CFD modelling and validation of the rotary lobe compressed air expander

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Abstract. The article covers the CFD analysis of the compressed air rotary lobe expander. Compressed air engines are commonly used in the explosion hazard zones, where the others power sources cannot be applied. The lobe construction seems to be cheaper solution in comparison with the classic turbine expanders at the corresponding power level.

In the paper the rotary lobe compressed air expander model was developed in the ANSYS CFX software. During implementation of the proposed numerical model the mesh motion problem was solved, which was challenging due to relatively small gaps between lobes and the housing as well as due to high rotary speed of lobes. What is more, the compressible fluid model was applied. Therefore, these conditions make the undertaken problem more complex and calculation more time consuming. Finally, the model was validated applying the catalogue data of the producer.

In the paper description of the developed model was presented. Then the results of simulations with their validation were shown which proved the correctness and accuracy of the model. The developed tool might be very useful for the further analysis of the expander including its optimization in terms of improvement of its efficiency or energy gain.

1 Introduction

The rotary lobe compressed air expander which is the object of the investigation in this paper in most cases is applied as the part of air motors. Air motors can successfully replace conventional, e.g. electric motors in explosive areas or in areas, where the suitable medium is available. The most common applications for air motors are elevators drives in the mines, coal mining locomotives, earth drilling rigs and units for winches [1,2]. Similar types of steam-based expanders can be applied in micro organic Rankine cycles (ORC).

The main advantage of gas expanders is the lower cost of application comparing to conventional turbine motors at the corresponding power level, combined with arduous and long-life systems. Other features determining the usage of air expanders are adjustable torque and output speed, the ability to stop under load without damage as well as a small number or lack of springs and pins [2]. Also, it is possible to install the air motor in wet, dirty and harsh environment due to its sealed housing and relatively compact structure.

In this paper the model of the rotary lobe compressed air expander was developed with the consideration of industrial solutions. As shown in Figure 1, the model used for the present simulation consists of two rotors moving inside the expanders housing. The torque is developed by the larger rotor situated in the lower part of the housing and then transferred to the rotating shaft by a

synchronizing gear train. To preserve high efficiency of the expander, air gaps between the rotors and housing are relatively small which allows rotation process without physical contact. The frictionless work might result in the long maintenance free operation without downtime in industrial applications [1].

2 CFD model of the Rotary Lobe Expander

During the substantive preparations for modelling of the expander, no literature was found referring to rotary lobe expanders. However, literature on modelling related devices of a different type of operation or using a different medium is available. There are CFD models of rotary lobe pumps described in [3,4]. In particular the CFD modelling approach applied in [3] is similar to the one applied in the presented paper. Moreover, the relatively high cost of calculations in such simulations was also underlined. The turbulence and gas models applied for similar simulations are described in [5–9]. There are also works related to models [10–12] and patents [13–17] of screw rotors, with lobes shape optimization taken into account. The lobes shape optimization is one of the next steps in the future development of the proposed rotary lobe expander model.

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2.2 Geometry

In the first step a full 3D geometrical model of the expander was created. The model includes the fluid domain geometry, which corresponds to the housing of the expander and the geometries of two relevant rotors. The rotors were designed with different shapes (i.e. 2- and 5-lobes) having the proper proportions to allow synchronous operation, with consideration of face-clearance between lobes and the housing as well as center-clearance between the corresponding lobes.

Due to relatively high computing cost, the axial dimension of the expander was reduced to 1 mm, which corresponds to quasi-2D approach. This gives the opportunity to observe the main phenomena responsible for the expander operation. Also, it is associated with, e.g. neglecting pressure losses in the axial direction, which have an effect on the hereby presented results discussed further. The geometry of the rotary lobe air expander is presented in the Figure 1.

Figure 2 shows the geometry of the gas-filled expander chamber without rotating elements. In order to refine the grid, it was divided into five areas: the working area of the larger rotating circle, the working area of the smaller rotating circle, the central area the larger and smaller rotor engagement, the expander inlet as well as the outlet chamber and the outlet from the expander.

Fig. 1. Axial view of the rotary lobe expander geometry with rotor direction indicators. No. 1 – smaller rotor, no. 2 – larger rotor.

2.2 Mesh

The applied immersed solids method requires the generation of separate grids for the fluid domain and for solids. Thus, the three areas of discretization were designed: an area for the fluid domain and two areas for rotating wheels.

Fig. 2. Mesh of the fluid domain.

The density of the mesh was selected in relation to the values of gradients of local parameter such as pressure or velocity of the gas. The number of cells per unit volume was increased in specific areas, i.e. in the gaps between the walls of the expander and the lobes, in the area of meshing of the rotors. The grid was created while maintaining its quality criteria, e.g., aspect ratio and skewness on the reasonable levels.

The fluid domain was meshed with the sweep method (see Figure 2). The sweep method is adequate for complex geometries with little or no variation in a specific direction and requires definition of source and target boundary. The front plane of the 3D domain was selected as a source boundary.

Fig. 3. Mesh of the smaller (on the left) and larger (on the right) rotors.

The density of the fluid domain grid was enhanced in the area of engagement of the rotors, which can be seen in Figure 2. The mesh properties for the fluid domain are presented in the Table 1. The grids for rotors were also created using the sweep method (see Figure 3), refined in areas of solid-fluid contact. The grid properties for the rotors are also presented in the Table 1.

Table 1. Meshes properties.

Parameter	Value		
	Fluid domain	Larger rotor	Smaller rotor
Number of cells	495,548	190,582	84,560
Number of nodes	744,897	287,340	127,602
Average aspect ratio	2.92	3.00	3.23
Average skewness	0.11	0.14	0.11
Average element quality	0.83	0.83	0.81

2.2 Computational domains, boundary conditions and simulations settings

In the ANSYS CFX the computational area was divided into three domains. The fluid flow region was defined as a fluid domain. The other two domains identified as solids were corresponding to two rotors. The boundary conditions for the inlet and outlet were set as fixed pressure values while for all remaining walls of the fluid area the no-slip wall was assumed.

Table 2. Parameters assumed in the simulations.

Parameter	Unit	No. of simulation		
		1	2	3
Speed of the larger rotating wheel	rev/min	1000	1000	1500
Speed of the smaller rotating wheel	rev/min	2500	2500	3750
Expanders inlet pressure	bar	4	8	3
Total simulation time	s	0.05	0.05	0.032
Time step	s	$5 \cdot 10^{-5}$	$5 \cdot 10^{-5}$	$3.2 \cdot 10^{-5}$
Number of time steps for a single simulation	-	1000	1000	1000
Number of revolutions of the smaller rotor during simulation	-	2.083	2.083	2
Number of revolutions of the larger rotor during simulation	-	0.833	0.833	0.8

In order to validate the rotary lobe air expander model, three simulations were conducted, and the obtained results were compared to the literature data. The value of pressure at inlet and rotation speed of the rotors took different values in the individual simulations.

To maintain the appropriate stability and quality of calculations, the time step values were adapted to the rate of change of geometry which corresponds to the rotation speed. The number of time steps has been set to achieve the stable solution and to observe the characteristics of the expander's operation. This refers to the execution of about two revolutions of the larger rotor.

In Table 2 the aforementioned parameters are presented. At the expander outlet fixed value of the pressure equal to 1 bar was assumed in the all simulations. Moreover, the inlet temperature was equal to 25°C and all walls of the fluid domain were adiabatic.

The catalogue data of the air expander producer does not provide detailed information about thermodynamic properties of the air (e.g. temperature or humidity). Therefore, the working medium applied in the presented simulations was the dry air with the thermophysical properties shown in the Table 3.

The flow inside the expander was assumed transient, compressible and turbulent. Therefore, ideal gas model together with the $k-\omega$ Shear Stress Transport (SST) turbulence model were applied during the simulations.

Table 3. Thermophysical properties of the working medium (dry air).

Parameter	Value	Unit
Molar mass	28.96	kg/kmol
Density	1.185	kg/m ³
Specific heat	1.0044	kJ/(kg·K)
Dynamic viscosity	$1.831 \cdot 10^{-5}$	kg/(m·s)
Thermal conductivity	$2.6 \cdot 10^{-2}$	W/(m·K)

2.2 Solver settings and residuals

As initially mentioned to investigate operation of the rotary lobe air expander transient calculations were carried out. The assumed main solver settings in the ANSYS CFX are presented below:

- Advection scheme: high resolution.
- Transient scheme: second order backward Euler.
- Residual type: root mean square (RMS).
- Convergence criteria: 10-4.

In order to conduct after-simulation analysis and to check the crucial parameters during the solution process the appropriate transient results were saved and transient monitors were created and registered. Selected variables which allowed analysis of the obtained results were saved for the whole computational domain for every time step. These variables were as follows: absolute pressure, total pressure in stationary frame of reference, velocity and temperature. Moreover, all together sixteen monitors were created for each simulation: two monitors for the average mass flow rate at the inlet and outlet to the fluid domain and fourteen monitors for the average pressure acting on the surface of half of each lobe, including both, smaller and larger rotors.

All simulations were carried out using the workstation with two processors (Intel® Xeon® CPU E5-2600 2.20 GHz) and with 64 GB RAM memory as well as utilizing of 8 to 16 computing nodes. The time required to carry out a single simulation equals approximately 26 h.

3 Results and discussion

In Figures 3-5 the average pressure acting on the surface of a half of each lobe, including both, smaller and larger rotors are presented. The obtained results are shown for simulation no. 2, which correspond to the inlet pressure of 8 bar and the rotation speed equals 1000 rev/min for the larger rotor. In each diagram a cyclical operation of the expander can be observed resulting in a regular change of the pressure on each monitored surface.

Fig. 4. The pressure variation for the smaller rotor for the pressure at the expander inlet of 8 bar and rotational speed of 1000 rpm.

Fig. 5. The pressure variation for the larger rotor for the pressure at the expander inlet of 8 bar and rotational speed of 1000 rpm.

After obtaining the mean pressure values acting on the surface of the half of the lobe, the force acting in the direction perpendicular to the axis of rotation for each of the lobe was determined [17]. Then the torques acting on

the individual rotors were determined as the sum of the component torques acting on the rotor from individual lobes. Such a procedure was necessary to obtain the actual value of the torque acting on the rotor, because at specific positions of the rotors, the force direction is unsuitable because it acts on the opposite direction to desired one.

The power of the modelled expander was calculated for the rotors as the product of the previously obtained torque values and the angular velocity – see Figure 6. The forces acting on the rotors and the power calculated on their basis show cyclicity resulting from the constant rotational speed of the rotors and the regular opening and closing of the space between the rotors and the walls of the expander.

Fig. 6. Calculated expander power acting on rotors for the system thickness of 1 mm, inlet pressure of 8 bar and rotational speed of 1000 rpm.

The amplitude of the force and power values for the smaller rotor is much greater than for the larger rotor. This phenomenon occurs due to fewer lobes and uneven distribution of forces acting on the smaller wheel at the inlet and outlet of the expander.

Table 4. Results comparison with literature data.

Parameter	Unit	No. of simulation		
		1	2	3
Calculated average power on rotors	kW	9.97	23.45	9.86
Calculated average power of expander	kW	8.55	20.12	8.46
Expander's power according to literature data	kW	9.53	19.03	8.29
Error	%	10.28	5.73	2.05

To obtain the power value of the expander received at the output from the device $\dot{W}_e$, the calculated expander powers acting on the rotors $\dot{W}_r$ should be reduced by mechanical losses (mechanical efficiency of the gearing

η_g , mechanical efficiency of the smaller rotor η_{R1} , mechanical efficiency of the larger rotor η_{R2}) and losses resulting from medium leakages in the expander axial slots η_l . Therefore, the expander power was calculated based on the following formula:

$$\dot{W} = \dot{W}_R \eta_g \eta_{R1} \eta_{R2} \eta_l \quad (1)$$

The assumed final value of the expander's mechanical efficiency is 0.8579. As previously mentioned, the calculated power of expander was obtained for the thickness of 1 mm. To obtain the total power, the calculated value for the thickness of 1 mm was multiplied by the corresponding length of the expander presented in literature data. The comparison of calculated power with literature data is shown in Table 4.

The exemplary contours of velocity magnitude, contours of pressure with path lines and contours of pressure with velocity vectors are shown in Figures 10-15 for simulation no. 1 and 2. One can notice the leakage phenomena in the gaps between the rotors as well as smaller rotor and housing.

Fig. 7. Distribution of velocity magnitude in the expander for 0.005 s inlet pressure of 4 bar and rotational speed of 1000 rpm.

Fig. 8. Distribution of velocity magnitude in the expander for 0.005 s, inlet pressure of 8 bar and rotational speed of 1000 rpm.

Fig. 9. Distribution of pressure and path lines in the gap between rotating wheels for 0.0025 s, inlet pressure of 8 bar and rotational speed of 1000 rpm.

Fig. 10. Pressure distribution and path lines in the gap between rotating wheels for 0.0025 s, inlet pressure of 4 bar and rotational speed: 1000 rpm.

Fig. 11. Distribution of pressure and velocity vectors in the expander for 0.0025 s, inlet pressure of 4 bar and rotational speed of 1000 rpm.

Fig. 12. Distribution of pressure and velocity vectors in the expander for 0.0025 s, inlet pressure of 8 bar and rotational speed of 1000 rpm.

2 Conclusions

In this paper the model of rotary lobe expander has been developed and then subjected to tests which validated the results obtained with data from literature. The model configuration and settings were selected in a way which ensured the highest possible accuracy and correctness of the obtained results as well as high computational efficiency. The validation activities were based on a device having a similar design for which geometrical and operational parameters are provided by the manufacturer. For the analysed cases (i.e. variable pressure of air inflow and variable rotational speed) satisfactory accuracy of simulated results with actual operational data was obtained. The maximal error was approximately 10 %.

It was shown that the developed model well simulates the operation of considered rotary lobe expander. Thus, it is ready to be used as a tool for conducting optimization in terms of geometry and parameters of the steam type expander.

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