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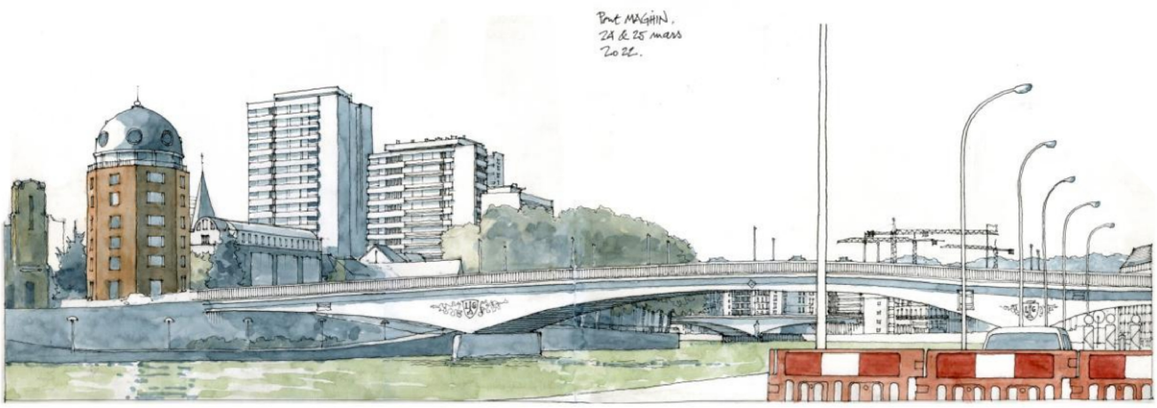

THEREOTICAL MODELLING AND EXPERIMENTAL INVESTIGATION OF A REVOLVING VANE EXPANDER

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Abstract. Theoretical and experimental studies on a Revolving Vane Expander (ReVE) are discussed in this paper. The concept and the working mechanism of this type of volumetric expander are examined and the benefits and drawbacks compared to other volumetric expander types are discussed. A brief insight into the modelling results of a simple Excel 1D-model is given. Furthermore, data of the expander measured with pressurized air are published and compared to the modelled data. Through the examination of the model and the comparison to the measurements, improvement potentials for the expander and the model are recognized.

Keywords. Volumetric Expander, Revolving Vane Expander, 1D-model, Organic Rankine Cycle

1 Introduction

Organic Rankine Cycles (ORC) are a proven technology for low-temperature waste heat recovery in the industry. [1] For applications below 10 kW_{el} volumetric expanders are most used due to their cost-effectiveness, as they are often adapted from mass-produced refrigeration compressors. One of the main disadvantages of volumetric expanders are the high frictional losses, which contribute significantly to the losses in this small power range. These frictional losses are largely caused by the high relative velocities of the rotating and non-rotating parts in contact of the expander. Furthermore, significant wear is the result. The Revolving Vane Expander (ReVE) reduces these frictional losses by reducing the relative velocities of the casing (outer cylinder) and the rotor in relation to each other. This advantage could make the ReVE a promising expander for the future in this power range. Figure 1 shows a cut through the expander. The three main parts of the expander are the vane (1), the rotor (2) and the casing, the so-called cylinder (3). The rotor is positioned eccentric to the cylinder of the expander. The vane connects the rotor and the cylinder via two bearing points but can retract and extend. The working fluid enters the expander axially via the hollow shaft and through a bore in the rotor to the working chamber (green arrows). The working fluid leaves the expander via the bore in the ejection chamber (red arrow). The expander rotates in the clockwise direction (black arrow). The working fluid enters the working chamber (green volume) under high pressure.

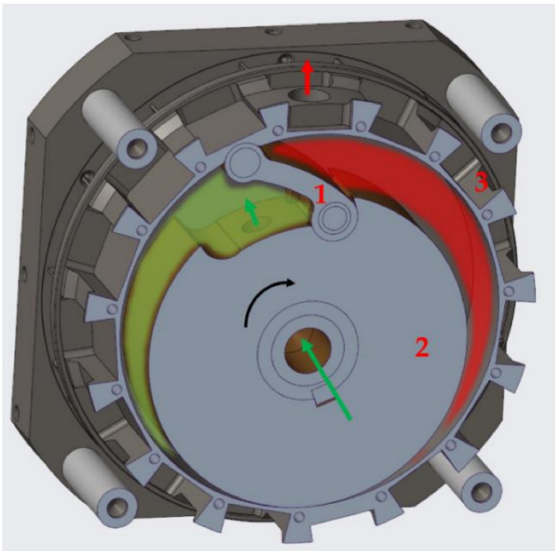

Figure 1: Cut through the Revolving Vane Expander in CAD

The high pressure causes a force on the vane which turns the entire geometry and the volume can expand in the clockwise direction due to the eccentric position of the rotor to the cylinder. At the same time, the working fluid of the previous expansion is pushed out through the outlet bore via the decreasing volume of the ejection chamber (red volume). As soon, as a full rotation has been completed, the fluid that was in the previous expansion chamber, is now in the ejection chamber and new fluid enters the working chamber over the inlet bore and the hollow shaft – the process repeats. Due to the firm connection of the rotor and the cylinder, the relative motion between these two parts is small, and therefore, friction and wear as well. Main disadvantage of the expander is the induced high forces on the bearings through the inertia of the corotating outer cylinder. Due to the eccentric position of the rotor to the cylinder, the cylinder has to accelerate and decelerate at each revolution, while the rotor remains at constant circumferential velocity. Furthermore, this type of expander requires a high level of manufacturing effort due to the many moving parts and a high level of manufacturing accuracy. Table 1 shows the basic data of the considered expander:

Table 1. Basic geometric data of the expander

Parameter	Value	Unit
Outer diameter	188	mm
Inner diameter	158	mm
Eccentricity	15	mm
Expander Length	50	mm
Swept volume	441	ccm
Inlet cross section	78.54	mm^2
Outlet cross section	314.16	mm^2

2 Modelling of the expander

This section only provides a brief insight into the results of the model. For more information on the modelling of the Revolving Vane Expander, the reader is referred to the work of Subiantoro et al., Naseri et al. and Ooi et al. [2–4] which have had a significant research contribution for this type of expander.

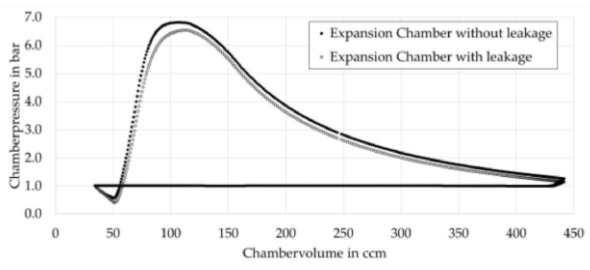

Figure 2. Modelled chamber pressure over chamber volume of the expander with and without leakage

Figure 2 shows the modelled pressure evolution in the working chamber and the ejection chamber of the expander over the chamber volume. The source pressure was set at 7.5 bar and the outlet pressure at 1 bar. The leakages in the chamber were modelled using measured values for leakage mass flow rates at different rotor positions and different chamber pressures. To implement these measurements, the expander was fixed in different positions and charged with different pressures. These measurements were used to generate a leakage mass flow rate matrix, which the model can use to determine the corresponding leakage mass flow rate for different positions and pressures. The curve in Figure 2 clearly shows that the course with leakage mass flow rate results in a lower working chamber pressure than the course without leakage. As can also be seen in the Figure, the chamber draws a vacuum, as the previously mentioned inlet opening cross-section can also be shifted to different positions in the model. Here in the modelled example, it is at 30°. By modelling the pressure in the chamber, it has already been established that a later inlet than 0° in the circumferential direction only has a negative effect on the expansion efficiency of the expander. This becomes clear, as due to the vacuum at the beginning of the expansion, no work output of the expansion chamber can be created. Even the opposite is the case. Due to the vacuum in the expansion chamber and the prevailing ambient – or even higher - pressure in the ejection chamber, a force acts against the direction of rotation on the vane.

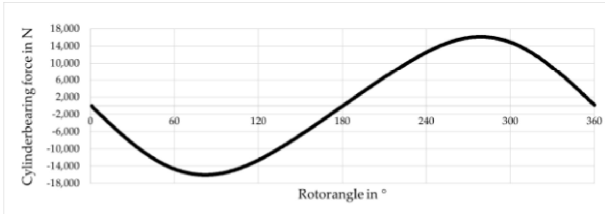

Figure 3. Cylinder bearing forces over rotor angle at 3000 rpm

Figure 3 shows the bearing load over the rotor angle. The diagram shows the load curve for 3,000 rpm. Here, the cylinder mass and geometry of the measured expander out of the CAD were used to calculate the inertia. The Figures show that the bearing load increases up to 16 kN - positive and negative. The course also shows that the maxima are at approx. 90° and 270°, the minimum at 180°. This seems logical, as the maximum eccentricity is at 180° and the maximum acceleration and deceleration of the cylinder take place before and after the maximum eccentricity. The maximum accelerations – positive and negative - and the inertia of the cylinder induce corresponding torques, which then cause forces on the cylinder bearing. With these amplitudes, it is highly questionable whether the expander could have been operated at this rotational speed at all.

3 Results and discussion

In this section, the results of the modelling are compared with the results of the measurement of the Revolving Vane Expander. The inlet position for the measurement was set at 30° and the length of the inlet opening was set at 40°. This position and width also correlate with the modelled curves just shown. The measurements were carried out with compressed air at the pressurized air facility *PDLT* at the OTH Amberg-Weiden [5].

Figure 4 shows the comparison of the mass flow rate of the measured expander with the values from the model over the rotational speed. At first glance, the courses are similar, with the curve of the modelling at 400 rpm being slightly lower than the curve from the measurement, but then rising more sharply. These deviations can best be explained by the transient effects of the inflow. The speed-dependent sliding of the inlet cross-section in the valve with the inlet opening bore in the rotor very probably results in transient effects that are not represented in the modelling. The transient effect of the constriction of the flow will be more pronounced at high rotational speeds than considered in the model. In the model, the inlet mass flow rate is determined for each degree step based on the prevailing opening cross-section, source and chamber pressure. What also comes apparent by the numbers in the Figure, which is the same for all the following Figures, is the considered rpm range. The range lasts from 400 rpm to 1,000 rpm, as for lower rpms than 400, the expander went into an area of strong strokes. Therefore, determination of mass flow rate, torque and rpm was very difficult. Rotational speeds above 1,000 rpm were not considered, as the expander began to beat strongly at these higher rpms and the test stand was not designed for these strong impacts.

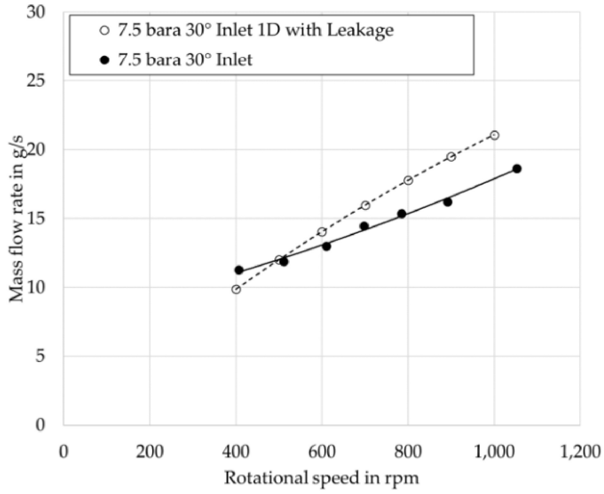

Figure 4. Modelled (open circles) and measured (full circles) mass flow rate over rotational speed

The torque distribution over the rotational speed of the expander in the model and measurement in Figure 5 shows a comparable course. Due to the higher mass flow rate after 300 rpm in the model, the pressure in the chamber of the expander is also higher and thereby the torque. Both curves show a similar slope, where the modelling shows about 2 Nm (400 rpm) to around 4 Nm (1,000 rpm) higher torque than measured. This slighter drop in torque in the modelling also correlates with the steeper increase in mass flow rate towards higher rotational speeds. For rpms lower than 300 rpm, where the modelled mass flow rate is lower than measured, the torque should also be lower. This is not the case, which implies, that the model still leaks some negative influences on the energy conversion.

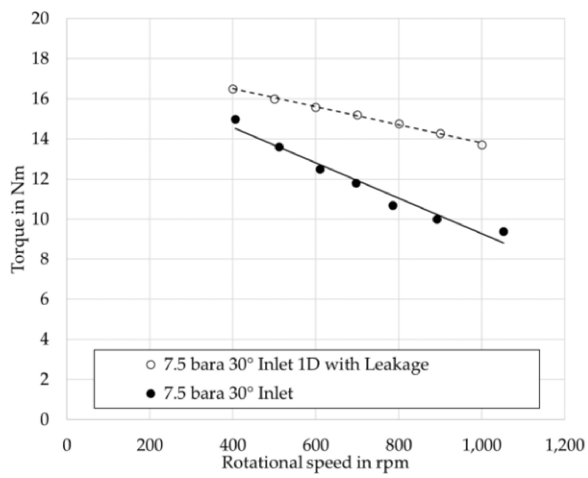

Figure 5. Modelled (open circles) and measured (full circles) torque over rotational speed

For a complete comparison, Figure 6 shows the modelled and measured isentropic expansion efficiency of the expander. The efficiency course of the two curves can be derived from the two previous Figures of the mass flow rate and the torque. What is immediately apparent is that the two curves are very similar. The curves are relatively flat over the entire rotational speed range. It is also clear that the model is consistently approx. 5 - 7 %-P above the measured expansion efficiency. Nevertheless, measured expansion efficiency is with about 45% at least competitive compared to available volumetric expander in this low power range (< 10 kW_{el}) [6].

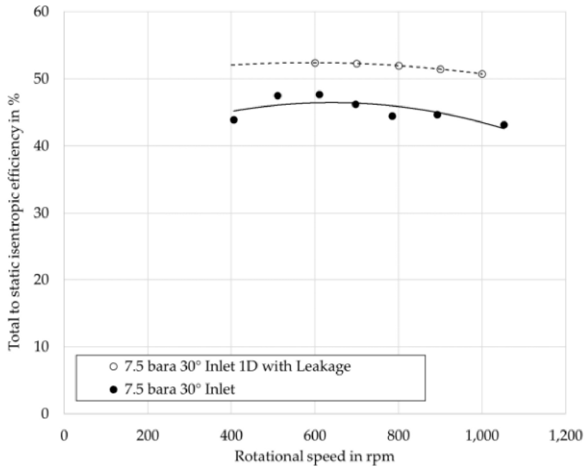

Figure 6. Modelled (open circles) and measured (full circles) isentropic expansion efficiency (ts) over rotational speed

4. Summary and conclusions

After this presentation of the Revolving Vane Expander and the comparison of the measurement results with the modelled results in this short paper, the following conclusions can be drawn. The first major drawbacks could already be identified through the simplified observation in the 1D Excel model. These are primarily the high torque amplitudes and the resulting forces on the bearings due to the inertia of the corotating outer cylinder. Furthermore, it was made clear that a later inlet position of the inlet valve after 0° only has a negative influence on the expansion efficiency of the expander. When looking at the modelling results compared to the measured results, it becomes clear that although the simple model with the empirical leakage modelling already provides a good guideline for the performance numbers, the model still needs to be further improved. A further, optimized Revolving Vane Expander is now to be designed and measured during the project. This expander should eliminate the negative points identified by the model and the measurement. Above all, the high inertia of the cylinder and the acceleration and deceleration. Thus, generate a higher power output and expansion efficiency. The 1-D model is to be further optimized on the basis of the results of the next expander. In the future it should be possible to extract the optimum design specifications from the model for certain applications and boundary conditions. Thereby it should be possible to provide corresponding highly efficient and reliable Revolving Vane Expanders for Organic Rankine Cycles.

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