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Numerical and experimental analysis of the intake and exhaust valves of a rotary expansion device for micro generation

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Abstract

The use of ORCs is growing in importance the last years, because this type of cycles permits to exploit energy sources which are characterized by low enthalpies (waste heat, low temperature geothermal, low concentration solar plants, etc.) and low installed power sizes (up to 50-100 kW). In this size range, volumetric machines are very attractive devices because they show better efficiencies than turbines. Volumetric expansion devices flow rate is not continuous as in turbines but on the contrary has a pulsating behavior and the effective flow area of the intake and exhaust ports plays an important role in determining the efficiency of such devices.

In this paper an analysis regarding the influence of the intake and exhaust valves features on the performances of a rotary expansion device for micro generation derived from a Wankel engine is presented. The analysis which is presented in this paper was carried out by means of both numerical and experimental techniques. CFD simulations of two different types of valves were performed to obtain the values of discharge coefficient and, subsequently, the results were validated at the fluid dynamic test bench. These solutions were also evaluated by testing the prototype using compressed air as working fluid under various operating conditions (pressure ratio, rotating speed). The delivered torque, the air mass flow rate and the indicated cycle were taken into account for the evaluation of the influence of the valves shape and timing on the device performances.

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Keywords: renewable energy; ORC; Wankel; volumetric expansion device; CFD;

1. Introduction

Small expansion devices are interesting for micro generation applications because they permit to exploit renewable sources such as geothermal, biomasses and solar energy. At the typical sizes of micro generation plants,

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volumetric expanders show better efficiencies than those of turbines; in fact low output powers (up to 10 kW) are covered by scroll expanders [1-8], while higher powers (25-100kW) are obtained by reciprocating expanders [9-10]. As a further matter of convenience, volumetric expanders show a greater tolerance respect to the vapor low title at the end of the expansion [11].

Amongst the variously proposed devices, in the range 10-50 kW very few examples of realizations can be found. For this sake, a modified Wankel engine was proposed in previously published work [12-16] showing some advantages which mainly rely on a low vibration magnitude level, a noticeable compactness, the capability of rotating at relatively high speeds and finally a quite long distance between suction and discharge ports which leads to a reduced heat flow between incoming and outgoing fluid. In this study the effects of both the operating conditions and the valves main fluid dynamic features on the device indicated cycle and delivered torque were experimentally analyzed by means of a properly built prototype.

2. Device characteristics

The device prototype (fig. 1) studied in this work was obtained by using the rotor, the shaft and the seals of an original Wankel engine, used in karts and ultra-light flight vehicles; the stator case on the other hand was newly designed and built.

The displacement was 316 cm^3, given by an eccentricity of 12.05 mm and an equivalent crank of 39.5 mm; the resulting volumetric compression ratio was equal to 12.7. The inflow and the outflow are controlled by rotary valves driven by pulleys connected to the engine shaft. A vernier system, located on each pulley, allowed the modification of the timing. Two intake and two exhaust valves were needed because each operative chamber completes two thermodynamic cycles during a full rotation of the rotor. In this work two different types of valves were analyzed. In both cases the admission and the recompression grades were equal respectively to 0.36 and 0.25 as a compromise between power and efficiency.

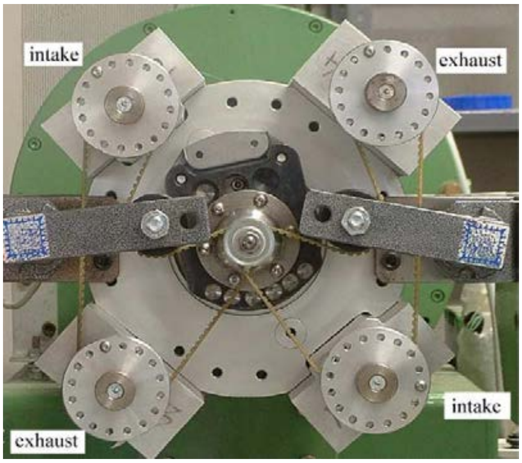

Fig. 1. Wankel expansion device.

3. Experiments

In this work the indicated cycle was chosen as the comparison parameter between the different operating conditions and valves configurations. The experimental analysis was carried out at the engine test bench, while fuelling the expander by compressed air and the experimental setup reported in fig. 2 was used. For the sake of ensuring a reliable measure of the air volumetric flow, a damping volume of 50L was employed between the flow meter and the expansion device, thus eliminating the pressure oscillations due to the device pulsating suctions. The inlet air pressure and temperature were also measured. The indicated cycle was obtained by two piezo-electric pressure probes positioned in one half of the stator case, because one thermodynamic cycle is completed during half rotation of the rotor. Two Kistler 6052 piezoelectric sensors, connected to a Kistler 4065 charge amplifier, were used to measure the indicated pressure while the angular position of shaft was measured by an angular encoder. The pressure traces were acquired with a sampling frequency of 1Ms/s per channel; the indicated cycle was obtained through a Labview program as the average of 50 cycles. The operating pressure was set to 5 bar while the rotating speed was varied from 500 rpm to 1500 rpm.

Before testing the whole expansion device, the valves discharge coefficient was measured at the fluid dynamic test bench in steady state conditions for several values of the opening grade. These experiments were performed with a pressure drop of 3 kPa, while the air mass flow rate was measured using a diaphragm built upon the requirements of the rule UNI-EN ISO 5167.

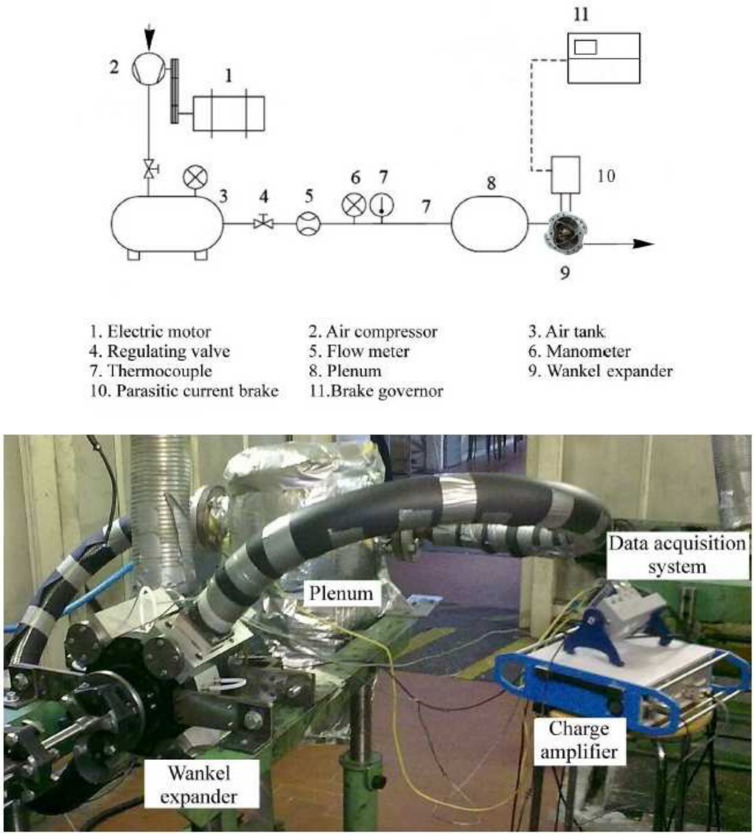

Fig. 2. Experimental apparatus.

4. Valves analysis

The experiments were carried out using two different types of valves to analyze the effects of the discharge coefficient on the indicated cycle. Three valves combinations employed during this analysis [tab. 1].

The first valve model, named a (fig.3, left), was characterized by an internal fluid volume with cylindrical shape. The discharge coefficient of this model was evaluated through CFD 3D simulations, considering several opening grades of the valve. The analyses were carried out in steady state, taking into account fully turbulent motion (k-ε model) and adiabatic conditions. These results were then validated at fluid dynamic test bench showing a sufficiently satisfactory agreement with CFD simulations (fig. 4). The discharge coefficient of the intake valve was higher than that of exhaust valve; on the other hand the intake valve showed the maximum value only at full opening featuring a smaller angular opening than the exhaust one (fig. 5). The higher discharge coefficient of the valve model b was explained by considering the absence of vortices which were responsible of the reduction of the discharge coefficient of the valve model a (fig. 6).

Table 1.Valves combinations employed in the present analysis.

Configurations	Inlet valve	Exhaust valve
1	a	a
2	b	a
3	b	b

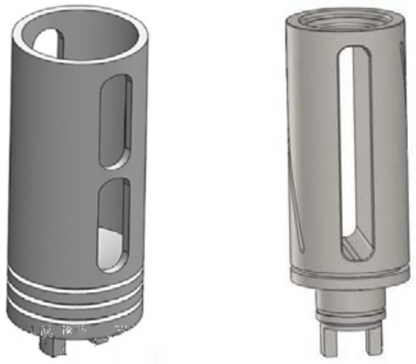

Fig. 3. Model of valve "a" (left) and "b" (right).

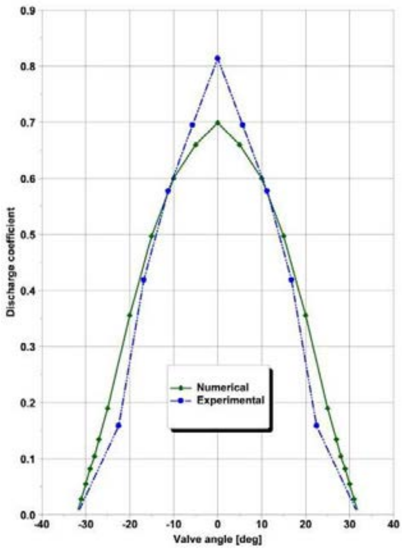

Fig. 4. Comparison between numerical and experimental intake valve discharge coefficient.

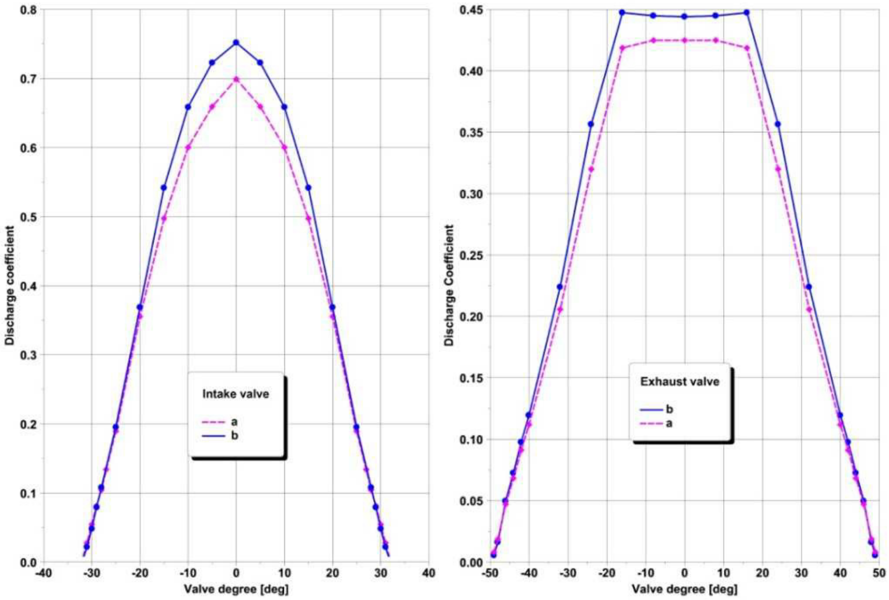

Fig. 5. Comparison of the discharge coefficient of the two valve types.

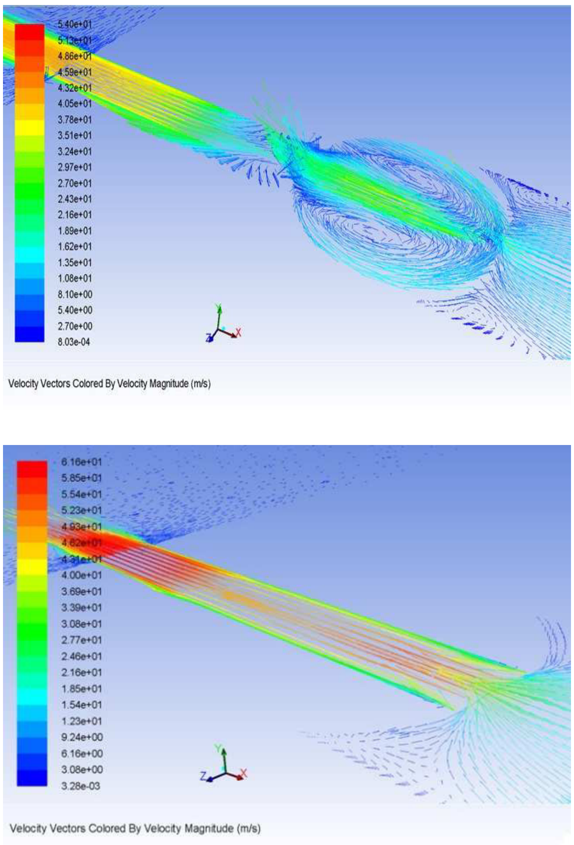

Fig. 6. Flow inside the valve cavity (model “a” on the top, model “b” on the bottom).

5. Analyzed cases

The results of the experimental campaign are summarized in this section under the general criterion of separating the effects of the various variables with the aim of presenting more generally valid results. All the indicated cycles were normalized by the intake pressure to allow a clearer comparison.

5.1. Effect of speed variation

As expected, the effect of the rotary speed is particularly relevant respect to the pressure losses across the valves. In particular, at 500 rpm (fig. 7), the introduction phase happened at nearly constant pressure while the increase in the rotary speed to 1000 and 1500 rpm reduced the in-chamber pressure during this phase. The global effect is therefore equivalent to the reduction of the introduction grade, thus leading to the decrease of the cycle area. The increase in the pressure losses was however partly compensated by the decrease of the residual pressure at the end of the expansion, thus limiting the decrease of the global efficiency of the device. This effect however practically disappeared when using the valves model “b” because of their higher discharge coefficient as discussed in the previous paragraph. On the other hand the rotary speed increase limited the entity of the internal leakages between one chamber and the others. This fact was particularly evident when considering the recompression, where the increase of the rotary speed had the effect of increasing the in-chamber pressure at the end of the recompression phase. The effect of the rotary speed increase is qualitatively the same whatever the valves configuration.

5.2. Valves shape effect

Different valves configurations (tab. 1) were analysed at constant speed (fig. 8) in order to investigate the effects of the valves shape. At 500 rpm, as expected, no particular differences in the introduction phase were observed. As the rotary speed was increased to 1000 and 1500 rpm, of course greater differences could be observed between configurations 1 and 2, while configurations 2 and 3 were equivalent from this point of view because the intake valves were the same. With a substantial analogy to the previous paragraph, using the valves with a better discharge coefficient had the same effect of increasing the introduction grade, thus increasing the indicated work but also the residual pressure at the end of the expansion phase.

The diagrams reported in fig. 8 also show another not negligible effect, that is to say the sudden decrease in the indicated pressure at nearly half of the expansion stroke. This effect may be explained as follows: when the rotor apex seal passed over the vane of the exhaust port, the observed chamber and the preceding were in communication so that a small leakage of operating fluid occurred (fig. 9). It is useful to note that during this period in the observed chamber the fluid was still expanding, while in the preceding one the fluid was being recompressed. Since the tolerance of the b-type exhaust valves was lower than the a-type (some hundredths of mm), the recompression pressure was higher in configuration 3 and due to the reduced leakage no particularly evident decrease in the indicated pressure was observed. As the rotating speed was increased, there was less time for the leakage to occur and this phenomenon was not so much evident than at low rotating speed for every studied configuration.

If we restrain the analysis only to the exhaust phase, the effects of the valves shape variation are more clearly observable. Though the pressure at the end of the expansion was higher with the configuration 3, due to the phenomena described in the previous lines, the in-chamber pressure dropped to the atmospheric value in a shorter time than in the other configurations. As a result, taking the configuration 1 as reference, the indicated work that had to be spent during the exhaust phase at 1500 rpm was reduced by 11.4% with configuration 2 and by 27% with the configuration 3 (fig. 10).

The influence of the exhaust valve timing was also carried out in the same way. When the opening advance was increased to 4 and to 8 degrees, at 1500 rpm the indicated work done was decreased respectively by 10.2 and 33.6% respect to the case with no advance (fig. 11) because of the lower average pressure during the exhaust (fig. 12). A further increase did not provide any advantage (here not reported for the sake of brevity). At lower rotating speeds the differences were obviously less noticeable. As a result, configuration 3 obviously provided higher values of delivered torque (fig. 13).

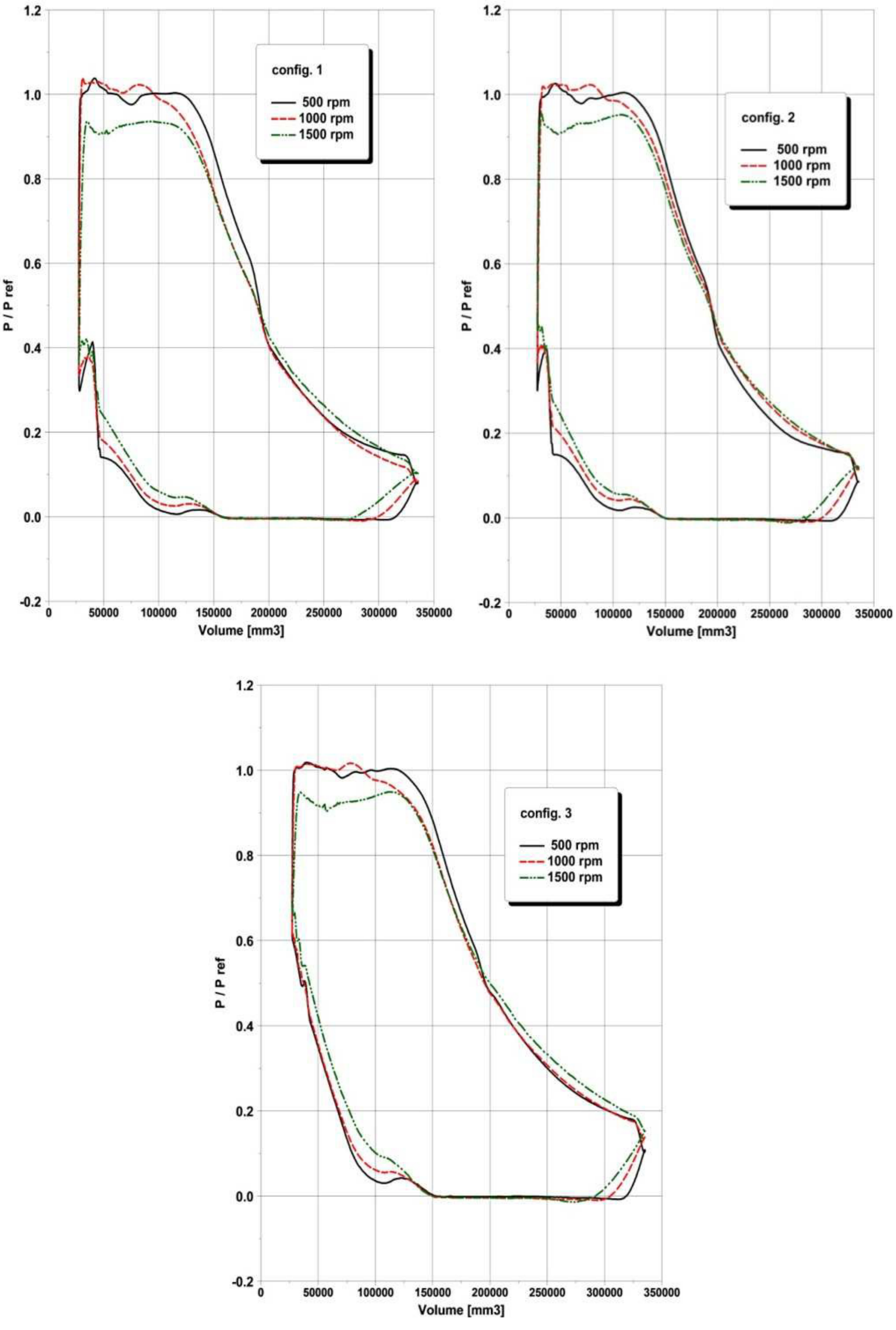

Fig. 7. Effect of the rotary speed increase for the various valves configurations.

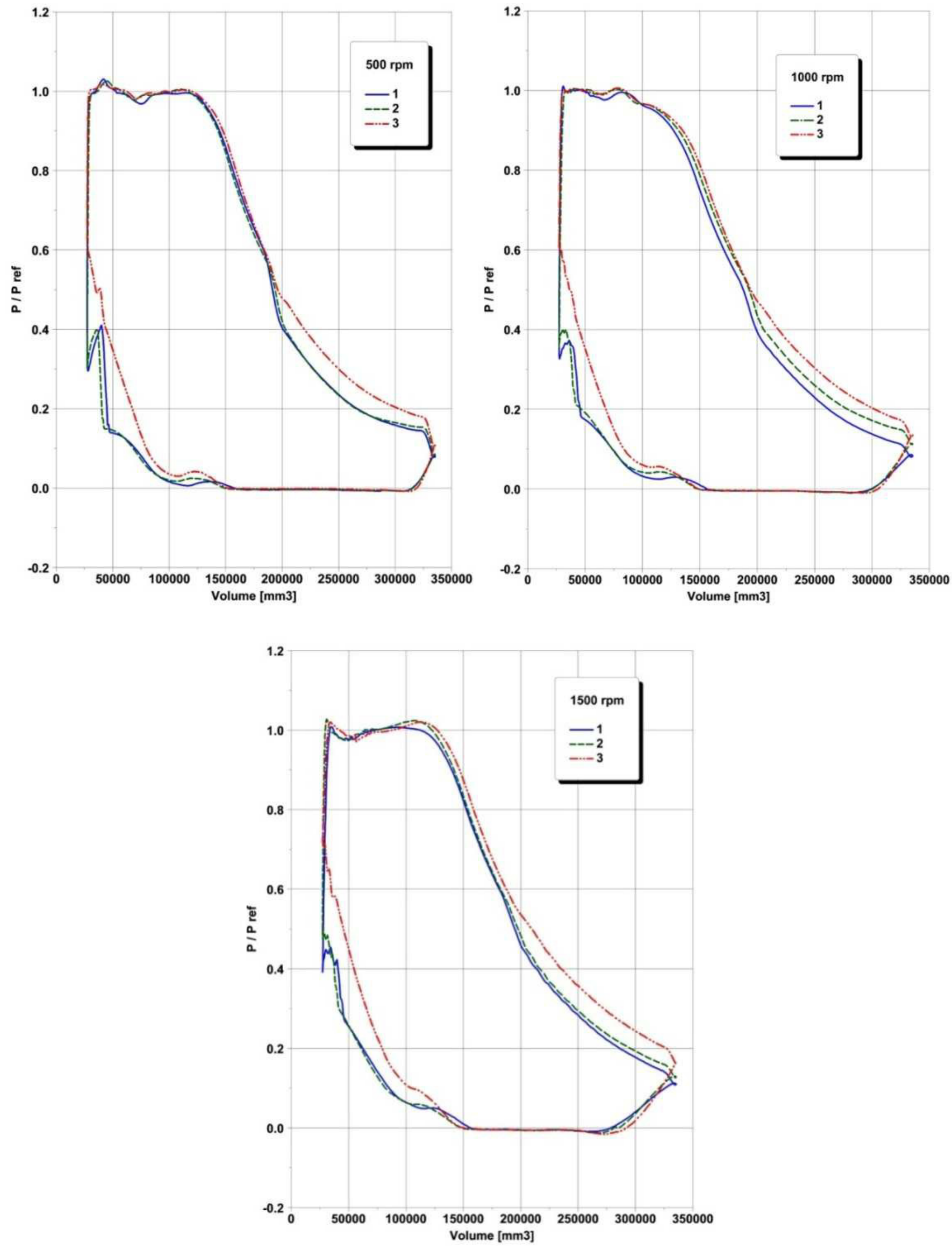

Fig. 8. Effect of the valves configuration at constant speed.

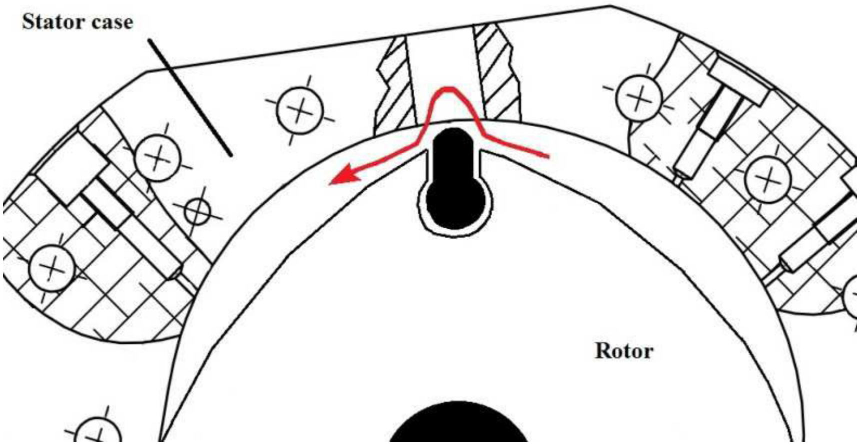

Fig. 9. Leakage between two chambers when the apex seal is in correspondence with a valve port.

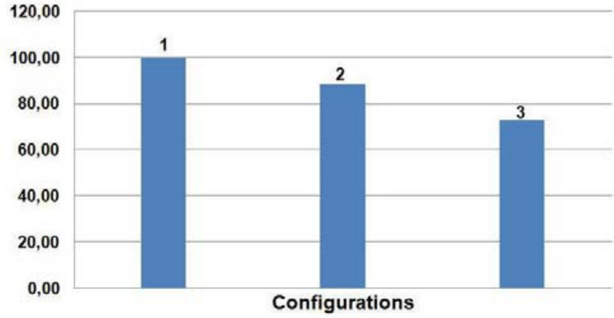

Fig. 10. Percentage of work done during exhaust phase at 1500 rpm for different configurations.

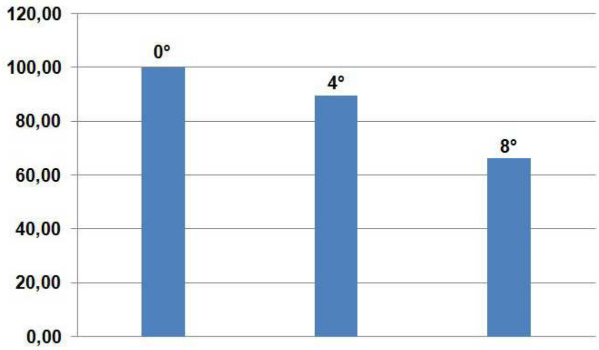

Fig. 11. Percentage of work done during exhaust phase at 1500 rpm with configuration 3 as a function of the exhaust valve opening advance.

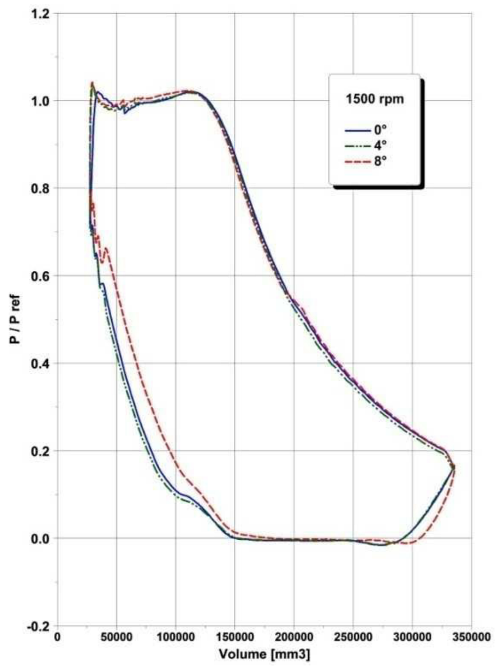

Fig. 12. Effects due to different values of the exhaust valve advance with configuration 3.

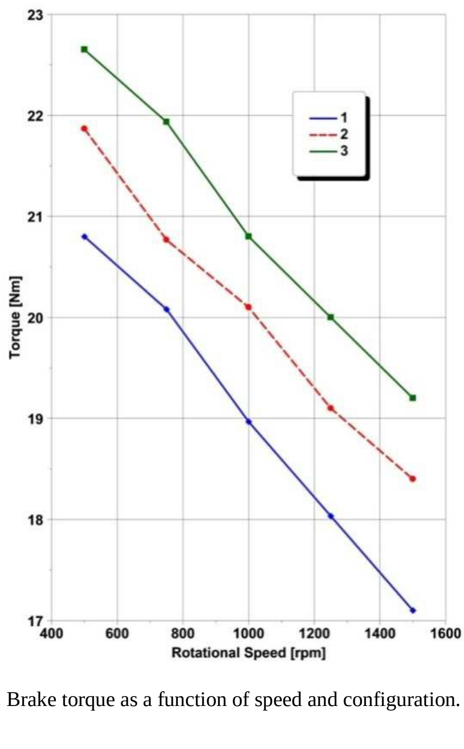

Fig. 13. Brake torque as a function of speed and configuration.

6. Conclusions

In this paper a rotary expansion device derived from a Wankel engine was tested using compressed air as working fluid. Particular attention was dedicated to the fluid dynamic phenomena happening across the introduction and exhaust valves, which were characterized both numerically with the CFD code Fluent and experimentally at the fluid dynamic test bench.

The increase of the device rotating speed obviously intensified the pressure drops across the valves, which had the effect not only of decreasing the work done or spent during the introduction and the exhaust phase, but also of reducing the effective introduction grade. The device efficiency reduction was therefore partially compensated by the more complete expansion of the working fluid.

As for the valves shape variation, the analyses performed at the engine test bench substantially confirmed the results that were obtained in steady state conditions, i.e. an improvement in the valves discharge coefficient has direct consequences on the device behavior in the real operating conditions. At constant speed the employment of the valves of the b-type had the result of reducing the pumping work done during the exhaust phase. A not negligible role was also played by the valves fabrication tolerance: reducing the leakages across the valves means increasing the recompression pressure for a given geometrical recompression grade and this effect has important consequences on the entity of the leakages between two adjacent chambers. In conclusion this analysis confirmed that the shape and the fabrication precision of the valves of such type of devices are critical points for both a correct operation and a high efficiency, due to the fact that one requirement of this kind of machines is the capability of rotate in a relatively wide range of speeds.

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