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Development of a 10 kW class axial impulse single stage turboexpander for a micro-CHP ORC unit

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Abstract. Development of micro ORC systems with 1-15 kW power output for micro-cogeneration and waste heat recovery at the Czech Technical University in Prague, University Centre for Energy Efficient Buildings (CTU UCEEB) has over ten years of history with many successes. These include 6 different ORC units, all with in-house designed rotary vane expanders (RVE) of many versions throughout this development. Among main advantages of the RVE belong relatively simple and robust design at low cost even at very small series of single-unit production and all that with acceptable efficiency. The ORC units operate with hexamethyldisiloxane (MM) working fluid at high pressure ratios and expansion ratios and the isentropic efficiency of RVE has a limit at these conditions around 60%, often however only at values around 50%. While this might be enough on a cost side for commercialization of this technology, in pursuit of higher efficiency solutions, different expander technology needs to be selected. A turbo-expander is a logical choice with prospect of higher efficiency. At the same time, a literature review has found a lack of reported detailed experimental data for micro (5-50 kW) turbo-expanders, possibly hindering global development towards economically feasible solutions. A project named Dexpand, "Optimised expanders for small-scale distributed energy systems" aims at these issues by objectives in designing, optimizing, manufacturing and testing several ORC expanders with MM and isobutane and their subsequent performance mapping and comparison. One major task is a design of a turboexpander for a 120 kWth biomass fired microcogeneration ORC unit currently operated at the CTU UCEEB. An axial impulse single stage turboexpander was selected as a suitable choice, providing a prospect of a decent efficiency at technically manageable rotational speed and size. This paper provides a detail of currently performed design activities, starting from boundary conditions specification, over development and optimization of a 1D model, preliminary 2D CFD calculations and finishing in a state of a robust and detailed 3D CFD model with a real gas model. Note that the working fluid, high molar mass organic vapour, is highly non-ideal in its behaviour and the flow conditions with pressure design ratio around 13 is highly supersonic (nozzle outlet isentropic Mach number exceeds 2). The current results based on 3D CFD indicate a prospect of an isentropic efficiency 71% at mechanical power output of 11 kW. Lastly, ongoing and future work is outlined, which includes aerodynamic optimization based on the developed 3D CFD model and construction design of the entire turbine assembly.

1 Introduction and overview

Organic Rankine cycle (ORC) power systems became an unrivalled technical solution and an industrial standard in several applications, such as low temperature heat utilization in geothermal systems, combined heat and power (CHP) systems in the scale of several MWs down to hundreds of kW or waste heat recovery (WHR) power systems down to dozens of kilowatts. ORC systems in small to micro scale domestic CHPs face mainly economical barriers with economy-of-scale and large cost per installed kilowatt in micro scale units. [1,2]

When focusing on micro scale ORC CHP with electrical output in the order of less than 10 kW, many laboratory units and prototypes have been built and tested. Regardless of these R&D efforts, these micro scale ORC power systems mostly have not seen commercialization, the rest have not yet been proven to be economically feasible or are until today very scarce on the market. Mainly because their installations face economical barriers with economy-of-scale, since downscaling the ORC power systems to micro scale results in high specific costs associated with low initial production quantities and large cost per installed kilowatt. [3] Another great reason which disfavours small scale ORC power systems is the lack of reliable, efficient and cost-effective expansion machines.

For medium and large-scale ORC power systems, a turbine is the state-of-the-art vapour expansion device. For the small to micro scale systems though, volumetric expanders dominate the market. Main reason for that is the possibility to derive the design of a small volumetric

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expander from an off-the-shelf volumetric compressor, common machine in refrigeration and compressed air technology, connected with low investment costs and reliability. [4]

Turboexpanders for small scale ORC plants are nowadays rather scarce and seldom used. Other reason for that may be that volumetric machines are able to operate in a wide range of conditions with decent efficiency compared to narrow design points of turboexpanders. Apart from that, volumetric machines are bringing mostly drawbacks in an increased complexity of the system, additional lubrication system is usually needed, wear occurs on the contact surfaces and the sealing is rather complicated.

This contribution discusses a preliminary thermodynamic and aerodynamic design of a turbine for an already existing small-scale ORC CHP plant, currently in operation with a rotary vane expander (RVE) and hexamethyldisiloxane (MM) as a working fluid as described in Figure 1 in a process flow diagram. The replacement with a turbine solution promises not only an increase in efficiency of the expansion process but maybe even more importantly an increase in reliability of the operation of the expander by omitting any contact surfaces between the stator and rotor. As no wear on these surfaces occurs, an increase in the lifespan of the machine is expected as well. All of these results are the key turning points for a turboexpander development. [5]

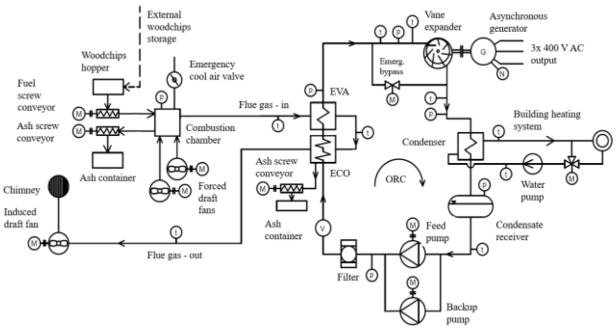

Fig. 1. Process flow diagram of the ORC CHP unit.

2 Turboexpander design

This chapter presents the design methodology and procedure for an ORC micro scale turboexpander. The design procedure of the ORC micro turboexpander follows several generations from the proof of concept unit for preliminary cold air testing towards the prototype of the turbine working with organic vapour.

2.1 Boundary conditions and turboexpander parameters

The boundary conditions to the turboexpander given from the cycle conditions are displayed in Table 1. Main relevant parameters of the cycle that determine the turbine design are the following: expander inlet and outlet pressure, inlet temperature, nominal mass flow rate and the nature of the working fluid; these cycle boundary conditions are supposed to be conserved with a new expander as well.

Table 1. Turboexpander design boundary conditions defined by the cycle performance.

Parameter	Value	Units
Expander inlet pressure p_{in}	650	kPa
Expander inlet temperature T_{in}	190	°C
Expander inlet superheating T_{SH}	10	K
Expander outlet pressure P_{out}	50	kPa
Working fluid	MM	-
Working fluid mass flow rate m_{wf}	0.3	$kg.s^{-1}$

According to Table 1 one may observe that the design pressure ratio for the turboexpander is around 12, and the inlet temperature of the superheated organic vapour is 190°C. For these inlet conditions, the speed of sound is 123 m/s, and it is quite obvious from this preliminary estimate that the fluid flow at the nozzle outlet for an isentropic enthalpy drop of around 50 kJ/kg will be highly supersonic. For ORC turbines with such high Mach numbers (above 1.4), it is necessary to design the nozzles as uniquely-shaped convergent-divergent to respect the nature of the supersonic fluid flow. [6]

The working fluid selection – hexamethyldisiloxane (MM) impacts the design of the turbomachine heavily. Due to its nontoxicity, excellent thermal and chemical stability, and limited flammability, it can be possibly considered as the most suitable fluid for such high-temperature ORC from an engineering point of view. [7–9] Other reasons for its application were also a reasonable price, that it remains liquid under ambient conditions, solubility of oil and previous unpleasant experiences with hydrocarbons as working fluid.

The working fluid can be considered as a dry fluid – meaning it works with a positive slope of the saturated vapor curve (ds/dT). This brings several peculiarities – such as an ensured expansion into the superheated steam region without any danger of condensation nuclei or droplets formation.

Another factor is the high molar mass and the molecular complexity of the organic fluid which results in low volumetric flow rates (an order of magnitude lower compared to steam at the same p, T) and high volumetric ratios (though not as high as for steam), low enthalpy drops along the expansion line and very low speed of sound. These aspects highly influence the turbomachinery design.

The complex organic fluids show a very non-ideal behaviour especially in the region which is of the upmost interest in turbomachinery — the vapour single phase region. As the consequence, rather complicated real gas Equations of State (EoS) have to be utilized during the design phase (nowadays possible thanks to fluid properties libraries such as REFPROP or CoolProp). Siloxanes as members of so-called Bethe-Zel'dovich-

Thompson (BZT) fluids family exhibit in the single vapour phase region a negative fundamental derivative in gas dynamics Γ - therefore all the thermodynamic properties show high sensitivity to its values and thus for precise enthalpy calculation, many experimental data and specific EoS (e.g. Peng-Robinson-Stryjek-Vera EoS) are needed. [7,10,11]

The implications of that are severe and of various character. On one hand, it helps to design very compact and little loaded machines resulting in rather cheap turbomachinery. On the other hand, low speed of sound means that the presence of a supersonic flow is inevitable and the blade design is nonconventional with convergent-divergent nozzles. Therefore, the occurrence of shock waves and their interactions with boundary layer can be observed in the fluid flow.

High volumetric ratios for very complex molecules lead to difficulties in processing the large difference in the volumetric flow rate in a single-stage machine which often has to be compensated by a significant change of the blade height along the streamline, partial admission and very high flow deflection angles. The variation of blade height comes with a penalty in the form of lower stage efficiency as it leads to an increased vorticity of the flow caused by the perpendicular velocity component and therefore increased secondary losses. The losses are furthermore increased by the high deflection blade shapes.

For micro to small scale ORC applications - be it automotive internal combustion engine bottoming ORC, domestic CHP units or small scale concentrated solar power plants - the expander cost makes the major share of the cost of the entire unit. Therefore, the aim of the authors' design is not to maximize the isentropic efficiency of the machine at any costs but rather to optimize it for reasonable price while reaching sufficient efficiencies and reliability. A single stage turbomachine is thereby preferred in such systems over multistage optimised turboexpanders.

High rotational speeds are often avoided during the design of a steam turbine. On the contrary for ORC turbines this is not the case, as the blade loading is usually very low due to much lower enthalpy drop and the temperature ranges in which the turboexpanders operate are also much lower compared to conventional steam turbines. Hence, the blades are much less mechanically stressed and high rotational speeds are not such a large concern from the mechanical standpoint. Also, the turboexpander as a whole is generally smaller and it is possible to apply permanent magnet high speed generators.

For all of the above stated reasons, a design approach of an ORC turbine cannot be simply derived from the design methodology of a steam turbine and has to follow a completely different method taking into the account all the aspects, specifics and peculiarities of the organic fluid flow. Traditional zero-dimensional statistical diagrams such as Smith charts or Baljé map can provide only a very rough estimate for the isentropic efficiency, because they are calibrated for large scale steam turbines. [9] The effect of highly supersonic flows at the stator outlet and strong shocks, large blade height variation and high deflection buckets should not be neglected even in the rough estimate.

2.2 1-D meanline design of the ORC turboexpander

This chapter discusses the preliminary thermodynamic and aerodynamic design of an axial impulse stage turbine for the given ORC CHP unit and boundary conditions. The 1-D meanline design is based on energy and mass balance in each state of the turbomachine and with several assumptions:

- Steady states at the inlet, stator outlet and rotor outlet
- The thermodynamic properties are calculated for midspan diameter
- Conservation of rothalpy principle is applied
- Conservation of moment of momentum
- Cycle boundary conditions are to be conserved and not affected by the replacement of the expander
- The kinetic energy of the working fluid at the inlet is neglected
- Any heat losses from the turboexpander to the environment are neglected
- All geometrical properties have to be kept within the limit of manufacturability
- The kinetic energy at the outlet is considered not to be any recovered
- Rotor buckets designed with constant channel width and equal but opposite relative flow angles
- The incidence and deviation angles are assumed to be zero
- Loss correlations adopted from turbine drives of rocket turbopump

The model was created using MS Excel with VBA and REFPROP add-in for fluid properties.

Since the designed turboexpander is meant to be affordable and simple to manufacture, assembly and operate, the desired combination of high power output and a compact design makes the impulse turbine a favourable choice compared to the reaction turbine. These may offer higher efficiency, at the cost of a larger machine. An axial configuration is chosen for the simplicity of manufacturing and easier design of the turboexpander. Also, partial admission can be easily adopted for axial impulse stage turbines. Finally, there were previous experiences from the experimental trials with cold aerodynamic tests of axial impulse stage turbines during the works on the master's thesis of the author.

High pressure ratios, transonic flow velocities and compressible flow effects along with the proximity of the saturation curve lead to the complicated combination of a non-ideal gas and compressible flow. In the commonly and commercially adopted 1-D meanline models, one of the two usually assumes the absence of the other. In the designed model, real fluid behaviour is therefore assumed, with a non-ideal gas model, the heat capacity ratio thus cannot be considered constant and is calculated for each state of the turbine.

For the initialization of the turboexpander design, some of the design variables had to be chosen with respect to the rotational speed, diameter of the machine and some other geometrical components. Another very important parameter to be estimated in the first step of the design is the isentropic efficiency of the turboexpander. These initial values and guesses are reported in Table 2. Some of these parameters were kept constant throughout the design phase and some were optimized, such as the rotational speed of the turbine and the blade height.

Table 2. Chosen input parameters and guesses; parameters denoted with * were later optimized – optimized values in brackets.

Parameter	Initial value	Units
Rotational speed n*	24000 (28000)	rpm
Midspan diameter Dmid	100	mm
Nozzle outlet flow angle α2	13	°
Isentropic efficiency guess	70	%
Partial admission guess e*	58.5 (97.5)	%
Blade height ratio*	0.1 (0.06)	-
Minimum blade height h_min	5	mm
Rotor blades aspect ratio AR	2	-

To better illustrate the design of the axial impulse stage, Figure 2 below shows the resulting velocity triangles of the turbine.

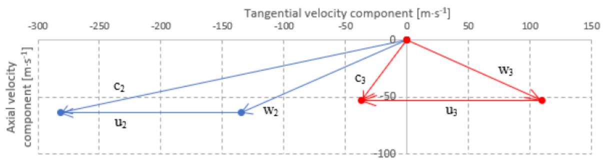

Fig. 2. Velocity triangle of the designed ORC impulse stage.

2.3 Design of the flow components

The stator is designed as a series of convergent-divergent de Laval nozzles. As the flow suffers sever real gas effects and is highly supersonic, the resulting nozzle geometry is very unconventional, cambered and unlike any stator blade geometries that could be found in an airfoil atlas. The geometry of the nozzle is thus designed as a combination of the divergent part, which for such highly supersonic flow impacts the efficiency of the nozzle the most and therefore more effort is put into designing it, and the convergent part.

The rotor blades were designed as constant channel width buckets with geometry derived from [12]. The shape of the buckets is again unconventional and highly cambered since it experiences large flow deflection. Thus, conventional airfoil for turbine rotor blades cannot be used, and the geometry is designed specifically for this type of application. The blades are simple cylindrical as they are easy to manufacture with standard technology. To reflect the increase in pressure within the rotor wheel due to the friction losses of the supersonic fluid flow, an increase in crossflow area is designed by means of linearly increasing blade length along the chord line of the blade. The aspect ratio was chosen as 2 as recommended by Agromayor and Nord in [13] for this type of axial impulse supersonic turbines in ORC. The blades were designed as unshrouded due to the manufacturability constraints.

3 Numerical simulation

Figure 3 shows the geometry of the computational model domain. The model consists of two computational domains S and R. Domain S contains 2 stator blade channels in the shape of a Laval nozzle. The rotating domain R contains 7 intermediate blade channels typical of a straight turbine stage. Both domains have the same tangential pitch of $2\pi/5$. This means that the computational model includes 1/5 of the flow section of the turbine.

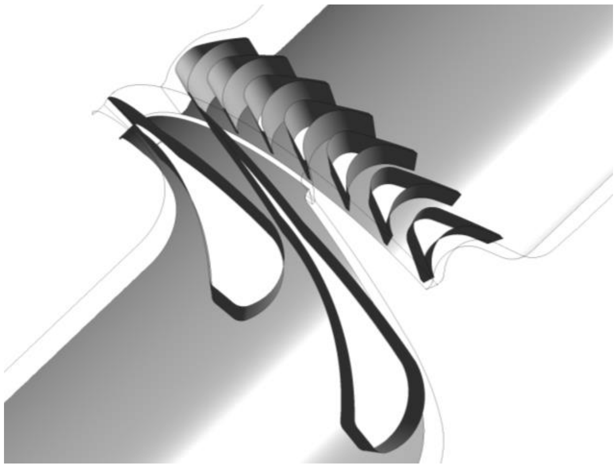

Fig. 3. Computatiomal model of the element of the single stage axial impulse turbine.

3.1. Numerical model settings and boundary conditions

The computational domains are connected using the Frozen rotor interface model. The rotating domain R is set to a very high speed of 28000rpm. The SST k-ω turbulent model is used for the simulation. Since a very high Mach number is expected in the flow section of the turbine, a High-Speed Wall Heat Transfer Model is also used. A pressure-inlet type boundary condition with total pressure and total temperature is set at the input of the computational model. At the output of the calculation model, a pressure-outlet boundary condition with static pressure is used. The set input and output parameters of the working medium are shown in Table 1.

3.2. Fluid properties of the organic vapour

The real gas properties of hexamethyldisiloxane (MM) must be included in the CFD analysis, as it plays an important role in the quality and accuracy of the analysis. An automatic generator of RGPgen v2.0 (Real Gas Properties) for ANSYS CFX was used in this simulation. This RGPgen, developed by Numsolution, can generate a set of material properties of any working medium in a form that is directly accepted by ANSYS CFX.

3.3. Mesh generation

The networks of both computational domains are prepared in ANSYS ICEM. Both domains are meshed with hexahedral cells and the computational model contains a total of 24.4 million hexahedral cells. The surface network detail of both computational domains is shown in Fig. 4.

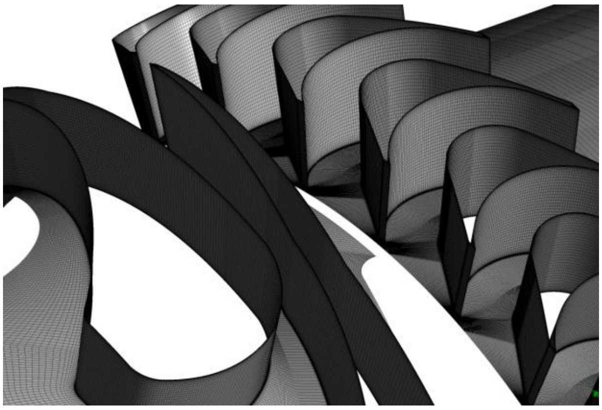

Fig. 4. Close-up onto the generated mesh of the stator-rotor interface.

4 Results & discussion

Results of the numerical simulation are reported within this chapter. These results are to be validated by an experimental campaign as the next goal of the project to develop a turbine for the aforementioned ORC CHP unit.

To describe the nature of the fluid flow through the turbine stage, contours of the Mach number in the stationary frame on midspan position are presented in Figure 5. Siloxane vapour at the admission pressure enters the stator nozzles, where in the convergent part of the nozzle increases its speed up to the speed of sound in the nozzle throat. Divergent part of the nozzle is designed as a linear segment increasing the nozzle area up to the designed outlet nozzle area, achieving Mach 2 at the outlet.

Significant shockwave structures at the stator vanes trailing edge are apparent. These shockwaves greatly determinate the fluid flow at the rotor inlet. There are again strong oblique shockwaves at the leading edge of the rotor buckets. These oblique shockwaves affect the fluid flow inside the channel – interaction of the shockwave with the boundary layer at the suction side of the bucket results in a flow separation, therefore secondary losses occur in the rotor wheel.

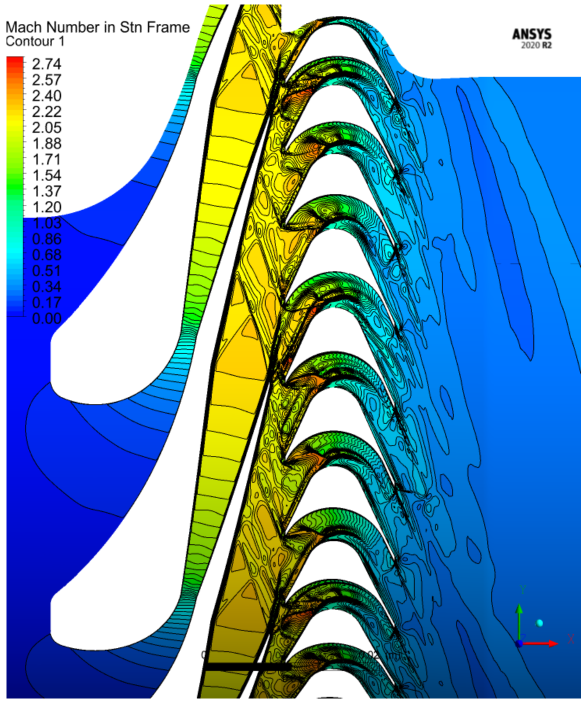

Fig. 5. Midspan contours of Mach number in stationary frame.

To assess the fluid flow in the spanwise direction, components of the velocity vector along the blade span are plotted, as well as the velocity angles of the fluid flow in Figure 6 below. Thanks to the optimization of the rotor blade outlet flow angle, the effect of the flow separation at the suction side of the rotor blade was decreased, as can be observed on the rather monotonic and flat behaviour the courses of the velocity compounds exhibit along the span at the leading and especially the trailing edge.

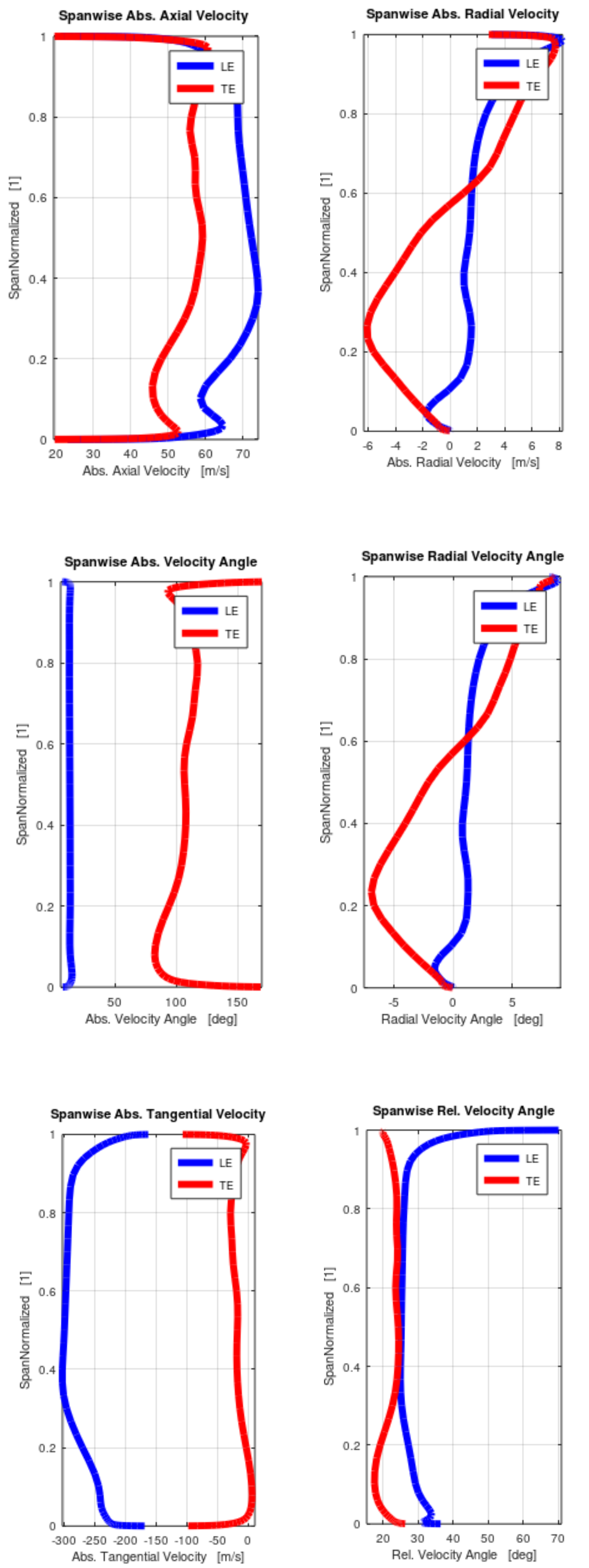

Fig. 6. Variation of the averaged components of the velocity vector along the span of the rotor blade at the leading edge and the trailing edge.

Results of the 3D CFD model were finally compared with the result of the optimized 1D model with loss correlations. Some of the parameters are listed in Table 3 below.

Table 3. Comparison of the achieved parameters inbetween the 1D meanline model and the CFD simulation.

Parameter	1D model	CFD model	Units
Nozzle outlet/Rotor inlet			
Mach number	1.95	2.01	-
Static pressure	55	64.3	kPa
Static enthalpy	298.9	296.2	kJ/kg
Rotor outlet			
Mach number	0.46	0.48	-
Static pressure	55	54.9	kPa
Static enthalpy	302.7	298.2	kJ/kg
Isentropic efficiency ηis	69.2	71.1	%
Mechanical power output	10.8	11.2	kW

4.1. Conclusion and future works

This paper presented an overview of the current progress in the DEXPAND project focused on the development of optimised ORC expanders for various solutions, in this case an axial impulse stage turbine for a biomass CHP ORC plant, and especially the numerical CFD simulation carried out in ANSYS CFX with real gas fluid properties. The rated mechanical power output of the turbine according to the CFD simulation is assumed to exceed 11 kW and total-to-static isentropic efficiency of over 71%.

Future works in this regard consist of further optimization using a Concepts NREC AxCent tool, designing the whole turbine assembly, manufacturing and assembling it and finally delivering an experimental campaign at the biomass CHP ORC unit at CTU UCEEB with a prospect of replacing the current volumetric expander.

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