the oil flow rate at the crankshaft exit.

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