VOLUMETRIC EXPANDER VERSUS TURBINE – WHICH IS THE BETTER CHOICE FOR SMALL ORC PLANTS?

Andreas P. Weiß

Competence Center for CHP Systems,
University of Applied Sciences Amberg-Weiden,
Kaiser-Wilhelm-Ring 23, 92224 Amberg, Germany
e-mail: a.weiss@oth-aw.de
Web page: http://www.oth-aw.de

ABSTRACT

Since the steam turbine replaced the steam reciprocating engine by the end of 19th century it has been the only expander type in Clausius Rankine Cycle (CRC) and Organic Rankine Cycle (ORC) power plants at least above 1 MW_{el}. Positive displacement expanders like scroll or screw machines have often been applied for smaller units – in particular below 100 kW_{el}. One reason for this is that in cooling or compressed air technology these machines are cheaply available as compressors which can be "easily" converted to expanders. In contrast, up to now small turbines are rather seldom in this market segment. One goal of this paper is to discuss whether there are others reasons than those already mentioned to justify the choice of volumetric expanders for small ORC plants and to clarify whether small turbines provide benefits which could not have been used in the past just due to the lack of appropriate machines.

The paper briefly introduces the working principles of positive displacement and turbine expanders and evaluates them concerning their application in small ORC-plants. In the author´s opinion, the advantages of turbines outweigh their disadvantages. Nevertheless, in the following the decision between e.g. impulse or reaction type, axial or radial, single or multistage turbine has to be made. The paper discusses and explains the "pro and cons" of these turbine types. This paper aims to identify the best expander for a given application and in addition evaluates the different expanders with regard to their suitability for a so-called "micro-expander-construction-kit" which should help to design and build an appropriate expander for any given application out of a wide range of boundary conditions and working fluids. Here, the single stage impulse turbine was identified as the best compromise

1. INTRODUCTION

By the end of 19th century the steam turbine had superseded the steam reciprocating engine in power generation as well as vessel propulsion because of its superiority with regard to power density and higher allowable steam temperatures and thus higher cycle efficiencies. Since that time, the turbine has been dominating the power generation at least above 1 MW power output. It is generally accepted that turbines outclass volumetric expanders regarding large power output and processing huge mass flows. However, on the lower end of the power generation range, i.e. 1 MW or even below 100 kW power output the situation seems to be different. For small ORC or CRC units very often volumetric expanders are applied (Figure 1, see Branchini et. al., (2013)). Many publications e.g. Glavatskaya *et.al* (2012), Lemort *et. al.* (2013) postulate that for small power output or rather small mass flow a piston, screw, scroll or rotating vane expander would be the better choice regarding efficiency, rotational speed, size, costs etc.. This statement will be discussed in the following.

The author is convinced that besides the above-mentioned reasons there is another very simple reason for the frequent use of small volumetric expanders for small plants: These machines were cheaply available in the past from refrigeration or compressed air technology where they acted as compressors. Compared to small compressors, small turbines appear rather seldom. There is one exception: Small radial inflow, (axial outflow) turbines which are typical for automotive turbochargers. However, these turbines are designed for rather low expansion ratios (ER). Furthermore, they are only available as turbocharger units equipped with oil bearings which rely on the internal combustion engine´s oil system. Thus, the application of a turbocharger turbine as ORC expander is a bit elaborate.

The goal of the paper is to determine whether turbines can be a reasonable choice for small ORC units in the range of 3 to 100 kW_{el}. This question is in particular interesting for the development of a „micro-expander-construction kit“ for small ORC expanders which has to cover different temperature levels, mass flow rates, as well as fluids.

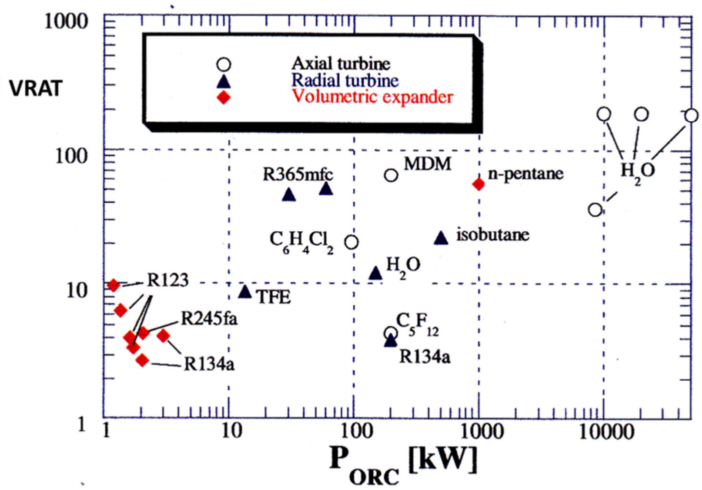


Figure 1: Actual VRAT values of existing ORC expanders with specified fluids (Branchini *et. al.*, 2014)

Figure 1 suggests that the volumetric expanders seem to dominate the power range below 10 kW. Furthermore, a first limitation of the volumetric expander can be identified: It is obviously restricted to small volumetric expansion ratios VRAT < 10. This is due to their built-in volume ratio. This geometrical volume ratio is for screw or scroll expanders about 5 and in the range of 10 for piston expanders (Lemort *et. al.*, 2013). However, higher expansions ratios may be advantageous e.g. for automotive waste heat recovery where small heat fluxes are combined with rather high temperatures or temperature differences, respectively. High temperature differences in an ORC usually result in high volume flow ratios for the expander.

2. VOLUMETRIC VERSUS DYNAMIC EXPANDER

2.1 Selection Criteria for Small Expanders

There are many criteria which may influence the choice of an expander for an ORC plant (Table 1). The design engineer tends to focus on efficiency, whereas for the „end-user“ the return of investment is the most important issue. Thus, beside efficiency, costs are a major criterion, which are strongly influenced by the design of the expander, its complexity, number of parts, the expected wear maintenance etc..

Table 1: Selection criteria for small ORC expanders

Economic Criteria	Technical Criteria
• costs	• efficiency
• availability on market	• rotational speed (bearing, generator)
• reliability	• lubrication (pollution of working fluid)
• maintainability	• sealing
	• power level (volume flow rate)

	• working fluid
	• wear
	• complexity
	• adaptability (fluid, VRAT etc.)

Furthermore, if the focus is not only on one ORC unit for one set of boundary conditions but a „micro-expander-construction-kit"-system with the intention to cover different levels of heat source/heat sink temperature, power output and different fluids, the adaptability of the expander design has also to be taken into consideration.

2.2 Comparison of Working Principles

The working principles of a volumetric and a dynamic expander are quite different (Figure 2). Volumetric expanders use the expansion work directly by changing the volume of a working chamber. Therefore, they deal with high pressures, big forces and small velocities of flow and machine parts. The built-in volume ratio and the swept volume are the main design parameters of volumetric expanders which limit their reasonable application range. The built-in volume ratio determines the specific work and the volume flow ratio (VRAT) which can be implemented per stage. A multi-stage arrangement for high VRAT is conceivable, but also elaborate due to additional piping, clutches etc.. The swept volume in combination with the rotational speed results in the volume flow rate which can be processed. Furthermore, both parameters determine the required size of the expander. Usually, due to their relative low rotational speed volumetric expanders can drive a standard generator directly without a gear. Part load (p. l.) e.g. reduced mass flow in an ORC can easily be handled by adjusting the rotational speed. The working chamber of the volumetric expander must be closed. Hence, it needs a contact sealing, which generates friction losses and wear and requires lubrication.

In a first step, dynamic expanders i.e. turbines convert the vapor´s internal energy into kinetic energy by means of nozzles. Therefore, pressure and forces are rather small but flow velocities are high. In a second step the kinetic energy is converted into mechanical work by turning the flow within the rotor blading. The circumferential speed u of the wheel has to be in the magnitude of the flow velocity i. e. high. The high circumferential velocity u in combination with a small diameter D leads to a necessary rotational speed usually in the range of 10,000 to 100,000 rpm or even more (u ~ n*D). Thus, turbines cannot be coupled directly to a standard generator. In addition, a gear or a high-speed generator must be used. In a turbine the fluid volume change during expansion is not just implemented by changing a chamber volume but by simultaneously increasing flow velocity and area. High expansion ratios can be implemented even in a single stage if supersonic flow is accepted, which leads to lower achievable efficiency. This is the reason why a turbine design with fixed main dimensions (e. g. diameter, length) can cover a wide range of boundary conditions (mass flow rate, expansion ratio etc.) just by adapting nozzle length and area, blade height and/or degree of admission. Partial admission (p. a.) is a means to handle part load (see chapter 3). In a turbine there are no contact seals. Hence, no lubrication is necessary. However, there is a certain leakage which cannot be avoided. Due to the high flow velocities, the absence of valves and the continuously working principle, turbines can process high volume flows in a small construction volume.

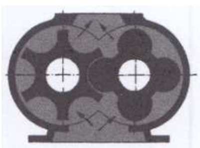
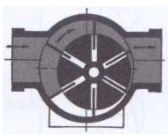
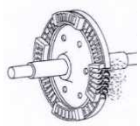
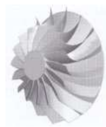
volumetric expanders				dynamic expanders		
$work = \int vdp$				$work \sim u^2$		
						
piston	screw	scroll	vane	axial	cantilever	radial

Figure 2: Expander types

2.3 Construction Types of Volumetric Expanders (Figure 2)

Piston: The classical volume expander is the reciprocating piston expander. It can show high expansion efficiencies (e.g. 70% in Eilts *et.al.* (2012)).The achievable volume ratios of volumetric expanders are in the range of 10 (Lemort *et. al.*, 2013) or slightly higher. However, it needs a lot of bearings and in addition inlet and outlet valves which makes the design complex and costly. Liquid in the cylinder can cause damage. Thus, the piston expander should not be applied for wet expansion. The machine and the flow are oscillating. Hence, the machine needs balancing and is prone to vibrations.

Screw: The screw expander expands the fluid continuously. It does not need any valves but at least four bearings for the two rotors. The rotors are not in contact with each other. Lubrication is required for sealing purposes. Even lubricated the necessary rotational speed is the highest for volumetric expanders. Without lubrication the rotational speed must be high (> 10,000 rpm). Therefore, standard generators are not suitable. Possible volume ratios (VRAT) are in the range of 5, efficiencies of around 50% (Eilts *et.al.*, 2012) might be acceptable. A certain amount of wetness can be handled by a screw expander.

Scroll: A scroll expander is a comparatively simple device: it consists of two spirals, one of which is rotating. It can be mounted directly on the shaft of the generator avoiding any additional bearing. Volume ratio is below 5 (Lemort et. al., 2013). Wang *et. al.* (2009) reported measured efficiencies in the range of 70% even for a quite small machine (< 1 kW). Droplets are no problem for a scroll expander.

Vane: The rotating vane expander is working continuously with a rather small rotational speed. Built in volume ratios are rather small (VRAT < 5). The vanes are in contact with the casing. Lubrication is required, which can spoil the working fluid. Furthermore, high friction losses and wear have to be expected. Rotating vane air motors are well known and widely used in industry. Their efficiencies are usually in the range of 30-40%. However, Badr *et.al.* (1984) report measured efficiencies of 80%.

Dry Runners: All the discussed volumetric expanders are available as dry runners, i. e. without lubrication, to avoid the spoiling of the working fluid. Usually, dry runners suffer from higher friction losses and leakages. Therefore, their efficiency is lower than that of their lubricated counterparts.

2.4 Construction of Small Dynamic Expanders - Turbines

Turbines are simple devices (Figure 2), comparable to volumetric expanders in terms of design. The turbine shaft needs two bearings. For small single stage turbines the rotor wheel can be mounted directly on the shaft of the high-speed generator. Because of the absence of contact seals, no lubrication is needed which could spoil the working fluid. Droplets at the end of expansion cause erosion in turbines. However, most of the applied organic working fluids show an isentropic or even dry saturation vapor curve. So, generally droplets are no problem in ORC applications. Small turbines suffer from a high relative surface roughness, big relative clearances and a big relative trailing edge thickness, etc.. Thus, they do not achieve efficiencies in the range of their bigger counterparts. All these statements hold true for axial and radial turbines of reaction or impulse type as well. The advantages and disadvantages of the different types of turbines will be discussed in more detail in chapter 3.

2.5 Which Expander Type for the "Micro-Expander-Construction Kit?

Table 2 summaries the results of this first evaluation. As long as efficiency is not the main focus or the only issue of consideration and as long as high speed generators are available, the turbine can compete with any volumetric expander. From the author´s point of view its main advantages are its simplicity, the possibility to adjust one basic turbine design quickly to different boundary conditions (e. g. VRAT) without changing the overall size and finally, that lubrication in contact with the working fluid can be avoided. Already Quoilin *et. al.* (2012) concluded that a turbine does have the broadest application map of all expander types. Hence, the author´s research group (Weith *et. al.*, 2013) decided to build up the ORC "micro-expander-construction kit" based on turbines.

Table 2: Evaluation of small expander types

expander type	η	VRAT	n	p. l.	size	adapt-ability	lubri-cation	wear	wet-ness	vib-ration	com-plexity	Σ
volumetric												
piston	**2**	**1**	**2**	**2**	**0**	**0**	**0**	**1**	**1**	**0**	**0**	**9**
screw	**1**	**0**	**1**	**2**	**0**	**0**	**0**	**2**	**2**	**2**	**0**	**10**
scroll	**1**	**0**	**2**	**2**	**1**	**2**	**0**	**1**	**2**	**2**	**2**	**15**
vane	**0**	**0**	**2**	**2**	**1**	**1**	**0**	**0**	**2**	**2**	**1**	**11**
dynamic	**1**	**2**	**0**	**1**	**2**	**2**	**2**	**2**	**1**	**2**	**2**	**17**

3. COMPARISION AND ASSESSMENT OF DIFFERENT TURBINES

Although the decision was made in favor of a turbine expander there are still many different types of turbines e. g. impulse or reaction turbines, axial or radial turbines and radially inflow or outflow turbines which can be considered. In the following, these turbines will be compared and evaluated regarding their applicability as a basis for the “micro-expander-construction kit”.

3.1 Impulse versus Reaction Turbine

Figure 3 compares the blading and the velocity triangles of an impulse stage and a 50% reaction stage. In an impulse stage the nozzles convert the entire required stage enthalpy drop Δh_{is} into kinetic energy. Thus, the nozzle exit velocity c_1 is very high. The rotor blades turn the flow without changing the magnitude of velocities ($|w_1|= |w_2|$). The pressure p in the rotor blading remains constant. In a 50% reaction stage the conversion of the stage enthalpy drop is equally distributed between nozzle and rotor blades. Thus, the nozzle exit velocity c_1 is not as high as in the impulse stage. The following acceleration ($|w_2| > |w_1|$) and pressure drop in the rotor blades has the same magnitude as in the nozzle blades. These differences in velocity triangles result in certain differences in stage characteristics:

- Higher velocities mean higher losses: the efficiency potential of an impulse stage is lower than that of a reaction stage.
- Thanks to the constant pressure via the rotor blading, impulse stages can be designed to work with partial admission (p.a.). This means that a portion of the total arc of the annulus is blocked off. Hence, the flow impinges only on-parts of the rotor blading. Partial admission is an option to implement part load with reasonable efficiency or to build turbines for very small power output without requiring blading heights that are too small to be manufactured with sufficient accuracy. Additionally, in this respect the impulse rotor blading benefits from the circumstance that it is subjected to the minimum pressure in the ORC plant and thus works with the maximal volume flow rate occurring in the cycle.
- The pressure drop via the reaction rotor blading generates non-negligible axial thrust. It either has to be balanced or the bearing must be able to withstand it.
- Furthermore, due to the pressure drop, the reaction stage efficiency is more sensitive to radial clearances.
- Applying the simplified Euler equation (1) for turbo machines ($u_1= u_2$)

$$\Delta h_{blading} = u * (c_{u1} - c_{u2}) \tag{1}$$

it becomes obvious (Figure 3), that

$$\Delta h_{blading,impulse} = 2 * u^2 \text{ and } \Delta h_{blading,reaction} = 1 * u^2 \qquad (2), (3)$$

Since $\Delta h_{blading} \approx \Delta h_{is}$ it follows that

$$u_{opt,impulse} = \frac{1}{\sqrt{2}} * \sqrt{\Delta h_{is}} \text{ and } u_{opt,reaction} = 1 * \sqrt{\Delta h_{is}} \qquad (4), (5)$$

i.e. for an identical stage enthalpy drop Δh_{is} the impulse stage requires only a significantly lower optimal circumferential speed ($u_{opt,imulse} = u_{opt,reaction} /\sqrt{2}$) than the reaction stage. This is a big advantage of the impulse stage applied as a small expander.

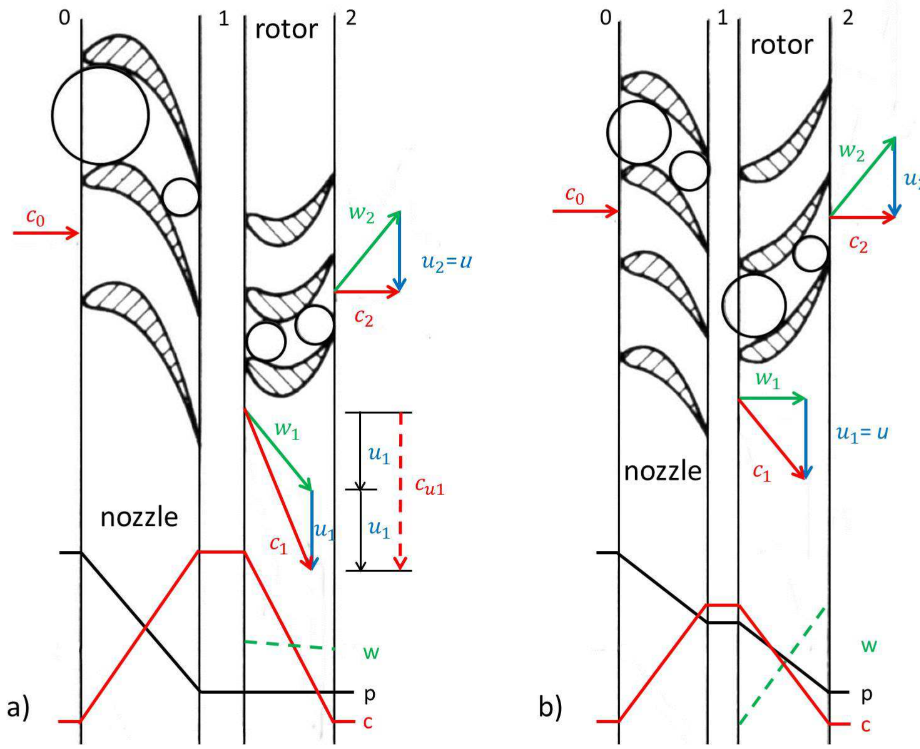
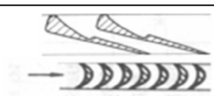


Figure 3: Comparison impulse (a) and reaction stage (b)

It has often been stated in literature - e. g. Bao and Zhao (2013) - that radial inflow turbines are better suited for low mass flow rates and high pressure ratios than axial ones. This is correct in principle. Nevertheless, for radial inflow turbines which are usually designed as reaction stages a maximal expansion ratio in the range 8 to 10 is reasonable (Moustapha *et. al.*, 2003). For higher values not only the nozzles but also the impeller would choke. That is why Bao and Zhao (2013) or Quoilin *et. al.* (2013) limit the rotor relative exit Mach number to 0.85 in their considerations. However, impulse stages can even cope with supersonic relative Mach number ($Ma_2 \approx Ma_1$) with acceptable efficiencies of 70%-80% (Vernau, 1987). As a consequence, very high stage expansion ratios ER >100 (Rinaldi *et. al.*, 2013)) or volume flow ratios (VRAT) can be put in practice.

Table 3: Impulse versus reaction turbines

turbine axial – cantilever - radial	impulse	reaction
		
efficiency potential (ts) ≈	80%	90%

turbine	η	VRAT	n	axial thrust	p. a.	leakages	minimal power/size	Σ
impulse	**0**	**2**	**2**	**2**	**2**	**2**	**2**	**12**
reaction	**2**	**0**	**0**	**0**	**0**	**0**	**0**	**2**

To summarize (Table 3), for small power output the impulse turbine (axial or radial) is obviously the more flexible and also a simpler approach for a micro ORC expander. The classical radial (reaction) inflow turbine might be more efficient for certain tasks but is probably less suitable for an ORC “micro- expander-construction-kit”. The impulse turbine can be easily adapted to a wide range of mass flow rate, fluids, inlet and exit pressures just by changing the nozzle area (-ratio), the blade height and/or the degree of admission.

3.2 Axial versus Radial Turbine (Figure 4)

Equation (6) below is valid for all types of turbines and shows the Euler work $\Delta h_{blading}$ processed in the rotor as difference between the squared rotor inlet (1) and exit (2) velocities (compare Figure 3). For an axial turbine, in particular for a small one with short blades the Δu^2-term is almost or exactly zero.

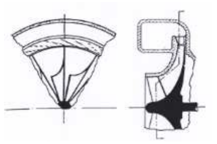
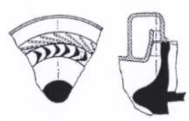
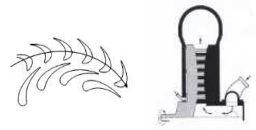
$$\Delta h_{blading} = \frac{1}{2} * [(c_1^2 - c_2^2) - (w_1^2 - w_2^2) + (u_1^2 - u_2^2)] \quad (6)$$

If the flow through a turbine wheel is subjected to a significant change in radius e.g. a change in circumferential velocity, the Δu^2-term contributes a substantial part to the overall enthalpy conversion like in a radial inflow reaction turbine. As a result, the radial inflow turbine can process higher expansion ratios than an axial stage without getting transonic or supersonic.

The radial inflow cantilever turbine benefits from this Δu^2-effect as well – however to a lesser extent. The author´s research group has recently developed small cantilever „quasi-impulse“ turbines which do not work with acceleration in the rotor ($|w_1| = |w_2|$). Nevertheless, caused by the Δu^2-term there is a small amount of reaction. Thus, the nozzles are slightly relieved. The Mach numbers at nozzle exit and rotor inlet remain smaller than for the axial counterpart. Although a small amount of reaction is used, this type of cantilever turbine can be applied using partial admission. Of course, this benefit does have a disadvantage: the cantilever „quasi-impulse“ turbine requires a slightly higher circumferential speed.

The significant advantages of radial outflow turbines are mainly twofold:

1. a flow direction from a smaller to a bigger radius corresponds to an area increase of the flow path which is helpful for expanding organic fluids with high volume flow ratios (VRAT)
2. centrifugal flow direction easily enables a multi stage arrangement if the expansion ratio of one stage is not sufficient.

turbine type			
axial	radial inflow	cantilever inflow	cantilever outflow
			

turbine type	η	VRAT	n	axial thrust	p.a.	multi stage	complexity	Σ
axial ($r \approx 0$)	**1**	**2**	**2**	**2**	**2**	**2**	**2**	**13**
radial inflow ($r \approx 0,5$)	**2**	**0**	**0**	**0**	**0**	**0**	**1**	**3**
cantilever inflow ($r > 0$)	**2**	**1**	**1**	**1**	**2**	**0**	**2**	**9**
cantilever outflow ($r \approx 0,5$)	**2**	**2**	**0**	**0**	**0**	**2**	**1**	**7**

Figure 4: Axial versus radial turbine

One main disadvantage of the radial outflow turbine stage is the fact that the Δu^2-term is working against the others terms (equation 6). I.e. in a radial outflow turbine the velocities (absolute, circumferential) and Mach numbers must be higher than in its inflow competitor for the same enthalpy drop.

The construction of a single stage axial turbine or a radial inflow cantilever turbine can be very simple. The manufacturing of the nozzle rings and the integral wheels does not need a 5-axis milling machine like for a radial inflow turbine. At least for small expanders the design and arrangement of a radial out-flow stage seems to be more challenging. Therefore, it was decided to rely on the single-stage impulse turbine for the “micro- expander-constructing-kit”.

3.3 The Micro Turbo Generator Concept

Figure 5 displays the developed micro turbo generator concept which relies on the “micro expander construction kit”. Its main features are:

- hermetically sealed turbine-generator (3 -100 kW_{el} , implemented with 5 manufactured sizes)
- single stage axial impulse turbine (10.000 – 70.000 rpm)
- integrally manufactured turbine wheel (Ø 50 – 250 mm)
- permanent magnet high-speed generator
- turbine wheel directly mounted on generator shaft: just one set of bearings required, no gear, no coupling
- roller bearings, slide bearings or aerodynamic bearings - depending on task
- compact design, low material usage
- design can be easily adapted to different boundary conditions, fluids etc.

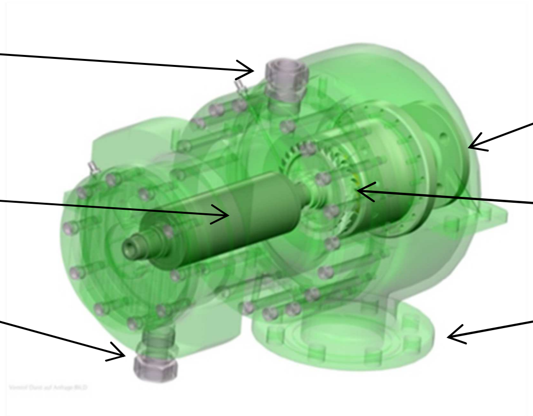


Figure 5: The micro turbo generator concept

3.4 Test Results of Developed Small Turbines

Based on the introduced concept several small axial and cantilevered turbines for steam, air and different organic fluids (Table 4) have already been built and successfully tested.

Table 4: Test results of built small turbines

	turbine type		fluid	∅D	n	VRAT	p.a.	P	η_{ts}
#				m	rpm	-	%	kW	%
❶	impulse	axial	steam	0,05	70000	3,9	30	1,0	40[1]
❷	impulse	axial	r245fa	0,08	21000	3,4	90	11,0	70[2]
❸	impulse	axial	cyclopentane	0,12	30000	16,0	55	11,0	65[3]
❹	impulse	axial	air	0,08	49000	6,4	60	5,4	60[1]
❺	low reaction	cantilever	air	0,08	49000	6,4	50	5,7	63[1]

[1]: brake efficiency [2]: including generator losses [3]: based on measured inlet/outlet temperatures and pressures

The small axial steam turbine ❶ was developed for automotive waste heat recovery. The r245fa- and the Cyclopentan-turbine (❷,❸) were designed and built for small waste heat recovery ORC plants as bottoming cycles for biogas engines. The air turbines (❹, ❺) are just demonstrators to investigate the pro and cons of the cantilever design compared to the axial design. The cantilever designed showed the expected higher efficiency. Nevertheless, the axial impulse design provides many other advantages (see Figure 4) as discussed. Due to different constraints (e.g. maximum rotational speed of the bearing technology) none of the turbines could be designed to operate with its optimal circumferential speed (u_{opt}). Therefore, the efficiencies are rather too low but still acceptable for the given application.

4. CONCLUSIONS

For small and micro ORC plants costs per kW are usually high and it is questionable if those small units will work economically. Therefore, all components, especially the expander must be simple and cheap in series production. From the author´s point of view, this requirement rules out the reciprocating piston and screw expanders. The rotating vane expander has already been produced in big numbers for compressed air application. Due to the scrubbing (sealing) vanes it shows usually poor expansion efficiency and high wear. Using a scroll expander mounted on the standard generator shaft a very simple expander unit can be implemented. However, the scroll expansion ratios are limited to the lower end. Thus, only smaller temperature differences can be processed efficiently in an ORC if a scroll expander is to be applied.

The single stage axial impulse turbine can cope with small volume rates and high expansion ratios. Thus, a wide range of boundary conditions and working fluids can be covered with the introduced „micro-expander-construction-kit“. Combined with a high-speed generator a compact, simple and cost-efficient turbo expander unit can be put into practice.

NOMENCLATURE

c	absolute velocity	(m/s)
D	diameter	(m)
ER	expansion ratio	(-)
h	enthalpy	(J/kg)
Ma	Mach number	(-)
n	rotational speed	(rpm)
P	power	(W, kW)
p	pressure	(N/m^2)
u	circumferential velocity	(m/s)
v	specific volume	(m^3/kg)
VRAT	volume flow ratio	(-)
w	relative velocity	(m/s)
Δ	difference	
η	efficiency	(-)

CRC	Clausius Rankine Cycle
ORC	Organic Rankine Cycle
p. a.	partial admission
p. l.	part load

Subscript

0	nozzle blading/stage inlet
1	nozzle blading outlet, rotor blading inlet
2	rotor blading/stage outlet
el	electric
is	isentropic
opt	optimal
ts	total to static
u	in circumferential direction

REFERENCES

Badr, O., O´Callaghan, P. W., Hussein M., Probert, S. D., 1984,
Multi-Vane Expanders as Prime Movers for Low-Grade Energy Organic Rankine Cycle Engines
Applied Energy 16, p129-146

Bao, J., Zhao, L., 2013,
A review of working fluid and expander selections of organic Rankine cycle
Renewable and Sustainable Energy Reviews 24, p 325-342

Branchini, L., De Pascale, A., Peretto, A., 2013,
Systematic comparison of ORC configurations by means of comprehensive performance indexes
Applied Thermal Engineering 61, p 129-140

Eilts, P., Seume, J., Brümmer A., 2012,
Zwischenbericht über das Vorhaben 1060 – C02-Sonderforschungsprogramm
Expansionsmaschine, *FVV Informationstagung 2012, Heft R558*, Bad Neuenahr

Glavatskaya, Y., Podevin, P., Lemort, V., Shonda O., Descombes G., 2012,
Reciprocating Expander for an Exhaust Heat Recovery Rankine Cycle for Passenger Car Application,
Energies, 5, p 1751-1765

Lemort, V., Ludovic, G., Arnaud, L., Declaye, S., Quoilin, S., 2013,
A comparison of piston, screw an scroll expander for small Rankine cycle systems
Proceedings of the 3rd International Conference on Microgeneration and Related Technologies, Naples, Italy

Moustapha, H., Zelesky, M. F., Baines, N. C., Japikse, D., 2003,
Axial and Radial Turbines
Concepts NREC, ISBN 0-933283-12-0

Quoilin, S., Declaye, S., Legros, A., Guillaume, L., Lemort, V., 2012
Working fluid selection and operation maps for Organic Rankine Cycle expansion machines
International Compressor Engineering Conference at Purdue, July 16-19

Rinaldi, E., Buonocore, A., Pecnik, R., Colonna, P., 2013,
Inviscid stator/rotor interaction of a single stage high expansion ratio ORC turbine
ASME Organic Rankine Cycle 2013, *2nd International Seminar on ORC Power Systems*, October 7-8, Rotterdam 2013

Verneau, A.. 1987,
Supersonic Turbines for Organic Fluid Rankine Cycles from 3 to 1300 kW
von Karman Institute for Fluid Dynamics, Lecture Series 1987-07, Brussels

Wang, H., Peterson, R. B., Herron, T., 2009,
Experimental performance of a compliant scroll expander for an organic Rankine cycle
Proc. IMechE Vol. 223 Part A: J. Power and Energy, pp 863-872

Weith T., Heberle F., Weiß A. P., Zinn G., 2013,
Development of a Small Scale ORC for Waste Heat Recovery
ASME Organic Rankine Cycle 2013, *2nd International Seminar on ORC Power Systems*, October 7-8, Rotterdam 2013

ACKNOWLEDGEMENT

This project was funded by the Bavarian State Ministry of Education, Science and the Arts.