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Numerical and Experimental Examination for Oil Pump System Using a Simplified Uncoupled Simulation Model

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ABSTRACT

Compressors with variable capacity are employed in domestic applications where better performance, fine temperature control, low power consumption and low noise levels are required. This is achievable not only by the use of an electronic inverter drive capable of operating at different frequencies, but also by technology applied in mechanical, electrical, acoustic and thermodynamic subsystems, making possible to control its cooling capacity, fully meeting the product requirements.

One fundamental condition of operation is to ensure proper lubrication of mechanical components, for 100% of compressors, operating throughout its speed range, ensuring full operation throughout its lifetime. To do that, several methods are used to pump the oil towards the bearings. This task is easily performed when the compressor's crankshaft has high kinetic rotating energy. On the other hand, it becomes a challenge when compressor operates at lower speeds. In this case, computational techniques, using numerical methods in commercial simulators, have aided on designing oil pumping devices that maximizes the oil flow rate.

Through numerical and experimental techniques, this study aims to propose a simplified, uncoupled simulation model for the compressor oil supply system. The primary response variable is the volumetric oil flow rate of a helical pump, regarding the following analysis factors: a) the immersion depth and b) shaft rotation. Numerical results from uncoupled proposed model have shown good agreement with experimental data.

1. INTRODUCTION

Reciprocating compressors require oil supply on bearings to ensure proper lubrication and, consequently, saving energy consumption, avoiding mechanical losses and wear. In the last decades Computational Fluid Dynamics (CFD) has assisted on oil pump design of Variable Capacity Compressors (VCC) where oil pump must work in a large range of speeds, usually from 1000rpm to 4500rpm. This requires a large quantity of simulation and lab tests to efficiently design the oil supply system. Several CFD and experimental works has been done to provide an estimate of steady state oil flow rate and climbing time for specific oil supply system regarding the entire supply oil system (Luckmann *et al.*, 2008; Alves *et al.*, 2010; Alves *et al.*, 2012).

Wu *et al.* (2010) perform CFD and analytical analysis testing a reed centrifugal pump. Wu shows the reed pump will result in failure to pump oil to the oil system, when the compressor operates under low rotation (1000rpm). Kim *et al.* (2002) presents an analytical approximation for oil flow rate for a spiral groove by using an analogy with potential and resistance circuits, applied in each component of the oil supply system, comparing its results with numerical simulations. Kim shows his method, handling with several speeds, is efficient to predict oil flow rate having reasonable agreement with experimental tests. However, for each oil pump design it is necessary to have a respective analytical solution. According to his method, by analytics expressions, is it possible to quantify the influence of all elements on the supply system. More recently, Alves *et al.* (2010) present analytical and CFD solutions for a helical pump, comparing results with experimental data with good agreement. His simulation of the overall system, performed on a Pentium D-930, takes 336 hours. He also presents a semi-analytical method taking few seconds to predict oil flow rate, with maximum error of order of 12%, however some important simplification on semi-analytical method, mainly on low speed, had been assumed, for example, the gap between pin and sleeve pump not being taken in account.

The Present work aims to propose a simple, uncoupled CFD model for reciprocating compressor oil supply system, aiming to predict the oil flow rate efficiently over the lower VCC compressors speed range. Figure 1 shows a typical reciprocating compressor's oil supply system. The analysis has been performed experimentally on entire system and numerically on component highlighted, called "oil pump" in an uncoupled way, i.e., instead to simulate all oil supply system, the model takes in account only the components 5a and 5b of a helical pump as shown on Figure 1. Experimental verification has been made with lab tests detailed on Section 3. Primary response variable is the volumetric oil flow rate respect to the following analysis factors: a) the immersion depth and; b) shaft rotation. The immersion is eligible for varying the hydrostatic pressure on inlet pump, being a factor for evaluating the proposed numerical boundary condition detailed on Section 2.3.

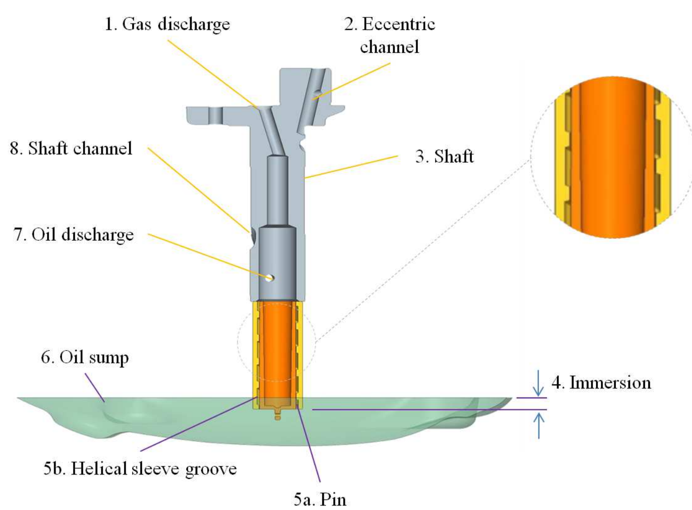

Figure 1: Oil supply system and helical oil pump detail.

2. NUMERICAL SIMULATION

Main assumptions on the mathematical model are: a) isothermal; b) constant density and viscosity; c) no interfacial tension; d) laminar flow; e) uncoupled oil pump with respect to oil pump system. The geometry of oil system evaluated is able to pump at low and high speeds, however present work is dedicated to low speeds. Thus by hypothesis it is assumed an "infinite conductivity" in all remaining system, i.e., the shaft channel (8) and eccentric channel (2) resistance is neglected and system performance is defined by oil pump.

2.1 Fundamentals of the mathematical model

The CFD results were obtained by using a laminar, homogeneous, free surface model on Ansys-CFX®. According to Ansys® help system (2013a), the method used to solve fundamental equations are the volume of fluid (VOF) method. A free surface refers to an interface between a gas and a liquid, where the difference in the densities between the two is quite large. Due to the low density, the inertia of the gas is usually negligible, so the only influence of the gas is the pressure takes place at the interface. Hence, the gas region do not need to be modeled, and the free surface is simply modeled as a boundary with constant pressure. The VOF method determines the shape and location of free surface based on the concept of fluid volume fraction χ. In general, the evolution of the free surface is computed either through a VOF advection algorithm or through the following equation:

$$\frac{\partial \chi}{\partial t} + \mathbf{u} \cdot \nabla \chi = 0 \tag{1}$$

In the homogeneous model, a common flow field is shared by all fluids and the transported quantities φ_α, except volume fractions, are the same for all phases, that is, $\varphi_\alpha = \varphi$ for $1 \le \alpha \le N_\alpha$. The bulk transport equations can be derived by summing the individual phase transport equations over all phases to give a single transport equation for φ:

$$\frac{\partial}{\partial t}(\rho\varphi) + \nabla \cdot (\rho \mathbf{u} \varphi - \Gamma \nabla \varphi) = S \tag{2}$$

where $\varphi = 1,\ \Gamma = 0,\ S = 0$ recover the mass balance equation and $\varphi = \mathbf{u},\ \Gamma_\varphi = \mu,\ S_\varphi = -\nabla p$ recover the laminar momentum equation. In addition the homogeneous model uses the following mixtures variables:

$$\rho = \sum_{\alpha=1}^{N_\alpha} \chi_\alpha \rho_\alpha, \quad \mathbf{u} = \sum_{\alpha=1}^{N_\alpha} \chi_\alpha \rho_\alpha \mathbf{u}_\alpha, \quad \Gamma = \sum_{\alpha=1}^{N_\alpha} \chi_\alpha \Gamma_\alpha . \tag{3}$$

Therefore, the system of equations is made up by primary five variables u, v, w, P and χ. The governing equations are (1) and (2) resulting on five equations. In the present work, the flow is air-oil and the one of volume fractions can be computed from restriction volume equation $\chi_{gas} = 1 - \chi_{oil}$. Details on modeling VOF and homogeneous model can be found on Ansys® help system (2013a). Barbosa et al. (2008) has also provided details about governing equations and VOF solution method.

2.3 Domain and boundary conditions

Figure 2 shows the proposed rotating domain at a rate of **Ω** [rad/s]. The volume domain is where fluid can flow, being the oil pump "solid's negative", as illustrated on Figure 1.

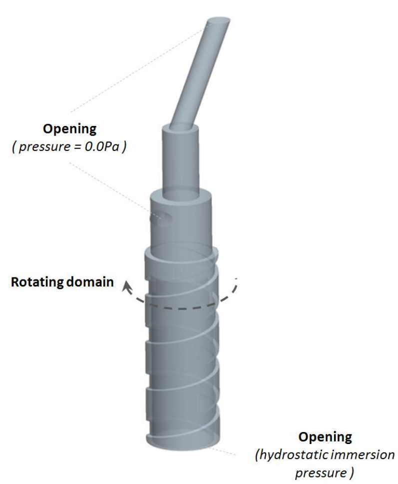

Figure 2: Helical pump – solutions domain.

The center pin – component 5a on Figure 1 – is modeled as a cylindrical wall inside the pump defined as a counter rotating wall having opposite rotation -Ω [rad/s]. The bottom face inlet is defined as an opening with hydrostatic pressure prescribed, i.e., pressure relative to immersion height. The oil volume fraction defined as 1. Discharge side outlet is defined also as an opening with 0Pa prescribed pressure and volume fraction to be calculated considering zero gradients.

Similarly the top gas outlet is stated as opening with pressure 0Pa and gas volume fraction equal to 1. All other faces are defined as no slip walls. Interpolation was defined as pure upwind and residual problem convergence is achieved when all conservation equations reaches 1.0e-4 residual. Inlet flow rate and outlet flow rate is monitored and steady state is achieved when the inlet/outlet volume balance is smaller than 1%. Mesh has about one million of elements to capture efficiently high near wall velocity gradients.

3. EXPERIMENTAL VERIFICATION

Experimental lab tests were performed to validate the models employed in the simulation. Figure 3 shows the bench test with the following components: 1) compressor kit (crankcase, motor, crankshaft and oil pump); 2) oil colleting tube; 3) graduated cylinder; 4) oil; 5) heater; 6) oil reservoir; 7) thermocouple and; 8) oil pump. Some additional components used on the experiment are not shown in this figure.

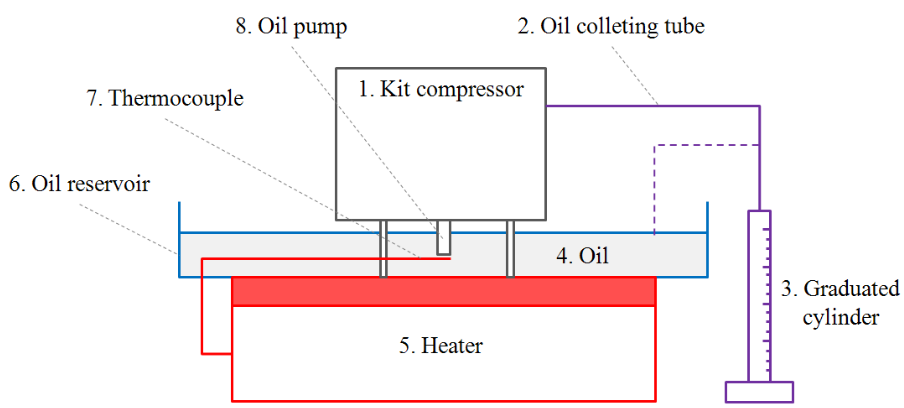

Figure 3: Oil flow rate bench test.

The mechanical kit compressor (1) has its rotating speed controlled by a frequency inverter in such a way is it possible to operate in a wide speed range with around 5rpm uncertainty. The oil pumped by the oil supply system is collected directly to a graduated cylinder – with 1ml of resolution – when measuring is being performed; otherwise oil flow returns to reservoir, both ways transported by colleting tube (2). Immersion pump is controlled by marks on oil pump matching with oil reservoir height.

Before measuring, the entire system must be in thermal equilibrium. The thermocouple located in a vicinity of the oil pump registers temperature continuously, with an uncertainty around 1°C, and the heater has an automatic system control that always tries to keep temperature in a given magnitude. All tests were performed at (50 ± 1)°C, once system achieves equilibrium. In this work, it is adopted a simple approximation to calculate the uncertainties, considering only the repeatability of the entire system for each setup test, neglecting all other uncertainties acting on measure system.

4. TEST PLAN

Both numerical and experimental tests will be performed according to the present plan. As previously stated, the output variable will be the oil flow rate. The input variables are the pump's immersion and speed rotation. It must be emphasized in present analysis the pump's height and all other geometric pump's variable like pitch, channel depth etc. were kept constant. It is not objective of this work to quantify oil flow rate variation with respect to these geometric variables. Results presented here are supposed valid for a generic helical pump, independent of its intrinsic geometric parameters.

Non-dimensional rotating speeds, both numerical and experimental, are evaluated with respect to its maximum value. The experimental tests were performed according to Table 1. For each test setup, nine oil flow rate measurements were carried out, calculating average, standard deviation and uncertainties with 90% of confidence interval. The numerical tests were performed according to the variables levels as shown on Table 2. The levels were combined into enhanced centered face central composite Design Of Experiment (DOE), totalizing seventeen deterministic test simulations performed on Ansys-CFX® in order building a response surface with determination coefficient closer to 1.0 and maximum absolute error closer to 0.0. Details about the probabilistic method can be found on reference Ansys® help system (2013b).

Table 1: Experimental lab test setup variables.

Test setup	Speed [-]	Height [mm]
1	0.5	3.0
2	0.5	11.5
3	0.5	20.0
4	1.0	3.0
5	1.0	11.5
6	1.0	20.0

Table 2: Numerical variables levels.

Level	Speed [-]	Height [mm]
1.0	0.5	3.0
2.0	1.0	20.0

As previously commented, for each setup test, nine measurements were performed and average, standard deviation and uncertainty were calculated considering a Student t-distribution. The "t" value employed was 1.83 once the interval confidence regard is 90%. The uncertainties on dimensional measuring system were neglected for being considered small.

5. RESULTS AND DISCUSSIONS

As remarked, experimental lab tests were performed with entire oil supply system and compared with uncoupled simulation oil pump results. Figure 4a shows numerical and experimental oil flow varying with rotating and discharge height. All experimental results overlap each other, showing the minimal influence of immersion pump on oil flow rate. The delimited gray area shown in Figure 4b represents estimatives for experimental uncertainties, with 90% of confidence interval, along rotation and immersion pump range. Numerical results were gathering from a standard response surface – full 2nd order polynomials – with determination coefficient equals to 0.99977 and maximum absolute error equals to 2.45. Table 3 summarizes experimental results and uncertainties calculation. The numerical uncoupled simulation model shows good agreement with experimental results and higher sensibility for immersion pump as discussed below.

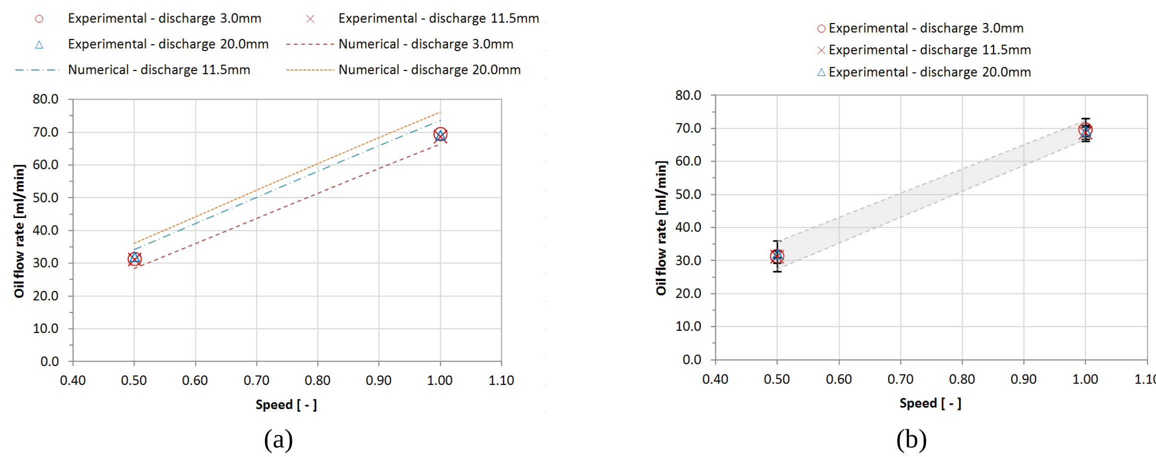

Figure 4: Oil flow rate over dimensionless rotation and discharge height: (a) experimental and numerical results; b) experimental uncertainties and interval of confidence for oil flow rate varying with rotation and immersion.

Perhaps the main reason of the numerical sensitivity to immersion height is relative to boundary condition at both inlet and outlet oil pump region. The outlet neglects any resistance along oil supply system and inlet considers only hydrostatic pressure. Despite relative sensitivity respect to immersion as shown on Figure 2a, numerical curve was achieved with seventeen numerical simulations of oil pump, having grid with around one million of elements and taking about two hours each simulation on Intel Core i7-3720. Taken the need of several simulations when oil supply system is designed for VCC compressors, the proposed uncoupled model may be a good approach to assist on designing the oil pump.

Table 3: Summary of experimental results.

Test setup	Speed [-]	Height [mm]	Average oil flow [ml/min]	Standard deviation [ml/min]	Uncertainty [ml/min]
1	0.5	3.0	31.333	2.500	4.583
2	0.5	11.5	31.111	1.042	1.909
3	0.5	20.0	32.111	0.551	1.010
4	1.0	3.0	69.500	1.875	3.437
5	1.0	11.5	68.611	1.042	1.909
6	1.0	20.0	69.222	0.908	1.665

6. CONCLUSIONS

The Present work evaluated an uncoupled simulation model proposed by varying immersion's pump height and compressor speed, comparing results with experimental tests. The CFD model did not consider the oil supply system as a whole, but only the oil pump. The following conclusions can be stated:

- Simplified uncoupled numerical proposed model shows good agreement with experimental data and efficient computational effort;
- Immersion shows minimal effects on experimental oil flow rate and small sensibility on numerical results.

NOMENCLATURE

Roman

N	number of phase	(–)
u	vector mixture velocity	(m/s)
N	number	(–)
S	0, minus pressure gradient	(–, Pa/m)
h	dimensionless discharge height	(–)

Greek

χ	volume fraction	(–)
Ω	angular velocity	(rad/s)
ρ	density	(kg/m^3)
φ	unit, mixture velocity	(–, m/s)
Γ	null, viscosity	(–, Pa s)

Subscript

α	phase oil or gas

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