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SIMULATION OF TWO STAGE TWIN SCREW COMPRESSOR INCLUDING LEAKAGE FLOWS AND COMPRASION

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Abstract: This paper presents the methodology of creating the numerical model for a sample screw compressor provided by Sullair for research purposes. The desired simulation approach should be able to deliver sufficient accuracy at a feasible effort in terms of computational time and manpower to create the numerical model.

Keywords: Polyethylene terephthalate bottle blowing system, water-flooded single-screw compressor, dehumidification, pressure loss, isentropic efficiency.

Introduction:

Computational Fluid Dynamics (CFD) is a common and validated simulation method in research and industry for the analysis of fluid systems. In the past years, it has proven to become more and more applicable for modeling the flow physics inside positive displacement (PD) machines. The working chamber and thus the discretized flow domain of PD machines are changing in time, characterized by complex thermodynamics. Compressible fluids, real-gas properties and leakage flows with trans- or supersonic characteristics are phenomena which have to be accounted for in order to properly model the behavior of the machine. As CFD methods evolve in general, but also for the application of PD machines in particular, the numerical model can replace a prototype during early stages of the product development. It is a dry running two stage twin screw compressor running with air at a rated power range between 190 and 300 kW. The two stages are gear driven by the main shaft at rotational speeds between 1180 and 2100 rev/min. Each stage features different rotor profiles, where the first stage has a 4-5, the second stage a 7-9 lobe combination. The total pressure ratio of the two stages combined is up to 10:1. To enhance the performance of the compressor, discharged air from the first stage is cooled down before entering the second stage. A specific meshing method is used to model the size-changing working chambers between rotors and casing, where only hexahedral cells are used and mesh topology is constant. The model accounts for radial and axial clearances between rotors and stator, where rotors and stator are connected with interfaces. The transient simulation results are compared to experimental measurements for torque, and flow rate. Also discharge pressure and temperature after first and second stage are compared to the experimental results. In addition, the possibilities of the simulation are exemplified by the gathering of time- and space-resolved monitor points like temperature or pressure at distinct points within the compressor. Apart from direct comparison to the experiment, also a sensitivity study regarding the change of housing clearances is presented, as leakage flow has severe impact on the compressor performance. These clearances and the resulting leakages are often not exactly known whereas they also vary because of manufacturing tolerances or deformations due to the load on rotors and stator. Here, the numerical simulation can serve as a helpful tool to estimate the sensitivity and change of machine characteristics, which is hard to determine in the scope of experiments.

Given that approach, it has to be considered that both compressor stages are modeled in one simulation setup, where the physical time step in the transient simulation is equal for the whole model

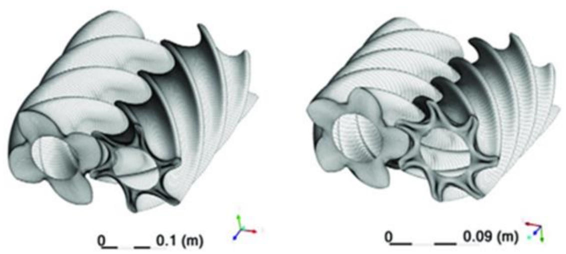

Figure 1. Grids for first stage (top) and second stage (bottom) of the compressor

Thus, angle increments of the generated grids cannot be chosen arbitrarily for each stage. The ratio of angle increments for both stages has to represent the speed ratio of the two male rotors. The tooth pitch angles and angle increments for both stages are given in the following table:

Table 1. Angle increments according to the speed ratio of the modeled two stage screw compressor

	Stage 1	Stage 2
Speed ratio (Stage 2/Stage 1)	1.53	
Tooth pitch angle	90°	72°
Number of grids per pitch angle	90	47

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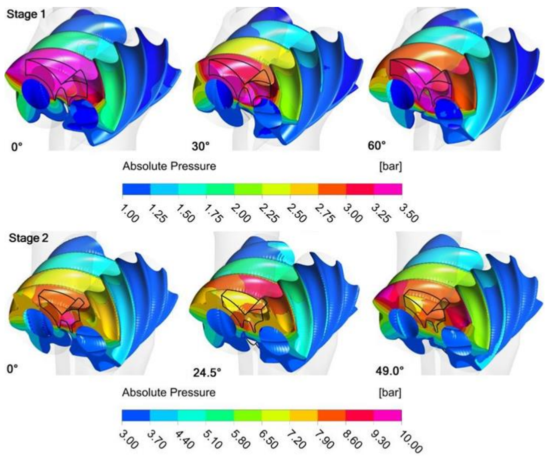
Angle increment 1° 1.532°
Angle increment ratio (Stage2/Stage1) 1.532

The case is set up for three different operating points (OP), where the rotational speed of the stages is varied as well as inlet (first stage suction port) and outlet temperature (second stage discharge port, temperature is only used in case of backflow) according to the conditions indicated by experimental data. In addition, one case with smaller housing clearances (uniformly decreased by approximately 20%) and smaller intermesh clearance (closest distance between the rotors decreased by approximately 50%) for each stage is simulated, while axial clearances remain unchanged. Table 2 shows the four operating points taken into account:

Table 2. Boundary conditions for the performed simulations

Case	Main shaft speed [rev/min]	Inlet Pressure [bar(a)]	Outlet pressure [bar(a)]	Inlet temp. 1^{st} stage [C]	Inlet temp. 2^{nd} stage [C]	Outlet temp. 2^{nd} stage [C]
OP1	1490	1.0	7.98	30.8	31.9	136.1
OP2	1790	1.0	7.98	28.6	34.1	143.0
OP3	2100	1.0	7.98	27.0	37.9	150.8
OP4 (Decreased radial clearances)	1790	1.0	7.89	28.6	34.1	143.0

For both stages of the screw compressor, the compression inside the working chamber is visualized in Figure 2 by the instantaneous pressure distribution on the rotor walls. It is shown over one pitch angle of the corresponding stage. At the depicted rotor positions (0° to 60° for the first stage, 0° to 49° for the second stage), the pressure in the rotor chambers formed by male and female rotor lobes increases as the volume of the individual chambers is decreasing. Once the lobes reach the control edges, the connection of the chambers with the discharge port is



established and compression process is finished.

Figure 2. Pressure distribution at 1790 rpm for first (top) and second stage (bottom) over one pitch angle

Conclusion

This paper shows a CFD approach to simulate the sample dry screw compressor provided by Sullair. This compressor consists of two compressor stages with an intermediate cooler. In the presented approach, both stages as well as the interstage cooling are modeled within one setup, allowing a direct coupling of the stages without the need to specify boundary conditions at the outlet of the first stage or at the inlet of the second stage respectively. The cooler is replaced with a simple duct, where the temperature drops (i. e. a specified suction temperature for the second stage is reached) is realized with an energy sink.

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