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3D CFD ANALYSIS OF AN OIL INJECTED TWIN SCREW EXPANDER

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ABSTRACT

Small scale Organic Rankine Cycle (ORC) systems have a big potential for waste heat recovery in the market. Due to the smaller volume flows inside these systems, non-conventional expansion technologies such as screw expanders become more interesting. Recent economic studies have shown the important role of screw machines in such cycles. However, in order to get a better understanding of the expansion behaviour in an ORC, appropriate simulation models of screw expanders are necessary. The flow inside an oil-injected twin screw expander is modeled in detail with 3D CFD (Computational Fluid Dynamics) calculations. These simulations are challenging because of the deforming domain and the narrow gaps between the screws or between a screw and the casing. The deforming mesh motion is handled by an in-house code which generates a block-structured grid with the help of the solutions of the Laplace problem. The oil-phase was modeled with an Eulerian multiphase model and the working fluid is treated compressible. The performance of the screw expander is strongly affected by the oil-injection which provides lubrication and a better sealing of the gaps. Therefore, the different types of leakages inside the screw expander are studied and monitored. As the result of the simulations, knowledge about the flow process and the losses inside the oil-injected screw expander is built up.

Keywords: expander, twin screw, oil-injection, Organic Rankine Cycle

NOMENCLATURE

n number of blocks
nr number of radial nodes
nt number of tangential nodes
nax number of grids in the axial direction

INTRODUCTION

The most widely used solution for generating power from low-grade heat is the Organic Rankine Cycle (ORC) [1]. Thermodynamic machines for the expansion of the organic fluids are the key element for the power generation in the ORC. Two main types of such machines are turbo and positive displacement types. The screw expander is a special type of rotary displacement machine, in which a number of screws rotate inside two overlapping cylinders.

The current generation of screw expanders is in fact designed as compressors used the other way around [2–4]. Though they exist already long time, expander technology for small power ranges is still under development. Continuing demand for further improvements has led to the need of 3D CFD analysis of the flow inside the screw expander.

The starting point for the study in this paper, is the twin screw compressor operating in reverse mode. The twin screw compressor consists of a male and female rotor. During both compression and expansion, the spaces between the screws form working chambers in which gas or vapour is trapped.

When a compressor is operated as an expander, so in reverse

mode, the rotors are rotating in the opposite direction and the inlet port becomes the outlet port and vice versa. When fluid is expanded it will affect the rotors and power will be generated through the shaft. Although the operation looks rather similar, some differences can be remarked immediately, especially in the filling process. When the rotors rotate, a volume is formed in which the fluid is filled and expanded. This volume grows for a certain time. Therefore, the inlet port should be designed in such a way that filling occurs only during a small part of the growing phase of the volume. Once the volume is disconnected from the inlet, the volume increases until the maximum volume is reached. Consequently, the inlet port determines the built-in expansion ratio. Since the expander has a built-in pressure ratio, the outlet pressure can be higher or lower than that of the expanded fluid (over or under expansion can occur), which leads to flow losses.

Another type of losses are leakage flows in the clearances between the rotors and between the rotors and the housing. Clearances between the rotating screws exist in order to compensate machining tolerance, thermal expansion and force deflection. In order to lubricate the rotor motion and seal the gaps, oil is injected to the compression/expansion chambers. Advantages of using oil injected machines are cheap screw elements (no need for a sophisticated sealing or special rotor coating) and no need for synchronizing gears (as the direct contact between the rotors is avoided).

Computer models for oil injected screw compressors with gas flow, heat transfer and inlet and outlet port losses are studied in [5–10] and for scroll expanders in [11]. Influence of the oil injection upon the screw compressor working process with different parameters that were varied (oil flow rate, inlet temperature, droplet atomization, position of the oil in the casing at which the oil was injected, etc) were studied in [12]. In all these mathematical models, the internal flow is approximated by the changes in different parameters. Moreover, the experimental flow analysis is hampered by the complex geometry of such machines. Therefore, the numerical flow calculations could prove to be very useful for detailed analysis of the internal flow inside the twin screw expander.

In this paper, a complete 3D CFD calculation for an oil injected twin screw expander is performed. To simulate the two phase flow (working fluid with the oil), a multiphase model was used. In order to calculate the leakage flows with high accuracy, excellent grid resolution in the gap region was preserved. Each of these aspects is explained further and results of the calculation are presented.

CFD ANALYSIS

The calculations were performed for an oil injected twin screw expander. The geometry is shown in Figure 1. There are four male lobes and six female lobes with unsymmetrical rotor profiles. Design parameters are listed in Table 1.
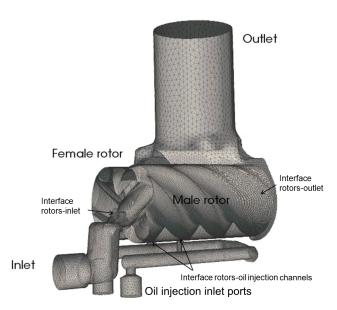

FIGURE 1: 3D GRID FOR THE OIL INJECTED TWIN SCREW EXPANDER

Grid generation for 3D CFD calculations of the screw expander

In the rotary positive-displacement machines, the volume of a working chamber is continuously changing. As a consequence of a rotor rotation, the boundaries of the flow domain move in a very complex manner.

To facilitate the grid generation for the numerical simulation of 3D flow processes, the domain is decomposed into four blocks, namely inlet, outlet, oil-injection channel and the casing (see Fig. 1).

The grid blocks in the inlet and outlet port, and oil injection channel are stationary, since the only moving boundaries to the flow domain are the rotors surfaces. The number of cells for the inlet zone is 81505, for the outlet zone 168237 and for the oil injection channels zone 104001.

The grid in the casing is built by stacking two-dimensional structured quadrilateral grids in slices of the casing. Grid cells are defined by the same faces and nodes during the entire simulation, what means that the grid nodes move (ALE, Arbitrary Langrangian-Eulerian method). To generate the two-dimensional grids, a grid generation algorithm developed in [13, 14] was used. The grid generator only creates the grid in the casing of the machine. To reduce the calculation time, it is preferable to generate a set of grids before the actual flow calculationis started. Therefore, a set of 60 structured grids is generated, one for each degree rotation of the female rotor. For each time step in the calculation, a new grid can be constructed by interpolating between two supplied grids.

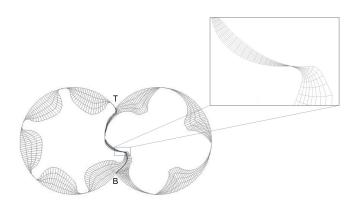The size of the various gaps between the rotors and between

FIGURE 2: 2D GRID IN A SLICE OF THE TWIN SCREW EXPANDER WITH DEFINITION OF THE DIVISION LINE [14]

Design parameters and operating conditions	
Outer diameter of the rotors	69.29 mm
Length of the rotors	128.5 mm
Rotational speed (male)	6000 rpm
Rotational speed (female)	4000 rpm
Inlet pressure	9 bar
Outlet pressure	1 bar
Pressure at oil injection ports	6 bar
Volume fraction of oil at injection ports	0.8

TABLE 1: Design parameters and boundary conditions

the rotors and the casing are extremely small (order tens of microns versus rotors diameters of about 70 mm). For the high numerical accuracy, the grid has to be of high quality in the whole domain and in every rotational position. A slice of the 2D structured grid with a zoom on the resolution in the gap region is shown in Fig. 2. The division line shown in Fig. 2 starts and ends in the cusps (B-bottom cusp and T-top cusp).

The total number of nodes in the casing is $N = n * nr * nt * nax = 2 * 4 * 285 * 250 = 570000$ cells.

All blocks are joined together by using sliding interfaces between the inlet, outlet, the oil injection channel and the casing. The complete 3D mesh of the twin screw expander consists of 923743 cells.

Boundary conditions

The boundary conditions are presented in the Table 1. The fluid, which expands within the screw expander, is air described by the equation of state for an ideal gas. Mineral oil with the ISO grade of 46 is used for the lubrication. The dependence of the oil viscosity on temperature is predicted with the ASTM D341 standard [15] which is based on Walther's equation [16] . At the inlets (inlet port and two oil injection ports) and the outlet, standard pressure boundary conditions were used.

Mutiphase modeling and Solver

The governing differential equations for the fluid flow inside the screw expander geometry are solved by a commercial package with use of user-defined functions to handle the grid movement and multiphase flow.

This mathematical model consists of set of momentum, energy and mass conservation equations, which are accompanied by the equation of state and a turbulence model. The $k - \varepsilon$ model with the standard wall functions was used [17]. To handle the

relative motion of the rotors inside the casing, the rotor zone is connected with the inlet, outlet and oil injection channels by non-conformal interfaces.

For modelling the oil injection into the chambers, both fluids are treated as separate but interacting phases. Because of that, an additional set of mass, momentum and energy conservation equations for the second phase is incorporated and a volume fraction equation is solved. Size of the oil droplets ranged from 10^{-8} to 0.1 mm.

The heat transfer between the phases is modelled with the correlation of Ranz and Marshall [18, 19].

The spatial discretization that is used for the calculations is first-order upwind. The temporal discretization is first-order implicit. The solver calculates the pressure and velocity simultaneously.

PERFORMANCE ANALYSIS

In this section, results of the CFD calculations are presented. Contours of the pressure inside the screw expander are shown in Fig. 4. It is possible to clearly distinguish the different working chambers formed by the rotor lobes.

Mass flow rates at the inlet, outlet and the oil injection ports are plotted over one complete revolution of the oil injected screw expander and are shown in Fig. 3. A negative value in Fig. 3 represents the mass flow rate that is going out of the expander.

For efficient sealing, parameters like the amount of oil and the location of the oil injection ports should be considered and chosen well. The volume fraction of oil on the wall of the male rotor and oil injection channels is shown in Fig. 5. Locations with a high amount of oil can be clearly distinguished. Furthermore, it is shown that the high concentration of oil is high in the regions of the clearances (sealing and tip clearances in Fig. 5). Oil is mainly concentrated in the gaps between the rotor and the housing and between the meshing rotors.
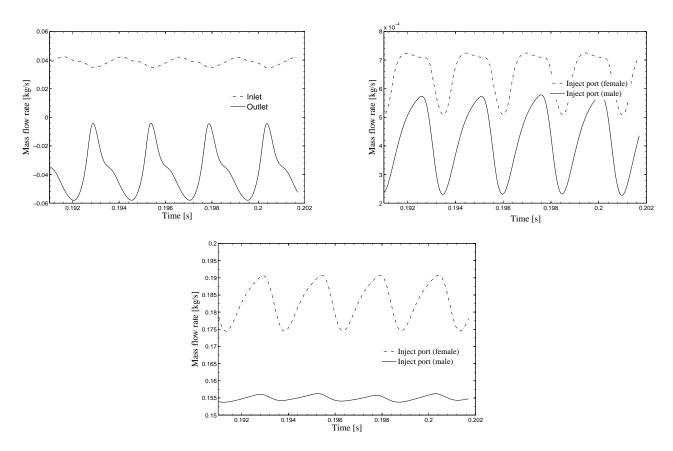
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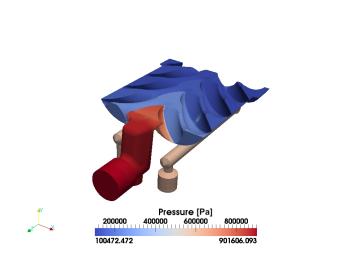**FIGURE 3**: MASS FLOW RATES AT THE INLET, OUTLET AND OIL INJECTION PORTS FOR THE AIR (TOP) AND THE OIL (BOTTOM)
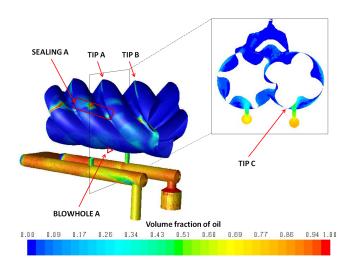

FIGURE 4: PRESSURE INSIDE THE SCREW EXPANDER

FIGURE 5: DISTRIBUTION OF OIL THROUGH THE SCREW EXPANDER
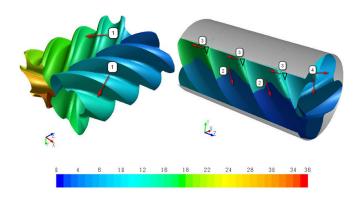
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FIGURE 6: CHAMBER NUMBER CONFIGURATION WITH 4 TYPES OF THE LEAKAGES

LEAKAGES

In this paper special focus is on a study of mass flow rates through the clearances which are forming leakage paths. These leakage flows have a very large impact on the expander's efficiency. It is paramount that these leakage flows are assessed correctly. While the impact of these leakage paths on the expander performance is studied in [11, 20], analysis of the flow inside the gaps with monitoring the time dependent change in for example mass flow rate, are lacking in the published literature. Therefore, it is important to have a model which can monitor different fluxes through every clearance that is formed during the operation. In the screw compressor/expander there are four types of leakage paths. It is possible to identify every one of them as it is shown in Fig. 6 with further explanations:

1. Clearances between the rotor **tips** and the housing allow flow from one chamber to another one which is at a lower pressure (number 1 in Fig. 6);
2. Between the meshing rotors, a **sealing** line is formed, where leaking occurs (number 2 in Fig. 6). The sealing line can be defined as the length along which the rotors seal the fluid between high and low pressure areas;
3. **Blowholes** are triangular gaps that are formed at the cusps between two rotors and the housing (number 3 in Fig. 6). Because of manufacturing reasons, it is impossible to have a sealing line that reaches to the cusp of the housing. As a consequence, when looking at a 2D cross section at the end of the sealing line, a triangular region (blowhole) is formed between this end and the nearest positions of the tip clearances between both rotors and the housing. Blowholes are essentially defined by the rotor profile geometry [21].
4. Leaks through the clearances at the **end planes** occur both on the inlet and outlet side (number 4 in Fig. 6);

Chamber numbering and mass flow rates for all leakages

The use of structured rectangular grids has two advantages. Firstly, in a structured grid, it is possible to store the entire grid in a matrix. A second advantage is that the grid can be aligned with the flow field in the gaps between rotor and the rotors and the housing. [14]

If the positions of all cells are known, it is possible to detect locations with the leakage paths. For that a few assumptions were made which ensure detection of all leakages:

1) Chamber numbering

    (a) male and female parts of the grid in the casing are divided by the division line shown on the Fig. 2
    (b) the grid for male and female parts are numbered (odd numbers for female, even for male)
    (c) numbers are rising from the inlet to the outlet port

2) Location of leakage paths

    (a) leakage between the tip and the housing occurs between chambers of either the male or female rotor (two odd or two even numbers) and it is not on the division line
    (b) sealing leakage occurs between the meshing rotors on the sealing line
    (c) blowhole leakage is detected in the cusp, between male and female cells

With respect to the previous assumptions, locations of the leakage paths were defined after every time step. Fluxes through them were monitored for both the air and oil phase. In this paper, leakages at the end planes were not studied.

From the results of the 3D CFD calculation, mass flow rates through all leakage paths and for one time step are shown in Fig. 7. For every leakage that occurs in the screw expander it is possible to read the location (combination of chamber numbers) and mass flow rate that is going through. Few conclusions can be drawn from the results:

- the leaking of the air through the rotor tips and the housing is higher on the high pressure side (described with the small chamber numbers, Fig. 7.a)
- in the blowhole region, oil mass flow rates are low what indicates that oil lubrication is less sufficient (Fig. 7.c)

With respect to Fig. 5, it is possible to determine exact locations of the leakages in Fig. 7. For example, if the TIP A from Fig. 5 is a leakage between the rotor tip and the housing (chamber numbers 10 and 12 in Fig. 6), the exact mass flow rate can be read from Fig. 5 and the other way around. To determine the other types of leakages, the same principle is used and examples are shown in Fig. 5 and Fig. 7.
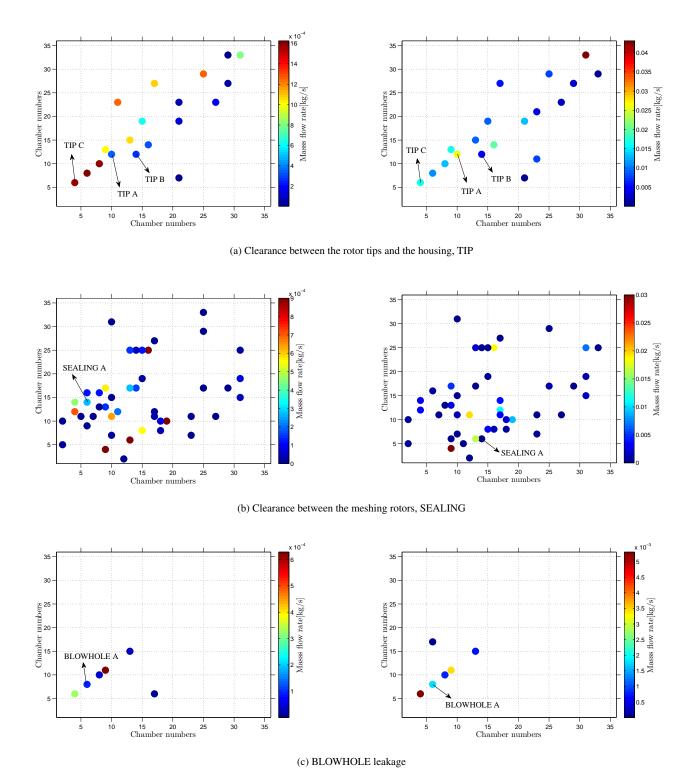
(a) Clearance between the rotor tips and the housing, TIP

(b) Clearance between the meshing rotors, SEALING

(c) BLOWHOLE leakage

FIGURE 7: MASS FLOW RATES THROUGH THE CLEARANCES. LEFT-AIR , RIGHT-OIL

DISTRIBUTION OF THE OIL IN THE OIL INJECTED EX-PANDER

Fig. 8 shows the volume fraction of the oil in the oil injected expander in different time steps. The location of the oil injection ports is the same as in the set up for the oil injected compressor. The mass flow rates of the oil through the oil injection ports are shown on the Fig. 3(bottom). These amounts are relatively similar to ones that are used for an oil injected compressor. Under these conditions, the results show that an area with a high fraction of oil is formed near the outlet during every cycle. In this area, the fraction of oil rises to very high levels. When the oil is trapped in a volume that is decreasing, high pressures and temperatures occur. This can cause an overheating of the oil which can lead to decrease in the oil viscosity and insufficient lubrication, but also reduction of power output. This result, highlights two things that should be changed when changing from oil injected compressor to oil injected expander. First, the location of the oil injection ports should be moved closer to the inlet port. Second, the amount of the oil should be significantly smaller in the oil injected expander than in the oil injected compressor.

CONCLUSION

With growing demand for use of the twin screw expanders in the industry, there is a need for detailed study of the flow inside the expander and methods that will result in advances of the operation and design. In this paper, a full 3D CFD calculation for the oil injected twin screw expander was performed. The mass flow rates through leakage paths are presented. From the results of the numerical simulation, problems that can occur when changing from oil injected compressor to the expander are highlighted. In the future, air should be replaced with the refrigerant (R245fa, R134a, etc) with appropriate real gas equation. Another challenge is to simulate the formation of the oil film on the surfaces inside the expander.

ACKNOWLEDGEMENTS

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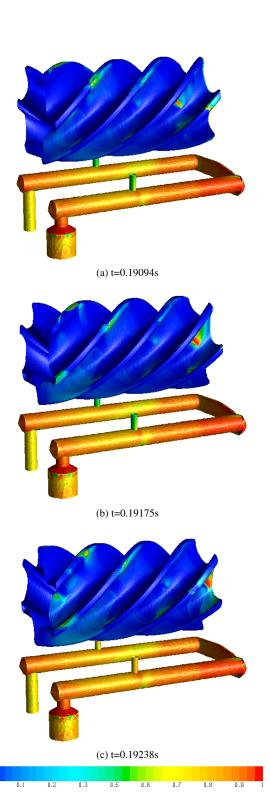
(a) t=0.19094s

(b) t=0.19175s

(c) t=0.19238s

FIGURE 8: OIL DISTRIBUTION FOR DIFFERENT TIME STEPS

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