Design and test of a 10kW ORC supersonic turbine generator

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Design and test of a 10kW ORC supersonic turbine generator

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Abstract: Manufactures are searching for possibilities to increase the efficiency of combustion engines by using the remaining energy of the exhaust gas. One possibility to recover some of this thermal energy is an organic Rankine cycle (ORC). For such an ORC running with ethanol, the aerothermodynamic design and test of a supersonic axial, single stage impulse turbine generator unit is described. The blade design as well as the regulation by variable partial admission is shown. Additionally the mechanical design of the directly coupled turbine generator unit including the aerodynamic sealing and the test facility is presented. Finally the results of CFD-based computations are compared to the experimental measurements. The comparison shows a remarkably good agreement between the numerical computations and the test data.

1. Introduction

Automotive manufacturers are permanently searching for possibilities to increase the efficiency of internal combustion engines to reduce the fuel consumption and the emission of greenhouse gases. The investigation of energetic losses in a combustion engine showed that about one-third of the chemical energy is lost as thermal energy through the exhaust gas [1]. A possibility to recover some of this thermal energy is an organic Rankine cycle (ORC).

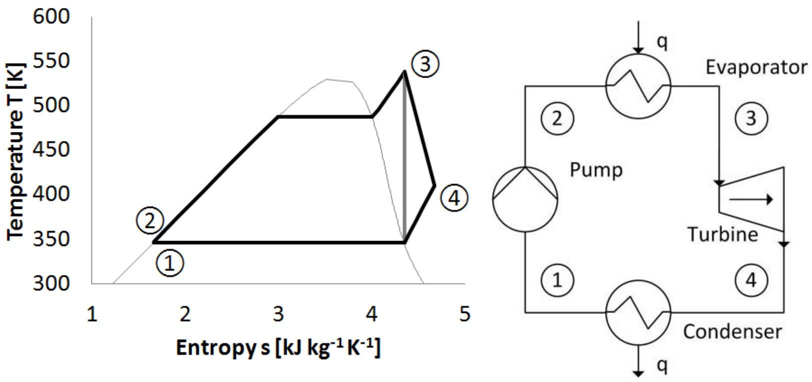

Figure 1: Basic organic Rankine cycle with ethanol [2]

The four working processes of an ORC are shown in figure 1. First a pump raises the pressure of the liquid working fluid ①→②. Afterwards the fluid is heated, evaporated und superheated in an evaporator ②→③. The superheated steam is expanded in a turbine ③→④, generating the desired power. At last the fluid enters a condenser where it is liquefied and returned to the pump ④→①.

2. Operating conditions of the ORC

The intention of this investigation is the development of a turbine generator unit for a waste heat recovery system of a 12.8 l diesel engine with a maximum power output of 375 kW. The design point is a medium engine load point with an exhaust gas mass flow of 0.249 kg/s at a temperature of 615 K. Table 1 shows the design boundary conditions for ethanol heated by this exhaust gas flow. Ethanol was chosen as working fluid for this ORC cycle because a prior investigation of Kunte and Seume [3] showed that this is the most promising fluid for this boundary conditions.

Table 1. Design boundary conditions for ethanol

Parameter	Units	Target
Fluid		Ethanol
$\dot{m}_{S,in}$	kg/s	0.052
$T_{S,in}$	K	530
$p_{t,S,in}$	bar	40
$p_{R,out}$	bar	0.81

In contrast to the prior investigation a mass fraction of 5 % water is added to the ethanol in order to improve the corrosion resistance of the titanium rotor. The water changes the fluid properties and therefore the boundary conditions of the ORC have to be adjusted. The new boundary conditions of the ORC for a mixture of ethanol and water are summarized in table 2.

Table 2. Design boundary conditions for a mixture of ethanol and water

Parameter	Units	Target
Fluid	Ethanol (95%mass), Water (5%mass)	
$\dot{m}_{S,in}$	kg/s	0.045
$T_{S,in}$	K	539
$p_{t,S,in}$	bar	40
$p_{R,out}$	bar	0.81

The temperature of the fluid at the turbine inlet must be raised from 530 K to 539 K to avoid condensation and droplet impact erosion which would damage the turbine rotor. Due to the increased process temperature, the higher thermal capacity, and the higher enthalpy for the vaporization of the water the mass flow is reduced to 0.045 kg/s. The maximum inlet and outlet pressure are mainly depended on the turbine design and remain unchanged.

3. Design of the turbine generator unit

The boundary conditions of the ORC are quite ambitious for turbines due to the combination of a low mass flow with a very high pressure ratio. The high pressure ratio results in high flow velocities and dissipative compression shocks if expanded in a single stage turbine. Whereas the low mass flow requires small wheel diameters and low blade heights, which lead to increased blade losses such as wall friction and tip gap losses. In addition the small turbine diameter combined with the high flow velocity results in a high rotational speed which increases the complexity of the bearing and the coupling of the machine. Together with the demand for a compact design and low production cost with regard to the automotive application, the right choice of the turbine type is crucial to the overall performance of the system. In a prior study Sourell et al. [4] investigated different turbine types for these boundary conditions. They concluded that an axial impulse turbine is best suited for this application. The advantages of an axial impulse turbine compared to other turbine designs, e.g. a radial inward turbine, are comparable low speeds which simplify the coupling and bearing of the generator, simple manufacturing due to the 2D blade design and the capability to expand high pressure ratios in one stage which allows a compact design. Furthermore an axial impulse turbine can be used with partial admission.

Especially the partial admission is advantageous in this application. Partial admission means that just a part of the circumference is provided with flow and the rest of the circumference merely ventilates. On the one hand the partial admission causes additional losses, which decreases the turbine efficiency. On the other hand partial admission allows increased blade heights which decreases the relative gap height and hence the tip gap losses. As a result the additional partial admission losses are usually overcompensated by the reduced tip gap losses, especially for low mass flows. However the main advantage is that the partial admission can be used to regulate the turbine mass flow rate. The presented turbine design features a variable partial admission, which can be changed between 20%, 40%, 60%, and 80% of the turbine circumference. As a result, the turbine operating range and therewith the range of high efficiency is significantly extended. Without a regulation it would be impossible to reach high efficiencies for all desired operating points of the truck engine. The regulation strategy of the turbine is shown on left side of figure 2. The degree of partial admissions depends on the number of opened stator passages. For low mass flows of less than about 0.03 kg/s only two stator passages are opened. This accounts for 20% partial admission. Every time the stator inlet pressure exceeds 40 bar two additional stator passages are opened until all eight passages respectively 80% of the circumference are opened.

Eventually it was decided that a single stage axial impulse turbine with partial admission is best choice for the described boundary conditions. In an impulse turbine the conversion from pressure energy to kinetic energy only takes place in the stator. Hence, there is no pressure gradient in the rotor and the flow is only redirected. This means that the stator has to expand a pressure ratio of up to 49 with the described boundary conditions which leads to a supersonic flow at the outlet of the stator. To minimize the expansion losses the stator passages are formed like Laval nozzles. The working principle of a Laval nozzle can be described using the right side of figure 2. The flow enters the Laval nozzle at subsonic velocity ① and is accelerated until it reaches the speed of sound in the throat ②. Afterwards the flow is accelerated to supersonic speed in the divergent part of the nozzle ③. The Laval nozzle is chocked for all examined operating points. Hence the mass flow through the nozzle is only dependent on the inlet temperature, the inlet pressure, and the cross-section of the nozzle throat. Components or fluid conditions downstream of the nozzle, including the rotational speed of the turbine do not influence the mass flow. The shape of the Laval nozzle stator are designed with the help of an in house code which uses the typical equations for Laval nozzles to calculate the cross sections of the trapezoid shaped passages. The rotor blades are designed with a commercial design tool. The cross sections of the rotor passages are constant from inlet to outlet to avoid any acceleration. The radii of the blade leading and trailing edges are thickened to 0.2 mm to improve the mechanical integrity against abrasion. The blade profiles shapes of the stator and the rotor are shown in figure 2.

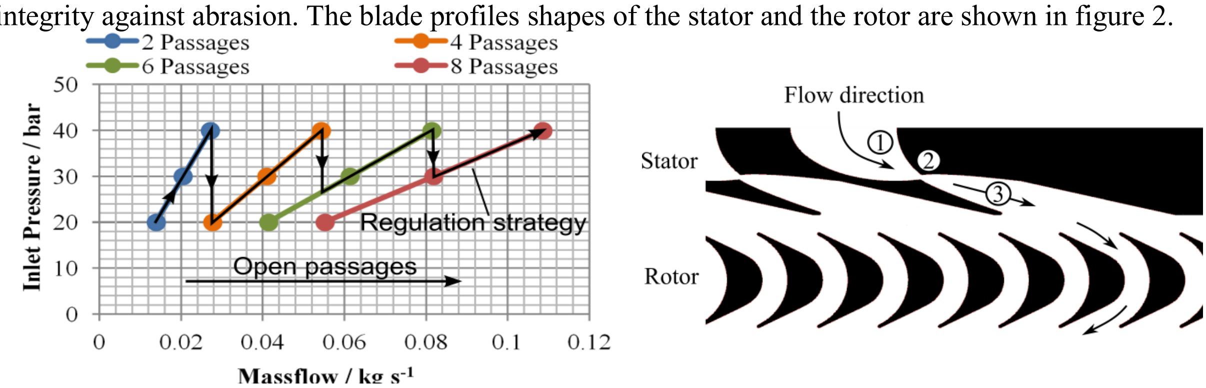

Figure 2: Left: Regulation strategy; Right: Blade profile of the stator and rotor

The geometrical parameters of the single stage impulse turbine, designed with the process described above, are listed in table 3. The turbine is designed for a partial admission of 40% at a rotational speed of 100,000 min^{-1}. At this operating point the predicted turbine power output is 8.0 kW. With the maximum degree of partial admission of 80% a turbine power of 18.3 kW is achieved at the maximum rotational speed of 110,000 min^{-1}. The rotor has 33 blades whereas the stator has 8 passages. The blade height is only 3.4 mm with a shroud diameter of about 63 mm.

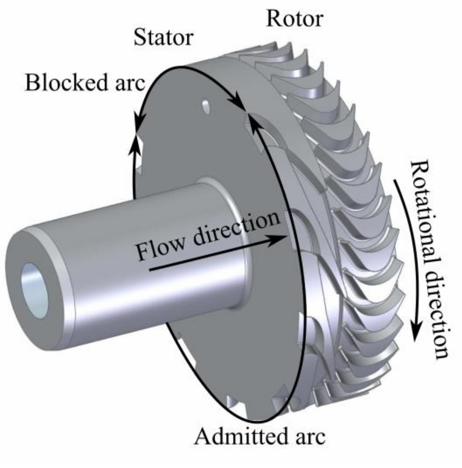

Table 3. Parameters of the investigated turbine

Parameter	Unit	Value
D_{Shroud}	m	0.0631
h	m	0.00343
δ	m	0.00013
$n_{Operation}$	min^{-1}	60,000...110,000
n_{Design}	min^{-1}	100,000
N_S		8
ε_{Design}		40%
ε		20%, 40%, 60%, 80%
N_R		33

Figure 3: Model of stator and rotor

The turbine consists of the a stator with Laval nozzles and a turbine wheel with impulse blades, as shown in figure 3. The stator is made of steel and the rotor blisk is made of titanium. The lighter weight of the titanium is beneficial for the rotor dynamics compared to a steel blisk. To avoid corrosion a small part palladium is added to the titanium. Both turbine parts are made via milling.

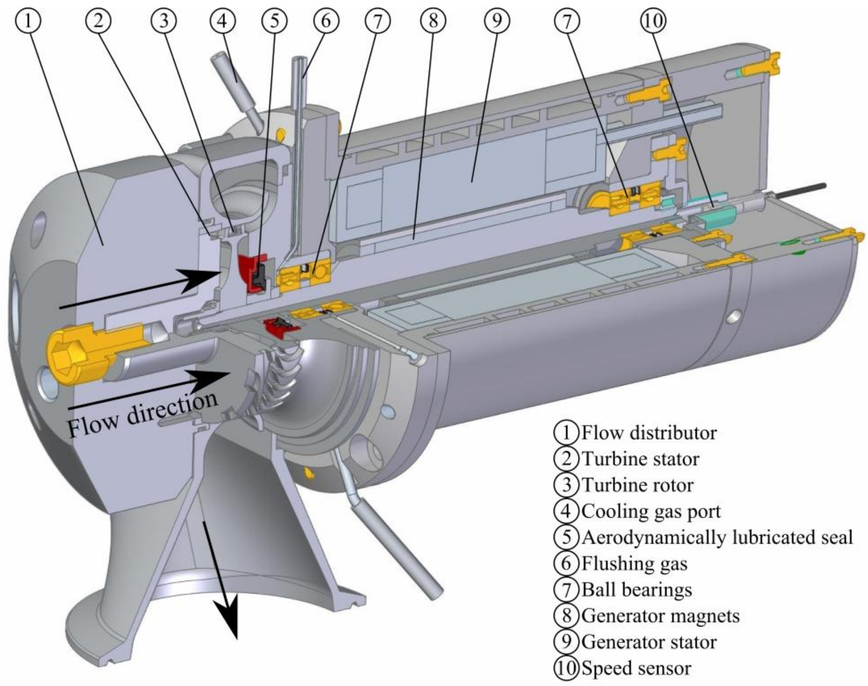

Figure 4: Cut section view of the mechanical design

A cut section view of the mechanical design of the turbine generator unit is shown in figure 4. The ethanol steam enters the flow distributor ① through three individual pipes which are necessary for the different degrees of partial admission. Inside the distributor are channels which guide the flow to the individual stator nozzles ②. Each nozzle is sealed by an O-Ring to avoid bypass flow. The flow is accelerated inside the stator and enters the rotor ③ with supersonic speed. The rotor converts the kinetic energy of the flow into rotational energy of the shaft. Downstream of the rotor the flow is redirected radially inside a diffuser. Finally the flow is collected in a plenum which guides the flow to the outlet.

The turbine rotor is mounted in an overhung position on a shaft equipped with permanent magnets ⑧. Through the magnetic field the mechanical power is transferred to the generator stator ⑨. The shaft is supported by two pairs of high precision ball bearings ⑦ with lifetime lubrication. In order to avoid a contamination of the ethanol cycle with air the shaft is equipped with an aerodynamically lubricated face seal ⑤. This face seal transports small amounts of ethanol from the cycle into a collecting space. The collecting space is permanently flushed with nitrogen through a port ⑥.

4. Testing of the ORC turbine generator

The designed and manufactured ORC turbine generator unit is tested on a multipurpose test bench of the University of Hannover. The basis of test bench, which is shown in figure 5, is a turbocharger test bench with a dual burner for the examination of passenger vehicle and commercial vehicle turbochargers. For ORC measurements an ORC-module is connected to the large burning chamber which provides the hot exhaust gas for the cycle. The design of the ORC-module is focused on versatility and robustness rather than size and weight. Therefore only approved components that fulfill all governmental regulations are used instead of small and lightweight parts for automotive applications. However, the modular design of the test bench allows an easy exchange of components meaning that not only ORC turbines but all parts of an ORC cycle can be tested under different conditions. For this purpose the test bench can be operated with several working fluids and not only the turbine but also each component of the cycle is highly instrumented.

Figure 5: ORC test bench of the University of Hannover

Nevertheless, the major task of the ORC test bench is the performance evaluation of expansion machines for ORCs. A simplified scheme of the test bench including the and the main measurement locations is shown in figure 6.

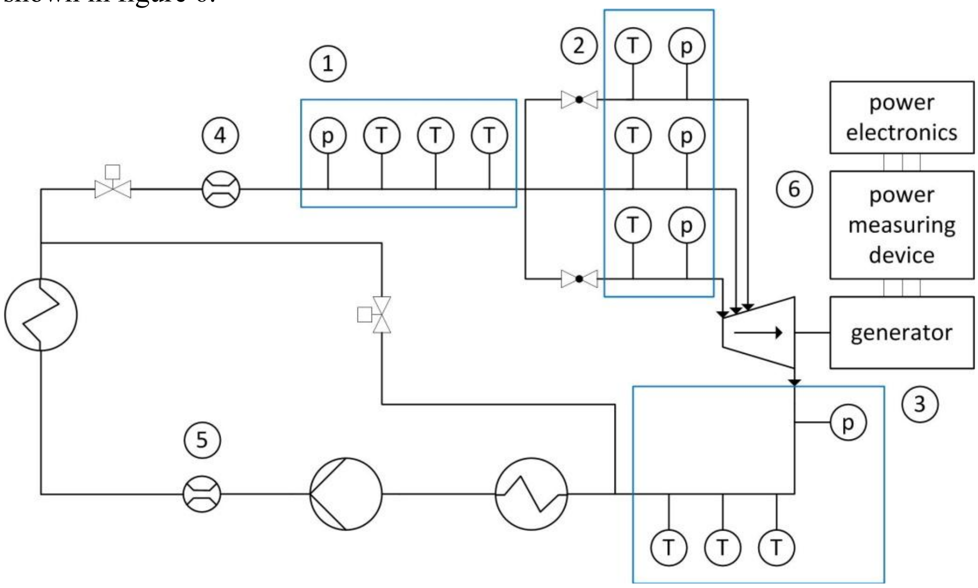

Figure 6: Simplified scheme of the test bench and the main measurement positions [2]

Instrumented tubes at the inlet ① and outlet ③ of the turbine generator unit are used to measure the temperature and pressure of the fluid. Additional temperature and pressure probes are implemented inside the turbine directly in front of the stator ② to reduce the influence of thermal losses, which are impairing the measurement accuracy of the efficiency. The mass flow is determined in the vapor phase right in front of the turbine ④ with a flow meter and in the liquid phase behind the pump with a mass flow meter ⑤. The test bench is equipped with a bypass for the start procedure as well as for security reasons. The bypass is located between the evaporator and the turbine and splits the flow into two parts. One part flows through the turbine and the other part flows directly to the condenser. The flow meter in front of the turbine measures only the flow which passes through the turbine whereas the mass flow meter in the liquid phase measures the complete mass flow from the pumps. If the bypass is completely closed both meters measure the same turbine flow.

The electric power of the generator is measured with a high accuracy power analyzer for high frequencies ⑥. The analyzer is located between the generator and the power electronics which allows the determination of the turbine isentropic efficiency, if the mechanical efficiency of the expansion machine is known.

All shown experiments are conducted with a constant static outlet pressure of 0.81 bar. The inlet temperature is set in dependency of the variable inlet pressure to prevent condensation. This is achieved by setting the inlet temperature 10 K above the temperature needed to avoid condensation at the turbine outlet, assuming an ideal isentropic expansion inside the turbine. The rotational speed can be set freely, however only the results of the rotational speed with the highest efficiency are shown in this paper.

5. Numerical simulations

For the numerical performance prediction of the ORC turbine a combination of CFD calculations and empirical methods is used. For the first step the aerodynamic performance is calculated by means of CFD simulations. Secondly empirical calculations are used to take losses due to the partial admission into account.

The steady-state CFD calculations for the performance evaluation of the turbine are performed with Ansys CFX (ver. 14.5) using the SST turbulence model. The fluid is a mixture of 95% ethanol and 5% water which is modeled by the Aungier Redlich Kwong real gas model of Ansys. The computational domain consists of one stator and three rotor passages followed by an axial-radial diffuser. To match the pitch of all components the rotor has been geometrical scaled so that all parts cover 36° of the circumference. The stator and rotor and the rotor diffuser domain are connected with an axisymmetric frozen rotor interface which allows the interaction of the shocks between rotor and stator. The alternative mixing plane interface was not used because a test showed non-realistic reflections of the shocks at the interface. Mass flow rate and total temperature are used as inlet boundary conditions and static temperature is used as outlet boundary condition. All values for the boundary conditions including the rotational speed of the turbine are derived from the experimental results (see section 4).

The grid generation is performed with Ansys ICEM CFD for all parts. All numerical grids are structured grids with a y-plus value of about 30 so that the near wall velocity profile is computed using wall functions. A grid-independence study is performed with grids of 0.19 million, 2.6 million and 9.13 million nodes. The study shows that the efficiency deviation of the medium grid compared to the fine grid is 0.2%, which is assumed to be sufficient for this investigation.

As a result of the rotational periodicity condition of the numerical model no partial admission losses occur in the CFD calculations. To take the partial admission losses into account, the calculated efficiency values of the CFD are reduced using the empirical approach of Aungier [5]. The losses of the empirical model are composed of two parts. The first part is the windage loss that occurs in the non admitted arc of the rotor by reentry of the outlet flow into the inactive passages. The second part is the end of sector loss that occurs if a rotor blade passage enters or leaves the admitted part of the stator.

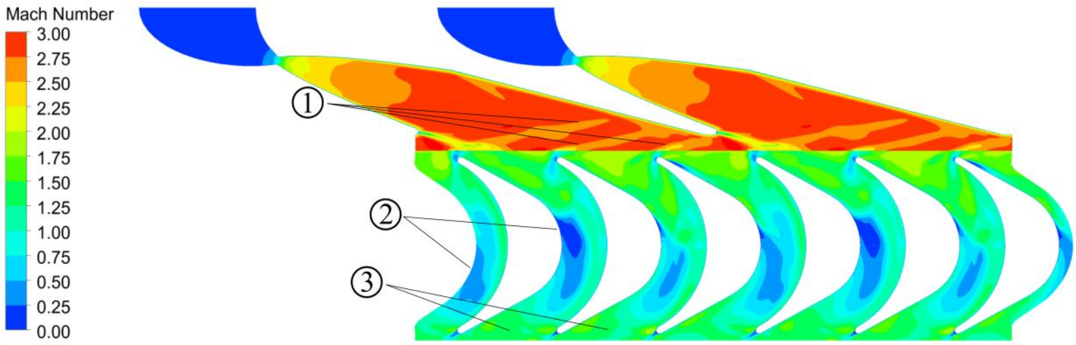

Figure 7: Mach Number at 50% span for an inlet pressure of 40 bar

Figure 7 shows the relative Mach number at 50% span of the stator and rotor at a rotational speed of 100,000 min^{-1}, which is the optimal speed for this operating point. The flow in the divergent part of the nozzle is highly supersonic (Mach number > 3). Close to the outlet of the nozzle several shocks ① occur which reduce the outlet velocity of the nozzle. The flow is even more decelerated by additional shocks at the leading edge of the rotor. This leads to a transonic flow with subsonic regions in the rotor ②. At the outlet of the rotor ③ the flow is slightly accelerated by a free expansion due to the remaining pressure gradient relative to the outlet pressure of 0.81 bar.

6. Comparison of experimental and numerical results

A comparison of the experimental and numerical results, achieved with the setup described above, is shown in the figures 8, 9, and 10 (for boundary conditions and experimental setup see section 4). All figures show the results for a degree of partial admission of 20% and 40%, which means that two respectively four passages of the turbine stator are opened. The results for two opened passages are shown in blue and the results for four passages in brown. The lighter colors represent the results of the CFD and the darker colors of the experiments.

In figure 8 the mass flow through the turbine is shown in dependency of the inlet pressure. The deviation between the numerical and experimental results is about 0.3% to 0.5% for two passages and 1.3% to 2.0% for four passages. The good agreement between both results is partly due to the behavior of the supersonic Laval nozzle used in the stator. For a given nozzle geometry the mass flow through the stator and with it also through the whole turbine is only depended on the inlet pressure and temperature. Whereas other values like the rotational speed do not influence the mass flow characteristic. For this reason the rotational speed of the turbine could be chosen to meet the highest power output and efficiency for each shown operating point without influencing the mass flow.

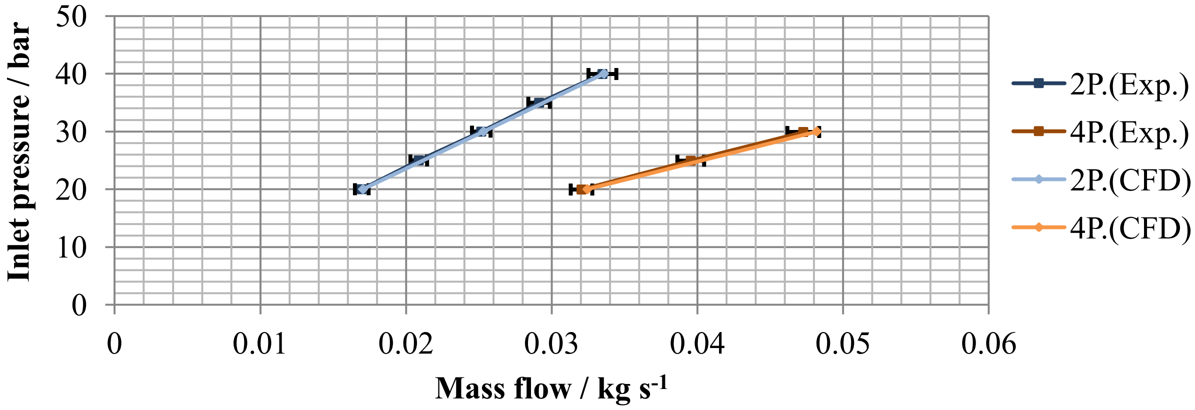

Figure 8: Mass flow of the turbine in dependency of the inlet pressure

Figure 9 shows the aerodynamic turbine power output. The turbine reaches a maximum power output of about 8 kW with four opened passages at an inlet pressure of 30 bar. With two passages the maximum power output is about 5 kW at an inlet pressure of 40 bar. The deviation from the turbine power between the numerical and experimental results is between 4.7% and -4.0% for two opened passages and between 3.8% and 2.9% for four opened passages. Based on these results it is estimated that the turbine reaches a maximum power output of up to 19 kW with all eight passages opened.

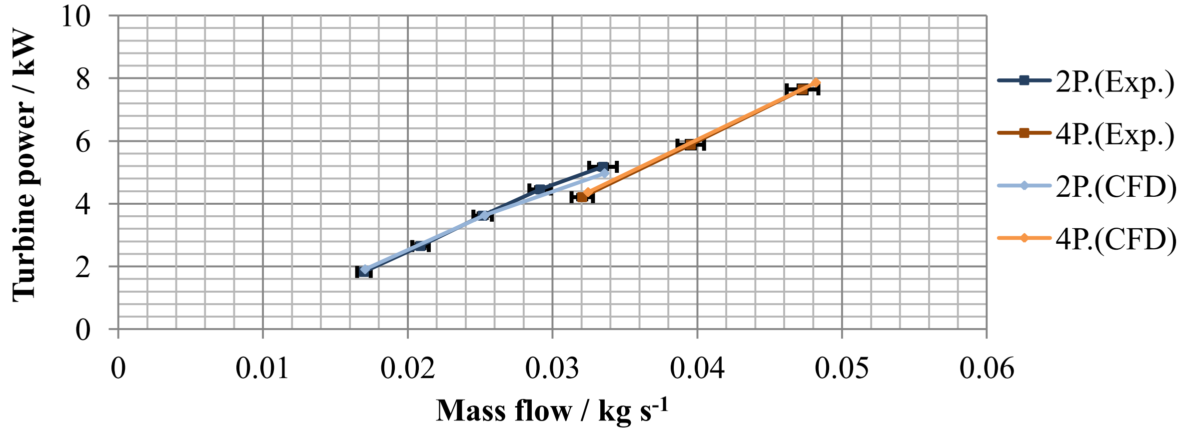

Figure 9: Turbine power output

The aerodynamic total-to-static efficiency of the turbine is shown in figure 10. The experimental total-to-static efficiency increases from 43.1% at the lowest mass flow and two opened passages to 57.0% at the highest mass flow and four opened passages. The deviation between the numerically and experimentally evaluated efficiencies is significantly larger for two passages with 2.1% to -3.3% than the deviation of four passages with 1.2% to 0.2%. Notably is the different trend of the numerical and experimental data for two opened passages. The numerical calculations predicted a decreasing efficiency of the turbine above an inlet pressure of 30 bar with two opened passages. Whereas the experimental data shows an increasing efficiency up to the maximum inlet pressure of 40 bar. No explanation for this discrepancy is found so far.

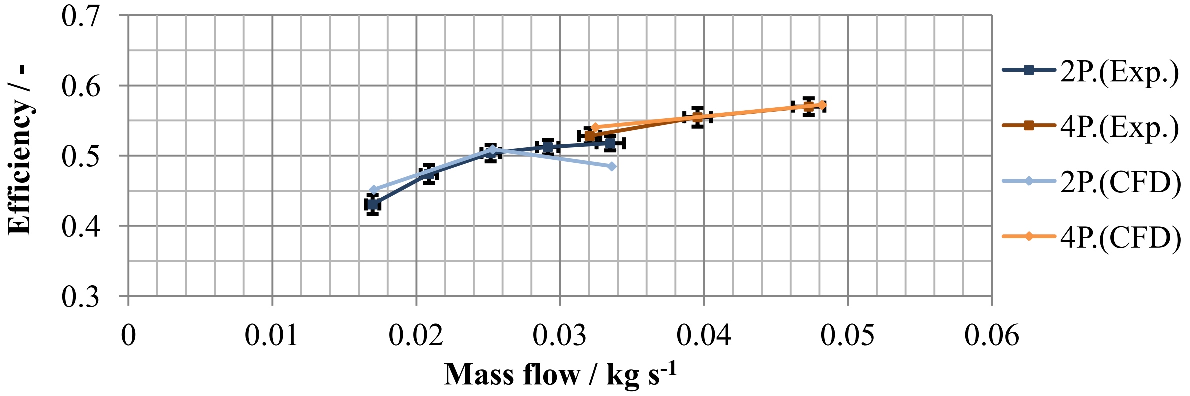

Figure 10: Efficiency of the turbine

7. Conclusions

An ORC supersonic turbine generator unit for an automotive applications has been developed at the TFD. The turbine is designed for challenging boundary conditions of a pressure ratio of up to 49 and a low mass flow of 0.045 kg/s. The turbine is designed as a single stage impulse turbine to achieve a compact design and to allow partial admission which is used to achieve feasible blade heights. The stator consist of Laval nozzles which accelerate the flow to supersonic speed. The rotor consist of an impulse turbine with zero degree of reaction. To cover a wide operating range the turbine is regulated using different degrees of partial admission between 20% and 80%.

A prototype was designed and manufactured. This prototype consists of two main components, the supersonic impulse turbine with variable partial admission and the high speed generator. The high speed generator and the turbine are mounted on one shaft. This leads to a compact and lightweight machine suitable for automotive applications.

The prototype was experimentally examined at the ORC test bench of the University of Hannover. The test bench is designed for a variety of ORC fluids and components. The objective of the experimental investigation was the validation of the numerical calculations and the functional verification of the ambitious design. The numerical calculations are a combination of CFD calculations and empirical methods to take into account the partial admission losses. The experimental results showed the proper functionality of the variable partial admission and of the high speed generator in a operating range between 2...8 kW.

The comparison between the numerical calculations and the experimental data showed a very good agreement. The deviation of the calculated mass flow is in average less than 1%. The measured turbine power output is up to 8 kW with a degree of partial admission of 40% and an inlet pressure of 30 bar. It is estimated that the turbine reaches a power output of up to 19 kW with its maximum degree of partial admission of 80%. Based on experimental measurements, the turbine reaches an aerodynamic total-to-static efficiency between 43.1% and 57.0%. Whereas the maximum deviation between the numerical and experimental efficiency is only 3.3%.

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