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Analysis of Oil Pumping in a Reciprocating Compressor

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ABSTRACT

The immediate provision of lubricant oil to the radial and thrust bearings and to the piston-cylinder gap in reciprocating refrigeration compressors is a crucial reliability issue following compressor start-up. The aim of this paper is to present a hydrodynamic analysis of the lubricant oil pumping system of a reciprocating compressor using Computational Fluid Dynamics (CFD). In the present system, oil is pumped from the sump through a reed-type centrifugal pump located at the bottom end of the shaft. The oil flows initially as a climbing film on the internal surface of the shaft before it is directed to the external surface where it flows along a helix channel carved on the shaft wall. The model was implemented on the commercial package Fluent using the Volume of Fluid (VOF) Method to resolve the free-surface oil flow. Parameters regarding the oil behavior during start-up, such as the time required to reach steady-state operation and the associated oil mass flow rate are explored in the manuscript.

1. INTRODUCTION

Two-phase vapor-liquid flows are encountered in every component of the vapor compression refrigeration cycle. In the heat exchangers (condenser and evaporator) and in the expansion device, the two-phase flow occurs as a result of the phase change of the refrigerant from vapor to liquid and *vice-versa*. In the hermetic compressor, despite being in a superheated vapor state from suction to discharge, the refrigerant can be combined with the lubricant oil to produce a variety of two-phase flows in many parts of the compressor.

In addition to the more obvious role of lubrication, the oil performs many other tasks in the compressor. Among these one may cite cooling, sealing, protecting against corrosion, acting as a hydraulic fluid, reducing the noise level and maintaining low equalizing pressures during the off-cycle (Prata and Barbosa, 2008). Immediately after the electrical motor starts-up, as a means of avoiding the contact between the sliding parts which may result in excessive friction and wear, it is essential that the lubricant oil is delivered instantaneously to the radial and thrust bearings and to the piston-cylinder gap. Thus, the ideal oil pumping system design is that in which there is an optimum compromise between the time taken for the first oil particle to leave the sump and reach the bearings (henceforth referred to as the ‘climbing time’) and the oil flow rate needed for the specific dynamic loading conditions which the bearings may be subjected to.

Cost requirements impose that the oil pump should be simple and efficient and thus, in some household reciprocating compressors, use is made of the actual rotation of the shaft as the driving force to overcome gravity and friction as oil is forced through the helical groove that is machined on the shaft, to feed the shaft bearings. From the top of the shaft, oil is then expelled to the internal environment reaching the upper part of the compressor shell and falling back to the sump in the lower part. To properly design the pump and the channels and passages for a successful lubrication system, a detailed understanding of the oil flow is required, which is made more complex due to the necessity of addressing the issues of refrigerant out-gassing as well as the gas-liquid interface dynamics.

Only recently has the oil pumping problem in hermetic compressors been receiving more attention in the literature. The research is motivated by the increasing demand for more efficient lubrication systems to reduce energy consumption and improve reliability. Previous experimental work, such as those by Asanuma *et al.* (1984) and Drost and Quessada (1992), treated the problem with less emphasis on elaborate physical and mathematical models, relying on experiments and empirical models.

Computational Fluid Dynamics (CFD) has been applied successfully in the simulation of the oil pumping system of scroll compressors by many researchers. Bernardi (2000) studied the oil supply system of a scroll compressor using the Volume of Fluid (VOF) method to characterize the free-surface flow inside the channels of the oil pumping system. The conditions evaluated by Bernardi were such that the shaft speed was maintained at 60 Hz and the oil was a POE ISO 15. A deviation of -17% between the calculated and experimental oil flow rates at the top of the shaft was reported by the author. Cho *et al.* (2002) have conducted a CFD study of the oil flow in geometry very similar to that of Bernardi (2000) in which the effects of the shaft speed and the oil temperature have been investigated in some detail. The mean deviation between the experimental and numerical results was of the order of 15% . More recently, Cui (2004) investigated the effect of temperature, electric motor acceleration during the compressor start-up and the volume of oil in the compressor sump (carter) on the oil flow rate. Although absolute results of the oil flow rate have not been informed, the simulations indicated that the climbing time is inversely proportional to the volume of oil in the sump.

The objective of the present work is to perform a CFD analysis of the oil pumping system of a reciprocating compressor. In this system, the lower part of the shaft, which remains immersed in the oil sump, is connected to a centrifugal pump operated with a reed at its inlet. The reed, which rotates together with the shaft, provides the driving force for pumping the oil. The commercial CFD package Fluent has been used and the two-phase free-surface gas-liquid flow has been resolved through the VOF method. Of particular interest to the present analysis is the time required for the oil to reach the bearings once the compressor is started. Results are presented in terms of phase fraction contours representing the oil interface evolution as a function of time. The calculated oil flow rate has been compared with an experimental datum.

2. MODELING

2.1 Fundamentals of the VOF Method

The phase interface position calculation in the VOF method consists of evolving the material volume associated with each phase through each computational cell as a function of time and of reconstructing the interface at each subsequent time step according to the solution of the equations of motion. In the Fluent package, a solution algorithm similar to that presented by Scardovelli and Zaleski (1999) is employed and the reader is referred to their work for further details regarding the fundamentals of the VOF method.

The movement of the two-phase interface is described by the phase volume fraction equation given by

$$\frac{\partial \chi}{\partial t} + \vec{u} \cdot \nabla \chi = 0 \quad (1)$$

For an incompressible and isothermal flow, the equations of motion are given by

$$\nabla \cdot \vec{u} = 0 \quad (2)$$

$$\rho \frac{\partial \vec{u}}{\partial t} + \rho \vec{u} \cdot \nabla \vec{u} = -\nabla p + 2\nabla \cdot \mu \tilde{D} + 2\sigma \kappa \delta_s \hat{n} \quad (3)$$

where $\tilde{D}$ is the rate of deformation tensor. The last term on the RHS of Eq. (3) is due to surface tension and, in the literature, there are models specifically devised to deal with this term — for instance, the CSF (*Continuum Surface Force*) model (Fluent, 2005). In the VOF method, the physical properties, namely density and viscosity, are not continuous through the interface and are calculated respectively by the following expressions,

$$\rho = \rho_g \chi + \rho_l (1 - \chi) \quad (4)$$

$$\mu = \mu_g \chi + \mu_l (1 - \chi) \quad (5)$$

2.2 Geometry and Boundary Conditions

The geometries of the compressor and of the oil pumping system are illustrated in Figs. 1 and 2, respectively. The computational grid, as can be seen from Fig. 3, has been devised so that the pressure in the upper reservoir and the carter are equal. The grid is comprised of approximately 400,000 tetrahedral volumes whose sizes have been distributed so that the grid refinement is increased close to the solid walls and interfaces between adjacent domains.

Figure 1. Compressor geometry.

Figure 2. Geometry of the oil pumping system.

The boundary conditions have been established so as to allow the existence of three distinct domains: the oil sump or carter (static), the pump-shaft (moving) and the upper reservoir (static). In the course of the simulation, the oil which has been moved to the upper reservoir does not return to the carter, although the pressure between the two static domains is maintained the same. The boundary conditions for the mass and momentum equations were specified as follows: (i) no-slip between the oil and the bottom of the oil sump, (ii) prescribed pressure at the upper surface of the oil sump domain and at the base of the upper reservoir, (iii) angular velocity of the walls of the pump-shaft domain set equal to the speed of the electrical motor (moving-mesh domain), (iv) all remaining walls and interfaces between domains are stationary.

The initial condition is such that the oil is stationary prior to the start-up of the electric motor. The shaft has a constant speed from the time it is set into motion, i.e., its initial acceleration is infinite.

2.3 Interpolation and Coupling Schemes and Convergence Criteria

A second-order upwind scheme (Versteeg and Malalasekera, 1995) has been utilized for discretizing the equations of motion together with a body force weighted scheme for the pressure interpolation. The *PISO* (Pressure-Implicit with Splitting of Operators) algorithm (Issa, 1986) was used for the pressure-velocity coupling for it handles very well the small time steps which had to be employed in the present simulation. The convergence criteria for all velocities and mass were set at 10^{-4} RMS.

2.4 Conditions of the Simulation

The following simplifying assumptions have been adopted:

- There is no refrigerant dissolved in the oil;
- The flow is isothermal;
- The physical properties of the fluids (oil and refrigerant) are constant;
- The acceleration of the electrical motor at start-up is infinite;

v. The radial clearance of 250 μm in the middle part of the shaft, between the journal bearings, has been neglected (see Fig. 4).

Figure 3. The computational grid.

Figure 4. Detailed geometry of the shaft and feeding channels.

The conditions of the numerical simulation are as follows:

- a. Shaft speed: 3600 rpm (60 Hz);
- b. Fraction of the pump's tip that remains submerged in the oil sump: 15 mm;
- c. Working pressure: 1.2 bar;
- d. Working temperature: 75°C;
- e. Refrigerant: R134a;
- f. Lubricant oil: POE ISO 10;
- g. Surface tension: 0.012 N/m;
- h. Refrigerant density: 4.289 kg/m³;
- i. Refrigerant viscosity: 13.78 x 10⁻⁶ Pa.s;
- j. Oil density: 922.02 kg/m³;

k. Oil viscosity: 3.64×10^{-3} Pa.s.

Double precision has been utilized in all computations. The time step has been set at $25 \mu\text{s}$. The default constants of the $k\text{-}\varepsilon$ turbulence model provided by the commercial code have been maintained.

3. RESULTS

Figures 5.a-h show the evolution of the oil flow in the oil pumping system (volumetric fraction contours) at eight distinct instants along the first second following the electric motor start-up. Only the first two seconds (real time) of the oil pumping system operation were simulated. From the time evolution of the volumetric fraction contours, the regions with the most significant flow restrictions can be visualized.

Figure 5.a shows the initial condition of the problem, where the lubricant is stationary in the sump. As the pump-shaft starts moving and the oil flows into the reed pump, it starts its ascending motion in the inner part of the hollow shaft. After approximately 0.6 s, the lubricant reaches the inlet of the helix channel, where it starts feeding the radial bearings. As the lubricant reaches the top end of the shaft ($t \sim 0.8$ s), under the influence of its angular momentum (Figs. 5.g and 5.h), the oil is directed to the walls of the upper reservoir. In the real situation, the fraction of the oil flow reaching the upper region of the compressor is responsible for lubricating the piston-cylinder gap. The contact between the oil and the piston is made when the latter is at the bottom dead center and its lower portion becomes exposed to the internal environment (crankcase).

Figure 6 shows the volumetric flow rate of the oil as a function of time at different sections of the oil pumping system. The flow reaches steady-state within the first two seconds of operation. The simulation time for the first second of operation was 541 hours and for the whole two seconds was 1034 hours. Two parallel P4 3.0 MHz processors have been utilized in the simulation.

Figure 5. (continues on the next page).

Figure 5. Volumetric oil fraction as a function of time. (a) $t = 0$ s, (b) $t = 0.05$ s, (c) $t = 0.2$ s, (d) $t = 0.5$ s, (e) $t = 0.65$ s, (f) $t = 0.8$ s, (g) $t = 0.9$ s, (h) $t = 1.0$ s.

As can be seen from Fig. 6, the calculated volume flow rate at steady-state is significantly lower than the experimental datum obtained from the compressor manufacturer (Ribas, 2006). It is postulated that this discrepancy is due to the fact that the region with an increased radial clearance of $250\ \mu\text{m}$ on the external surface of the shaft has been neglected. In the simulations discussed above, we have opted for not considering the radial clearance region because this simplification resulted in a significant reduction of the number of grid elements and, consequently, of the computation time.

Figure 6. Oil flow rate as a function of time at several points of the pumping system.

As an attempt to verify the real effect of the radial clearance region on the oil flow rate, a supplementary CFD study was conducted in which only the clearance-helix channel region was considered (as highlighted in the right hand side sketch of Fig. 4). In this CFD simulation, the region between the helix channel inlet and the channel outlet at the top of the shaft (including the radial clearance region) has been discretized. At the channel inlet, the pressure has been set at the same value as the average liquid inside the reed pump (obtained from the complete simulation). At the top of the shaft, the pressure was set at 1.2 bar (crankcase pressure). The surface of the shaft (i.e., the inner surface of the channel) was set in motion at 3600 rpm (60 Hz) and the outer surface (representing the bearings) was held stationary.

Figure 7 presents the results from the supplementary simulation, in which the oil flow rate is plotted as a function of time for the helix channel inlet and outlet at the top of the shaft. As can be seen, after steady-state is achieved, the calculated flow rate becomes much closer to the experimental datum of 250 ± 20 ml/min, indicating that the radial clearance region between the bearings should not be disregarded in the CFD simulation.

Figure 7. Oil flow rate as a function of time (modified geometry).

5. CONCLUSIONS

This paper presented results from CFD simulations of the lubricant flow in the oil pumping system of a reciprocating compressor. The commercial CFD package Fluent was employed and the free-surface oil flow has been resolved with the VOF method. Important design parameters such as the ‘climbing time’ and the steady-state oil flow rate have been assessed. The following major conclusions can be drawn: (i) It is essential to be as realistic as possible in terms of the model geometry, as even apparently minor simplifications (such as neglecting the radial clearance region between the shaft bearings) can have a great impact on the calculated oil flow rate; (ii) The computational cost of a full CFD simulation including heat transfer and out-gassing effects is still prohibitive for these effects to be taken into account in design; (iii) More experimental data is needed in order to validate the models.

NOMENCLATURE

p	Pressure	(N/m ²)
t	Time	(s)
u	Velocity	(m/s)
χ	Volumetric phase fraction	(–)
μ_g	Gas dynamic viscosity	(Pa.s)
μ_l	Liquid dynamic viscosity	(Pa.s)
ρ_g	Gas mass density	(kg/m ³)
ρ_l	Liquid mass density	(kg/m ³)
σ	Surface tension	(N/m)

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