CUSTOMIZED SMALL-SCALE ORC TURBOGENERATORS – COMBINING A 1D-DESIGN TOOL, A MICRO-TURBINE-GENERATOR–CONSTRUCTION KIT AND POTENTIALS OF 3D-PRINTING

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ABSTRACT

Due to the various possible applications in the ORC waste heat recovery business i.e. different heat sources, heat flow rates, temperature levels, pressure levels and working fluids, it is not appropriate to design and build one standard turbogenerator to stock. Therefore, the authors developed a very flexible "micro-turbine-generator-construction-kit (MTG-c-kit)" by means of which a customized turbine generator can be designed and built for any required power output between 1 - 200 kW, for a wide range of working fluids and boundary conditions – quickly and cost efficiently. The architecture of this construction kit and its main features are introduced briefly in this paper. However, more focus is put on the developed 1D turbine design tool. It allows to design and optimize a single stage turbine for any fluid and any boundary conditions very quickly. The implemented models are discussed by means of a specific design examples of axial impulse and radial inflow cantilever turbines. Experimental results of two micro turbines, which were designed and built according to the above-mentioned approach, are presented and the agreement and discrepancies between measured and design data are discussed.
In order to further accelerate and cheapen those customized micro turbine generators for small and low temperature applications, we provide an outlook of possibilities of additive manufacturing methods like 3D-printing of plastic turbine wheels for example. These new possibilities provide potential to improve the competiveness of small-scale ORC in waste heat recovery. A prototype of a simple turbine design with plastic 3D printed wheels has been manufactured and first tests are presented.

1. INTRODUCTION

Distributed generation is playing an increasingly important role in modern power systems. The reasons behind that include environmental (local renewable resources, avoiding grid transmission loss), economic (local cheap resources as biomass or waste heat), strategic (independence in volatility periods, resilience of virtual power plants) and other matters. Systems with small power output also bring the possibility of energy production to a wide range of potential customers. Many technologies considered for these systems require an expander as the key component for producing work. Examples of these technologies include micro/small Rankine cycles (RC), organic Rankine cycles (ORC) and other novel thermodynamic cycles for biomass combustion and waste heat recovery, micro solar plants, reversible heat pumps, small compressed air energy storage (CAES) or pressure recovery in gas and steam lines instead of throttling.

1.1 Small micro-expanders

There are two fundamental choices of expanders. The first is a volumetric expander including piston, scroll, screw, rotary vane or Wankel types. The advantage is that these expanders may often be derived from compressors. They are considered to be well fitted for small volumetric flow rates, while achieving decent efficiency and reasonably low rotational speed. (Quoilin *et al.*, 2013; Imran *et al.*, 2016)

The second option is a dynamic expander, i.e. a turbine. It is standard for applications with a higher power output than about 500 - 1000 kW_{el}. In smaller applications they are rather rare. Their major types are radial inflow turbines (90 ° IFR), radial cantilever turbines and axial turbines.

For small systems, generally single stage turbines are considered. Currently 90° IFR turbines are often favoured, for example (Macchi and Astolfi, 2016; Alshammari *et al.*, 2017; Pini *et al.*, 2017). The reasons for this include its mass application in turbochargers for internal combustion engines and the theoretically achievable high efficiency. The design point of these turbines is at a very high rotation speed. The issues for the limited number of actual commercial applications of this turbine type include the availability, costs and life span of bearings for high rotation speeds, together with a complex geometry, where the customization of very small series is too expensive and time-consuming.

In order to overcome the drawbacks of turbine development and application, we proposed a concept, allowing both modularity and uniformity of general turbine design together with customization to specific needs at the level of blading modifications. This resulted in the micro-turbine-generator-construction-kit (MTG-c-kit) for power output of 1-200 kW_{el} described in this work.

1.2 Additive manufacturing for micro turboexpanders

Furthermore, in order to allow even faster development and manufacturing of the customized turbine blade wheels, especially for the smallest applications with low temperatures, additive manufacturing (AM) methods including plastic materials are further proposed in this work. In order to show the current state-of-the-art, a review of major technologies and their micro-turbine applications are shown below.

Among the first AM technology was stereolithography (SLA) based on UV curing of resin. Most materials have temperature limits around 70 °C but special resins for “weak loading” up to 289 °C exist. Fine tuning of properties as well as improved precision and speed are expected from novel modifications such as continuous liquid interface polymerization “CLIP”. An experimental kW-scale radial inflow air turbine made almost entirely from resin by SLA (Rahbar *et al.*, 2017) has been reported. Note that maximal rotational speed in the experiments was below the nominal design point, suggesting issues either in bearings or the material itself. Regardless, fine precision of this method (layer thickness up from 25 μm) and thin sharp structures (trailing edges of blades) are still an issue. Circular tolerance can reach over 0.5 mm on a 120 mm wheel, when typical production practices (parts at a given angle within the production area) are performed. Support structures are also needed for SLA and leave burrs, which need to be manually polished.

Fused Deposition Modelling (FDM) is the most common AM technology among private and “home” applications. However, the quality of its prints can only rarely fulfil requirements of turbomachinery, especially on rotating parts due to product inhomogeneity. The minimum layer thickness is around 80 μm (typical < 150 μm) but thermal effects cause large deterioration of accuracy. The circular tolerance on the outer circumference of 120 mm testing wheel with professional manufacturing still exceeded 0.5 mm. Support structures for overhanging structures are necessary and leave burrs after removal. Generally, the surface quality requires additional treatment due to high roughness of the printed parts. In terms of materials and their temperature resistance, materials with a temperature resistance of over 80 °C, exceptionally above 100 °C (Ultra ABS) can be obtained. This method has been applied for prototype 90° IFR turbines in (Hernandez-Carrillo, Wood and Liu, 2017), but experimental tests are yet to be reported. Additionally, this technique finds application in less stressed parts, where surface quality is not critical, such as radial inflow turbine casings (Hernandez-Carrillo, Wood and Liu, 2017; Hernandez, Liu and Wood, 2017). Suggestions to use this method mostly for stator components can be supported also by studies of NASA, which tested possibility of manufacturing guide vanes for turbojet engines by FDM from materials as thermoplastics or Ultem® with carbon fiber. (Grady *et al.*, 2015) (Chuang *et al.*, 2015).

Selective Layer Sintering (SLS) of polymer powder, typically containing nylon and composite mixtures, provides an interesting option for small turbomachinery. The height of individual layers is possible starting from about 60 μm. Together with no need for supporting structures, it provides decent surface quality, even without further polishing. Tests on 120 mm wheels showed the best circular tolerance out of the tested AM technologies, being usually under 0.25 mm. The temperature resistance is typically around 80 - 100 °C, a little bit more in case of composite materials. Even though turbomachinery applications were not found in available literature, we find it features as accuracy and rigidity promising technology for micro low temperature applications.

Lastly the Direct Metal Laser Sintering (DMLS) technology is presented, which works on a similar principle as SLS, but with metal powders. Minimal layer height is similar but of the demand for support structures or large resulting roughness limits application for turbine wheels without post-processing. On the other hand, it has often been used for 90° IFR turbines such as (Arifin *et al.*, 2015; Pini *et al.*, 2017), though experimental data have not yet been reported. The possibility of creating cooling channels in this type of turbine has been succesfully explored for high-temperature automotive radial turbomachines. (Zhang *et al.*, 2018a, 2018b)

The potential of this technology for the MTG-c-kit along with the trial turbine design for AM wheels and the very first experimental tests are included in the last part of this work.

2. 1D TURBINE DESIGN TOOL

In order to comprehensively and reliably assess and compare different turbine concepts, a computational tool has been developed for following considered turbine concepts:

- Axial impulse turbines
- Axial two-wheel velocity compounded Curtis turbines
- Radial inflow cantilever quasi impulse turbines
- Radial inflow axial outflow reaction turbines (90° IFR)

It has been developed so that the main geometry data and the efficiency of a turbine can be assessed automatically and quickly with following inputs to the model:

- Working fluid and mass flow rate
- Pressure at the inlet and outlet and temperature at the inlet
- Wheel mean diameter and rotational speed

It is based on 1D meanline model with an in-house developed appropriate simple loss model for axial impulse, Curtis and radial cantilever turbines, which will be introduced in the following. For 90° IFR turbines, the open access loss model introduced in (Moustapha *et al.*, 2003) is applied.

There are some well-known turbine loss models in literature e.g. by (Craig and Cox, 1970; Kacker and Okapuu, 1982; Traupel, 2001) etc. All of them are mainly developed for and based on data from subsonic MW-class multistage steam and gas turbines working with full admission. These sophisticated loss models take into an account a huge number of flow and geometry parameters for predicting total pressure loss or enthalpy dissipation per blade row. However, this work focuses on single stage supersonic ORC micro turbines in the few or several kW-class, which often have to operate with partial admission. (Klonowicz *et al.*, 2014) showed that the classical loss models can lead to differences in turbine stage efficiency in the magnitude of more than 10 percentage points, especially for low specific speed turbines. This fact is not only a problem in terms of turbine performance, but also in terms of turbine geometry design: the required flow area depends on thermodynamic flow condition, which is directly influenced by losses. Thus, the authors looked for alternative loss models for these rather simple, highly loaded small turbine stages. They found some correlations in literature dealing with turbine drives of rocket turbopumps. These correlations have been continuously adjusted by in-house measurements.

Generally, in small-scale ORC applications, the required turbine pressure ratio leads to supersonic Laval nozzles. The velocity coefficient of the supersonic nozzle is calculated by Equation (1) which is derived from data given in (Reynolds, 1966).

$$\varphi_{No} = \left(1 - \left(0.0029 * Ma_{1,is}^3 - 0.0502 * Ma_{1,is}^2 + 0.2241 * Ma_{1,is} - 0.0877\right)\right)^{0.5} \tag{1}$$

The velocity coefficient $\varphi_{No} = c_{No}/c_{No,is}$ is a pure function of the isentropic nozzle exit Mach number $Ma_{1,is}$ i.e. of the nozzle pressure ratio.

The correlation for the blading velocity coefficient $\varphi_{bl} = w_{exit}/w_{inlet}$ of the pure impulse blading is also taken from former turbopump development (see (Beer, 1965)). The basic or profile velocity coefficient is a function of the blade relative inlet Mach number Ma_{r1} and the blade deflection angle $\Delta\beta$, Equation (2).

$$\begin{aligned} \varphi_{bl} &= 0.957 - 0.000362 * \Delta\beta - 0.0258 * Ma_{r1} + 0.00000639 * \Delta\beta^2 \\ &+ 0.0674 * Ma_{r1}^2 - 0.0000000753 * \Delta\beta^3 - 0{,}043 * Ma_{r1}^3 - 0.000238 * \Delta\beta * Ma_{r1} \\ &+ 0.00000145 * \Delta\beta^2 * Ma_{r1} + 0.0000425 * \Delta\beta * Ma_{r1}^2 \end{aligned} \tag{2}$$

Ventilation losses play a significant role for the considered small turbines due to partial admission which is often necessary. The degree of admission can be increased and the ventilation losses reduced if the blade height is reduced. However, secondary losses due to small blade aspect ratio (blade height h to blade chord s) are increased. To estimate ventilation and secondary losses, based on the investigation of (Watzlawick, 1991), the basic blade velocity coefficient is adjusted by a correlation assuming that secondary losses are linearly coupled to basic profile losses and inversely proportional to the blade aspect ratio.

Very often, partial admission has to be applied to avoid too small wheels and the corresponding high rotational speeds or too short blades, respectively. An internal design rule defines the range of blade height to mean diameter h/D_{mean} from 0.025 to 0.10. Thus, the minimum blade height for a 50 mm wheel is 1.25 mm which actually has already been put into practise. The original correlation (Equation (3), part in squared brackets) for the ventilation power P_V of a single wheel was taken from (Pfleiderer and Petermann, 2004) and has been adjusted to our design architecture . The leading factor “1.85/2” has been derived from in-house measurements on a 5 kW air driven test turbine. Its architecture corresponds to the MTG-c-kit.

$$P_V = \frac{1.85}{2} * \left[(1-\varepsilon) * \rho_{exit} * \left(\frac{n}{60}\right)^3 * D_{mean}^4 * 4.5 * h_{blade} \right] \text{ [W]} \tag{3}$$

The loss model approach is identical for an axial impulse, an axial Curtis and the radial inflow cantilever turbine. Of course, the static enthalpy drop due to the centrifugal pressure field is taken into account for the cantilever turbine. This centrifugal pressure field does not lead to an acceleration in the rotor blade channel, therefore the cantilever turbine is called “quasi impulse” (see (Weiß, 2015)).

The 1D-meanline-turbine-design tool calculates efficiency and power output based on the above described simple loss model. The loss model for nozzles and buckets allows the prediction of the static thermodynamic conditions of the working fluid at the nozzle throat, nozzle exit and at the blading inlet and outlet by applying REFPROP fluid properties (Lemmon, Huber and McLinden, 2010). Hence, the velocity triangles at mean diameter and the required cross areas can be determined in the form of nozzle angle, height and width, blade height and angle at inlet and outlet. Based on these few geometry specifications, an experienced designer is able to build up the 3D model of the turbine in CAD. With regard to manufacturing, prismatic blades with the height-diameter-ratio h/D_{mean} from 0.025 to 0.10 is used as mentioned before.

The 1D-meanline-turbine-design tool can be applied manually or in a semi-automatic mode. In the latter, the designer can define a parameter space in which, for example, the diameter, the rotational speed and/or the degree of admission can be varied with a given step size and the main results and input data of each individual design are stored. The designer can then choose the best compromise for the given task.

With the available 3D CAD model of the turbine, the next design step, the analysis of the turbine flow by means of Computational Fluid Dynamics (CFD), can be carried out. Like the simple 1D design tool, the 3D CFD calculations apply real gas data for the ORC working fluid taken from REFPROP. The analysis of the 3D flow field and the integral data of CFD allow the evaluation of the turbine performance and the identification of areas for improvement regarding the flow path. The geometry modifications are implemented in CAD and checked again by CFD before the final turbine is built.

3. MICRO TURBOGENERATOR CONSTRUCTION KIT

Previous development work (Weiß, 2015) suggested, that a single stage axial impulse turbine mounted on a permanent magnet high-speed generator is the best i.e. most flexible compromise. Compared to a (radial-inflow) reaction turbine (90° IFR), the axial single stage impulse turbine is able to process unlimited pressure ratios theoretically, requires lower rotational speed, does not produce axial thrust (from a pressure difference across the rotor) and can be designed with partial admission, which easily allows the implementation of smaller turbine outputs with the same dimensions and rotational speed. Therefore, the axial single stage impulse concept with optional partial admission has been chosen as standard for further development of the MTG-c-kit. Further design requirements for successful market applications are following:

- Hermetically sealed design
- Permanent magnet high-speed generator; power output range 1 - 200 kW_{el}
- Compact design (turbine wheel directly mounted on generator shaft - no additional bearings, gears or couplings, low material usage)
- As little design modifications as possible over changes in boundary conditions (pressures, temperatures, power rating, fluid)

The resulting design comes with following features:

- 5 different sizes over the above given range of boundary conditions
- Rotational speed range 10,000 – 70,000 rpm
- Integrally manufactured turbine wheel (Ø 50 – 250 mm)

The architecture of the micro-turbine-generator-construction-kit is depicted in Figure 1.

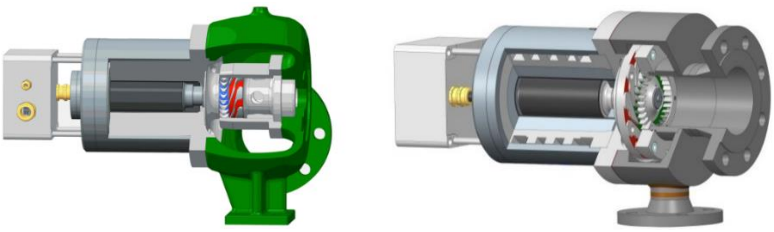

Figure 1: Architecture of the MTG-c-kit - left: standard axial impulse turbine, right: recently developed radial quasi impulse cantilever design (source:DEPRAG)

Design example for specific boundary conditions

From the authors point of view (see (Weiß, 2015; Weiß. *et al.*, 2018)), a single stage axial impulse turbine is the simplest and most flexible turbine concept for the requested variety of ORC applications and therefore the best compromise for the MTG-c-kit. However, if higher efficiency is required, a radial inflow cantilever turbine, following the “quasi-impulse” concept might be an alternative which fits in the MTG-c-kit architecture (see Figure 1) as well.

To prove these considerations, the authors designed a number of models based on the before mentioned two turbine types for the same duty by means of the 1D-meanline-turbine-design-tool and tested them in the ORC research plant of the Centre of Energy Technology at the University of Bayreuth, Germany (see (Weiß *et al.*, 2018)). The boundary conditions and the major design results of this work are given in Table 1.

Table 1: Turbine main design data

Parameter	Unit	axial impulse turbine	radial cantilever turbine
working fluid	-	Hexamethyldisiloxane (MM)	
wheel diameter D_{mean}	mm	120	
rotational speed n	rpm	24,000	28,000
degree of admission ε	%	100	100
pressure ratio PR (ts)	-	18.75	
degree of reaction r	%	0	13
nozzle exit Mach number Ma_1	-	2.11	1.98
rotor relative inlet Mach number Ma_{r1}	-	1.14	0.87
expected shaft power	kW	12 - 13	
predicted total-to-static isentropic efficiency (1D loss model) $\eta_{ts,1D}$	%	67.0	74.5

The main goal of this research project was to prove the predicted efficiency gain (1D: 74.5% to 67%) of the radial cantilever turbine compared to the standard impulse turbine. Figure 2 depicts examples of the semi-automatic design calculations for the axial impulse turbine (left) and the radial cantilever turbine (right). Each data point represents one 1D-meanline-turbine-design (not all are shown). The wheel mean diameter, the rotational speed and if necessary, the degree of admission were varied.

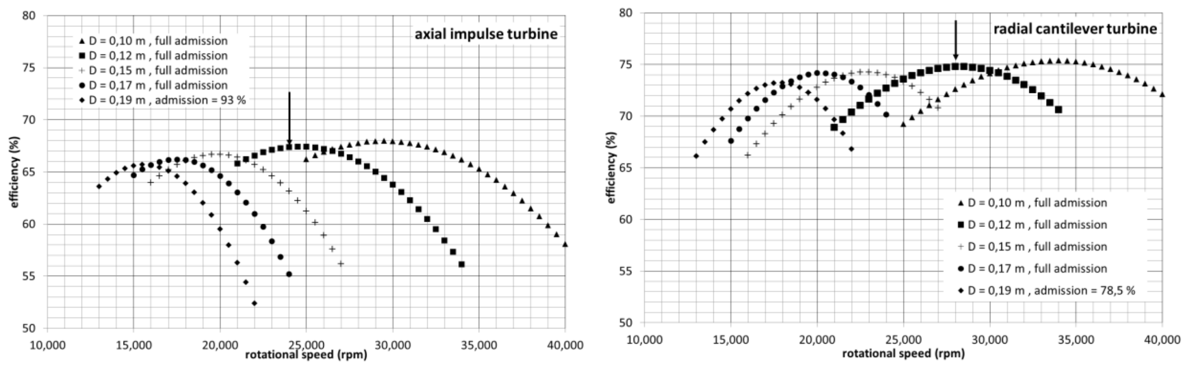

Figure 2: Predicted total-to-static isentropic efficiency of the 1D-meanline-turbine-design calculations

With regard to comparability, both machines should be identical with the exception of the nozzles and the wheel. The maximum allowable rotational speed of the identical generator is 30,000 rpm. Therefore, the axial impulse turbine, which was designed first, was equipped with a 120 mm wheel running at 24,000 rpm design speed ($\eta_{ts,1D} \approx 67\%$). The cantilever turbine needs 28,000 rpm for maximum efficiency ($\eta_{ts,1D} \approx 75\%$), due to the slight reaction across the wheel. Both turbine designs where simulated and optimized by CFD see e.g. (Syka and Weiß, 2018)) before they were manufactured.

The complete experimentally determined turbine characteristics for both turbines measured in the ORC test plant at University of Bayreuth can be found in (Weiß *et al.*, 2018), including the description of the test facility. Figure 3 presents the main results. The comparison of the efficiency envelope curves over pressure ratio for both turbines. As expected, the cantilever turbine exceeds the axial impulse turbine in efficiency (76.8% to 73.4% peak efficiency). However, while the axial turbine achieves maximum efficiency at design pressure ratio (PR = 18.75), the performance of the cantilever turbine at design point is not really superior, but at a higher pressure ratio of PR = 22.4 it is. Furthermore, it is obvious that the 1D-meanline-turbine-design tool significantly underpredicted the efficiency of the axial impulse turbine (67% instead of 73.4%). This is surprising, because many axial impulse turbines have already been designed applying the 1D-meanline-turbine-design-tool. The turbine were tested and the loss model in the tool has been continuously adjusted. In contrast to this, the MM cantilever turbine is just the second example of its type which has been tested.

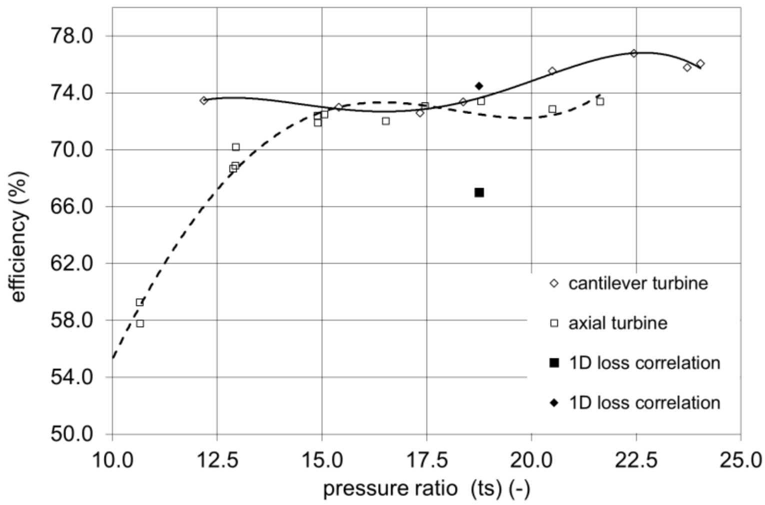

Figure 3: Measured total-to-static isentropic efficiency envelopes over pressure ratio for the axial impulse and the radial cantilever turbine

Although this paper focuses on the thermodynamic behavior of micro turbines, we want to give at least a rough figure about the cost of our turbines so that the efficiency of the turbines can be coupled with investment costs in the future. The turbine manufacturer of our turbine (DEPRAG SCHULZ GMBH u. CO) states that costs are about 1000 €/kW$_{el}$ (including the high speed generator). However, this figure depends on the number of turbines ordered.

4. POTENTIALS OF ADDITIVE MANUFACTURING

Regardless of the modularity and other favourable features of the MTG-c-kit, it has more potential for wider applications. Low temperature and low power systems can be named as the field with major obstacles. In order reach its potential for such applications, the following aspects should be tackled for the MTG-c-kit:

- Reduction of specific costs €/kW$_{el}$ by at least 20%, especially for low power applications
- Reduction of manufacturing lead times, including for tests of multiple blading design
- Improvement of the design and customization of blading for specific applications
- Further reduction of the number of components (e.g. wheel with shroud in a single piece)

Additive manufacturing (AM) has the potential to contribute to improving these issues. It is already being applied for large turbines in the industry. In micro power and low temperature applications, the authors see further interesting potentials of some technologies and materials (including plastics). Based on the literature review as well as on in-house tests an overview of AM applicability for micro turboexpanders was created, presented in Table 2.

In order to explore the possibilities and limits, manufacturing trials with the main AM technologies were performed on a 120 mm trial air turbine. The turbine is designed as a simple proof of concept for the purposes of first experiments (different from MTG-c-kit, now significantly lower pressure ratio). However, the performance can be applied later to the standard design as well. The design of the turbine and a photograph of the assembled central part are shown in Figure 4. The degree of admission of the test turbine was reduced to 6.7% due to the power limit of the electrical control unit.

Performance from the first tests of the air turbine with nylon SLS wheels without any surface treatment is shown in Figure 5. It shows that the AM parts without any surface treatment provide much worse performance than predicted (around 50%). However, it has to be taken into account that the design speed of the fully admitted wheel is about 20,000 rpm. Thus, we would expect about 20% efficiency for full admission and full speed. Further investigations will however include performance comparisons of parts from several technologies, effects of various surface treatment options, long term operation under specified conditions as well as material suitability tests for given working fluids and temperatures (chemical stability, swelling etc.).

Table 2: Summary of AM technologies for small turbines

Technology	**FDM**	**SLS**	**SLA**	**DMLS**	**EBM**
Possible applications	Low demand parts, other with limits	Any part	Critical stator parts, rotor with limits	Critical parts	Critical parts
Materials	ABS, Ultra ABS, PETG, composites…	Nylon based	Resins	Metals (steel, Al, Ti…)	Metals (steel, Al, Ti…)
Max. T	~ 100 °C	~ 80 °C	> 200 °C	>1000 °C	>1000 °C
Advantages	Cheap, widely available	Cheap, no support, best tolerances	Good resolution and surface quality	Most available metal AM technology	Fast, well-tuned properties
Issues	Rough surface in as-printed, need of supports, non-uniform properties	Limited T ~ 80 °C	Most expensive plastic method, needed supports, fine features and tolerances are issue	Expensive, sensitive to fine-tuning, Rough surface in as-printed, need of supports	Very expensive

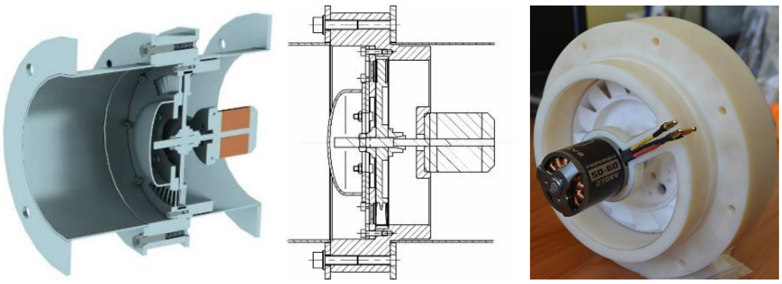

Figure 4: Design of the air turbine for tests of AM components and photograph of its assembled central part

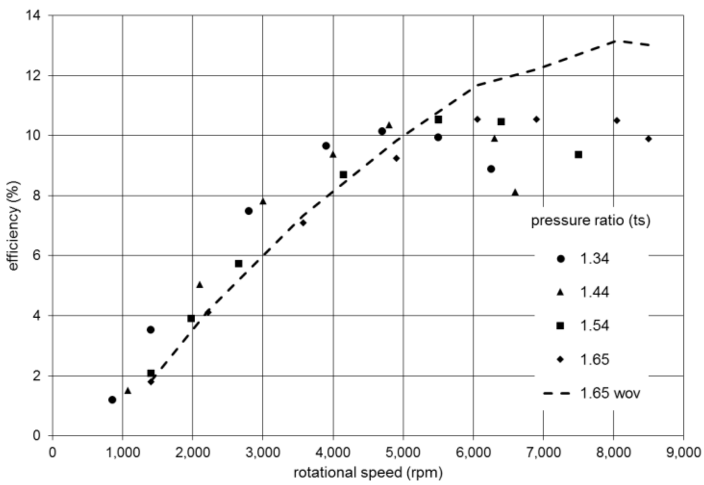

Figure 5: Experimentally determined isentropic total-to-static efficiency of the first 3D printed plastic turbine (wov = without ventilation losses, calculated by Equation (3))

5. CONCLUSIONS

This work has shown how impulse turboexpanders for micro ORC applications (1 - 200 kW_{el}) can provide a flexible, efficient and also cost-effective alternative to radial turbines and volumetric expanders. Basic fluid-dynamic design can be performed well with 1D meanline models after adopting suitable loss coefficients. This approach is implemented in the presented design tool. The tool can be partly automated, to provide a range of possible designs for the choice of the best solution. The physical design performed with a rather simple 1D approach combined with CFD verification and optimization can result in efficient micro-turbines as is shown on a case of axial and (quasi-impulse) radial cantilever ORC turbines.

The economic feasibility of the expander plays an important role. Therefore, a simple design concept with hermetical design, permanent magnet high speed generator and five sizes covering a large range of boundary conditions is applied. Integrally manufactured stator and rotor wheels are hereafter the only varied component. In order to further increase the economic application range, additive manufacturing and plastic materials are proposed especially for low temperature and low power applications. A brief review of prospective technologies along with first design and experimental results of an air turbine for proof-of-concept tests is shown in the last part of this work.

NOMENCLATURE

β	relative flow angle	(°)
ε	degree of admission	(1)
d	diameter	(m)
h	blade height	(m)
Ma	Mach number	(1)
n	rotational speed	(rpm)
P	power	(W)
Δ	Difference	(1)
φ	velocity coefficient	(1)

Subscripts

0	nozzle blading/stage inlet
1	nozzle blading outlet, rotor blading inlet
2	rotor blading/stage outlet
bl	blade
No	nozzle
r	relative
ts	total-to-static
V	ventilation

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