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CFD SIMULATION OF A SCROLL COMPRESSOR
OIL PUMPING SYSTEM

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ABSTRACT

A CFD (Computational Fluid Dynamics) analysis of the oil pumping function of a scroll compressor is proposed where the internal oil free surface shape is determined. The numerical simulation is validated by laboratory tests on a dedicated test rig. The results demonstrate how powerful this approach can be for a physical understanding of the lubrication process: more than a design tool, the CFD model proves to be a visualisation tool that increases understanding of free surface shape inside the oil pump and the crankshaft.

INTRODUCTION

For design engineers, journal bearing lubrication has always appeared to be a major problem in rotating machinery. The two main reasons are that:

1. It is a complex experimental task to determine the required amount of oil necessary to feed any new bearing to avoid seizing.

2. Building an analytical model of the oil flow rates delivered to the bearings gives rough estimates that can only be used for pre-dimensioning the oil supply system (Asanuma et al.[1], Drost et al.[2], Itoh et al.[3]). It is nearly impossible to clearly understand what local mechanisms affect lubrication.

A new way to answer the last point is CFD simulation. The goal of the present study is to understand and to quantify how the bearings are fed with oil for a new scroll compressor oil pumping system without building a laboratory prototype.

Data like oil velocity, pumping head and pressure losses are found at any given point in the model which makes the oil pumping function easy to optimise in a short time.

A complete study is proposed from the CAD design to the post-processed results which suggests how this method can be used to compare various crankshaft and oil pump geometries with the aim of improving bearings lubrication.
1. BUILDING THE MODEL FROM CAD TO CFD

As everything begins with CAD, we first have designed the mechanical parts with Pro-Engineer application (crankshaft, oil pickup tube, baffle). The next step was to assemble these parts and to extract the inner volume which is actually the fluid domain - figures 1 and 2 illustrate the CAD inputs and outputs. Once the fluid domain model has been built, it is possible to mesh it and export the mesh to a CFD solver. The CFD package that has been used is STAR-CD which is a leading code on its market - figure 3 is a close view of the mesh directly obtained from the internal mesh capabilities.

2. CFD MODEL DESCRIPTION

Simulations are based on several choices and assumptions. First of all, the internal oil flow is assumed to be laminar and the oil free surface deformation during the transient flow is simulated from compressor start. The latter choice leads to a two phase flow calculation with interface (Volume Of Fluid technique) in a rotating reference frame. No exchange between gas and oil at the interface is considered (out gassing, solubility). The thermal exchanges are also neglected so that viscosities and densities are kept constant all along the runs.

The following comments refer to figure 4. Static pressures are set as boundaries at the several orifices of the model. At the inlet of the pickup tube, the pressure (PIN) equals the hydrodynamic pressure, that corresponds to the oil level in the compressor (typically 900 Pa). At the lower and the upper bearings a back pressure is applied (PLB and PUB) which is equal to the bearings pressure loss. A model of such a boundary is given by Drost et al. [2] but it is not applicable to any bearing geometry. Thus this model could be viewed as a first order of magnitude that has to be redefined in the experimental validation phase. In the present case the back pressure can be considered equal for the two bearings LB and UB (same dimensions, same clearances).

The orbiting scroll bearing orifice (OSB) as well as the venting hole outlet are set to atmospheric pressure as is the case in the validation rig.

All the simulations have been carried out for a 60 Hz rotational frequency and with a ISO15 Polyol Ester oil (heavy fluid). The light fluid is air.
3. NUMERICAL RESULTS

The CPU time required to reach a steady state flow rate at the orbiting scroll bearing (OSB) is roughly one week on a bi-processor Sun Ultra-II work station.

Oil Free Surface Shape :

An overall view of the oil free surface can be seen in figure 5. The flow conditions correspond to a steady state.

A parabola shape is predicted inside the pickup tube, as it is described in references ([1], [2] and [3]), which is locked to the baffle. When the oil enters the main channel, the parabolic branch is centrifuged so that a thin film crawls up to the top of the crankshaft. Three section slices enable the oil film thickness to be controlled along the main channel.

Flow rates at the bearings :

Figure 6 shows the transient oil flow rates at each bearing. For a better understanding, the data are divided by their respective steady state values.

One can firstly see that, even if the oil reaches the various orifices quickly, it nevertheless takes about 1 second to reach a constant feeding of the bearings.

At a closer look, there is a rush of oil inside the lower bearing and the upper bearing orifices during a short time (100 ms) which is followed by a dramatic drop (100 ms) and by a smooth increase until a steady state is reached. The rush is due to the centrifugal force (radial component): once the orifice is full, the oil flows out and the vertical flow velocity magnitude increases, the oil is thus forced to the top of the crankshaft and the orifice is then emptied. As the oil continues to rise, the oil film continuously thickens at the inlet of the orifice. As a consequence the oil flow rate increases steadily until it reaches an equilibrium.

The orbiting scroll bearing is only fed by ascendant oil flow: this rush phenomenon does not exist at all. There is a delayed oil delivery which quickly reaches a steady state.

As the time for oil to reach the bearings is a key factor for compressor life time (mainly in start and stop running conditions), one can understand from this close analysis how it is possible to change any geometrical parameter of the oil pumping system and to extrapolate what can be the impact on the bearings life time.
4. EXPERIMENTAL VALIDATION

The experimental validation was carried out on a dedicated test rig similar to the one used by Drost et al. [2]:

When the test compressor is turned on, the oil reaches the top of the crankshaft and flows out by the orifice OSB to the oil receiver. The oil flow rate can then easily be measured. The comparison between experiments and simulation is based only on the oil flow rate that reaches the orbiting scroll journal bearing (Q_{OSB}):

<table>
<thead>
<tr>
<th>@60 Hz</th>
<th>( Q_{LB} )</th>
<th>( Q_{UB} )</th>
<th>( Q_{OSB} )</th>
<th>( Q_{total} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured oil flow rate (g/s)</td>
<td>Not measured</td>
<td>Not measured</td>
<td>40.3</td>
<td>Not measured</td>
</tr>
<tr>
<td>Simulated oil flow rate (g/s)</td>
<td>0.3</td>
<td>0.5</td>
<td>33.5</td>
<td>34.3</td>
</tr>
<tr>
<td>(% of ( Q_{total} ))</td>
<td>(0.9%)</td>
<td>(1.5%)</td>
<td>(97.6%)</td>
<td>(100%)</td>
</tr>
</tbody>
</table>

The simulation accuracy is 17% on \( Q_{OSB} \). This value could certainly be reduced by mesh improvement and back pressures adjustments at the boundaries; yet, it can be considered as very good for such a tool.
CONCLUSION

This article opens a new way of investigating lubrication problems. The new free surface capabilities of CFD commercial packages allow design engineers to investigate lubricant paths in detail.

The proposed study focused on the validation of this approach and showed how transients can be investigated locally. A next step will be to validate precisely the bearings back pressure models. As a result, this leads to a deeper understanding of how oil enters the bearings for fixed or variable speed application.

ACKNOWLEDGEMENTS

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REFERENCES


FIGURES

Fig. 1 - Mechanical parts (CAD input)

Fig. 2 - Fluid domain (CAD output)

Fig. 3 - Pickup tube Mesh (detailed view)
PIN = pressure at inlet
PLB = pressure allowor bearing
PVH = pressure al vonUng hole
PUB = pressure at upper bearing
POSB = pressure at orbiting scroll

PIN = Patm + 1100 Pa

Fig. 4 - Model Boundaries

Fig. 5 - Oil free surface shape

Fig. 6 - Transient flow rates at the bearings