EXPERIMENTAL SETUP OF A SMALL SUPERSONIC TURBINE FOR AN AUTOMOTIVE ORC APPLICATION RUNNING WITH ETHANOL

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ABSTRACT

Waste heat recovery by bottoming Organic Rankine Cycles (ORC) is a promising method to increase the efficiency of automotive transportation based on combustion engines. The efficiency of these ORCs is significantly dependent on the efficiency of the expansion machine. For this reason, a supersonic impulse turbine with a variable partial admission was developed for such an ORC. The design of the blade profiles and the design parameters for this turbine are presented. The high rotational speed and the ambitious working fluid were the most challenging aspects in the design process of the expansion machine, hence the mechanical design of the expansion machine, the methods of power conversion, the seals and bearings are shown. The presented prototype has been manufactured and is been experimentally investigated at an ORC test facility at the Institute of Turbomachinery and Fluid Dynamics. This test facility provides two possibilities to measure the efficiency, which will be discussed; firstly, the determination of the turbine power with the enthalpy drop from inlet to outlet; secondly, the direct measurement of the power of the generator with the power electronics and a motor analyzer.

NOMENCLATURE

D	[m]	diameter
h	[m]	blade height
$\dot{m}$	[kgs^{-1}]	mass flow
N	[-]	number
n	[min^{-1}]	rotational speed
p	[bar]	pressure
		pressure probe
T	[K]	temperature
		temperature probe
δ	[m]	blade tip clearance
ε	[-]	partial admission

<u>subscripts</u>

in	inlet
out	outlet
R	rotor
S	stator
Shroud	diameter of the shroud
t	total thermodynamic conditions

<u>abbreviations</u>

FFKM	perfluoro-elastomer
ORC	Organic Rankine Cycle
PTFE	polytetrafluoroethylene

1 INTRODUCTION

Automobile manufacturers are currently focused on reducing fuel consumption and emissions. This can be achieved by increasing the efficiency of the combustion engine. The investigation of the energy flow in such a combustion engine showed that a significant part of the added chemical energy is lost as thermal energy in the exhaust gas (Bourhis and Leduc (2010)). A possible way to increase the efficiency of these combustion engines is to recover some of that energy. Different systems for such a recovery are conceivable, e.g. thermo electric systems, a Stirling motor, a Joule Cycle or an Organic Rankine Cycle (ORC). Span et al. (2011) performed a thermodynamic investigation of the last three processes and compared the achievable power output. They identified the ORC as the most suitable for the temperature levels encountered in the engine exhaust.

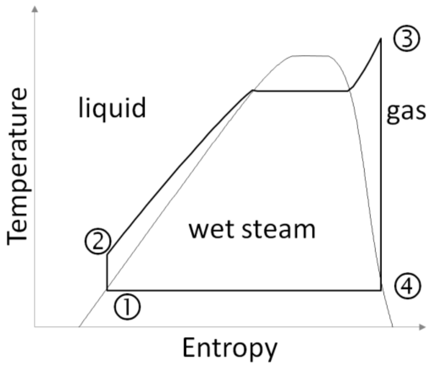

Figure 1: Isentropic ORC, schematically

The ideal isentropic temperature-entropy plot of such an ORC is schematically shown in figure 1. The efficiency and power output of such an ORC is highly dependent on the working fluid used, on its boundary conditions, and the expansion machine. The working fluid has to be selected with consideration of the allowable temperature and pressure levels. For this reason, suitable fluids must be determined for each individual application. Additionally, the overall efficiency of the ORC is highly dependent on the efficiency of the expansion machine. As the ORC application in automobiles is still under development, no standard expansion machine has established yet. Potential expansion machines for this application are: Piston expanders (Glavatskaya (2012)), screw expanders (Oomori and Ongino (1993)) and turbines (Patel and Doyle (1976), Freyman et al. (2008)). Piston expanders and screw expanders use the displacement principle, thus they seem to be more appropriate for the high pressure ratio and the low mass flow. However, supersonic impulse turbines are also able to be operated at these boundary conditions (Kunte and Seume (2013), Verneau (1987), Uusitalo et al. (2014)). In addition, they promise a more compact design. Experimental data of such turbines are rarely accessible in the literature, because the introduction into the commercial market has not yet occurred. A supersonic impulse turbine is being developed for this application at the Institute of Turbomachinery and Fluid Dynamics Hannover, in order to determine the performance potential experimentally and to validate the numerical simulations.

The scope of this work is to present the aerodynamic and the mechanical design of the turbine. The supersonic flow requires the use of special blade profiles. These blade profiles and the resulting turbine parameters are described. The resulting turbine is mounted on one shaft with the generator, which requires a special high-speed generator and demands suitable bearings and seals. For this reason the chosen overall design of the expansion machine is presented. This expansion machine will be investigated by experiments to determine the performance. However, the challenge of an accurate performance determination is also well known from turbochargers, where heat losses at the turbine housing can falsify the measured performance. Based on that experience, two independent methods are provided in the test facility for an accurate determination of performance, in term of power and efficiency.

2 The Boundary conditions of the ORC

Table 1: Operating conditions of the turbine

Parameter	Units	Design	Min	Max
Fluid	ethanol (95%mass), water (5%mass)			
$\dot{m}_{S,in}$	kg s^{-1}	0.045	0.006	0.091
$T_{S,in}$	K	539	458	539
$p_{t,S,in}$	bar	40	10	40
$p_{R,out}$	bar	0.81	0.81	0.81

The diesel engine of a truck with a cubic capacity of 12.8 liters and a maximum power of 375 kW was chosen as heat source. An operating point of medium load with a power of 228 kW is the selected design operating point. In prior studies, Kunte and Seume (2013) performed a thermodynamic analysis at this operating point to determine the optimal boundary conditions of the ORC. It was shown that using ethanol as the working fluid promises the highest power output of the ORC for the given exhaust gas conditions of the combustion engine. Deviating from the prior study, a mass fraction of 5% water is now added to the ethanol. This additive has no thermodynamic purpose, but it improves the corrosion resistance of the titanium turbine rotor, which can be damaged by stress corrosion in pure ethanol environments. Additionally, the inlet temperature of the turbine must be raised from 530 K to 539 K to avoid condensation and droplet impact erosion damage to the turbine rotor. The higher process temperature, the higher thermal capacity, and the higher enthalpy of vaporization of water reduce the mass flow from 0.052 kg/s to 0.045 kg/s. The maximum inlet pressure of 40 bar and the outlet pressure of 0.81 bar remain unchanged. Additionally a minimum and a maximum operating point were defined. These operating points have no corresponding operating point in the combustion engine, rather they represent the boundaries of the performance map of the turbine. The resulting thermodynamic boundary conditions are summarized in table 1.

2.1 Aerodynamic design of the supersonic impulse turbine

The chosen boundary conditions for the ORC are quite challenging for a turbine. Firstly, the mass flow is very low. This leads to a small turbine diameter and a low blade height and consequently effects like wall friction and tip gap losses will increase. Secondly, the pressure ratio of 49 is very high for a single stage turbine. As this turbine should be used in an automotive application, the premise was to use a single stage turbine to minimize the manufacturing cost and the size.

However, Verneau (1987) showed that supersonic impulse turbines expand high pressure ratios per stage at a good level of efficiency. In spite of the high pressure ratio, this type of turbine promises an acceptably low rotational speed, which can be used by a gear or a generator. Additionally, this type of turbine can be operated at partial admission. That means that only a part of the circumference of the stator is equipped with passages while the rest of the circumference is blocked. Although partial admission causes additional losses, it allows the blade height to be raised to an acceptable value. In this case, the reduced tip gap loss can exceed the losses due to the partial admission and thus the partial admission can raise turbine efficiency.

A supersonic impulse turbine with partial admission was designed to this end. Impulse turbines are characterized by degree of reaction close to 0. This means, that the whole pressure ratio is expanded in the stator and the subsequent rotor only redirects the flow without a change in speed. The blade profiles chosen for this turbine are shown in figure 2. The stator passages are formed like Laval nozzles, consisting of a convergent and a divergent part. The flow enters the stator in the convergent part of the nozzle at subsonic velocity ① and is accelerated to the speed of sound up to the throat ②. A further acceleration takes place in the divergent part of the nozzle ③. The design specifications from Humble et al. (1995) were used, to determine the radii of the throat and the shape of the

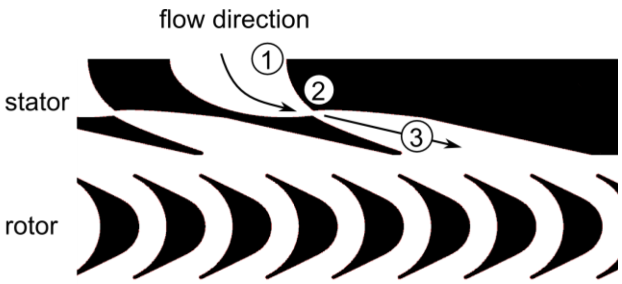

Figure 2: ORC Blade profile of the turbine

divergent part of the nozzle, which is formed like a bell nozzle. The rotor blades are formed like a redirecting blade row. The cross section of the passage is constant from inlet to outlet. The leading and trailing edges of the rotor should be as thin as possible because of the supersonic approach velocity of the rotor. However, very thin leading and trailing edges are not able to withstand the expected aerodynamic forces and to ensure the mechanical integrity of the blade. Already Boxer et al. (1952) have given geometrical specifications for such leading and trailing edges. However, the small dimensions of the turbine make the implementation of such geometries very challenging due to manufacturing tolerances. Thus a radius of 0.2 mm was chosen for the leading and trailing edges.

A model of the turbine design is shown in figure 3. The flow angles were determined by a flow coefficient of 2 and a 0 degree of reaction. This results in a shroud diameter of 63.1 mm and a rotational speed of 85,000 rpm at the design point. At this point, the turbine operates with a partial admission of 40 %. This leads to a rotor blade height of 3.43 mm. This low blade height can cause very high tip gap losses, if the radial tip gap is chosen too large. Hence, a minimum tip gap of 0.13 mm was determined after consideration of manufacturing tolerances, thermal expansion, and centrifugal forces.

The amount of thermal energy, provided by the combustion engine of the truck, varies over time. As a result, the mass flow of the ethanol-water-mixture varies, too. This would lead to a highly differing inlet pressure of the turbine, if the total area of the nozzle throats would be fixed. However, Kunte and Seume (2013) showed, that a variable partial admission of the turbine allows adjusting the overall area by opening additional nozzle throats, to keep the turbine inlet pressure within an acceptable range. This ensures to operate the Laval nozzles under nearly ideal conditions at a high efficiency. The designed turbine is able to vary its degree of partial admission stepwise between 20 %, 40%, 60% and 80 % with a predicted power of up to 19 kW. The design parameters of the turbine are listed in table 2.

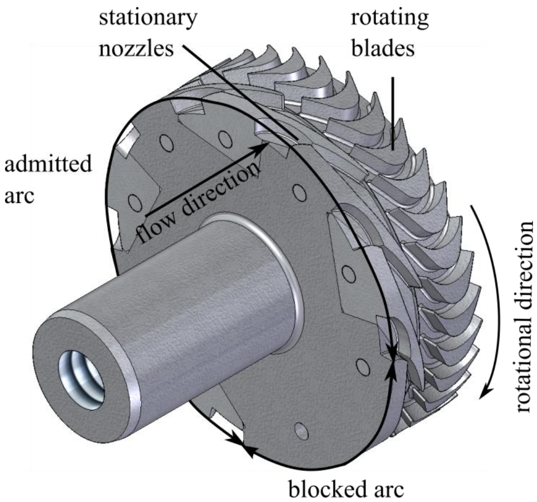

Figure 3: Model of the turbine

Table 2: Parameters of the investigated turbine

Parameter	Units	Values
D_{Shroud}	m	0.0631
h	m	0.00343
δ	m	0.00013
n_{Design}	rpm	85,000
N_s		8
ε_{Design}		0.4
ε		0.2, 0.4, 0.6, 0.8
N_R		33

2.2 Mechanical design of the expansion machine

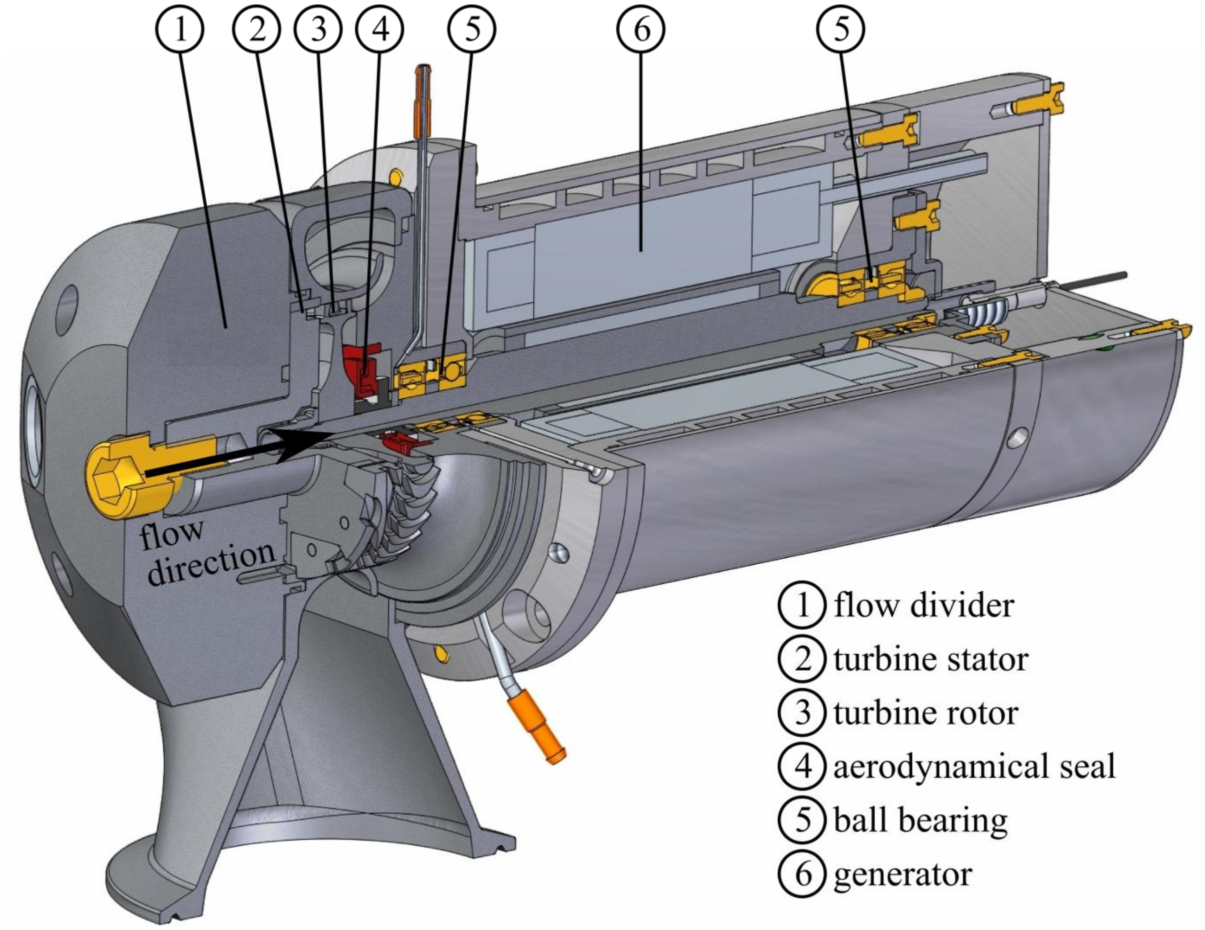

Figure 4: Mechanical design of the expansion machine

Due to the high rotational speed of the turbine, it is not easy to utilize the mechanical power gained. A direct coupling with the combustion engine is not possible because of the differing rotational speeds. Two possibilities were considered to allow the utilization of the power generated: Firstly, reducing the rotational speed with a gear box and secondly, directly coupling the turbine to a high-speed generator. A comparison of these options showed advantages of the high-speed generator over the gearbox. The generator has a more compact expansion machine design and generators suitable for these rotational speed and power levels are technically realizable. Additionally, suitable gears for this application are not available on the market as a standard unit and have no cost advantage over the generator. Hence, a turbine generator unit was designed.

This expansion machine is depicted in figure 4. The flow enters a flow divider ①, which splits the flow and guides it to every individual stator nozzle. The stator nozzles ② are attached to a stationary disc with an anchor, which connects the stator with the divider. Each of these stator nozzles is individually sealed on the contact surface to avoid bypass flows between the stator nozzles. After the stator, the flow passes the turbine rotor ③ which converts the kinetic energy of the flow into rotational energy. Behind the rotor, the flow direction is turned radially and the flow is collected in a plenum. The turbine rotor is mounted on the shaft, which transfers the rotational power to the generator rotor. The seal used is an aerodynamically lubricated axial face seal ④. The shaft is mounted on two pairs of high precision ball bearings 5. The turbine rotor is in an overhung position. This is disadvantageous for the rotor dynamics, however it makes sealing the ethanol cycle easier. The generator is placed in the middle of the bearings. These components will be described in detail in the following sections.

2.2.1 Seals

The ORC can be divided into a high pressure section and a low pressure section. In the high pressure section temperatures of up to 539 K and pressures of 40 bar must be sealed. Leaks must be avoided to prevent the formation of explosive gas mixtures outside the machine. On the other hand, the seals must avoid the penetration of air into the low pressure section of the turbine, which would contaminate the working fluid and thus deteriorate the cycle. In the high pressure section, only static seals are necessary. In addition to the static seals, dynamic seals are necessary on the rotor shaft in the low pressure section.

The O-rings in the high-pressure section as well as in the low-pressure section are made of FFKM. FFKM is resistant to ethanol and its use is permitted for operating temperatures up to 598 K, which is much higher than the maximum temperature of 539 K in the cycle. Additionally, FFKM is a flexible elastomer. For this reason, FFKM is suitable for the O-rings, which have to be stretched, for example for positioning in notches of a shaft seal. One disadvantage of FFKM is the high cost of this material. For this reason, a different sealing material was searched for the flat seals, which provides the required thermal and chemical resistance, but does not need the flexibility of FFKM. PTFE and graphite seals satisfy these requirements. Both PTFE and graphite are resistant to ethanol, but graphite has a higher thermal resistance of up to 673 K compared to 533 K for PTFE.

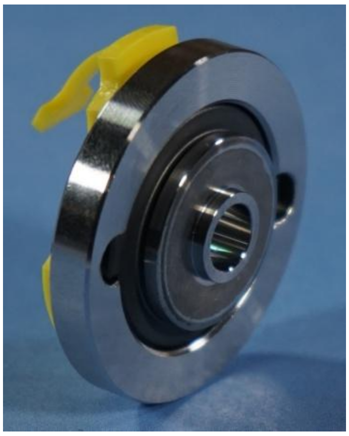
Figure 5: Aerodynamically lubricated seal

A generator housing, which is enclosed together with the ORC, is not an option because the ball bearings and their lubricant are not resistant to ethanol. Thus, the fast rotating shaft has to be sealed to separate the generator cavity from the turbine cavity with the ethanol. However, the sliding speeds are too high for contacting seals while non-contacting seals such as labyrinth seals have to be provided with purge air for a complete separation from the Rankine cycle. Therefore, an aerodynamically lubricated seal is used instead to seal the shaft. This seal was developed for turbocharger applications to minimize oil losses of the bearings (figure 5) (Simon et al. (2010)). This seal consists of three main components, a rotating steel element, a stationary graphite ring and the housing. If the shaft is not rotating, a spring pushes the non-rotating ring in axial direction against the steel element. When the shaft starts to turn, grooves on the front face of the rotating element induce a radially-oriented flow film between the stationary ring and the rotating element, thus separating the two elements. The shape of the grooves transports a small amount of ethanol against the pressure gradient into a cavity. This cavity is purged by a permanent nitrogen flow. This means a small loss of ethanol over the time, but prevents the contamination of the ethanol cycle. The housing material of the seal was modified, due to the fact, that the standard aluminum housing is not resistant to ethanol. For this reason, stainless steel housing was used.

2.2.2 Bearings

Figure 6: Ball bearings and spring

For the shaft, different bearing solutions were investigated. Three requirements were defined, which should be fulfilled by the bearing. Firstly, the bearing should work oil-free, respectively the bearings should have their own seals to avoid a pollution of the generator with oil. Secondly, the rotordynamics should be favorably affected, which prohibits the presence of harmful eigenfrequencies in the operating range. Thirdly, the bearing and their design should be low-priced. These demands can be met with high precision ball bearings. Ball bearings (HCB7000-C-2RSD-T-P4S) from the Schaeffler Group were used in a paired O-arrangement and are pre-loaded by a spring (figure 6). This arrangement was chosen, because it improves the rotordynamic behavior due to its higher bending stiffness. These bearings are also able to absorb the axial forces of the turbine, hence no additional axial thrust bearing is necessary. They are lubricated for life and self-sealed.

2.2.3 Generator

The generator used is a high-speed synchronous generator with a maximum power of 19.5 kWe and a maximum rotational speed of 110,000 rpm (ATE (2012)). The bipolar permanent magnet, mounted on the shaft is made of neodymium. The magnet itself is covered by a steel jacket, which resists the centrifugal forces. Due to the high power density of the generator, water cooling of the generator is required. Thus, the generator stator is mounted in a cooling jacket made of aluminum. This cooling jacket also functions as the generator housing.

3 Measurement methods for the efficiency determination of the turbine

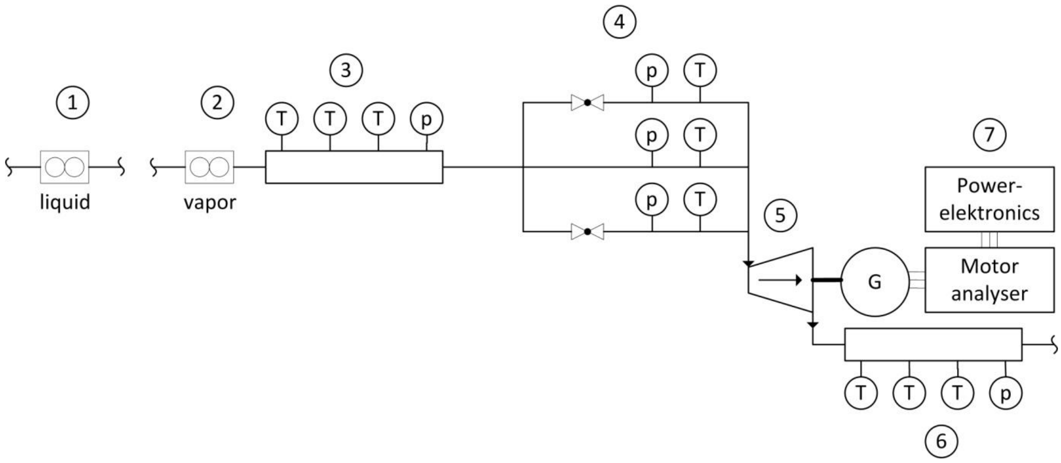

Figure 7: Instrumentation for the determination of the turbine efficiency

Instrumentation for the efficiency determination is shown in figure 7. Two mass-flow meters are positioned in the cycle. The first one is positioned in the liquid section of the test facility ①. At this position, the temperatures are low, which simplifies the measurement and increases the accuracy. However, the test facility has a bypass which can direct the flow past the expander. Therefore, a second flow meter is integrated, which measures the flow passing through the expansion machine ②. It is possible to operate the cycle without bypassing. In this case, the mass flow meters measure the same flow and can be compared. The inlet and the outlet of the expansion machine are equipped with three temperature probes and a pressure probe ③⑥. This arrangement ensures that the average temperature and pressure profiles are measured and enhances the accuracy of the efficiency determination. Additionally, the expansion machine is equipped with its own temperature and pressure probes. These probes are positioned in each of the tubes of the control unit for the variable partial admission directly in front of the turbine inlet ④. This unit consists of three tubes, which can supply a part of the stator nozzles with ethanol. One of these tubes is continuously opened and represents the minimal partial admission. The degree of partial admission can be controlled with ball valves, which open additional tubes to vary the degree of partial admission. Within this, the temperature and pressure probes have two purposes. Firstly, the pressure probes check the correct function of the ball valves by comparing the pressure level with the target pressure. Secondly, pressure and temperature losses occur in the pipes between measuring tube ③ and the turbine inlet. These losses can be detected by the additional probes in front of the turbine and the accuracy is improved. Using the measurement of these thermodynamic conditions, the energy flow out of the system can be determined. In adiabatic systems, this energy flow represents the turbine power from which the efficiency can be calculated.

However, it is known from experiments with turbochargers, that turbine applications are not adiabatic and heat losses can significantly affect the measured efficiency (Casey and Fesich (2010)). For this reason, a second possibility for the determination of the turbine power is implemented. The power electronics provides the possibility to measure the torque and power of the expansion machine, too (Sieb & Meyer (2013)). The power electronics are not intended for measurement applications, which require a very high accuracy. For this reason, an additional motor analyzer is required, which is able to determine the power with the desired accuracy. However, the measured power is not the turbine power, because friction losses in the bearings and in the seal occur. These losses have to be investigated and determined in order to consider them in the final determination of the turbine efficiency.

4 Description of the test facility

Figure 8: ORC test facility of the Leibniz University Hannover

The experiments will be performed on an ORC test facility (figure 8). This test facility is designed with approved components that comply with all governmental regulations, such as the pressure equipment directive or the ATEX directives. Size-optimized components, which are currently in development for automotive applications do not fulfill these requirements and are not used for the standard configuration. However, the test facility has a modular design, which enables the exchange of individual components. This gives the opportunity to investigate other prototype components of ORCs with this test facility, as well. The currently implemented instrumentation is designed to measure the operating conditions of each component. Additionally, the test facility is designed for several working fluids besides ethanol.

5 Conclusion

Automotive waste heat recovery is a promising opportunity for increasing the efficiency of vehicles powered by internal combustion engines. Prior studies have shown that an ORC is a most suitable candidate for such a recovery. However, the efficiency of these ORCs is significantly dependent on the efficiency of the expansion machine.

A supersonic impulse turbine has been developed at the TFD for this application. This impulse turbine is faced with challenging boundary conditions, firstly a high pressure ratio of 49 and secondly a very low mass flow of 45 g/s. In the aerodynamic design, special blade profiles were used which promise a high efficiency under these conditions. The stator is formed like a Laval nozzle with a divergent bell-shaped nozzle part. Additionally, a partial admission of 40 % was chosen for the design point to maintain acceptably high blade heights in spite of the small mass flow. Furthermore, automotive applications demand high variability in the expansion machine to cover the performance map of the combustion engine. For this reason, the degree of partial admission can be varied stepwise between 0.2. 0.4, 0.6, 0.8. This turbine promises an output power of up to 19 kW.

A prototype of this turbine was manufactured at the TFD to validate it experimentally. The high rotational speed and ethanol as a working fluid, made the design of the prototype challenging. The turbine and the generator are mounted on one shaft. A high speed generator was used because of the high rotational speeds. The shaft was mounted with two pairs of preloaded ball bearings, which are greased for life. An aerodynamically lubricated seal, developed for a turbocharger application, was used to seal the rotating shaft and avoid the leakage of ethanol. FFKM, PTFE, and graphite are used for static seals.

Additionally, instrumentation for the determination of efficiency was implemented which provides two options for the determination of efficiency. Firstly, the enthalpy drop from inlet to outlet can be measured. Secondly, the power output and the torque absorbed by the generator can be measured. This instrumentation is part of the ORC test facility of the Leibniz University at Hannover.

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