TESTING AND MODELING A VANE EXPANDER USED IN AN ORC WORKING WITH HEXAMETHYLDISILOXANE (MM)

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Introduction

For ORC systems with power output up to 10 kW, small-scale turbines are still expensive to manufacture and their use can be problematic in terms of high shaft speed or quality of inlet vapor. It is therefore preferable to use positive displacement expanders. The poster presents and analyses the measurements conducted on a prototype of a vane expander of own design. This vane expander characterized by a 1 kW power output operates in an ORC heat engine that uses hexamethyldisiloxane (MM) as a working fluid. The expander inlet temperature varies approximately from 135 °C to 150 °C, inlet pressure varies approximately from 200 to 300 kPa abs, isentropic efficiency from 0.4 to 0.58. A grey-box model, which is calibrated on the base of the measured data takes into consideration major losses of the expander: supply and discharge pressure losses, under and over-expansion, internal leakages and mechanical losses. The model is finally used to assess the impact of each source of losses on the overall performance of the expander.

Description of a test rig and a vane expander

A scheme of experimental test rig with organic Rankine cycle built at UCEEB is shown in figure 1. ORC is designed as single-circuit. Flue gas from diesel burner with temperature of approx. 390 °C is used as a source of heat. Evaporator is a spiral tube exchanger. Heat input to the cycle is approximately 20-23 kW. Generated vapor comes into a vane expander of own design. The expander is connected directly with the evaporator without any control valve. The expander drives a 24 VDC automotive alternator, which is connected with a photovoltaic inverter. Condenser is similar to the evaporator. It is a spiral tube exchanger designed with focus on low pressure drop. Condensation occurs inside the tubes. Condenser is air-cooled with the inlet temperature of about 30 - 35 °C. Water sprinkler for a possibility of cooling performance improvement was mounted above the condenser. Condenser is followed by condensate receiver under which is placed a gear pump regulated by frequency inverter.

Vane expander was selected because of simple design, low manufacturing costs, reliability and possibility of reaching good thermodynamic efficiency in case of optimal design. Design of tested expander is shown in figure 2. Expander is semi-hermetical; torque transmission is ensured by a magnetic coupling with permanent magnets. Nominal built-in volumetric expansion ratio is 3. Expander is designed to produce 1 kW of mechanical power with overall efficiency of 0.41 at the nominal conditions of the cycle (inlet 225 kPa, 135°C / outlet 23 kPa). Nominal shaft speed is 3020 RPM.

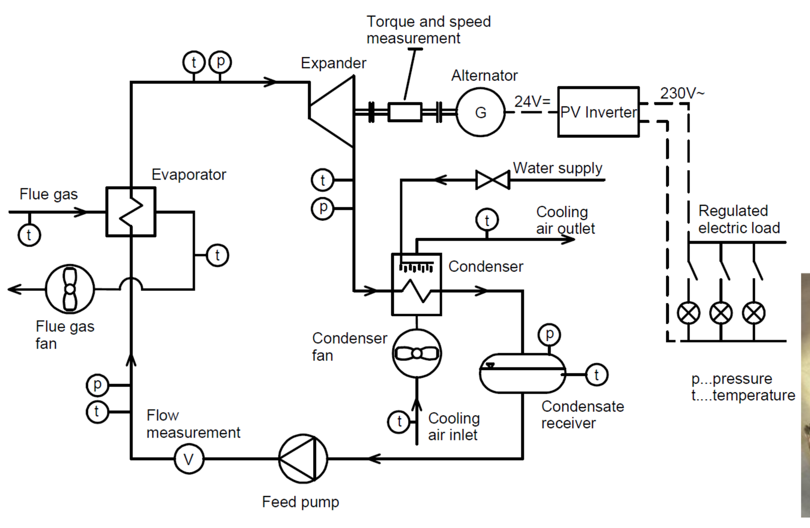

Fig. 1: Scheme of the ORC test rig

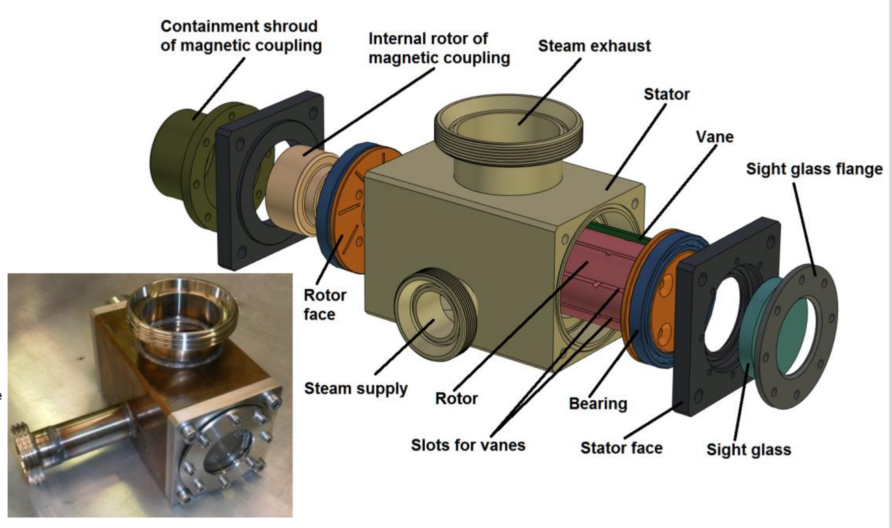

Fig. 2: Design of the vane expander

Modeling

A semi-empirical model was calibrated on the base of the measured data. This lumped-parameter model takes into consideration major losses of the vane expander such as supply and discharge pressure losses, under and over-expansion, internal leakages and mechanical losses (figure 3). The input variables of the model are the supply pressure, the supply temperature, the exhaust pressure and the rotational speed of the expander. The model then calculates the mass flow rate displaced by the expander, the delivered mechanical power and the exhaust temperature. The validation of the model was realized comparing the predicted and measured values for the mass flow rate, mechanical power and the exhaust temperature of the expander. The maximum deviation between the predicted and measured mass flow rate is 3 %, shaft power was predicted with an accuracy of 8%.

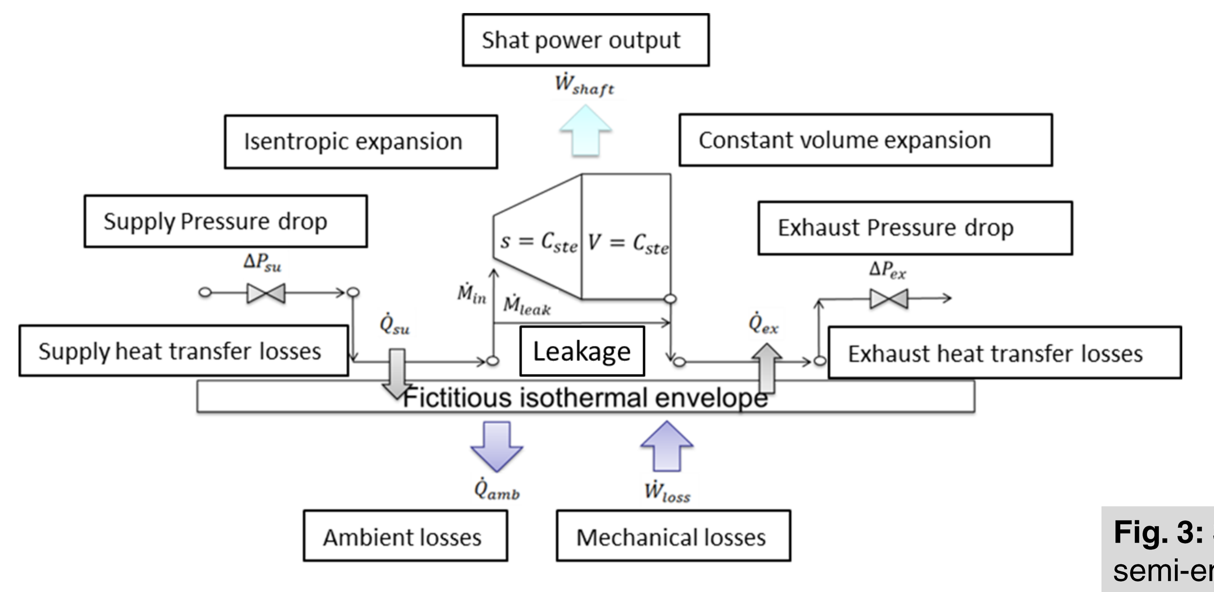

Fig. 3: Scheme of the semi-empirical model

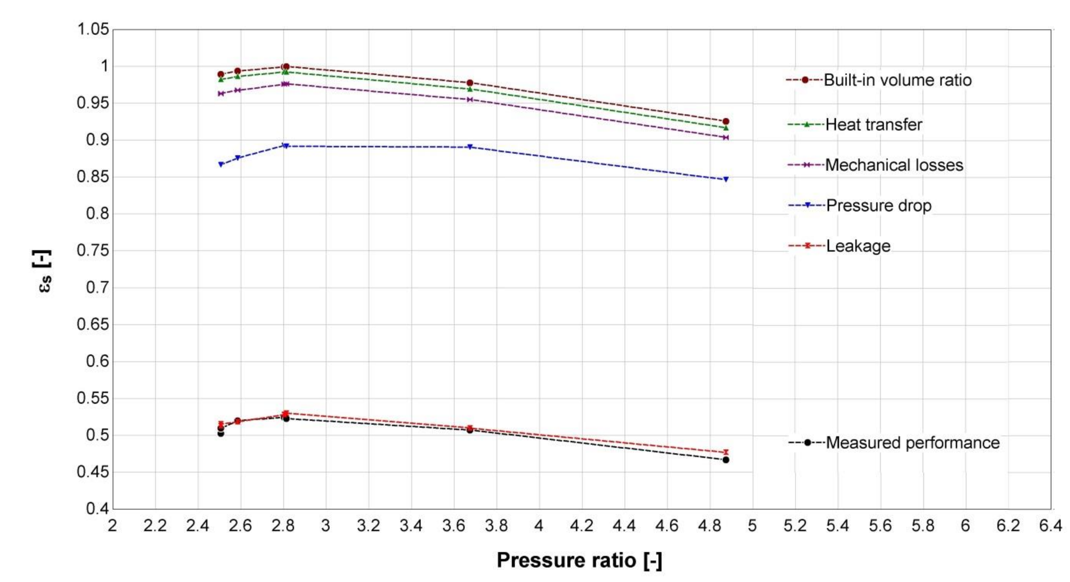

Fig. 4: Model analysis - evolution of the calculated overall isentropic effectiveness with the imposed pressure ratio for a rotational speed of 3000 RPM when taking into account the heat transfer, mechanical losses, pressure drop and leakages. The internal leakages are responsible for the major part of the performance loss.

Results of experimental campaign

Graph in fig. 5 shows the expander power output. It can be seen that the performance rises rapidly in the area where v_{ex}/v_{su} = 2.5-3.2. The expander isentropic efficiency (eq.1) is shown in figure 6. Rapid increase of efficiency can be seen in the same range of values v_{ex}/v_{su}. The highest values of expander efficiency are at v_{ex}/v_{su} = 3.2. This value approximately corresponds to the built-in volumetric expansion ratio of the expander ($r_{v,in}$=3). The efficiency falls in case that the value v_{ex}/v_{su} is higher than approximately 3.5. The points with superheating of 10-20 K are also highlighted in the graph in fig. 5 and fig 6. It can be seen that the higher superheating mean higher efficiency in all cases. However, this fact is needed to be confirmed by further measurements. Figure 7 shows the dependency of volumetric efficiency (eq.2) on pressure ratio and expander shaft speed. The value of volumetric efficiency rises with the increase of speed. This is caused by rising of the overall flow through the expander when the total leakage within the expander remains almost the same.

There is also declining trend of volumetric efficiency with increasing of pressure ratio within the expander. Higher pressure ratio leads to greater leakage of working fluid from the working chambers of the expander.

$$\eta_s = \frac{\dot{W}_{shaft}}{\dot{M}_{wf}\cdot(h_{su}-h_{ex,s})} \qquad \eta_{vol} = \frac{N_{rot,exp}\cdot c\cdot V_c}{60\cdot\dot{M}_{wf}\cdot v_{su}}$$

Eq. 1, Eq. 2: Overall thermodynamic efficiency and volumetric efficiency calculation

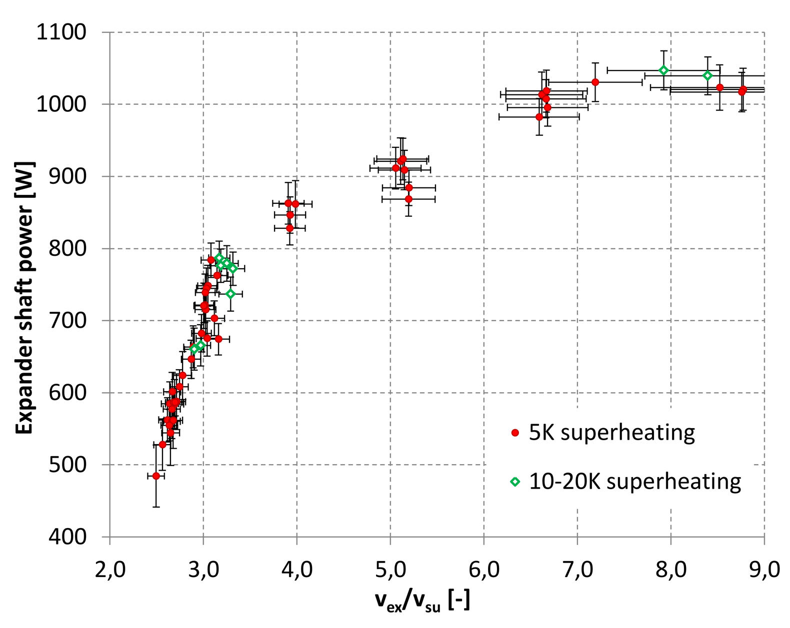

Fig. 5: Measured values of expander shaft power

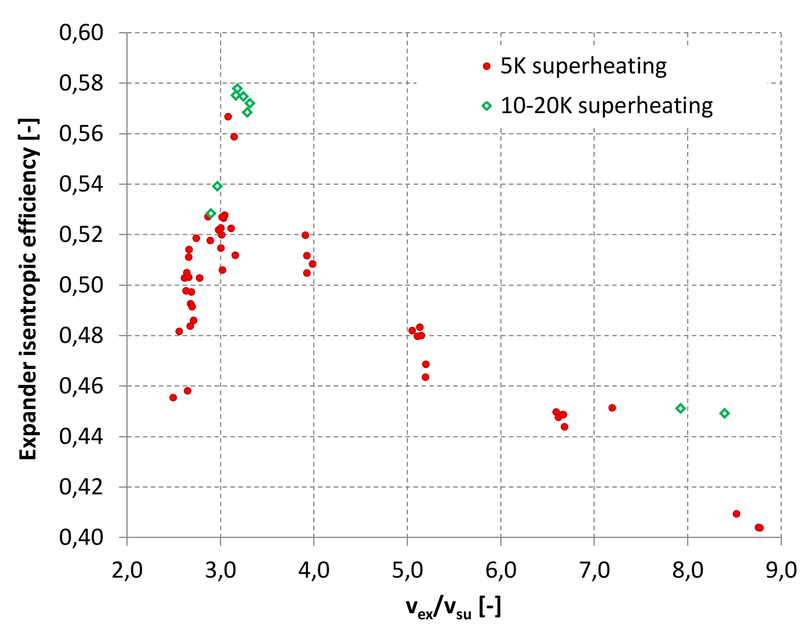

Fig. 6: Expander isentropic efficiency vs. vex/vsu

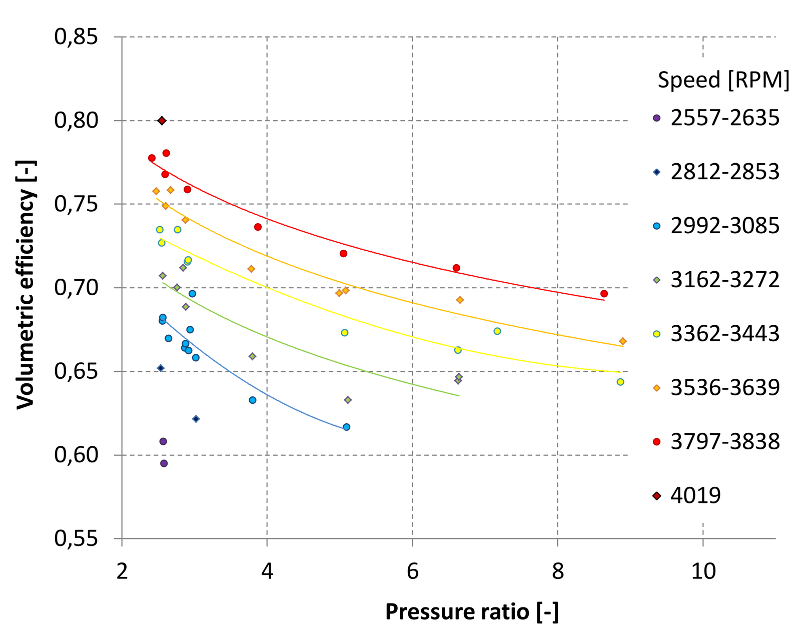

Fig. 7: Volumetric efficiency, 5K superheating

Conclusion

The vane expander reached the maximum shaft power of 1.05 kW with the isentropic efficiency 0.45. The maximum reached isentropic efficiency was 0.58 at 800 W of shaft power. It is obvious that highest efficiencies were obtained when the ratio v_{ex}/v_{su} roughly corresponds to the internal built-in volume ratio of the expander. Further attention should be paid to the optimal superheating which apparently affects the expander efficiency. Results of a semi-empirical model show good agreement between calculated and measured mass flow rate and shaft power respectively. The model shows that the internal leakage is responsible for the major part of performance loss. The supply pressure drop has also significant influence on the overall performance. Therefore, further work should be focused on investigation how to reduce these major losses.

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