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DESIGN, OPTIMIZATION AND EXPERIMENTAL INVESTIGATION OF VARIOUS SMALL-SCALE ORC EXPANDERS IN A BIOMASS COGENERATION PLANT

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ABSTRACT

This paper presents the design, optimization, and experimental investigation of three small-scale ORC expanders—one rotary vane expander and two turbines (a single-stage axial impulse turbine and a radial re-entry turbine called Elektra)—developed for an operational biomass-fired CHP unit. All expanders were tested in a 120 kW_{th} woodchip-fired ORC plant at CTU UCEEB. The rotary vane expander demonstrated broad operational flexibility, moderate efficiency sensitivity to boundary conditions, and over 10,000 hours of run time, though it requires periodic maintenance. The Elektra turbine, designed as a low rpm standard generator alternative, using in-house 1D meanline modeling and CFD optimization, achieved around 34.8% efficiency at 3,000 rpm. The axial impulse turbine, optimized via genetic algorithms, reached over 61% efficiency and 7 kW output at 14,000 rpm. Techno-economic analysis showed that the rotary vane expander achieved the lowest specific plant costs, while the axial turbine delivered the highest efficiency at increased cost. The Elektra turbine, despite its low capital cost, had limited performance and the highest plant-specific costs. Experimental results validated the design tools and offered comparative insight into expander suitability for small-scale CHP applications.

1 INTRODUCTION

When integrating Organic Rankine Cycle (ORC) technology into Distributed Energy Systems (DES), the choice of the expander plays a crucial role in determining overall system efficiency, costs, and operational flexibility. Volumetric expanders, such as scroll, screw, or reciprocating piston expanders, are commonly used in small-scale ORC applications due to their relatively simple construction, ability to handle wet fluids, and high efficiency at part-load conditions (Imran *et al.*, 2016). These expanders operate at lower rotational speeds, typically in the range of a few hundred to a few thousand revolutions per minute, and provide a favorable trade-off between efficiency, costs and mechanical complexity. Their ability to work efficiently with lower-temperature heat sources and their robustness against fluid properties variations make them an attractive option for waste heat recovery in small and medium-scale DES (Pantano and Capata, 2017). Turbines, in contrast, operate at high rotational speeds, typically in the range of a few thousand to tens of thousands of rpm. High-speed single-stage turbines are designed for applications that require higher peak efficiency and high-power density. However, their high rotational speed necessitates a high-speed generator and advanced bearing systems, often incorporating magnetic or gas bearings, to minimize frictional losses and ensure long-term reliability. These machines are often associated with higher peak efficiency, but exhibit a narrower operating range (Alshammari *et al.*, 2018). To investigate and compare the performance of these expansion machines, three different expander types, a single stage axial turbine, a radial re-entry turbine – called Elektra and a Rotary Vane

Expander (RVE) are compared experimentally in an operational biomass-fired ORC CHP unit. Whereas the RVE was originally planned as the expander when the system was designed and has already gathered most of the test hours on this system, the two turbine types are brand new designs from the system perspective. The aim of this paper is to give a brief insight into the design of the expanders, their optimization in the design process and their integration into the plant. Subsequently, the performance data of the expanders are compared with one another in order to draw a conclusion about the expanders in use in this specific ORC system.

2 CHP-ORC PLANT

For the measurement campaign of the three different expanders, a 120 kW_{th} biomass-fired Organic Rankine Cycle (ORC) Combined Heat and Power (CHP) unit was used. This unit has been developed as an upscaled version of a previously designed 50 kW_{th} plant. This system at CTU University Center for Energy Efficient Buildings (UCEEB) is utilizing woodchips as fuel. The ORC medium used in this plant is hexamethyldisiloxane (MM), in case of operation with RVE mixed with approx. 5 mass-% of lubricating oil. For the measurement campaign of the two turbine types, oil was drained and the working fluid was distilled to eliminate oil from the cycle. Experimental data from thousands of operational hours validate the system's performance and technical maturity level. A comparative analysis indicates that the ORC unit is viable in applications such as municipal heating, sawmills or remote mountain lodges. Despite the scale-up, production costs remain relatively stable due to design optimizations, making this unit a commercially attractive solution for decentralized energy systems.

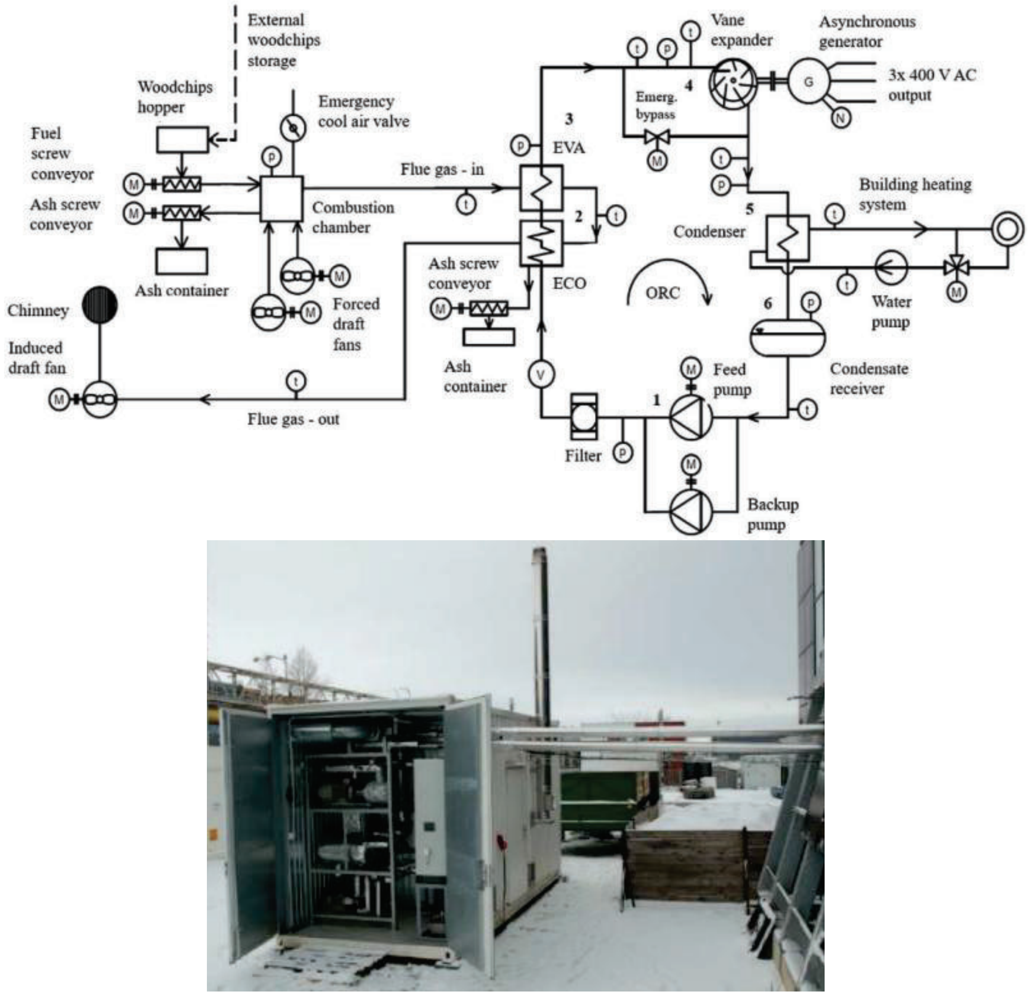

Figure 1. Piping and instrumentation diagram (top) and picture of the of the biomass-fired CHP ORC unit at CTU (bottom)

To make this unit ready for the measurements of different types of expanders, a generator VFD (Variable Frequency Drive) and other necessary power electronics was added in order to be able to test rotational speed variations of the turbines used. This was needed, because the unit is standardly fitted with the RVE that rotates at grid frequency (50 Hz). Figure 1 shows the piping and instrumentation of the unit with the RVE as expander and a picture of the open CHP container from the ORC side (Mascuch *et al.*, 2021).

3 EXPANDERS

The three different expander types are presented in detail in the following section. All three types were designed for and tested in the above-mentioned ORC CHP unit. Table 1 shows the different design boundary conditions of the various expanders. The Elektra turbine and the Rotary Vane Expander were designed for a thermal input of 120 kW_{th}, while the single-stage axial turbine was limited to 100 kW_{th} to avoid exceeding the nominal power rating of the necessary output high frequency power electronics.

Table 1: ORC Boundary conditions for the expander designs

Parameter	Single stage turbine	Elektra turbine	Rotary Vane Expander	Units
Inlet Pressure	550	650	580	kPa
Inlet Temperature	180	190	180	°C
Outlet pressure	55	55	55	kPa
Pressure ratio	10	11.8	10.6	-
Mass flow rate	0.25	0.3	0.33	kg s^{-1}
Rotational speed	18,000	3,000	3,000	rpm

3.1 Single stage axial turbine

The supersonic single-stage axial impulse turbine has been developed for a nominal electrical output of 8 kW_{el} at 18,000 rpm, the turbine aims for a design-point isentropic efficiency of 68.1%. The aerodynamic design utilizes an in-house developed one-dimensional meanline approach with velocity loss correlations for supersonic flow (Weiß *et al.*, 2020) and real-gas properties modeled via REFPROP in Python (Huber *et al.*, 2022). The optimization was performed using a Genetic Algorithm (GA) (see Eq. (1,2) to refine the key design parameters such as mean diameter, blade angles, and rotational speed.

$$max\ f(\vec{\boldsymbol{x}}) = \eta_{is}(\vec{x});\ \text{where} \quad (1)$$

$$\vec{\boldsymbol{x}} = [D_{mid}, n, \alpha_s, \frac{h}{D_{mid}}, \beta_r, e] \quad (2)$$

Figure 2 shows the cut-away view of the turbine with its casing, rotor, stator with nozzles and magnetic coupling part.

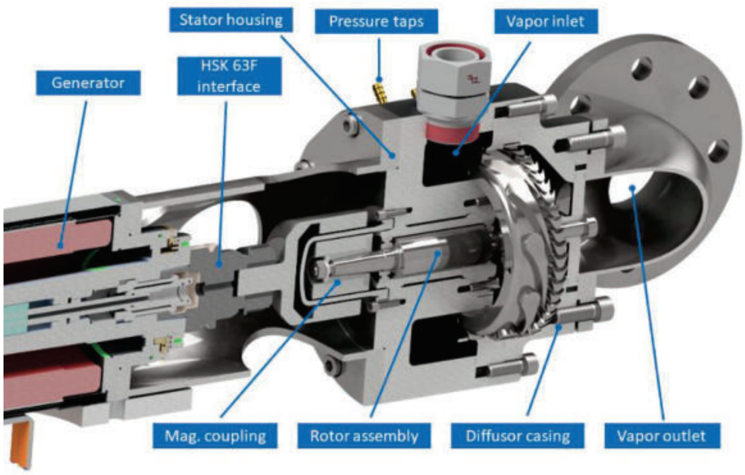

Figure 2. Cut-away view of the supersonic axial impulse turbine

The turbine features convergent-divergent nozzles for the stator designed using Method of Characteristics with real gas properties using an open-source Python script *open-moc* (Anand *et al.*, 2020) and constant-channel-width rotor blades designed with the vortex flow method (Paniagua *et al.*, 2014); (Goldman and Vanco, 1971).

Table 2: Key design parameters of the single stage axial turbine

Parameter	**Value**	**Units**
Midspan diameter D_{mid}	135	mm
Rotational speed n	18,000	rpm
Nozzle out. Mach number Ma_2	1.89	-
Nozzle out. flow angle α_2	13	°
Rotor in. flow angle β_2	23.5	°
Blade height h	5.5	mm
Number of blades Z_s, Z_r	10, 47	-
Isentropic efficiency $\eta_{is_{t-s}}$	68.1	%
Mechanical power output P_{mech}	8	kW

A hermetically sealed magnetic coupling transmits the power to a high-speed CNC spindle motor acting as a generator, delivering power to the grid through the VFD. The mechanical design accounts for high rotational speeds, employing standard off-the-shelf steel ball bearings lubricated for life with a high temperature grease, while the generator assembly is water-cooled. A more detailed explanation about the design and optimization can be found in (Spale*,* 2024). Table 2 shows the GA optimized design parameters of the manufactured turbine. Key design parameters of the turbine indicate a very compact, highly efficient turbine.

3.2 Radial re-entry turbine (Elektra)

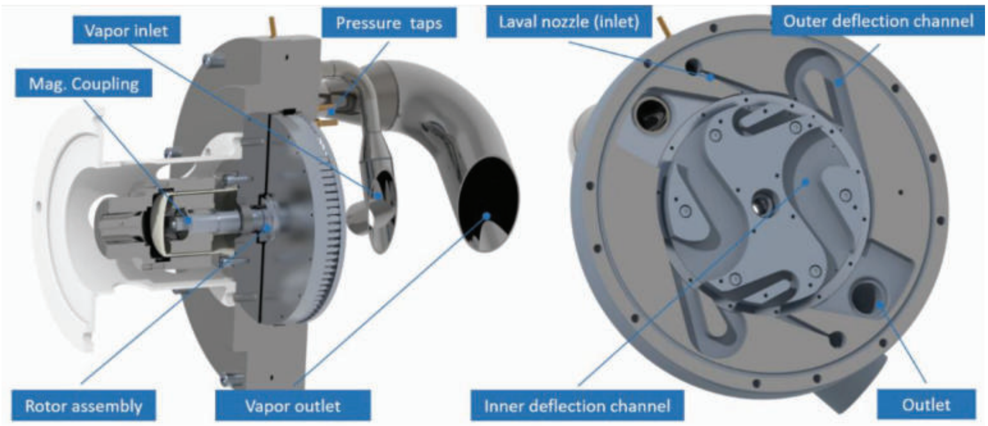

Figure 3. Cut away view and blade to blade cross section of the radial re-entry turbine

The Elektra turbine, a radial re-entry turbine, has been designed for a nominal output of 7 kW at 3,000 rpm. A series of computational fluid dynamics (CFD) simulations were conducted to refine its geometry, focusing on optimizing flow characteristics through its multiple wheel passes and deflection channels. The CFD process iteratively improved the turbine's design, increasing its predicted CFD isentropic efficiency from 44.5% in the initial version to 47.9% in the latest optimized configuration (Streit *et al.*, 2024). Figure 3 shows the design of the Elektra turbine in its manufactured state. The left part shows the turbine in its cut away view, the right the blade to blade cross section with the two nozzles and the deflection channels shown. Key modifications included reducing the nozzle exit area to mitigate overexpansion, optimizing deflection channel geometry to minimize flow separation, and adjusting the outlet area to enhance flow distribution. Despite these improvements, the CFD simulations indicate a potential efficiency shortfall compared to the original target of 50%, especially considering unsteady flow effects, strong flow deflections in the channels and high relative Mach numbers in the blade channels. Table 3 shows the design key parameters of the CFD-optimized and manufactured turbine.

Table 3: Key design parameters of the radial re-entry turbine (Elektra)

Parameter	Value	Units
Outer wheel diameter D_{out}	255	mm
Rotational speed n	3,000	rpm
Wheelpasses WP	4	-
Wheel diameter ratio D_{in}/D_{out}	0.9	-
Blade channel width	3.5	mm
Blade relative inlet angle	30	°
Blade relative outlet angle	150	°
Number of blades, Z_r	84	-
Isentropic efficiency (CFD) $\eta_{is_{t-s}}$	47.9	%
Mechanical power output P_{mech}	≈7	kW

The key design parameters of the Elektra turbine directly show the completely different approach of this turbine type compared to the single stage axial turbine. The Elektra turbine requires four-wheel passes, but also the large impeller of 255 mm in order to achieve a design speed of 3,000 rpm for the given boundary conditions. The Elektra turbine, just like the axial turbine, is connected via a hermetically sealed magnetic coupling to a generator that feeds into the grid via the frequency converter. This frequency converter is actually not necessary for the Elektra turbine, as the design rotational speed of the turbine is 3,000 rpm and could therefore feed directly into the grid via the generator. However, it was applied for these investigations to determine the efficiency map of the turbine as a function of rotational speed.

3.3 Rotary Vane Expander

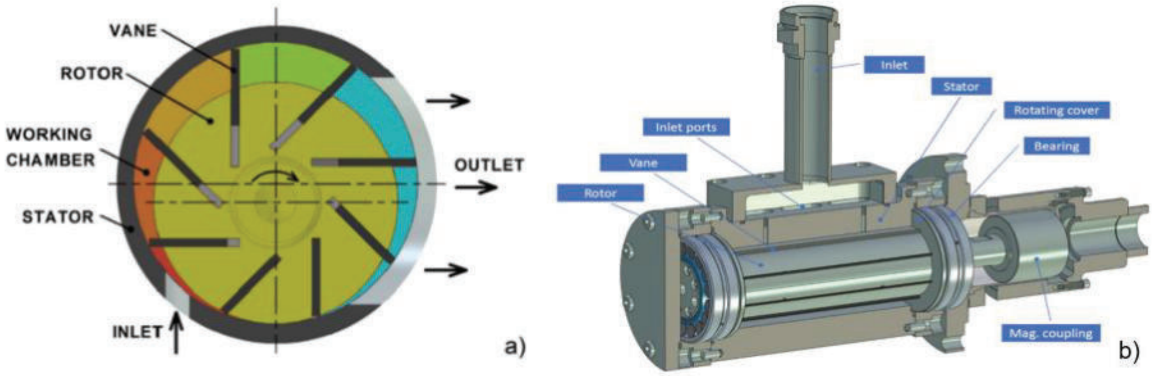

Figure 4. Working principle (a) and cut-away view (b) of the RVE

The design of the rotary vane expander is based on many years of experience in the development of these type of expanders at UCEEB. It is a positive displacement machine with a simple design and favorable production costs. On the other hand, RVE is generally expected to have lower efficiency compared to other positive displacement expanders, mainly due to vane friction losses and leakage losses. The challenge of these machines is also to solve the long-term reliability and service life. For the geometric design of the machine, a genetic algorithm using a 1D thermodynamic model was used, which was verified on a previous type of expander (Vodicka *et al.,* 2019). The model accounts for mass and energy balances within the working chamber and leakage pathways, incorporating vane dynamics and friction losses with ±5% of prediction accuracy. The vane expander was designed to run at 3,000 rpm with the possibility of direct connection to a generator using a magnetic coupling. Figure 4 shows the RVE working principle on the left and a 3D cut-away view on the right. Table 4 shows the key design parameters of the optimized RVE.

Table 4: Key design parameters of the Rotary Vane Expander (RVE)

Parameter	Value	Units
Stator diameter	85	mm
Rotor diameter	73	mm
Eccentricity	5.9	mm
Rotational speed	3,000	rpm
Chamber length	204	mm
Vane thickness	1	mm
Vane width	24	mm
Number of chambers	8	-
Expansion ratio	2.94	-
Initial chamber volume	26.3	cm^3
Isentropic efficiency according to the model	55.6	%
Mechanical power output according to the model	8.6	kW

4 MEASUREMENT CAMPAIGN AND COMPARISON

After the three different types of expanders have been briefly introduced and their design and optimization described, the measurement results of the expanders are compared with each other. The measurements of the expanders took place at different times, whereby the following illustrations only show directly comparable points. For all three expanders, the points with the highest efficiency or the highest power were always compared with each other. The efficiency of the expanders was calculated as follows:

$$\eta_{is} = \frac{P_{mech}}{\dot{m}_{MM}(\dot{V}_{MM}, \rho_{liq}(p_{liq}, T_{liq})) \cdot \Delta h_{is}(p_{in}, p_{out}, T_{in})} \tag{3}$$

and the thermal input:

$$\dot{Q}_{th,in} = \dot{m}_{MM}(\dot{V}_{MM}, \rho_{liq}(p_{liq}, T_{liq})) \cdot (h_{evap,out}(p_{evap,out}; T_{evap,out}) - h_{evap,in}(p_{evap,in}; T_{evap,in})) \tag{4}$$

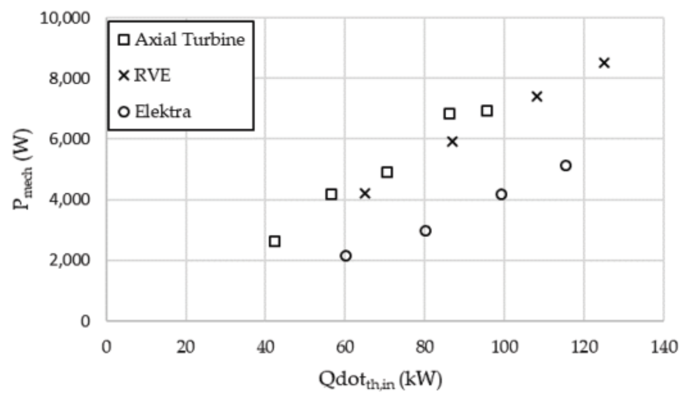

Figure 5. Mechanical power output over thermal input for all three expander types

Figure 5 shows the mechanical power (shaft power) output of the three expander types over the thermal input. The ORC system is designed for 120 kW_{th}, but can also run at partial load as it follows the thermal load demand of the building hot water system, for which the rotational speed of the woodchips screw conveyor is adjusted to lower speeds. It was therefore possible to run a series of expander measurements for these different thermal inputs. All three expander types show a relatively linear increase in power output with an increasing thermal input. The Elektra turbine, despite its advantages in low rpm and potentially high maintenance life, clearly lies behind the RVE and the axial turbine in mechanical power output. The single stage axial turbine in comparison to the RVE shows a relatively similar curve over the measured thermal input. The axial turbine could almost achieve its designed power output, the Elektra turbine is about 30 % below its design point. At high rotational speeds (> 15,000 rpm), the axial turbine power fell significantly, probably due to strong shock-induced losses, connected with the audial effects and a visible increase in vibration velocity measurement, thus not achieving peak performance at nominal rpm.

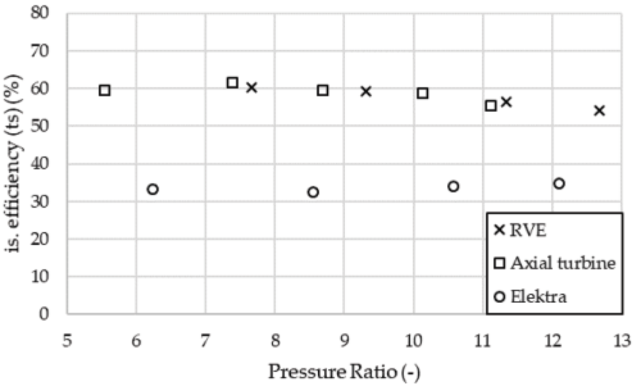

Figure 6. Total to static isentropic expansion efficiency over pressure ratio for all three expander types

Figure 6 shows the isentropic expansion efficiency over the pressure ratio for all the three expanders. None of the three expander types shows a strong dependence of the efficiency on pressure ratio, which is a good indication of a flexible operation under variable thermal loads. The RVE shows a slight decrease to higher pressure ratios, while the Elektra turbine shows a slight increase up to a pressure ratio of approx. 12 – its design pressure ratio (PR). The axial turbine shows its highest isentropic efficiency at a PR of about 7.3, which is lower than its design PR of 10, probably due to high shock losses at higher rpms, corresponding to the optimal ratio of the circumferential velocity U and nozzle outlet velocity c, at the design PR. The RVE has an efficiency curve very similar to that of the axial turbine, with a peak value of 60.3% at a PR of approximately 7.8.

Table 5. Performance parameters of the three expander types tested in the ORC Plant

Parameter	**Single stage turbine**	**Radial re-entry turbine**	**Rotary Vane expander**	**Units**
Peak expansion efficiency ($P_{mech.}/P_{is.}$)	61.7	34.8	60.3	%
Peak mech. power output	7*	5.1**	8.5**	kW
Weight (incl. generator)	52.5	117	122	kg
Specific power output	0.13	0.043	0.07	kW/kg
Costs (incl. generator; 2022 EUR)	15,000***	5,700	10,500	€
Specific expander costs referenced to 100 kW_{th}	2,142	1,366	1,418	€/kW
Plant costs (CAPEX)	115,000	105,700	110,500	€
Specific plant costs referenced to 100 kW_{th}	16,428	25,347	14,932	€/kW

*@100 kW_{th} **@120kW_{th} ***(incl. VFD + power electronics)

To compare the three expanders in terms of cost-effectiveness within this ORC system, key performance parameters are summarized in Table 5. The single-stage axial turbine achieves the highest peak efficiency and, due to its relatively low weight, also delivers high specific power. However, its cost—especially when including the generator and the necessary VFD and power electronics—is the highest among the three, resulting in the highest specific cost per kW. Nonetheless, due to its high efficiency, it results in a moderate specific plant cost. In contrast, the Elektra turbine exhibits low peak efficiency and power, mainly due to high secondary losses in the deflection channels. Its heavy construction, despite aluminum components, leads to the lowest specific power. However, its simple design (e.g., 3-axis milled channels) makes it the least expensive option, despite its low efficiency it has the lowest specific costs among three expander types. Nevertheless, due to the poor performance, it results in the highest specific plant costs. The rotary vane expander (RVE), the longest-tested in this system, shows favorable performance. Its efficiency and mechanical power output closely matches the axial turbine. It’s simple design and high specific power output also results in lower specific costs than the axial turbine and in the lowest specific plant costs of all expanders tested. It's important to note that the cost comparison excludes maintenance. While lifetime testing confirms reliable operation for thousands of hours, long-term use of the RVE will likely require vane replacement due to internal friction. Additionally, it cannot operate oil-free, unlike the two turbine types, which are capable of maintenance-free, oil-free operation in MM working fluid, thanks to minimal internal contact (limited to bearings).

Table 6: Technical and operational comparison

Parameter	**Single stage turbine**	**Radial re-entry turbine**	**Rotary Vane expander**
Complexity/Design	-	+	-
Scalability	+	-	-
Maintenance	+	++	+
Noise/Vibration	+	++	-
System integration	+	++	++

The technical and operational comparison of the three expander types in table 6 – single stage axial, radial re-entry turbine and rotary vane expander – reveals clear differences in suitability depending on system requirements. In terms of complexity and design, the single-stage axial turbine is the most sophisticated, requiring precise aerodynamic profiling and high manufacturing accuracy. The radial re-entry turbine has a simpler design, although its larger size is generally not a major drawback in most applications. The rotary vane expander is the most straightforward in terms of basic geometry and use of standard components; however, the practical implementation demands highly precise machining, specialized surface treatments such as DLC coatings and substrate hardening to ensure durability. Regarding scalability, the axial turbine performs best, maintaining efficiency across a wider range of sizes. In contrast, both the radial re-entry turbine and the rotary vane expander exhibit limitations in scaling, due to major adjustment to the deflection channels of the re-entry turbine and increasing mechanical losses of the RVE. For maintenance, the radial re-entry turbine has clear advantages, featuring robust structure with a few wear-prone elements. The RVE requires more attention due to moving vanes and sealing surfaces. The axial turbine, although relatively low-maintenance, still involves components like high rpm bearings. The radial re-entry turbine excels in noise and vibration, due to very low rotational speed and no sliding contact surfaces. The axial turbine generates moderate high-pitched noise levels, where the RVE is the noisiest due to higher vibration, inherent mechanical metal-metal contact and pulsating flow. Another critical aspect is integration with power generation systems. Both the radial re-entry turbine and the rotary vane expander operate at low rotational speeds, allowing direct coupling with standard generators. This simplifies system architecture and reduces costs. In contrast, the axial turbine requires a high-speed generator and power output must be conditioned through expensive power electronics, which increases both system complexity and overall cost.

5 CONCLUSIONS

This paper provides a brief overview of the design, optimization and experimental investigation of three different expanders in a specific operational woodchips-fired small-scale ORC CHP system. The main points that characterize the measurements of the three expanders are as follows:

- Measurement of the single stage axial turbine indicated reliable and low vibration operation (100 hours). Peak power output was lower than expected (7 kW to 8 kW) and occurred at 14,000 rpm instead of the design point at 18,000 rpm.
  ➔ Discrepancy is attributed to potential shockwave formation at the stator trailing edge and choking the rotor blade channels. Further investigations through full-stage Computational Fluid Dynamics (CFD) and refining of the aerodynamic performance are necessary.
- Elektra turbine measurements showed reliable operation with low vibration for over 200 hours. Peak power output and efficiency was significantly below expectations with (5.1 kW vs 7 kW) and (34.8 % vs. 47.9 %). Nevertheless, peak numbers were achieved at the design point of 3,000 rpm.
  ➔ The large discrepancies of the measured power output compared to the designed values are largely due to the simplified CFD Simulations for the optimization of the deflection channels of the turbine. Further CFD optimizations of the flow guiding parts of the turbine are planned. Furthermore, optimization of the deflection channels should be carried out.
- The Rotary Vane Expander (RVE) is the most mature expander in this study, benefiting from several years of iterative experimental development. Based on a validated one-dimensional thermodynamic design model, the RVE demonstrated a nominal mechanical power output of 8.5 kW and an isentropic efficiency exceeding 60%. The current work confirmed the expander's operational flexibility and capacity for reliable long-term performance under real conditions.
  ➔ The primary challenge remains ensuring long-term durability and reducing the frequency of maintenance interventions. Recent advances in component design and surface treatments, particularly the application of DLC (diamond-like carbon) coatings, have already improved reliability. Future work will focus on further extending operational lifespan and minimizing maintenance needs, with particular attention to the optimization of surface coatings and the lubrication of bearing systems to enhance durability and reduce wear.
- A comprehensive techno-economic and operational comparison of the three tested expanders highlights distinct advantages and trade-offs. The axial impulse turbine achieves the highest isentropic efficiency (61.7%) and specific power output (0.13 kW/kg), making it ideal for performance-optimized systems. Despite the need for a high-speed generator and costly power electronics, its moderate specific plant cost (16,428 €/kW) enables a potentially faster return on investment (ROI). The rotary vane expander (RVE) demonstrates similarly high efficiency (60.3%) combined with the lowest specific plant cost (14,932 €/kW) making it the most cost-effective solution under balanced performance conditions. It enables direct coupling with standard generators but requires maintenance (vane wear, lubrication) and cannot operate oil-free. The Elektra turbine, a radial re-entry design, shows the lowest efficiency (34.8%) and specific power (0.043 kW/kg), resulting in the highest specific plant cost (25,347 €/kW).

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