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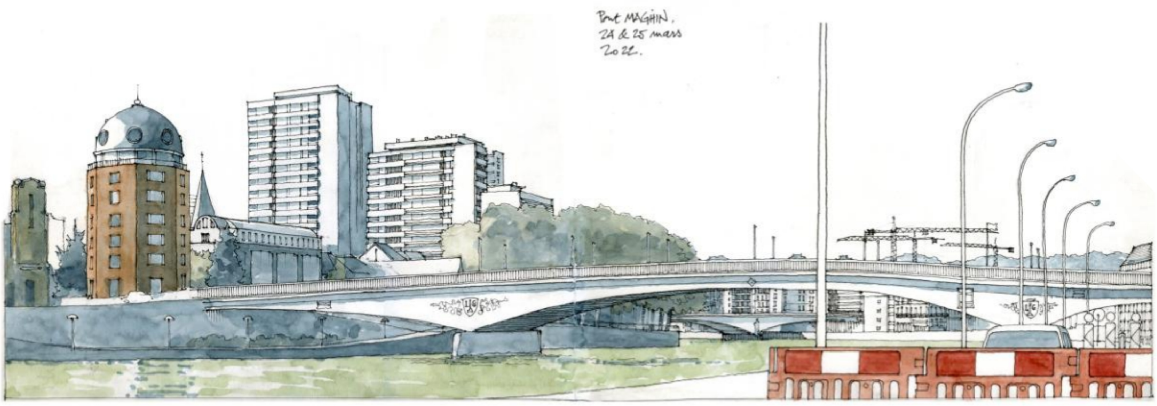

EXPERIMENTAL PRE-VERIFICATION OF A 7 KW$_{EL}$/3,000 RPM RADIAL RE-ENTRY TURBINE FOR SMALL ORGANIC RANKINE CYCLE (ORC) PLANTS

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Abstract. Experimentally determined efficiency characteristics of an uncommon velocity compounded radial four-fold re-entry turbine are discussed in this short paper. The small 7 kW turbine was designed to drive a 3,000-rpm standard-generator in a small ORC plant operating with hexamethyldisiloxane (MM). It was pre-investigated numerically and experimentally with compressed air as working fluid to check the overall functionality and to learn about complex flow phenomena. Due to the "off-design" working fluid, maximum total-to-static isentropic efficiency of only 27 % was achieved. However, the uncommon small turbine has the potential to substitute an existing standard axial impulse turbine with a similar rating which must work at 24,000 rpm driving an expensive high-speed generator.

Keywords. Radial re-entry turbine, Velocity compounded, Organic Rankine Cycle (ORC).

1 Introduction

Organic Rankine Cycles (ORC) are a well-established solution to recover low temperature waste heat in industry [1]. Most turbine driven systems work with power ratings from some 100 kilowatts to a few megawatts. Below 100 kW$_{el}$, volumetric expanders dominate because they are cheaply available as derivates of mass-produced refrigeration compressors [2]. However, turbines provide several advantages: they do not need any lubrication within the working fluid, they experience almost no wear, and they can be easily adjusted to different fluids and boundary conditions. The main disadvantage of those small (< 100 kW$_{el}$) turbines is the fact that they must work at rotational speeds of several 10^4 rpm [3]. Therefore, an expensive high-speed generator must be installed instead of a 3,000 rpm off-the-shelf generator along with the need for high-speed bearings, precision manufacturing and balancing. This drawback can be overcome by using a velocity compounded radial multi-fold re-entry turbine – a so-called Elektra turbine [4]. Figure 1 displays the architecture of the designed Elektra turbine: a convergent-divergent nozzle accelerates the fluid, then, the supersonic jet passes the wheel, four times in our case, at almost constant pressure. The velocity or kinetic energy of the jet is reduced from wheel pass to wheel pass (1st WP: 45%, 2nd WP: 30%, 3rd WP: 19%, 4th WP: 6%). This approach is called velocity compounding. Whereas for a simple impulse turbine, the optimal circumferential speed of the wheel would be half of the jet velocity, for the Elektra it is only one-eighth. Thus, the rotational speed can be reduced to 25 % compared to the impulse stage. In the discussed application, the Elektra wheel diameter was furthermore doubled compared to the existing 24,000 rpm axial impulse turbine. Thus, the necessary rotational speed could be reduced from 24,000 rpm to 3,000 rpm for the same duty [5].

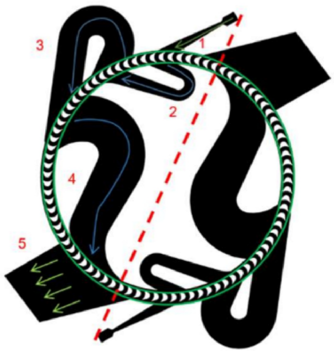

Figure 1: Architecture and flow path of the velocity compounded radial four-fold re-entry turbine (Elektra): 1 con-di-nozzle, 2 first deflection channel, 3 second deflection channel, 4 third deflection channel, 5 outlet. Two sets of nozzles and two sets of deflection channels are installed which are used in parallel with the same wheel. (red dotted line shows the symmetry plane).

2 Results and discussion

The preliminary efficiency characteristics of the 3,000 rpm Elektra turbine, designed for MM, were experimentally determined with compressed air at different pressure ratios and compared to computational fluid dynamics (CFD) results. Table 1 lists the MM turbine design data in comparison with the air pre-verification data. The compressed air test facility is described in [6].

Table 1: Turbine MM design and air pre-verification data

Parameter	Unit	Design	Pre-verification
Fluid	-	MM	air
Mass flow rate	kg/s	0.30	0.15
Inlet pressure	bar	6.50	6.00
Inlet temperature	°C	190	20
Outlet pressure	bar	0.55	≈1

First, Figure 2 compares the calculated absolute Mach number distributions for the given boundary conditions (Table 1) for MM (top, see [5]) and for air (bottom, see [7]). The simulation approach is explained in detail in [5]. The MM flow in the nozzles continuously accelerates to an outlet Mach number of about 2. The air flow accelerates even to higher Mach number but a strong normal shock within the nozzles leads to transonic flow at their outlets. The MM flow utilizes the cross-section of the deflection channels more effectively while the air flow forms rather singular jets. The overall Mach number level in the deflection channels with MM is significantly higher than with air. This clearly shows that the expansion of the air flow does not match the geometry of the MM turbine. Ultimately, this is also reflected in the calculated expansion efficiency, which is approx. 50 % for MM and approx. 34 % for air.

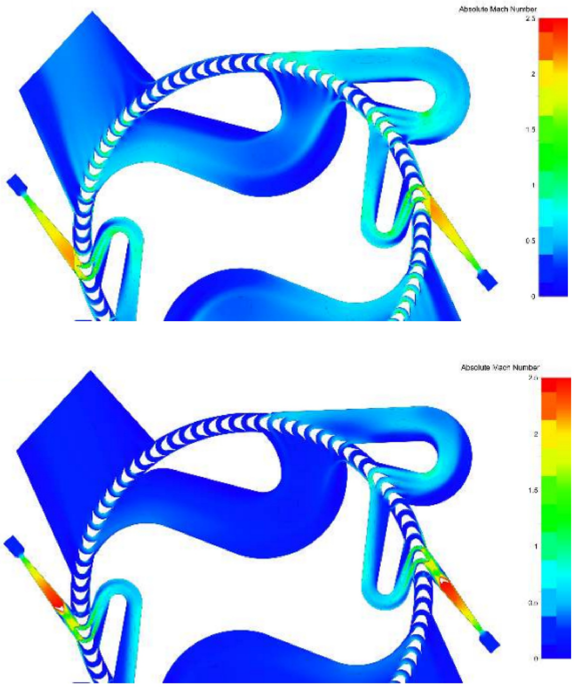

Figure 2: Absolute Mach number distributions at mid span for MM (top, [5]) and air (bottom, [7]).

The experimentally determined total-to-static isentropic efficiencies as function of rotational speed are depicted in Fig. 3. The maximum achieved efficiency for maximum PR = 6 was about 27 %, which is relatively low. Firstly, this is due to the wrong fluid – air instead of hexamethyldisiloxane (MM). The second adverse effect is the too small pressure ratio. The design pressure ratio with MM is PR = 13 (compare Table 1). On the compressed air test rig, PR = 6 was the maximal achievable pressure ratio. Finally, this is the tribute of the velocity compounding. A supersonic jet impinges on the wheel at very high relative velocity and is re-deflected three times (see Figure 2). These flow physics reduce efficiency significantly.

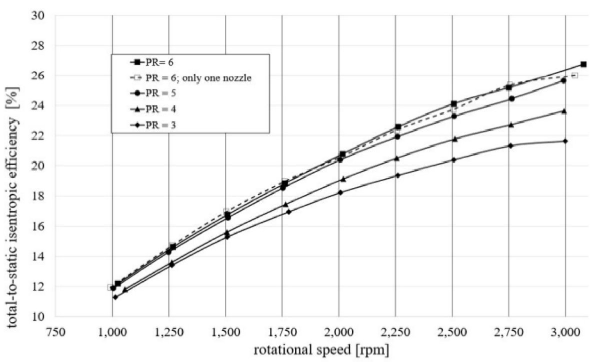

Figure 3: Total-to-static isentropic efficiency of the Elektra turbine for air at different pressure ratios (PR) as a function of rotational speed [7].

It is interesting that for PR = 3, maximum efficiency is achieved at 3,000 rpm. This is the design rotational speed for MM. For the higher-pressure ratios, maximum efficiency will be achieved at higher rpm which could not be tested due to limitations of the brake. At PR = 3 the enthalpy drop of compressed air is about 65 kJ/kg which is close to the MM design enthalpy drop (PR = 13) of 54 kJ/kg. With the increasing air pressure ratio, the enthalpy drops increases as well and achieves about 117 kJ/kg for PR = 6. The optimum rotational speed of a turbine is a function of the enthalpy drop. Because the enthalpy drop with air at PR = 6 is about twice the enthalpy with MM at PR = 13, the optimum rotational speed with air (PR = 6) is theoretically √2 times higher than that with MM. Finally, the dashed curve in Fig 3. displays efficiency with only one of the two nozzles open i.e. with only 50 % of the mass flow rate. Interestingly, efficiency is almost the same, what shows, that ventilations losses of the half non-admitted arc of the wheel are obviously very small at 3,000 rpm.

Fig. 4 displays the comparison of streamwise static pressure distribution – measurements versus computational fluid dynamics (CFD).

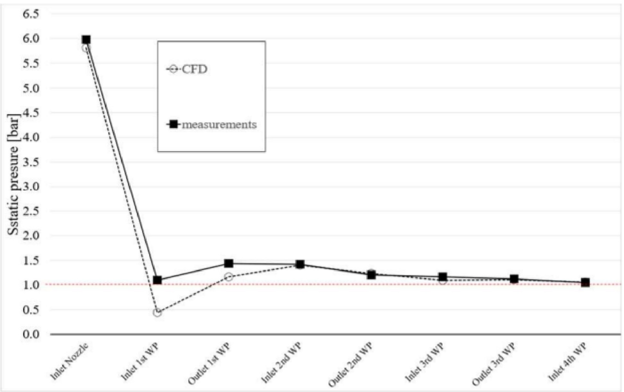

Figure 4: Streamwise static pressure distribution for air: measurement (full squares) versus computational fluid dynamics (CFD, open circles).

The qualitative shapes with a strong expansion over the con-di-nozzle, a pressure increase over the first wheel pass (WP) and an almost constant pressure distribution further downstream are the same. However, in the simulation (CFD), the nozzle expands to 0.5 bar, whereas the measurement shows about 1.0 bar nozzle exit pressure. Furthermore, static pressure is still higher at the outlet of the first wheel pass in the measurement. It cannot be excluded that these discrepancies are just due to the positions of the specific pressure taps in the measurement.

Finally, Fig. 5 compares total-to-static isentropic efficiencies as function of total-to-static pressure ratio (PR). As expected, the simplified frozen rotor CFD-simulation (see [7]), neglecting tip leakages and disc friction, shows about 5 p.p. higher efficiencies compared to measurements. Efficiency is rising with pressure ratio (PR) because the nozzle or the turbine is operated far below the design pressure ratio (PR_{Design} = 13).

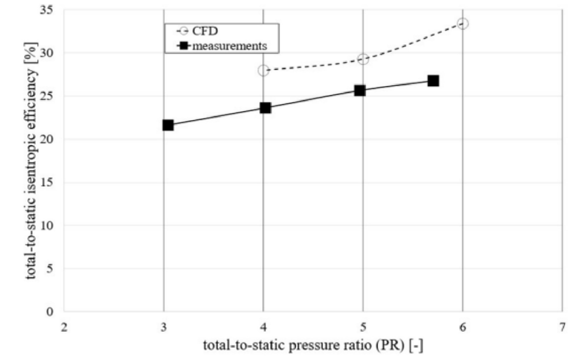

Figure 5: Total-to-static isentropic efficiency for air as a function of pressure ratio: measurement (full squares) versus computational fluid dynamics (CFD, open circles) [7].

3 Summary and conclusions

The short paper discusses the compressed air pre-verification of a small MM Elektra turbine. Of course, the gathered results can only give a tendency because the properties of the chosen test working fluid air differ significantly from those of the design working fluid hexamethyldisiloxane. However, this approach helped the authors to understand the flow phenomena of the Elektra turbine better and to develop and adjust their design approach.

The 50% isentropic expansion efficiency, forecasted by CFD, would be quite competitive with regard to volumetric expanders with similar power ratings [2]. Imran *et al.* 2015 lists many measured efficiencies of small 3,000 rpm ORC expanders. The mentioned maximum efficiencies vary from 30 % to 70 % for rotary vane, 10 % to 75 % for piston, 22 % to 84 % for scroll and 45 % to 80 % for screw expanders. All these positive displacement expanders need lubrication in the working fluid because they suffer from mechanical wear. Furthermore, some expander parts must be regularly replaced due to wear (e.g. vanes, gaskets etc.). They are prone to vibrations and noisy operation. All these disadvantages could be avoided by an Elektra turbine.

In 2025, the Elektra will be finally tested in an operational MM ORC plant [8] at UCEEB in Prague. There, it will substitute a 3,000 rpm rotary vane expander which is achieving about 56% expansion efficiency.

Acknowledgement

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