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EXPERIMENTAL INVESTIGATION OF A SUPERSONIC MICRO TURBINE RUNNING WITH HEXAMETHYLDISILOXANE

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Experimentally determined efficiency characteristics of a supersonic micro turbine are discussed in the present paper. The micro turbine is a representative of a "micro-turbine-generator-construction-kit" for ORC small scale waste heat recovery. The isentropic total-to-static efficiency of the 12 kW turbine reaches an excellent design point performance of 73.4 %. Furthermore, its off-design operating behaviour is very advantageous for small waste heat recovery plants: the turbine efficiency keeps a high level over a wide range of pressure ratio and rotational speed.

Keywords: turbine, Organic Rankine Cycle (ORC), waste heat recovery, small-scale

Introduction

Waste heat or pressure recovery systems in industry can save a lot of energy. Big scale waste heat recovery systems of several megawatts are almost state of the art. However, in a very small scale below 100 kW_{el} only very few solutions are available on the market although there is a big potential in industry [1]. In many decentralized combined heat and power plants which burn natural gas or bio gas in combustion engines (< 500 kW_{el}), the waste heat in the exhaust gas and/or cooling water has not been used so far. Here, a micro turbine generator directly installed in the exhaust flow of the combustion engine (Fig. 1a) or as an expander in a bottoming steam or organic Rankine cycle (Fig 1b) could increase electrical efficiency up to 10 % points.

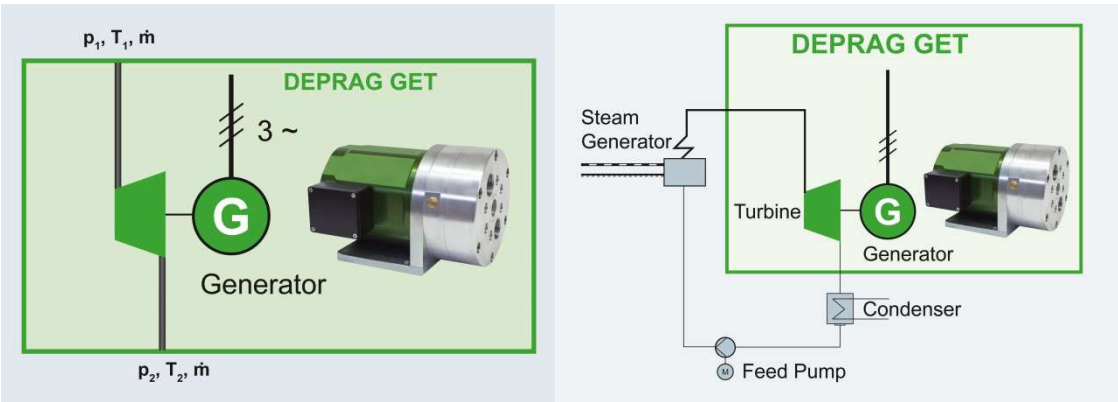

Fig. 1: Different application principles of a micro turbine generator [2]

Regarding all possible applications, the authors are convinced that there is a growing market for micro turbine generators below 100 kW_{el}. Thus, the company DEPRAG SCHULZ GMBH u. CO. and the University of Applied Sciences (UAS) Amberg-Weiden have been collaborating successfully for many years in developing micro turbines which drive high speed generators.

In this paper, recent tests of a 12 kW supersonic micro turbine running with hexamethyldisiloxane are presented and discussed. The investigated turbine generator serves in an ORC plant which aims to convert the waste heat of a 250 kW IC-engine into electricity.

2. ORC research plant working with hexamethyldisiloxane

The ORC (Organic Rankine Cycle) research plant is located at the Center of Energy Technology at the University of Bayreuth and was designed to investigate waste heat recovery from a 250 kW biogas engine with an exhaust gas temperature of about 500°C. Process simulations and theoretical investigations showed that for this rather high exhaust temperature, hexamethyldisiloxane is a suitable working fluid [3].

The layout of the research plant is shown in Figure 2. It is heated by a 120 kW propane gas burner which substitutes the piston engine. Its layout is that of a classical ORC plant consisting of a feed pump, an evaporator, an expander, a recuperator and a condenser. However, the evaporator is directly heated by the exhaust gas and does not use an additional thermal oil loop. The expander is the discussed micro turbine driving a high speed generator.

The thermodynamic circuit is working very stable. The heat flux can be easily varied between 50% and 100%. This reliable operating behaviour allowed the detailed experimental investigation of the micro turbine at different speeds, mass flow rates, pressure ratios etc..

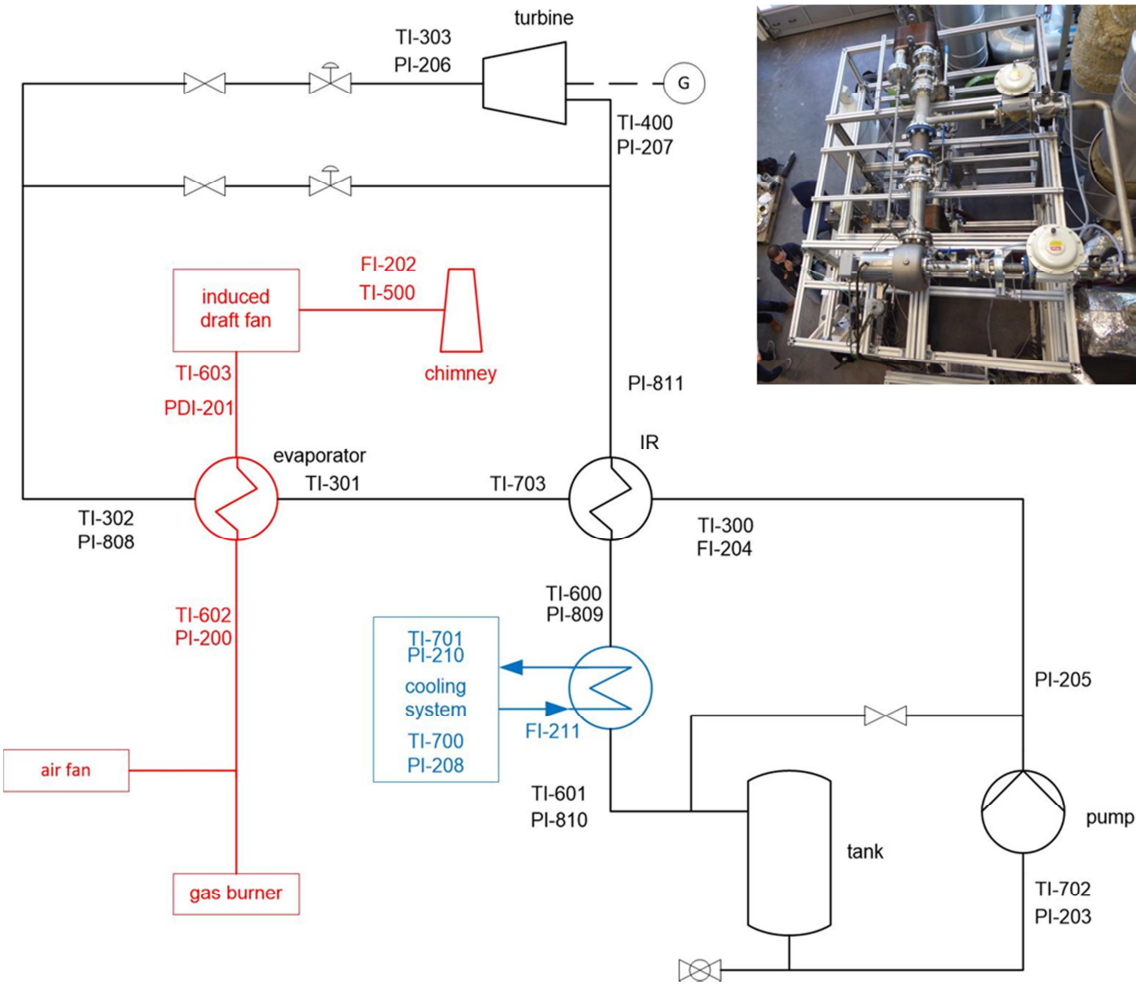

Fig. 2: ORC plant lay-out and a photograph (top view) of the plant at the University of Bayreuth

The generated electrical power is logged via the 25kW-feed-in unit. The mass flow rate is measured by a Coriolis device ("FI204" in figure 2). Pressures (PI) and temperatures (TI) are measured upstream and downstream of each component. Thus, the efficiencies of all components can be calculated. For full load i.e. high mass flow rates, the turbine efficiency can be calculated reliably on the basis of the measured turbine exit temperature. However, for small part load the heat losses are not negligible. Therefore, to avoid the usage of measured exit temperature, the applied electrical conversion chain was investigated separately, in advance. For this purpose, the generator was driven by an electric motor. A torque meter between motor and generator measured torque and rotational speed to get knowledge about the mechanical power input. Electrical power fed into grid was measured by the 25kW-feed-in unit. Thus, the overall electrical efficiency of the entire electrical conversion chain could be determined as a function of power (Figure 3). The weak influence of rotational speed was neglected. At its full load (25KW) the feed–in-unit achieves more than 90% electrical efficiency. In the following, the discussed total-to-static isentropic turbine efficiencies use actual enthalpy drop determined by measured turbine power and not via turbine outlet temperature.

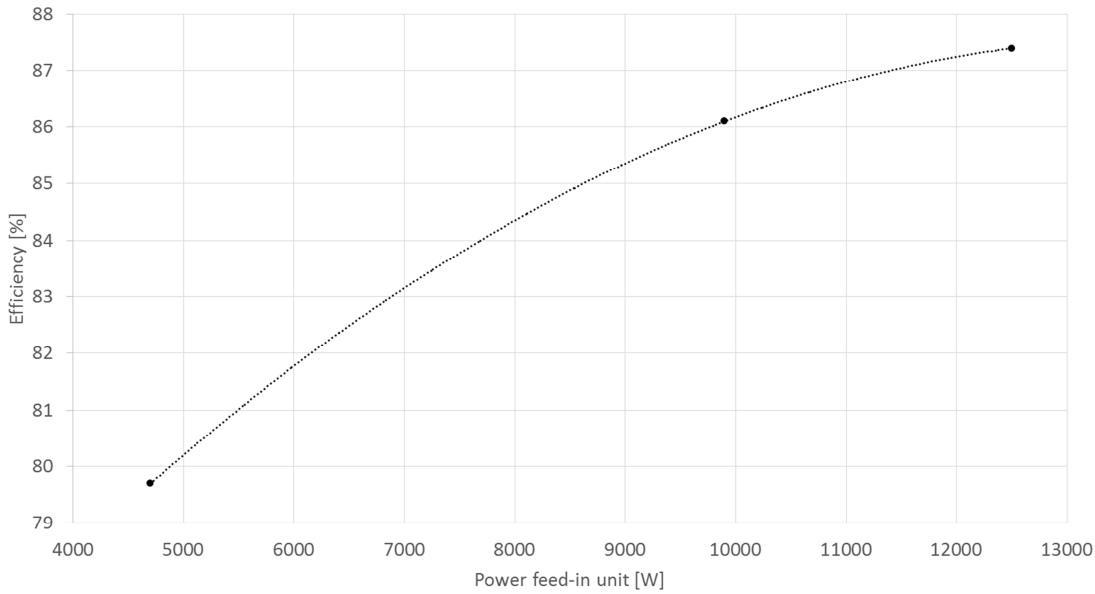

Fig. 3: Overall efficiency of the entire electrical conversion chain

3. The micro-turbine-generator-construction-kit – the tested 12 kW turbine

Due to the various possible applications in 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 some standard machines to stock. In fact, it is necessary to develop a very flexible "micro-turbine-generator-construction-kit" by means of which a customized turbine generator can be designed and built quickly for any required power output, any working fluid and any boundary conditions out of a wide range. The tested 12 kW micro turbine is built in accordance to the design principles of the addressed "micro-turbine-generator-construction-kit".

In [4] the author discussed this task in more detail and came to the conclusion 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, the axial single stage impulse turbine is able to process theoretically unlimited pressure ratios, requires lower

rotational speed, does not produce axial thrust and can be designed with partial admission, thus allowing the implementation of smaller turbines. Figure 4 displays the principal architecture. The characteristic features are listed in the following:

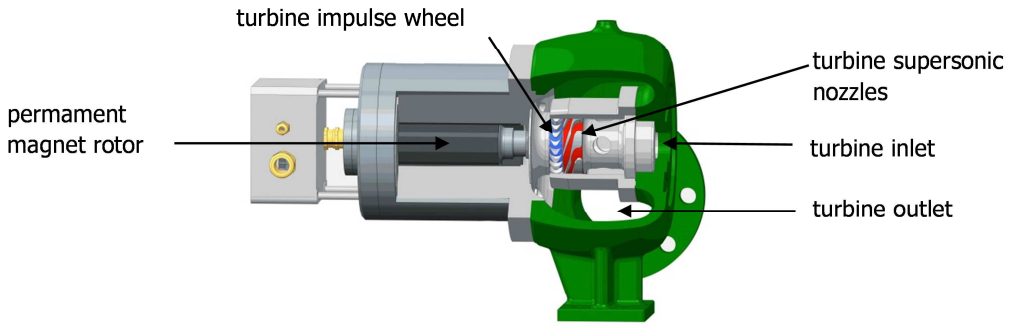

Fig. 4: Architecture of the "micro-turbine-generator-construction kit"

- hermetically sealed turbine-generator (3 -175 kW_{el}, implemented with 5 different- sizes)
- single stage axial impulse turbine (10000 – 70000 rpm) which is able to process very high pressure ratios (200:1 have already been tested; see e.g. [5]) and small volume flow rate (partial admission)
- integrally manufactured turbine wheel (Ø 50 – 350 mm)
- permanent magnet high-speed generator
- turbine wheel directly mounted on generator shaft: just one set of bearings required, no gear, no coupling
- compact design, low material usage

The main design data of the investigated single stage axial impulse turbine are listed below. Figure 5 presents a photograph of the turbine mounted on the top of the ORC plant.

wheel diameter	120 mm		
rotational speed	24000 rpm	pressure ratio (ts)	18.75
Laval nozzles	12	degree of reaction	0.0
rotor blades	32	degree of admission	full
mass flow rate	0.32 kg/s	**nozzle exit Ma number**	**2.11**
power output	12 kW	**rotor inlet Ma number**	**1.14**
inlet pressure	6.00 bar	nozzle Re number	9.3×10^5
outlet pressure	0.32 bar	rotor Re number	3.1×10^5
inlet temperature	176 °C		

The ORC plant is designed for rather high temperature waste heat. Higher maximum temperatures means higher necessary pressure (ratios). Thus, a single turbine has to work with supersonic flow even at rotor inlet.

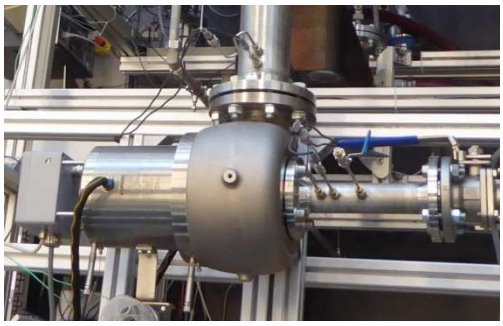

Fig. 5: The 12 kW micro turbine generator mounted on top of the ORC plant

4. Discussion of experimental results

The turbine efficiencies presented and discussed in the following are isentropic total-to-static efficiencies. Hence, total inlet pressure, total inlet temperature and static outlet pressure are used to determine the isentropic enthalpy drop (REFPROP database [6]). As described before, the actual enthalpy drop is not calculated by means of measured turbine outlet temperature but via the power output of the turbine (chapter 2).

Figure 6 displays the turbine efficiency characteristics as a function of rotational speed for different turbine inlet pressures i.e. mass flow rates. The condenser pressure varied only slightly during operation. Inlet pressure 6.1 bar repesents nearly design point i.e. 100% mass flow rate. 3.2 bar corresponds to about 50% mass flow rate (5 kW turbine power).

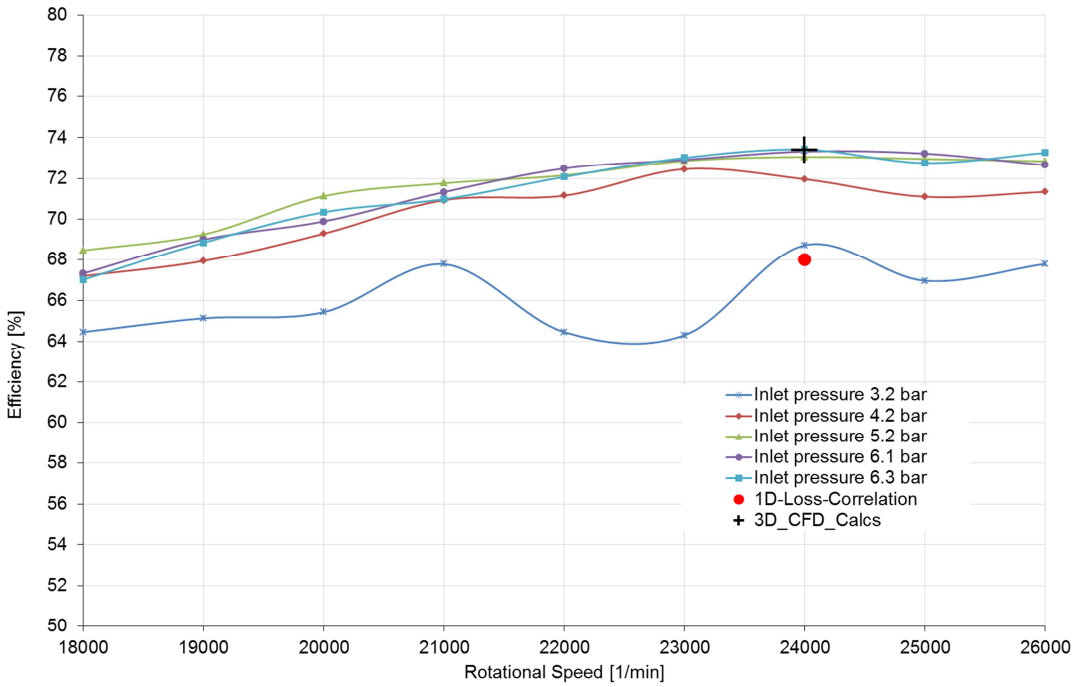

Fig. 6: Total-to-static turbine efficiency as a function of rotational speed

The higher the mass flow rate (inlet pressure), the smoother the course because measurement accuracy is increased, interferences are getting relatively smaller. At inlet pressure 3.2 bar (50% mass flow rate) the curve shape is not reasonable. The measuring points at 22000 and 23000 rpm are probably outliers. The measured power drops suddenly by about 250 W on a level of 5000 W. This is most likely not caused by the turbine flow.

At inlet pressure 6.1 bar (closest to design) the efficiency shows a flat maximum (73.4%) at design speed 24000 rpm. Design efficiency (1D-loss-modell) is 68.0%. Efficiency determined by 3D-CFD-calcs is 73.4%. 3D-calcs did not take into account shroud leakage, disk friction and bearing losses. So, both simulation approaches were too pessimistic – especially the 1D loss model which was used in design. The flat maximum occurs as well for the measured higher and lower inlet pressure down to 4.2 bar (about 2/3 of design mass flow rate).

Figure 7 bases on the same data like figure 6. However, now efficiency is plotted as a function of total-to-static pressure ratio. The percentage of corrected rotational speed n_{corr} serves as parameter.

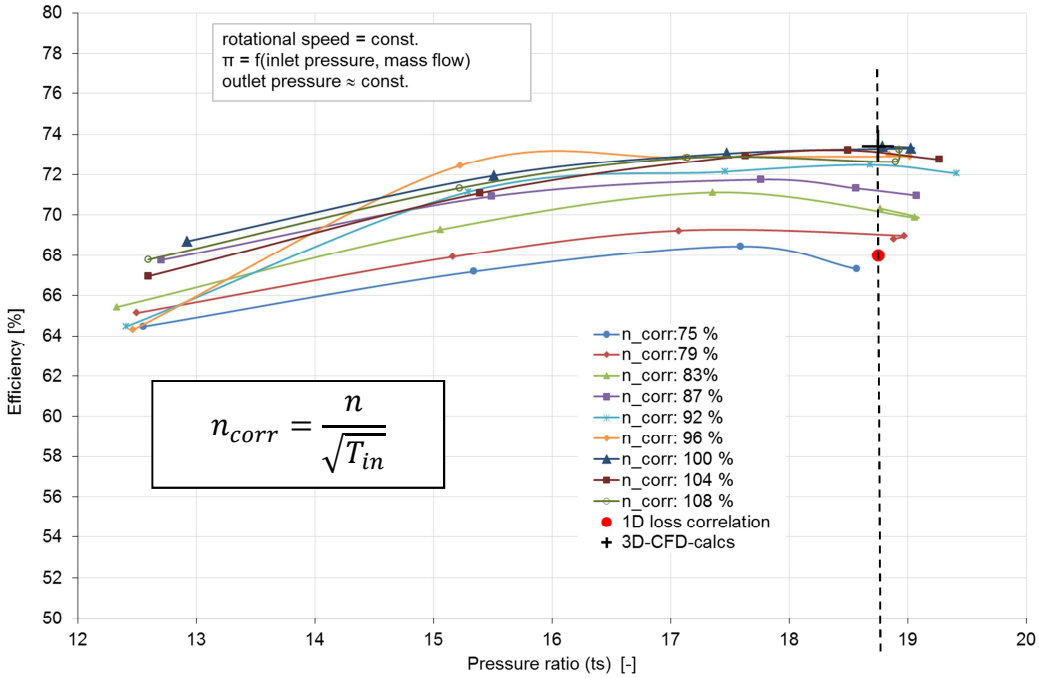

Fig. 7: Total-to-static turbine efficiency as a function of pressure ratio (ts = total-to-static; turbine inlet pressure varies, outlet pressure is almost constant)

The curves show very similar courses with exception of n_{corr}: 92% and 96%. This different shape is just caused by the data points at lowest pressure ratio. This data points correspond to the measurements at 3.2 bar inlet pressure and 22000 rpm or 23000 rpm (Fig. 6). As already mentioned – these points seem to be outliers.

The design pressure ratio of the Laval nozzles is $\Pi_{ts,\ design}$= 18.75. The maximum efficiency occurs there for n_{corr} > 90%. The maximum is rather flat i.e. the nozzle seems to react not very sensitively on changes in pressure ratio. May be, the increasing losses in nozzle at off design ($\Pi_{ts} < \Pi_{ts,\ design}$) are partly compensated by reduced losses in rotor blading due to the smaller rotor relative inlet Mach number.

This consideration is supported by Figure 8 which shows efficiency characteristics plotted again against pressure ratio. However, only data points at optimum rotational speed were used. Furthermore, additional measurements with lower vacuum were included.

Due to changes in ambient temperature, condenser pressure varied. Thus, pressure ratio was not only changed by inlet pressure but also by turbine outlet pressure. So, a bigger range of pressure ratios could be investigated. The inlet pressure varies with mass flow rate. I.e. also the overall power output varies significantly. Whereas, the power output changes only slightly, when back pressure varies. Figure 8 shows that efficiency is principally a function of pressure ratio no matter whether pressure ratio is varied by inlet or by back pressure. The figures are almost identical. The design pressure ratio of the Laval nozzle is $\Pi_{ts,\ design}$= 18.75. However, turbine efficiency is obviously almost constant in a range $16 \leq \Pi_{ts} \leq 22$. Rotational speed for maximum efficiency varies only slightly over this range.

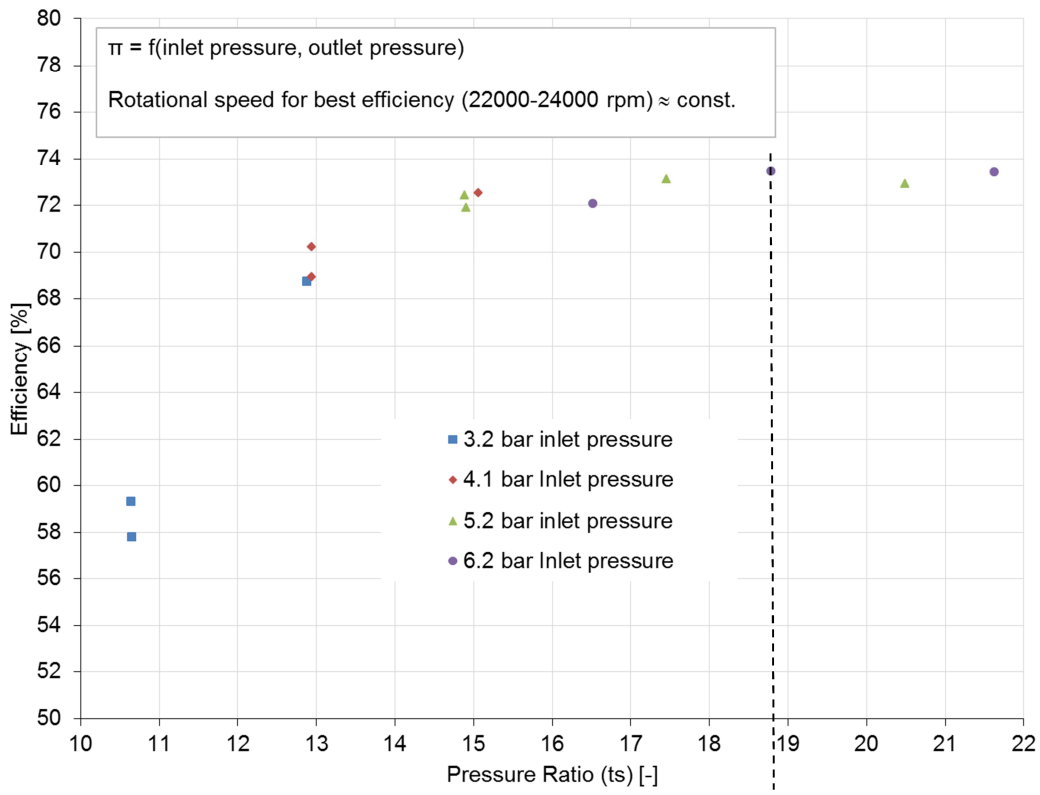

Fig. 8: Isentropic total-to-static turbine efficiency as a function of pressure ratio at optimum rotational speed (turbine inlet and outlet pressures vary)

Conclusion

The efficiency characteristics of a supersonic micro turbine running with hexamethyldisiloxane were measured in a research ORC plant and have been presented and discussed in the present paper.The 12 kW micro turbine is a representative of the introduced „micro-turbine-generator-construction kit“.

The achieved design point efficiency of 73.4 % (total-to-static) is an excellent figure, keeping in mind the small size of the turbine in combination with the high pressure ratio which it must process. The total-to-total isentropic efficiency is about 4% higher (based on design calculation). The efficiency characteristics are almost constant over a wide range of pressure ratios and, furthermore, show a rather flat maximum over rotational speed. Both qualities are very valuable for a micro-turbine-generator applied in any small waste heat recovery plant where exact operating condition can not be guarenteed or foreseen, respectively.

However, as discussed in [7], the authors work on an alternative turbine concept. This is a radial inflow cantilever turbine which provides a higher efficiency potential without losing partial admission capability. A representative of this concept will be tested in near future within the research ORC plant at the University of Bayreuth.

Nomenclature

n	rotational speed	[rpm, 1/min]
Π	pressure ratio (ts)	[-]
T	temperature	[K, °C]

CFD	computational fluid dynamics
IC	internal combustion
Ma	Mach
ORC	Organic Rankine Cycle
Re	Reynolds
rpm	revolution per minute

Subscript

corr	corrected
el	electric
in	inlet
ts	total-to-static

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