

Article

# **Comparison of Two Single Stage Low-Pressure Rotary Lobe Expander Geometries in Terms of Operation**

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**Abstract:** In the article the computational fluid dynamics (CFD) simulation and calculated operational parameters of the single stage low-pressure rotary lobe expander compared with the values obtained from a different geometry simulation are presented. Low-pressure rotary lobe expanders are rotary engines that use a compressed gas to produce mechanical energy, which in turn can be converted into another form, i.e., electric energy. Currently, expanders are used in narrow areas, but have a large potential in the energy production from gases of low thermodynamic parameters. The first geometry model was designed on the basis of an industrial device and validated with the empirical data. Simulation of the second geometry was made based on a validated model in order to estimate the operational parameters of the device. The CFD model included the transient simulation of compressible fluid in the geometry changing over time and the rotors motion around two rotation axes. The numerical model was implemented in ANSYS CFX software. After obtaining simulation results in the form of parameters monitors for each time step, a number of calculations were performed using a written code analysing the CFD program output files. The article presents the calculation results and the geometries comparison in terms of work efficiency. The research indicated that the construction of the device on a small scale could cause a significant decrease in the aforementioned parameter, caused by medium leaks in the expander clearances.

Keywords: rotary lobe expander; low-pressure gas; transient CFD

# 1. Introduction

In the era of energy recovery, one of the main trends in addition to renewable energy development is the search for solutions that enable energy generation from waste sources. There are many technological solutions enabling such energy production, e.g., direct combustion, pyrolysis, fermentation, conventional, and plasma gasification [1]. All of the previously mentioned technologies lead to thermal energy through the final combustion process of solid, liquid, and gas fuels [2]. In addition to the previously mentioned forms of energy recovery, there is also the possibility of the direct use of waste gases from industrial technological processes. At large industrial plants built, there are gas discharge stations with relatively low thermodynamic parameters. Previously, for economic purposes, the release of waste gases was considered reasonable. However, all effort is being made today to increase the energy efficiency of industrial processes and reduce energy wastage [3,4]. There are several device types that can be applied for aforementioned systems: Dynamic expanders, i.e., axial, cantilever, radial [5,6], or volumetric expanders i.e., piston, screw, vane, rotary lobe, and scroll [7–11]. Each of the devices has a different operating characteristic, which means it can be adapted to the system in terms of performance, operation time, start-up frequency, and the thermodynamic parameters of the working medium. A comprehensive comparison of expanders with an attempt to identify the best, in terms of power generation below 100 kW is presented in [6]. The author points out the pros and



cons of selected devices, e.g., a decrease in the efficiency of turbine expanders along with a decrease in power, high unit investment cost of screw expanders, or lubrication and friction losses of vane expanders. Unlike turbines in a higher power range, despite 30 years of research and experience, the best technology of low-power expanders has still not clearly emerged [5]. This fact may partly be caused by the still low demand for low-power generators powered by waste sources, high unit cost, and a combination of pros and cons of each expanders type. Research and experiments of expanders are still being carried out [5,9–17] in order to, for example, optimize the geometry. However, they are associated with relatively high costs of constructing the device often related to the required manufacturing precision [6]. There are also computational fluid dynamics (CFD) calculations on the expanders' operation [5,17,18], but since most of them are performed for time-varying geometry, these calculations are labour-intensive and require a lot of computing power. Furthermore, they should be validated with an operating device.

This work concerns rotary lobe expanders which are volume-type ones, where the torque is generated by two rotors whose power is transferred to the output shaft located in the axis of one of the rotors. The rotors are connected through synchronized gear which enables operation without friction. They can be used to produce mechanical or electrical energy from low-pressure waste gases or as part of a system with thermal energy obtained in the waste incineration process [6]. Rotary lobe expanders show a number of advantages that led to this expander type selection and to a series of CFD simulations afterwards. An additional feature considered by the authors is the presence of this type commercial devices [14,19–21] enabling the validation of the CFD model. To this day, rotary lobe expanders have been used for over 30 years as pneumatic motors in industry related to explosive environments, e.g., in mines or drilling platforms [21]. Since up to now mainly electric motors have been used in the considered power range, expander technology requiring the supply of compressed gas was only mechanically improved within the mentioned industry. However, after years of manufacturer's experience, the declared technological features can be considered reliable and encourage further research such as the optimization of the expander geometry. Rotary lobe expander features [19,21]:

- Both rotors move without physical contact, which eliminates friction and creates torque with mechanical efficiency exceeding 98%. It results in a long service life without maintenance and downtime;
- Minimal maintenance enhanced by hermetic bearings;
- The completely enclosed motor housing without vent holes allows for its usage in humid or polluted environments without the risk of corrosion inside the expander;
- Relatively low-pressure range of two to eight bars;
- Imperceptible vibration during operation, even at high speeds;
- Can be mounted in all directions with the shaft vertical or horizontal;
- Ability to stop under load;
- Work in hazardous condition provided by a sealed housing.

Expander modelling with the computational fluid dynamics methods involves the following aspects: Rotating machine modelling with computational domain geometry changing in time, transient case, and the main flow direction not being compatible with the rotors axes. The above-mentioned aspects combined may be the reason that CFDs of rotary lobe expanders have not been found in the literature. However, articles regarding models of devices with a similar type of operation or modelling methods have been found. The 3D semi-transient screw expander CFD analysis were conducted by Papes et al. [16]. The immersed solid method was introduced in rotary volumetric pumps and compressors by Voorde et al. at [22] and by Schiffer&Klomberg [23]. A single state Wankel expander CFD model was introduced in [17]. The screw rotors mathematical models are developed in articles [12,24,25] and patents [26–30]. The rotary vane expander CFD model with a customized grid generation methodology was developed by Binch et al. [31] and Montenego et al. [32,33].

This document is a continuation of work presented, among others in [18]. In the course of previous works, the CFD rotary lobe expander model was built based on industrial geometry. Then the series of simulations results were compared with empirical data. After a successful validation process, a new modified geometry was designed and CFD analysis was performed. Presented work was performed in order to select the appropriate expander geometrical parameters for the test stand. The new geometry model was subject to certain construction requirements such as expanders inlet pressure or rotational speed. The results comparison and analysis are presented in the graphical and tabular form.

## 2. Model Description

## 2.1. Flow Governing Equations

The numerical environment used for CFD calculations is based on continuity, momentum, energy, and state equations. The k- $\omega$  based Shear-Stress-Transport (SST) turbulence model was implemented. The k- $\omega$  SST model combines the advantages of the k- $\varepsilon$  model and the k- $\omega$  model by introducing an additional factor limiting the overproduction of turbulence kinetic energy in areas of strong positive pressure gradients (accumulation points, boundary layer detachment areas) [34,35] that occur in the expander chamber near rotating lobes. The k- $\varepsilon$  model well reflects turbulence in free flow and shear layers. Additionally, it has low sensitivity to inlet conditions for quantities describing turbulence. This is a desirable feature when creating the expander model because these quantities are not exactly known. The k- $\omega$  models turbulent flow in the boundary layer much better, which corresponds to areas near rotating lobes, while it is very sensitive to turbulent quantities in free flow (in the areas of inlet and outlet chambers), so the combination of features of both models is valuable in the expander model. The applied turbulence model is well described in [34,36–39]. The discrete form of the aforementioned equations has been implemented in the ANSYS CFX (18.2, ANSYS, Inc., Canonsburg, PA, USA) software, which was used to perform calculations.

The immersed solid model implemented in Ansys CFX software was also applied, which allows to model a transient state with solids that can move through a fluid domain. During simulation, CFX-Solver calculates which parts of the fluid domain coincide with the immersed solid and applies the momentum source to the fluid inside the immersed solid domain in order to force the flow to move along with the solid [35]. Additional conditions are applied at the boundary to improve flow behaviour near the wall of the immersed solid boundary. To calculate the immersed solid impact on the surrounding fluid the volume of fluid that corresponds to the solid volume is forced to move with the solid by using the source term in the momentum equation:

$$S_i = -\alpha C (U_i - U_i^F) \tag{1}$$

where  $i = 1, 2, 3, U_i, U_i^F$  are components of fluid velocity and components of forcing velocity due to the immersed solid, *C*,  $\alpha$  are momentum source coefficient and momentum force scaling factor. The momentum source coefficient is evaluated as the average of the three diagonal coefficients in the momentum equation [36]. The momentum source scaling factor value is a balance between accuracy and robustness. A higher value leads to a more accurate solution but is less robust and gives convergence difficulties during calculations. The default value introduced by Ansys CFX is 10 and was increased to 25 in presented calculations as the relatively high value was obtained with satisfactory compliance.

## 2.2. Geometries

The geometers of the rotary lobe expanders consist of two rotors of various shapes rotating in opposite directions, enclosed in a stationary housing. The designed geometries with the rotation direction indicated are presented in Figure 1. The first geometry based on the commercial device was used to carry out the comparison with the geometry designed according to new criteria. As mentioned, the new geometry model was subject to certain construction requirements for the planned test stand,

which have a lesser air consumption and speed ratio of 3000:1500 rpm resulting in a pitch diameter ratio of 2. The geometry parameters comparison is presented in Table 1. As easily seen, most of the significant parameters characterizing the compared geometries differ. In the previously modelled and validated geometry, which was reproduced on the basis of a commercial device, the smaller rotor was equipped with two lobes. This type of construction caused a maximum difference in pressure between the two halves of the lobe, which in turn resulted in a high value of the force acting on the smaller rotor. At the same time, a large pressure difference acting on a single lobe causes relatively large leaks on the radial expander clearance. Therefore, in the new geometry a larger number of smaller rotor lobes was implemented to prevent an excessive leakage of the medium. Another modification introduced in the new geometry was the change in the inclination of inlet and outlet channels.



Geometry No. 1

Geometry No. 2



Parameter	Unit	Geometry No. 1	Geometry No. 2
Number of lobes—smaller rotor	-	2	5
Number of lobes—larger rotor	-	5	10
Height of lobes—smaller rotor	mm	41.62	8.75
Height of lobes—larger rotor	mm	30.00	6.37
Angle between inlet and outlet channel	degrees	180	90
Inlet channel width	mm	60	8
Outlet to inlet channel width ratio	-	1	5
Inlet channel to radial clearances width ratio	-	3000	400
Radial clearance	mm	0.2	0.2

Table 1.	Geometry	parameters	comparison.
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A commercial device that was the basis for building the original geometry is often used as a drive in mine lifts where its main advantage is the possibility of working in two directions. However, this feature meant that the inlet and outlet of the geometry No. 1 had to have the same cross-sections. Since the new expander geometry is designed for purposes other than those used so far, it was decided to design an outlet chamber with a larger cross-section, thus allowing a medium expansion without throttling. The second geometry dimensions are much smaller due to the design associated with the expander test installation construction planned for the future. Due to the computational cost reduction, the thickness of the calculation domain is 1 mm.

## 2.3. Meshes

As the immersed solid calculation method requires the use of separate grids for the expander chamber and rotors, three independent calculation grids have been created for each geometry—see

Figure 2. The calculation grids were modelled according to the meshes quality criteria (e.g., skewness, element quality, and aspect ratio). The grids density was selected by the gradients of parameters criteria such as speed, density, and pressure. It was generated particularly carefully near the clearance between the rotors and the expander chamber as well as in the area of rotors interlocking (Figures 3 and 4).





1a





Figure 2. Computational grids of geometry No.1 (1a,1b) and geometry No. 2 (2a,2b). Fluid domains (1a,2a), rotors (1b,2b).



Figure 3. A zoomed view of superimposed meshes in the area of rotors interlocking.



Figure 4. A zoomed view of superimposed meshes in the clearance area.

The radial clearance for both geometries of  $2 \times 10^{-4}$  m was divided by 7 computational grid elements, the smallest of which was  $2 \times 10^{-5}$  m. The maximum size of cell wall as limited to  $2.5 \times 10^{-7}$  m<sup>2</sup>. Geometry No. 2 mesh was modelled with the same criteria and sizes as the No. 1 geometry. The sensitivity analysis of geometry the No. 1 mesh was carried out with the following total number of elements: 1,763,339; 1,356,414; 770,690; and 678,207 with the result presented as a percentage of the power obtained at the highest number of cells: 100%, 98%, 97%, and 92%. The authors decided to implement a grid giving a score of 97%, choosing between the results reliability and the calculation time. The resulting cell number of geometry No. 1 was 770,690 and consisted of domain rotor cells (495,548), as well as larger rotor cells (190,582) and smaller rotor cells (84,560). The cell number of geometry No. 2 domain, larger, and smaller rotor equalled respectively 335,735; 121,920, and 61,640.

#### 2.4. Boundary Conditions and Simulations Settings

The immersed solid modelling method requires the definition of a fluid and the rigid solid domains. In the analysed case, the expander chamber was defined as a fluid domain and rotors as solids with a specified rotational speed. At the domain inlet, a boundary condition was defined as a constant pressure. At the outlet, a pressure boundary condition of 1 bar was applied. The outlet type was defined as "opening", which allows the medium to move back into the domain and the boundary condition for the remaining walls of the fluid domain was adopted as adiabatic. The gas model was implemented as air ideal gas with thermodynamic properties set according to the Ansys CFX programme library. In order to validate the numerical model, four simulations were carried out for various operational conditions and then compared with empirical data provided by the expander manufacturer. The important operational parameters that changed in the individual simulations are presented in the Table 2.

After the validation process, two more simulations were carried out, one with geometry No. 1 and the other with geometry No. 2. The rotational speed of the validated geometry was 1500 and 3750 rpm for the larger and smaller rotor respectively. Rotational speeds of new geometry rotors were 1500

and 3000 rpm for larger and smaller rotor. The difference in the selected rotational speeds results from the ratio of the diameters (and the number of lobes) of the larger and smaller rotor.

The key adopted solver settings in the ANSYS CFX are: Advection scheme: High resolution; Transient scheme: Second order backward Euler; and Residual type: Root mean square (RMS); Convergence criteria: 10<sup>-4</sup>.

Parameter	Unit		No. of Simulation		
		1	2	3	4
Speed of the larger rotating wheel	rev/min	1000	1000	1500	1500
Speed of the smaller rotating wheel	rev/min	2500	2500	3750	3750
Expanders inlet pressure	bar	4	8	4	3
Total simulation time	s	0.05	0.05	0.03	0.032
Time step	s	$5 \times 10^{-5}$	$5 \times 10^{-5}$	$3 \times 10^{-5}$	$3.2 \times 10^{-5}$
Number of time steps for a single simulation	-	1000	1000	1000	1000
Number of revolutions of the smaller rotor during simulation	-	2.083	2.083	1.875	2
Number of revolutions of the larger rotor during simulation	-	0.833	0.833	0.750	0.8

Table 2. Validation simulations input parameters.

In order to calculate the expander power according to the method presented in the next chapter of this paper, monitors of appropriate values were set in each simulation. Mass flows at the inlet and outlet of the expanders as well as mean pressures on the surface of the halves of each lobes were monitored. Such criteria resulted in 16 monitors for validated geometry simulation and 32 monitors for the new geometry. Separated surfaces for pressure monitoring are shown in Figure 5. Values of crucial parameters were recorded at each timestep: Pressure, temperature, and gas velocity. Simulations were carried out using the Intel®Xeon®CPU E5-2600 2.20 GHz computing server. The time needed to perform a single simulation was ~26 hours with the utilization of 16 partition.



Figure 5. Separated surfaces for pressure monitoring at the halves of each rotor lobes.

### 3. Results, Validation, and Discussions

#### 3.1. Post-Simulation Data Analysis

Since the simulations were completed, in order to obtain the power values of each of the expander, a series of calculations based on the monitor data were made. The calculations were carried out with a program created in the MATLAB (R2017b, MathWorks, Natick, MA, USA) environment. Firstly, the mean pressure values acting on the lobe halves for each time step from the ANSYS CFX output files were loaded. Examples of average pressure waveforms are presented in Figure 6. Afterwards, the pressure difference  $\Delta p$  acting on each lobe for each time step and both rotors was calculated according to the Equations (2)–(5):

$$\Delta p_{t,r,l} = p_{h1,t,r,l} - p_{h2,t,r,l} \tag{2}$$

$$t = 1, 2, 3 \dots, T \tag{3}$$

$$l = 1, 2, 3 \dots, L \tag{4}$$

where  $p_{h1}$  and  $p_{h2}$  are the average pressures acting on lobes halves and *T*, *r*, and L are the number of timesteps, rotor, and lobe. Then, each lobe active surface area *A* was calculated as a projection of the lobe half area on the surface passing through the rotor axis and through the lobe center of mass (see Figure 7). The force *F* with direction perpendicular to the rotors axes and the momentum *M* acting on each lobe were calculated according to these equations:

$$F_{t,r,l} = \Delta p A_r \tag{5}$$

$$M_{t,r,l} = F_{t,r,l}R \tag{6}$$

$$M_{t,r} = \sum_{l} M_{t,r,l} \tag{7}$$

$$P_{t,r} = M_{t,r}\Omega_r \tag{8}$$

where R is the arm of each lobe calculated as the distance from the rotor axis to the center of lobe weight. For each rotor and each time step, the moments acting on individual lobes were summed up (Equation (7)). The power was obtained by multiplying the momentum acting on each of the rotors with rotor angular velocity (Equation (8)).



**Figure 6.** Average pressure waveforms for smaller and larger rotors. Simulation No. 2 according to Table 2.



Figure 7. Lobe active surface determination. Active surface marked in green.

After performing the above operations, graphs of the expander power were obtained as a function of time steps (and rotor positions), see Figure 8. On their basis, the average power of the expanders was determined, which was reduced by the mechanical losses specified in Equation (9):

$$W = W_R \eta_R \eta_{R1} \eta_{R2} \eta_l, \tag{9}$$

where: *W*—calculated power of the expander, *W*<sub>R</sub>—average calculated power on the rotors,  $\eta_g$ —mechanical efficiency of the gearing,  $\eta_{R1}$ —mechanical efficiency of the smaller rotor, and  $\eta_{R2}$ —mechanical efficiency of the larger rotor. Since the simulations were carried out for 1 mm of the expander thickness, the obtained power should be multiplied by the required axial dimension.



Figure 8. Power as a function of timesteps (rotor position). Simulation No. 2 according to Table 2.

## 3.2. Results and Validation

The average expander power calculated for geometry No. 1 compared with literature data is presented in the Table 3. The largest error occurred in the third simulation, in which the rotational speed was higher than in other simulations, 1500 and 3750 rpm, respectively, for the larger and smaller rotor. However, it was found that an incomplete expansion may occur at higher rotational speeds, therefore a test simulation was carried out assuming incomplete expansion of the working medium (i.e., up to 1.4 bar), and the results obtained were compared with the literature data. The result of this simulation is also shown in Table 3. Compared with the literature data, the calculation error was surprisingly small (0.18%), which can confirm the incomplete expansion occurring at the higher expander rotation speed.

Fable 3. Result	comparison	with the	literature	data	[21]	]
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Parameter	Unit	No. of Simulation			
		1	2	3	4
Calculated average power of expander	kW	8.55	20.12	12.64	8.46
Expander's power according to literature data	kW	9.53	19.03	11.01	8.29
Error	%	10.28	5.73	14.80	2.05
Calculated power of expander with consideration of incomplete expansion	kW	-	-	10.99	-
Error for incomplete expansion	%	-	-	0.18	-

After analysing the results of the four conduced simulations, a satisfactory accuracy compared with real operational data was obtained. The maximum error was about 10%. The authors of this article concluded that the developed model was ready for use as a tool for conducting tests in terms of the expander geometry and parameters. Then two further simulations were carried out, the first

using validated geometry and the second for new geometry No. 2. As the geometrical and operational parameters of the new geometry were imposed and they differ from the four simulations carried out previously, an additional simulation was carried out using geometry No. 1 and the model input parameters were selected as close as possible. The inlet pressure value for both simulations was 6 bar. Required rotational speeds for geometry No. 2 were 1500 and 3000 rpm for the larger and smaller rotor. As the same rotational speeds could not be obtained with geometry No. 1, (which results from the ratio of the rotor pitch diameters) the rotational speeds of 1500 and 3750 rpm were selected. A comparison of the rotary lobe expanders simulation results with two geometries, different rotational speeds, and different number of lobes cannot be unambiguous. Therefore, it was decided to compare the amount of gas consumed to the amount of energy produced. The comparison of calculation results (air consumption and output power) is presented in the Table 4.

Parameter	Unit	Geometry No. 1	Geometry No. 2
Air consumption	l/s	5.52	0.81
Calculated power	W	19,059	575
Air consumption to output energy ratio	1/J	0.2896	1.4028

Table 4. Results obtained with geometries No. 1 and 2 simulations comparison.

The simulation results are presented in the form of velocity vectors in Figures 9 and 10 and pressure projected on streamlines (Figure 11). Following conclusions can be presented after analysing the results. For the second geometry, a rapid decrease in efficiency was noticed. The second geometry was smaller, which with the same width of radial clearance caused a much larger share of gas leaks in the overall consumption. A several-fold decrease in efficiency which can be expressed by the ratio of air consumption to output energy was caused by high leaks. Indeed, the ratio of radial clearance to the width of the inlet channel was also several times lower for the second geometry (Table 1). This was caused by a minimum clearance limitation by the expander's performance which cannot be less than a fixed value of 0.2 mm.

The leaks can be noticed clearly on the velocity vectors (Figure 10) near the clearance between small rotor and expander's chamber as well as on the power trends (Figure 12). It can easily be seen that the power which is generated mainly by the pressure difference acting on the smaller rotor lobes in the case of the first geometry stays at a set high level during a part of rotation, while in the second geometry it dropped quickly after reaching the maximum value.



Figure 9. Velocity fields obtained as a result of geometry No. 1 simulation.



Figure 10. Velocity fields obtained as a result of geometry No. 2 simulation.



Figure 11. Pressure projected on streamlines.



**Figure 12.** Power as a function of smaller rotor position with a visible leakage impact marked. Geometry No. 1 (left) and geometry No. 2 (right).

## 4. Conclusions

The conducted research indicates that the construction of the device on a small scale could cause a significant decrease in efficiency, caused by medium leaks in the expander clearances. The increase in the share of losses resulting from leaks undermines the legitimacy of the expander construction on such a small scale. However, the authors claim that lowering the expander's operating pressure or using an additional seal may improve the expanders efficiency.

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