LAYOUT AND MECHANICS DEVELOPMENT OF A NOVEL PISTON-TYPE EXPANSION ENGINE FOR WASTE HEAT RECOVERY

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ABSTRACT

Within the wide range of possible waste heat recovery (WHR) technologies (cf. Feulner (2008)), the Organic Rankine Cycle (ORC) showed to be widely applicable and established on the market, but mainly for high power systems; if used for smaller applications, either efficiency or costs of the system are often not satisfying. This is mostly caused by the characteristics of the used expansion engine types, mainly being reciprocating piston engines or turbines. Whereas the first show good part load performance and are well suited for small power, they suffer from lubrication problems and life time issues. Turbines, on the other hand, are preferably used for high power and can do without elaborate lubrication, but suffer under part load conditions and are quite costly.

Especially for medium power systems, i.e. waste heat fluxes between 200 and 400 kW, a novel concept for the expansion engine within a Rankine steam cycle was conceived. The aim was to combine the advantages of reciprocating piston engines and of turbines at reasonable costs. The so-called rotational wing-piston expander uses two pivoting shafts, each holding two wing-like pistons, within one housing, that are performing a cyclic movement relative to one another. This way, four working chambers with varying volumes are resulting, each experiencing repetitive compression and expansion. The conversion of the cyclic changing angular velocity to a constant rotation at the output shaft is done via a non-circular gear. This solution offers the possibility of sealing the lubricated gearbox against the steam-flooded section containing the working chambers via rotational seals, being much easier than the sealing within a conventional reciprocating piston engine.

This paper starts with treating the design and layout of this novel expansion engine concept, followed by mechanics development. The strategy for creating a robust and effective expansion engine design as well as the most important findings and insights gained during the experimental investigations of the engine are shown.

1. Introduction

The present paper results from a research project called STEAM, aiming at the recovery of waste heat in order to convert this unused heat into another, high-grade energy type. This idea is well known and already used in specific applications. The use of (O)RC ((Organic) Rankine Cycles) is a well suited and well proven option for this purpose, applying a corresponding expansion engine which either provides mechanical power or, attached to a generator, electric energy.
For the output power range of 10-100 kW, existing commercial systems based on turbines or positive displacement expanders either show low efficiencies or long payback periods (>8 years) due to high costs, cf. Figure 1. As for the latter, which are better suitable for low power ranges, different types are available (cf. Figure 2). What can be noticed for all different types is the need for high pressure ratios in order to deliver high efficiency, compactness, flexibility (cf. Table 1) and the possibility to run the expanders without lubrication in the working medium, which is possible for Roots, Scroll and Screw expanders.

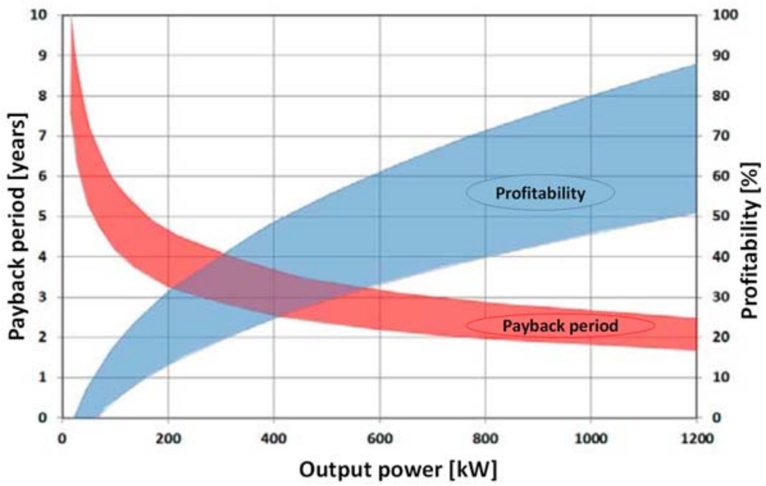

Figure 1: Payback time and profitability on output power for available systems (various working principles, depending on the output power) (Rettig (2018))

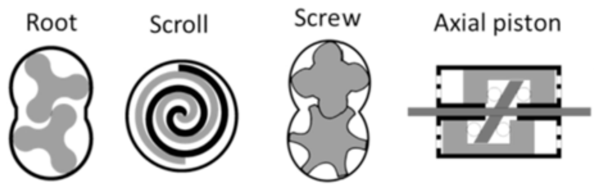

Figure 2: Common types of positive displacement expansion engines (Dumont et. al. (2018))

Table 1: Comparison of expander advantages and disadvantages (Dumont et. al. (2018))

	High Pressure and temperature	Wet expansion	Compactness	Flexibility	Efficiency
Piston	+	−	+	+	+
Screw	−	+++	+++	+++	+
Scroll	−	+++	++	++	++
Roots	−	+++	+	−	−

STEAM wants to fill the existing gap and deliver a Rankine Cycle for this rather low power range, starting with the development of a corresponding expansion engine (the rotational wing piston expander) for the application in stationary systems, delivering high efficiency at reasonable costs. The projected characteristics are high pressure and temperature capability, the possibility of coping with wet expansion, a compact design, and high efficiency.

A preceding project used a similar engine of a different power range (approx. 3 kW mechanical output power), with two main differences to the present expansion engine: The working medium was an organic one, and the wing pistons showed a rectangular cross-section with gap sealing only. The investigations showed a rather low efficiency, mainly due to high blow-by as a result of the piston shape and the gap sealing. Figure 3 shows the preceding prototype with the rectangular-shaped wing pistons. (cf. Lang et al. (2018a))

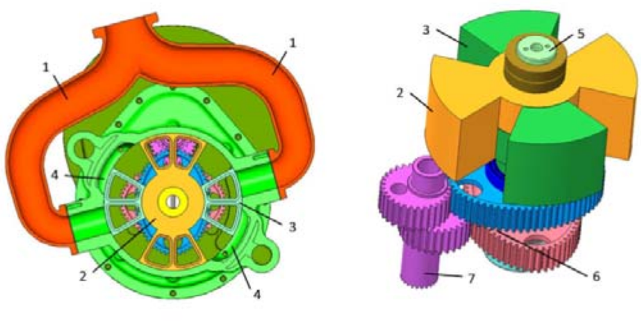

Figure 3: Rotating wing piston expander: First pre-prototype with rectangular-shaped wing pistons

1. Inlet port
2. Wing piston pair 1
3. Wing piston pair 2
4. Outlet port
5. Bearing of the rotational wing pistons
6. Non-circular gearing
7. Output shaft

This concept needed to be improved for the use with water as working medium as well as for an elevated efficiency. One main part was the redesign of the wing pistons, making the sealing of the working chambers easier (see elaboration in Chapter 2).

2. Rotational wing piston expansion engine – layout

The new, improved design should deliver the following characteristics:

- Displacement: 250 ccm/expansion chamber
- Nominal output shaft speed: 1500 rpm
- Pressure ratio: up to 15:1
- Built in volume-ratio ratio: 29:1
- Projected nominal output power (for 250 °C / 15 bar live steam): 15 kW
- Projected inner isentropic efficiency: approx. 80 %
- Projected power density (with respect to 4 working chambers): 15 W/ccm swept vol.

The layout and the main characteristics of the new generation wing piston expander will now be described. (cf. Lang et al. (2018a))

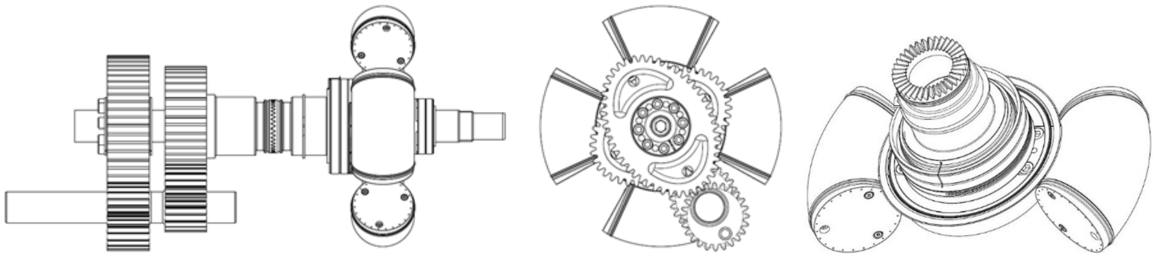

Figure 4: **Rotors and gearbox (left and middle) and single rotor (hollow shaft & hub & wing-pistons, right))**

2.1 Working principle

In the rotational wing piston expander (Figure 4), two concentric shafts (one hollow shaft and one inner shaft) with a pair of torus-shaped pistons on each of them (shaft & pistons = rotor, cf. Figure 4) within one common housing.

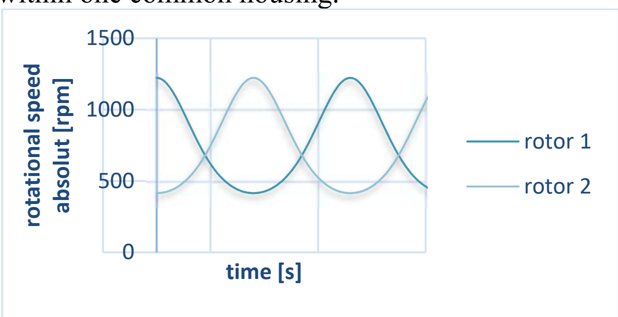

Figure 5: **Rotational speed of the two wing piston pairs (i.e. rotors) on time**

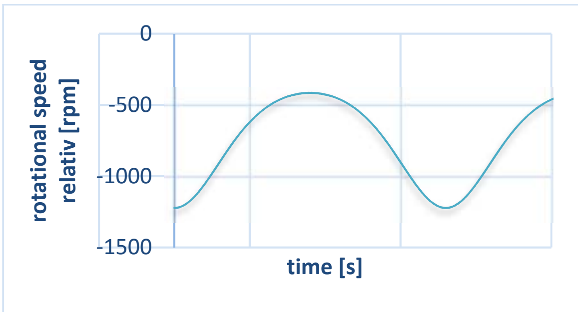

Figure 6: **Relative rotational speed of the two wing piston pairs (i.e. rotors) on time**

The two rotors rotate with varying cyclic rotation speeds relative one to another, as can be seen in Figure 5. This relative motion leads to four working chambers with varying volumes, each compressing and expanding alternatively due to the relative rotational speed of the two rotors, see Figure 6. By joining the two rotors via a special gear box, the resulting expansion work can be used on one single output shaft rotating at constant angular speed. The so called non-circular gear box consists of oval gear wheels in combination with eccentric circular cogs. As the two rotors do not only rotate relative to one another, but additionally both together in one defined direction, the gas exchange can be managed via port control without moving valves, similar to two stroke internal combustion engines. This enables a rather simple mechanical layout, reducing complexity and costs, but also guarantees high efficiencies.

2.2 Mechanical components

Hollow shaft & inner shaft

The hollow shaft as well as the inner shaft are each joined with two wing pistons via a hub. Pivot bearings enable rotational movement between the hollow shaft and the inner shaft, and thus between the two wing piston pairs. The two shafts are connected to the non-circular gearbox which defines the specific movement.

Gearbox

In order to convert the cyclic rotation with varying angular velocity of the rotors to a rotation with constant angular velocity of the output shaft, a non-circular gearbox is used; its design is very close to the gearbox which has already been used for the first, low-power, prototype (cf. Figure 3). This gear box consists of two oval gearwheels, one on each rotor, with a certain angle between them (cf. Figure 4). On the output shaft, corresponding gearwheels are mounted; in this case they are circular but eccentric. The positioning of the gearwheels on the shafts is variable in order to be able to compensate occurring tolerances. The gearbox is to be attached to a generator, with a flywheel in between.

Expander housing

The housing of the prototype expander uses a modular design, using covers and plates of simple shapes. The top and bottom plate contain the bearings of the rotors. The centre housing part contains the torus-shaped liner, as well as some holes for sensors. Additionally, this part can be conditioned (heated or cooled) for basic investigations concerning near-isothermal expansion, which could possibly enhance efficiency. The centre plate also contains the inlet and exhaust ports and the fitting for live steam feed and exhaust pipes.

Wing pistons

The pistons, also showing a torus-shape, are fitted to the main shaft via a hub. The detailed shape of the pistons is characterized by the seats of the working-chamber seals; on the other hand, the inertia of the rotors should be low, which is why the pistons are as light as possible, and the material is aluminum based.

3. Main challenges in mechanics development

The functional prototype treated in this paper should prove the concept in a small scale, i.e. aiming at a rated power of approx. 15 kW when supplied with 15 bar / 250 °C water steam at 1500 rpm at the output shaft. Nevertheless, a compact design and high power-density were important boundary conditions, in order to explore the limits of the concept. Apart from standard mechanics layout of some of the engine parts, being in some extent similar to the layout of conventional internal combustion engines, there are some special challenges occurring due to the interaction of high pressure forces and inertia forces with the special medium used (water steam); additionally, the absence of lubricant in the pressurized area as well as high temperatures pose high demands on the components. One of the most difficult compromises to find is the one between very tight sealings in order to minimize leakage, and the elevated friction generated by these sealings. That is why in this project, especially the sealings in the high pressure areas have been chosen with rather high emphasis on tightness to prove thermodynamic function, and can be modified after the first results on the steam test bench in order to optimize overall efficiency, but without sacrificing longevity.

Sealing design

One of the main critical areas of steam expanders is the sealing issue. It defines not only the inner efficiency by minimizing the blow-by, the friction, and the complexity of the design, but it is also crucial for the operating concept by separating the working medium from the lubricant. The demands for the sealings in this special concept are:

- Working pressure up to 15 bar
- Temperature of the working medium of up to 250 °C
- Working medium water (in high pressure areas)
- Lubricant-free operation (in high pressure areas)
- Mechanical load through port controlled gas exchange

The sealings of one rotor to another (i.e. between the two hubs), between the rotor hub and the housing as well as between the hollow shaft, the inner shaft and the housing, are all rotational, but require different approaches: some have to seal the high pressure steam from the low pressure steam, whereas some need to separate steam from lubricant.

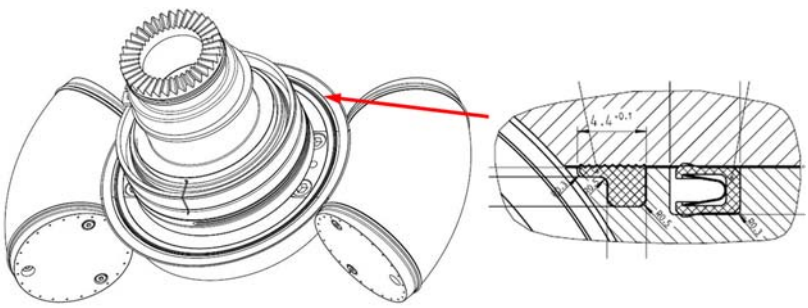

Figure 7: Axial seals on the hubs, preventing blow-by

Figure 7 shows the axial seals on the outside and inside, preventing blow-by of the operating medium into adjacent chambers. The geometries of the pre-seals are designed so that they are placed as far as possible outside, despite the larger friction radius, in order to achieve the lowest possible dead volume. In these axial seals, a two-stage concept of pre-seal and spring-loaded main seal is applied; the main seal has a smaller effect on the frictional torque despite the larger contact pressure due to the smaller friction radius.

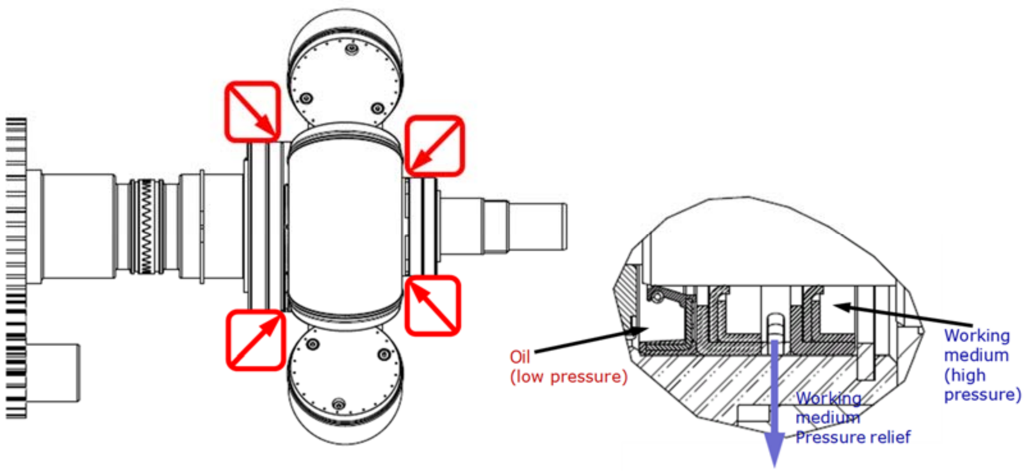

Figure 8: Shaft sealing package layout

Figure 8: The sealing package "shaft" is designed as a unit consisting of three sealing elements: A sealing with a radial shaft sealing ring, which is used to seal the oil, and two sealing bodies consisting of PTFE sealing lips, which can seal the high pressure of the working medium, and thus separate operating and lubrication medium. Within the two-stage sealing unit, a discharge hole to transport the leaked mixture from the machine is positioned.

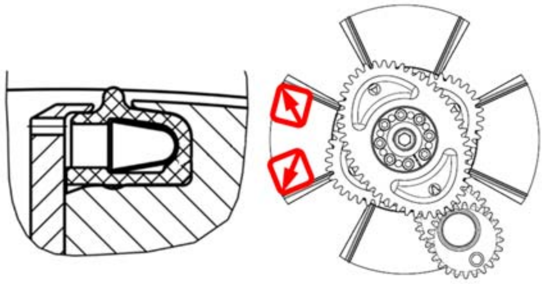

Figure 9: Piston sealing

Figure 9 shows the piston sealing. In the present case, the piston of the expander can be compared with a double-acting piston in an ICE. This means that the top as well as the bottom of the piston are pressurized. The hermetic prevention of dynamic leakage regarding the mass transfer and mixing is not as important as for other systems, as in the cylinder area there should be only working medium (as the lubricant area is separated via rotational seals). Anyway, regarding efficiency, dynamic leakage is of high priority. Due to the fact that the efficiency is known to depend on the system friction and this further on correlates with the dynamic leakage, there has to be a compromise and thus a certain leakage can be accepted. Seals on the piston are designed symmetrically with respect to their radial center plane. The chosen solution is a PTFE-based contact type seal.

The high-performance lip seal with metal spreading spring is supposed to defy the high thermal requirements with the help of special PTFE compounds. The expected pressure limit of 15 bar should also be met. The V-shaped bending springs stretch the PTFE compound body and thus ensure a static sealing effect. Due to boreholes, an intended effect of dynamic pressurization and thus increasing of the sealing effect depending on the working pressure occurs. This constructive measure should also counteract the blow-by effect.

Bearing layout

The bearing concept, both for the gearbox as well as for the rotors, is based on roller bearings. In order to accommodate axial forces due to pressure, pre-loaded axial ball bearings or deep groove radial ball bearings are used. In order to prevent solid contact between the rotors, the axial clearing has to be thoroughly set using shims.

Between the inner and the outer shaft and between the shafts and the housing, cylindrical roller bearings are used in combination with ball bearings.

Piston mounting

As the rotational speed of the single rotors within one revolution varies by over 200 % (cf. Figure 5), there are occurring high inertia torques. In order to keep them as low as possible, the pistons themselves are made of aluminium. For mounting them to the steel hubs of the shaft, a special mounting design using two fitting screws per piston is used. This mount has proved to be critical with respect to mechanical load, as can be seen in Figure 10: An FEM-simulation of the piston, applying the working pressure (15 bars) on one side of the piston, showed elevated equivalent stress values in the areas near the mounting screws. This could also be observed in first experimental tests using pressurized air, where cracks in this area occurred.

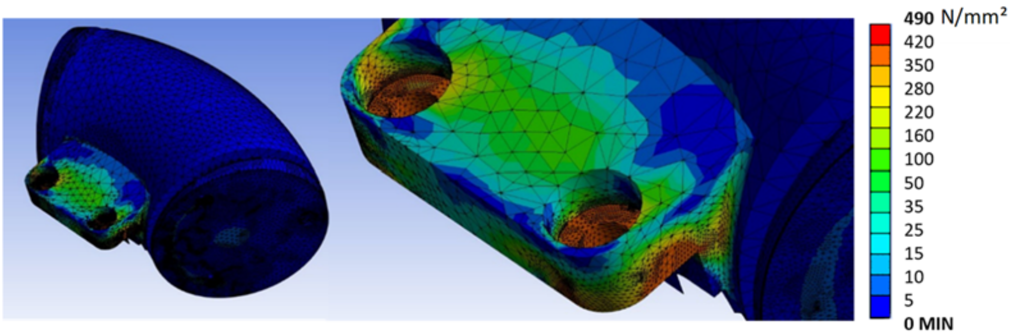

Figure 10: FEM-simulation: Equivalent stress on the piston mount

Therefore, the piston mounting has to be redesigned, applying more screws, a wider support and adapted geometrical shapes.

Friction minimization

In order to maximize the overall efficiency of the WHR system, the friction torque of the expansion engine is one of the crucial issues to be addressed. The main drivers are:

- Shaft seals
- Gearing losses
- Bearing losses
- Piston influences

In order to reveal the main influences, the expansion engine was investigated on the motoring test bench (Figure 11), using strip-down methods.

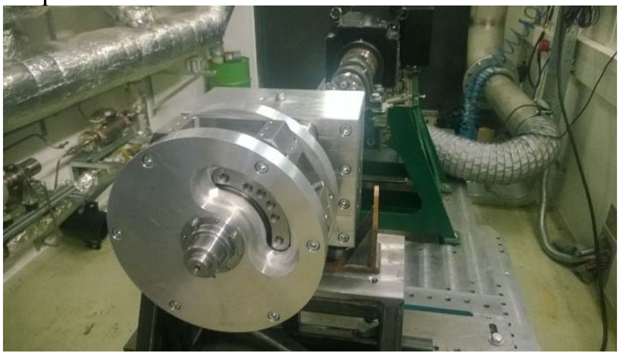

Figure 11: Expansion engine on the motoring test bench

The tests were carried out at an engine test bench: The expansion engine was motored at defined speeds (300 to 1500 rpm) with warm gearbox oil. At each speed, the torque was measured via a torque flange, positioned between the engine and the brake. Then, one single element (e.g. a seal) was removed, and the measurement was carried out again. This procedure was repeated, until only the rotor shafts without pistons and sealings were left. This way, the single contributions to the total friction could be determined.

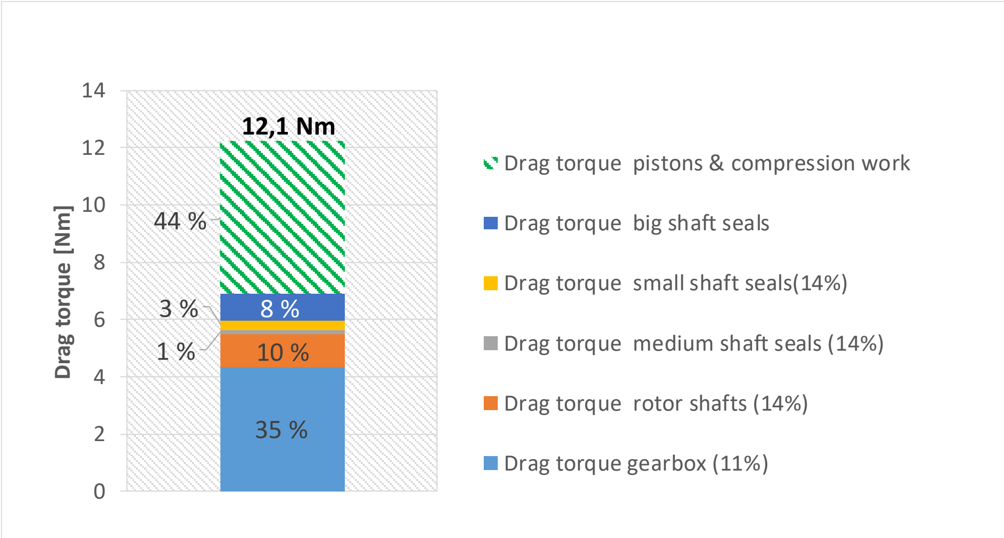

Figure 12: Contributions to drag torque of expansion engine at 1500 rpm output speed

The investigation results (Figure 12) show, that the gearbox influence is the biggest one, but as well the one that can be hardly influenced. The rotor shafts, mostly due to their bearings, add another relevant contribution, approximately as much as all of the shaft seals together. As for the piston induced friction, it is hard to tell the exact portion of the seals, as due to the engine design, the expander works as a compressor when motored. Therefore, the compression work is comprised. A pressure indication system, which has not yet been built up completely at the time of the investigations, would fix that problem, as it would allow the calculation of the inner work.

In the course of the measurement activities, it became clear that an accurate adjustment of the axial clearings as well as the bearing pre-loading is crucial for an overall low friction torque, as well as for the life time of the single components. As for the speed, the influence of the sealings on friction showed to be increasing strongly with higher speeds.
After the tests, it was decided that a new generation of sealings had to be designed in order to reduce tolerance influences and friction.

4. Summary/Conclusion

The presented concept of a rotational wing-piston steam expansion engine for the use in stationary WHR systems offers intriguing simplicity concerning the sealing of the lubrication area from the working medium area, as only rotational seals are needed. This way, closed media cycles are much easier to realize compared to a conventional reciprocating piston layout. For the sealing of the working chambers themselves, the promising concept of using PTFE-type contact sealings has been chosen.
The main challenges of mechanics development were, apart from the sealing concept, the overall friction minimization as well as the piston mounting. The latter is critical due to the material pairing (piston: aluminium; hub and shaft: steel) and the high loads, caused by the very compact design. FEM-simulations or the piston mounting showed that a redesign is needed. For the friction optimization, strip-down friction measurements on the motoring test bench have been carried out, leading to a redesign of the sealings.

The next steps after the expander redesign will be to investigate it on a superheated steam test bench at Graz University of Technology (as presented by Lang (2018b) and Lang(2017c)), which will not only evaluate one specific aspect, but the whole range of kinematic, tribological, thermodynamic, thermal, mechanical and corrosive influences. This way, a more holistic view will be possible, which should enable a further evolution of the expander design.

NOMENCLATURE

FEM	Finite Element Method
ICE	Internal Combustion Engine
(O)RC	(Organic) Rankine Cycle
PTFE	Polytetrafluoroethylene
WHR	Waste Heat Recovery

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