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Experimental and numerical analysis of the valve timing effects on the performances of a small volumetric rotary expansion device

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Abstract

Single stage expansion devices are currently studied for small scale size power plant, often in combination with Organic Rankine Cycles for the employment of solar, geothermal, biomass or waste heat energies.

A volumetric rotary single-stage expander was chosen in this study as expansion device for such type of plants. Its main characteristics and performances are discussed as a function of both the working conditions (fluid type, inlet temperature) and the working parameters (rotating speed, admission and recompression grades, valves advance).

These analyses were carried out with numerical and experimental techniques. The analysis of the effects of the working conditions on the expander performances was carried out through a numerical model created with the simulation tool AMESim. At the same time, a prototype was built and experimented with compressed air to validate the model used by means of air mass flow rate, torque and indicated cycle. This way the isentropic and mechanical efficiency are discussed.

The validation of the model was carried out by comparison with the experimental data collected at the engine test bench by operating the engine with compressed air. The indicated cycle, the air mass flow rate and the delivered torque were used as parameters of comparison. Moreover an experimental analysis at the fluid dynamic bench was carried out to validate the numerical 3D CFD model of the valves.

The part load performances of this expansion device were studied by hypothesizing different control strategies and comparing them in terms of efficiency reduction respect to the design point. The influence of valves advance was also discussed.

Keywords: renewable energy; ORC; micro-CHP; Wankel; volumetric expansion device; renewables; evacuated collectors

1. Introduction

Distributed energy generation actractivity is rapidly growing due both to the increased importance of issues such as primary energy consumption, climate change, environmental contamination and to their actractivity as regard of independence from the main grid, stability of energy prices, possibility of reuse of wastes and low-cost electricity generation.

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Small expansion devices could be of relevant interest for the realization of combined heat and power (CHP) distributed generation using a wide variety of energy forms: geothermal, biomasses, solar energy. Moreover the recovery of exhaust heat, for example from large reciprocating internal combustion engines, are matter of research as well.

Several types of expansion devices are currently proposed depending on the size of the plant: the smaller sizes are covered by Scroll expansion machines (up to 10 kW) [1–8], while reciprocating expanders are the object of research for relatively larger plants (25-100 kW) [9,10].

Rotary engines are in effects quite actractive from the point of view of smoothness, low vibration running and compactness, while expansion devices realization from reciprocating engines is made relatively easy by the large availability of the main components that are built in a very large number and at a very low cost by the automotive industry.

This study is focused on the analysis of the fluid-dynamic behavior of a rotary expansion device built on the basis of a Wankel engine [11,12] pursuing the idea of combining compactness, low vibration and noise operations and low cost construction. The Wankel engine is practically the only type of rotary engine that underwent an industrial development for automotive or ultralight aeroplanes purpose, altough the number of realizations is certainly of much minor importance than the reciprocating engine.

This machine is addressed to electrical generation in the size 10-50 kW depending on the operating condition and on the working fluid. Respect to the reciprocating expander, this kind of device offers a higher compactness and less vibrations. At the same time higher rotational speeds may be reached to the more lightweight mechanism and to the absence of parts in alternate motion.

Some first works of analysis were already published showing the potentialities of this device and a numerical study by means of a numerical model [13–15]. This analysis may be therefore considered as the continuation of this study that is now carried out by considering a real two-phase fluid instead of a perfect gas and going toward the evaluation of the control strategy of the device as well as the investigation of the influence of the admission and recompression grades on the engine performances. Opening advance and closing delay of the valves were as well considered as matter of interest.

Nomenclature

b Rotor axial depth (m)
e Eccentricity (mm)
N Rotating speed (rpm)
p Pressure (bar)
R Rotor radius (m)
T Temperature (K or ^oC)
V Volume (cm^3)

Subscripts
ad admission
d dead
ex exhaust
is isentropic
sat saturation
th thermal cycle
ud unit displacement

Greek
η Efficiency
γ Recompression grade (deg)
μ Dead space grade (deg)
σ Admission grade (deg)

2. Device main features and parameters

The proposed device, from the point of view of the limit cycle, has the same features of a conventional, reciprocating engine. The limit cycle of a volumetric expansion device is depicted in Fig. 1 and may be considered to be composed by the following phases: constant pressure admission (1-2), expansion (2-3), constant volume discharge (3-4), constant pressure discharge (4-5), recompression (5-6) and finally constant volume admission (6-1).

In both the cases of a reciprocating and a rotary device, the cycle is the same and may be characterized by several parameters, namely:

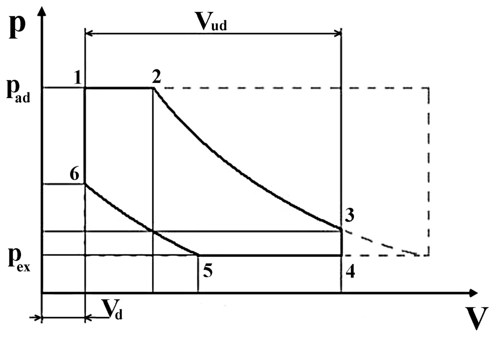

Fig. 1. Volumetric expansion machine cycle on the $p - V$ thermodynamic plane

- Dead space grade μ

$$\mu = \frac{V_d}{V_{ud}} = \frac{V_1}{V_3 - V_1} \tag{1}$$

- Admission grade σ

$$\sigma = \frac{V_2 - V_1}{V_{ud}} \tag{2}$$

- Expansion grade ε

$$\epsilon = \frac{V_3}{V_2} = \frac{1 + \mu}{\sigma + \mu} \tag{3}$$

- Recompression grade γ

$$\gamma = \frac{V_5 - V_6}{V_{ud}} \tag{4}$$

2.1. Prototype construction

The calibration and the validation of the numerical model which has been used in this work were carried out by comparing the results obtained with the experimental measures done at both the fluid dynamic test bench (for what concerns the discharge coefficient of the valves) and the engine test bench (performances, indicated cycle).

The tested prototype was built by modifying an internal combustion Wankel engine, that is commonly used in ultra-light flying vehicles and karts. For the construction of the prototype, only the rotating parts of the original engine have been used, that is to say the shaft, the bearings, the rotor and the seals. Conversely, the statoric case of the engine was newly designed and built. The displacement of the expansion device thus corresponds to the displacement of the engine, since it depends on the eccentricity, the radius and the axial depth of the rotor:

$$V_{ud} = 3 \cdot \sqrt{3} \cdot R \cdot e \cdot b \tag{5}$$

It was pointed out in literature that the geometrical compression ratio λ has a direct influence on the isentropic efficiency of the machine. Consequently, in order to increase the value of λ as possible, a rotor without the typical cavities of the combustion chambers was used. The resulting compression ratio is equal to 12.7, while the displacement is equal to $V_d = 316\,cm^3$. The value of the compression ratio is not very high if compared with reciprocating devices [9,10], however, as reported in the following paragraph, is more than sufficient when it is operated with organic fluids and relatively limited pressure ratios.

The transformation from Internal Combustion Engine (ICE) required the use of valves to regulate the inflow and the outflow of the working fluid. In this prototype the valves used were rotary; these valves are actuated by the engine shaft by means of a toothed belt and pulleys. It was necessary to introduce two intake and two exhaust valves since a full rotation of the rotor corresponds to two thermodynamic cycles and each chamber has two suction and two exhaust phases in the same period.

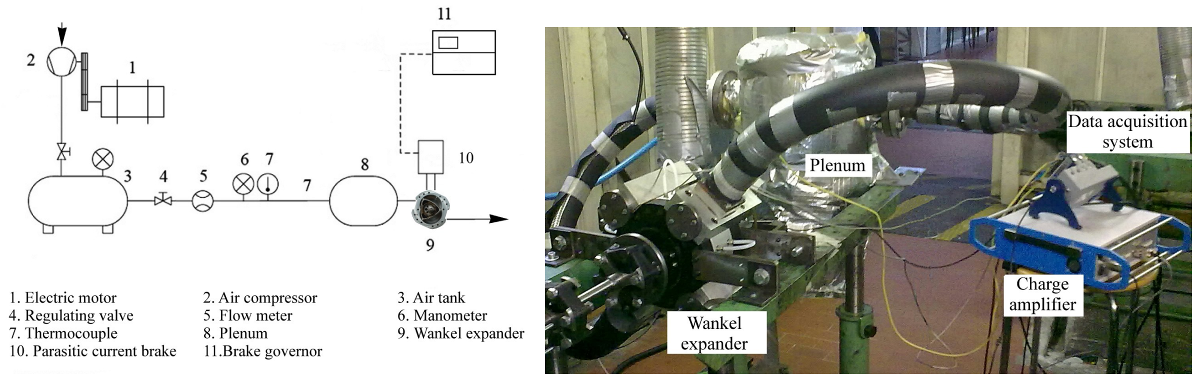

Fig. 2. Test fixture

3. Experiments

In this work the delivered torque, the working fluid mas flow rate and the indicated cycle were chosen as key parameters to calibrate and the validate the numerical model. The expansion device was therefore tested at the engine test bench. Compressed air was used during the experimental campaign; however experiments using steam as working fluid are to be done by using the experimental facility of the C.R.I.B.E. [16] and these results will be the subject of a future paper.

The results of the experiments are reported during the description of the simulations, in order to make a comparison between measured and simulated data when the validity of the numerical model is discussed. The testing equipment, whose schematic representation is shown in Fig. 2, allowed the measure of the air mass flow rate, the torque and the air consumpion of the prototype. A damping volume of $50\,L$ was put in between the flow meter and the expansion device in order to reduce the influence on the mass flow rate measure of the pressure oscillations due to pulsating suctions of the machine.

The acquisition of the indicated cycle required the mounting of two piezo-electric pressure probes on one half of the statoric case, since each chamber completes a single thermodynamic cycle during half rotation of the rotor. In fact, while in the reciprocating machines the head always faces the cylinder, in the Wankel engine the measurement is done on the same rotor side.

The indicated pressure was therefore measured by two Kistler 6052 piezoelectric sensors placed on the stator casing, while the angular position of shaft was measured by an angular encoder. The two piezoelectric transducers were connected to a Kistler 4065 charge amplifier.

All the signals were acquired with a sampling frequency of 10^6 samples per second by a compact NI-DAQ acquisition system. The signals were processed by a Labview program. Each indicated cycle was obtained as the average of 50 cycles to reduce the effects of the cyclic dispersion.

During the test the delivered torque and the air mass flow rate were also measured. The experiments were performed using inlet pressure values ranging from 3 to 5 bar, while the rotating speed was varied from 500 to 1500 rpm. The discharge coefficient of the valves was not only simulated but also measured with a fluid dynamic test bench. The measurements were carried out at the steady state under a pressure drop of $3\,kPa$. The air mass flow rate was measured through a calibrated diaphragm built upon the requirements of the rule UNI-EN ISO 5167 [18].

4. Numerical analysis

In this work a numerical analysis is presented for the evaluation of the performances of the Wankel expansion device when it is used with different types of fluids, operating conditions and controlling strategies. As regard of the investigated fluids, both dry and wet fluids were taken into account, while in any case (saturated or superheated) a subcritical pressure was used. In this analysis only a single stage expansion machine was considered, since the double stage solution was considered better suited for higher temperature drops. As regard of the operating conditions, various rotating speeds ranging from 500 to 3000 *rpm* were considered. Operation at variable admission and recompression grades were also considered, as well as the regulation by throttle valve.

The results were discussed in terms of both the isentropic efficiency and the thermal cycle efficiency. For the calculation of the thermal efficiency of the cycle, it is useful to make reference to Fig. 3. In this analysis the regeneration was not taken into account because, as written in the following paragraphs, it was hypothesized to use the waste heat for cogeneration purposes. The efficiency therefore was simply calculated as:

$$\eta_{th} = \frac{\dot{W}}{h_2 - h_1} \tag{6}$$

4.1. Modeling

The numerical analysis was carried out using a numerical model built with the simulation tools AMESim v.12.0. The simulation tool AMESim provide a wide variety of libraries including the models of the most commonly used pneumatical, electrical, hydraulic, mechanical and signal communication devices. Many other libraries, here not listed for the sake of brevity, are present as well. Among these, within the two-phase library are present many devices that were suited to model the volumetric machines operating with vapors.

A numerical model of the Wankel expansion machine was already presented in a previously published paper [13]. In this case however the elements of the two-phase library were used to evaluate the performances of this device when it is operated with organic fluids. The model used in this work simulates the behavior of the new prototype shown at paragraph 2.1, having the geometrical features above mentioned and in the following lines its features will be briefly recalled. The model simulates the in-chamber pressure as a function of the crank angle. The volume variation of the chamber is governed by the crank-conrod model, whose mathematical formulation was properly modified using the AMESet utility accordingly with the variation of volume of an epitrocoid.

The model takes obviously into account the effect of the inflow and the outflow of the working fluid by means of submodels that simulate the admission and exhaust valves. In the AMESim numerical model, the valves were modelled as orifices with constant area and variable discharge coefficient. These submodels naturally needed to be calibrated by imposing the discharge coefficient as a function of the opening angle.

The discharge coefficient was evaluated through CFD 3D simulations performed with the code Fluent 6.0 in steady state conditions and regarding only the valve and its housing (Fig. 4-a and b). These simulations were performed at several opening grades of the valves taking into account fully turbulent motion (k-ϵ model) and adiabatic conditions. The comparison between numerical and experimental results is shown in Fig. 4-c. The maximum difference between the numerical and the experimental results was lower than 10%; the results of the simulations performed by applying such a variation in the discharge coefficient showed that the final efficiency of the expansion device may be determined with an uncertainty of about 5%.

The numerical model was at first used to simulate the prototype operated with compressed air to evaluate the correspondence between the numerical and the experimental results. The results of the comparison between simulations and experiments is shown in Fig. 6-a and b. The prediction of the indicated pressure was retained sufficiently effective and, since no other submodel was involved in the simulations than the one regulating the evolution of the fluid pressure as a function of the volume variation, the numerical model used was considered predictive.

The whole process of calibration and validation of the numerical models required the exchange of informations between the various phases of the presented work. For the sake of clarity, the diagram of Fig. 5 briefly summarizes this process.

4.2. Analysed cases

As regard of the working conditions, a fixed condensation temperature was considered, equal to 80 °C, in order to take into account the possibility of cogeneration (for heating or cooling as well). The investigated saturation temperature range was 100-120 °C, as usually considered in other published works, since it allows the employment of waste heat or low-concentration solar energy.

In the following lines the results will be shown for a wet and a dry fluid, namely the R152a and the R600a, for which the main features were obtained by the NIST Webbook [17].

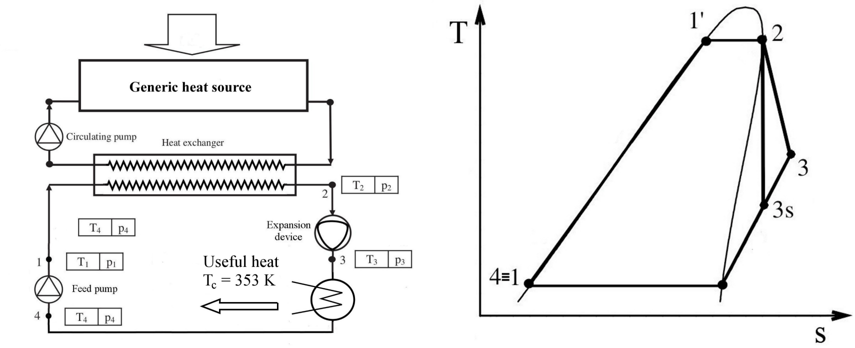

Fig. 3. Schematic of the plant (left) and $T - s$ diagram of the cycle (right)

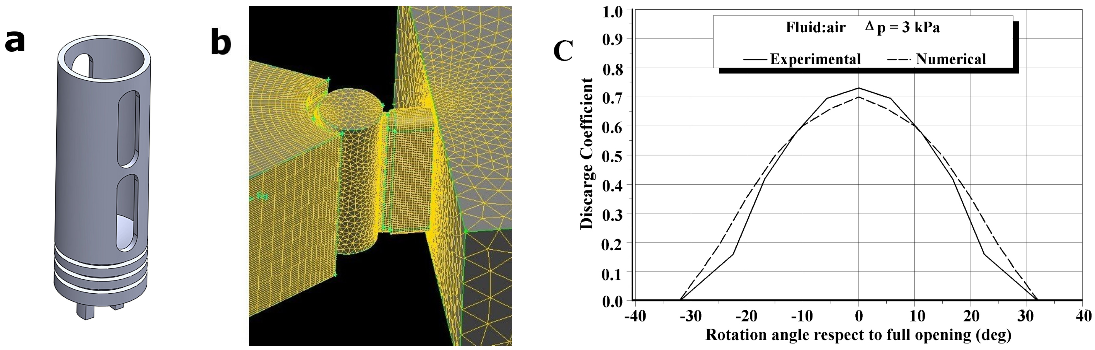

Fig. 4. CAD model of the admission valve (a), mesh (b) and comparison between experimental and measured discharge coeffcient (c)

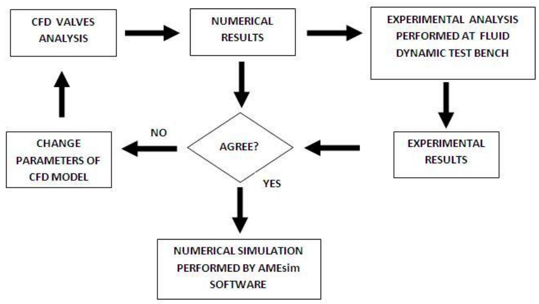

Fig. 5. Flow-chart of the calibration/validation process

4.3. *Variation of σ and γ influence*

One regulation that is commonly employed in the volumetric expansion devices is the variation of the admission grade σ. Usually in steam engines the reduction of σ is accompanied by an increase in the isentropic efficiency of the machine since the expansion is more complete and the residual pressure at point 3 is closer to the receiver pressure.

This may not be the case for a volumetric expansion device operated with ORCs, since the temperature and hence the pressure drops are quite lower. It follows (Fig. 7-a) that if the admission grade is too low (curve **a**), the cycle is overexpanded with an obvious work loss. On the other hand, high values of the admission grade, as widely recognized, lead to an underexpanded cycle (curve **c**).

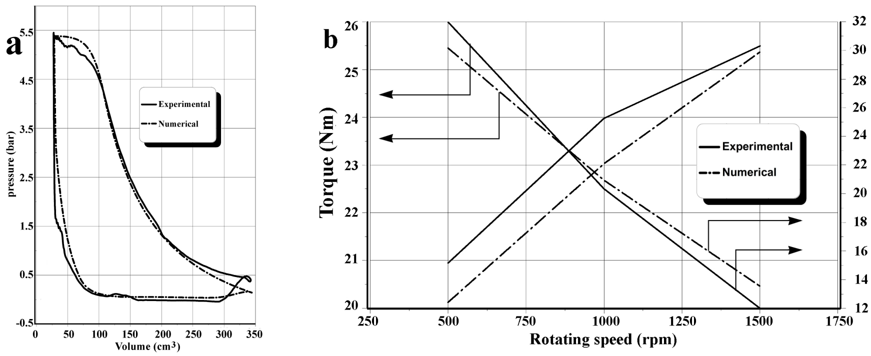

Fig. 6. Comparison between experiments and simulations in terms of indicated pressure (a) and torque and mass flow rate curves (b)

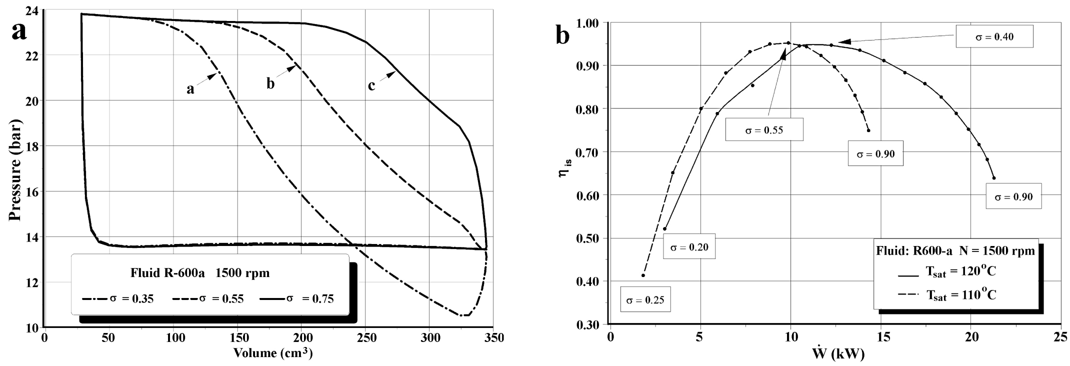

Fig. 7. Influence of the admission grade σ on the indicated cycle(a) and on the isentropic efficiency as a function of the saturation temperature (b)

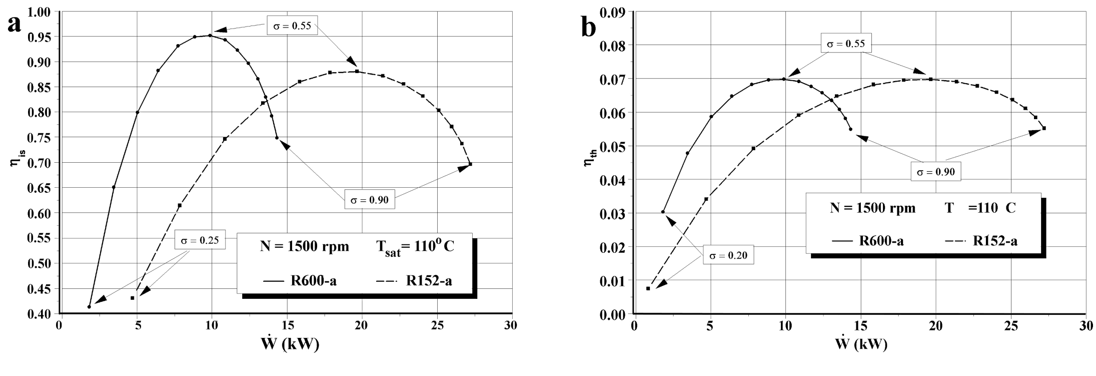

Fig. 8. Comparison between the R600a and the R152a in terms of isentropic efficiency (a) and cycle thermal efficiency (b)

Thus an optimal value of σ may be found (curve **b**) and in the cases of both the R600a and R152 with $T_{sat} = 110\,°C$ this value is around 0.55 (Fig. 8). The value of the optimal σ depends more upon the saturation temperature than the working fluid. Fig. 7-b, as an example, shows that the optimal value of σ decreases with the increase in the saturation temperature, as expected. In facts a lower value of σ is needed to get a complete expansion.

As regard of the recompression grade γ, the simulations showed that a small improvement may be attained by choosing the proper value, however the influence of this parameter on both the delivered power and the thermal efficiency of the cycle was much less important than the influence of σ. Fig. 9, as an example, reports the effects when using the R600a as fluid, with $T_{sat} = 120\,°C$.

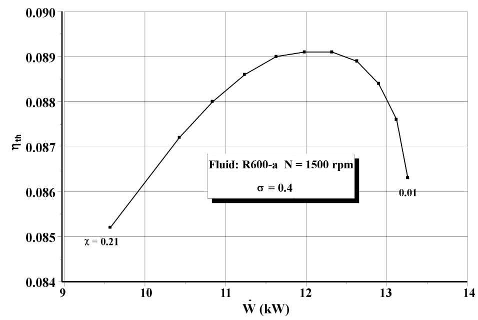

Fig. 9. Influence of the recompression grade γ on the thermal cycle efficiency

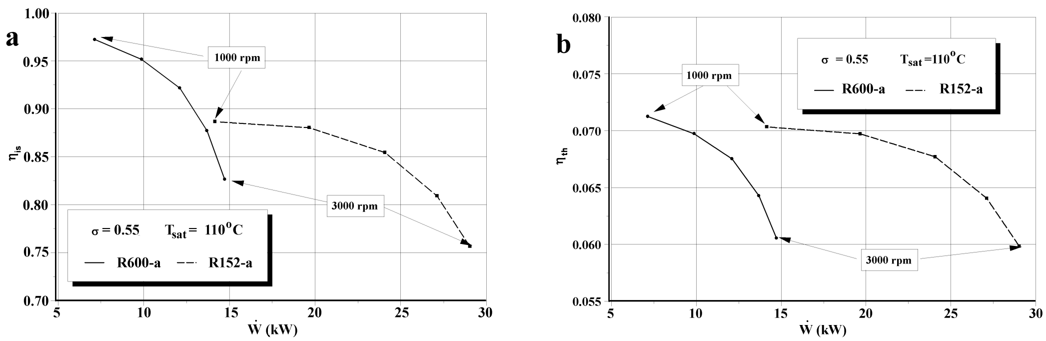

Fig. 10. Comparison between the R600a and the R152a in terms of isentropic efficiency (a) and cycle thermal efficiency (b)

4.4. Variable speed operation

The model was also used to simulate the regulation of the expansion device by variation of the rotating speed. Fig. 10-a and b report the behavior of the isentropic efficiency and the cycle thermal efficiency as a function of the delivered power for rotating speeds in the range 1000-3000 *rpm*.

As regard of the best choice about the admission grade σ to be applied when the regulation is performed through the variation of *N*, Fig. 11-a shows that higher values of σ are less sensitive to the variation of *N* as regard of the cycle efficiency.

This is due to the pressure losses across the intake valve, whose effect is a decrease of the fluid pressure inside the chamber at the end of the admission phase. Besides the obvious consideration that the pressure losses decrease the area comprised within the cycle, thus leading to a decrease in the delivered work, it is to be pointed out that, depending on the chosen value of σ, the cycle results might be overexpanded (Fig. 11-b) and a part of it is travelled counterclockwise thus leading to a further loss.

These regulation strategies were compared with the regulation through a throttling valve, that is the simplest way to regulate an expansion device. Fig. 12 confirms that this type of regulation is to be operated on machines that are expected to run at the design point for the most part of their life.

4.5. Timing advances effect

An investigation was carried out on the admission and recompression valve advances as a function of the rotating speed and the admission grade. As expected the most visible effect is at the highest rotating speeds, namely 3000 *rpm*, and with the highest values of σ. This mainly due to the shorter time available for both the admission and the exhaust processes (influence of *N*) and to the higher mass to be introduced (influence of σ). Fig. 13 shows that the admission valve advance giving the best performance is variable from zero (no advance) to 7-8°, while for the exhaust

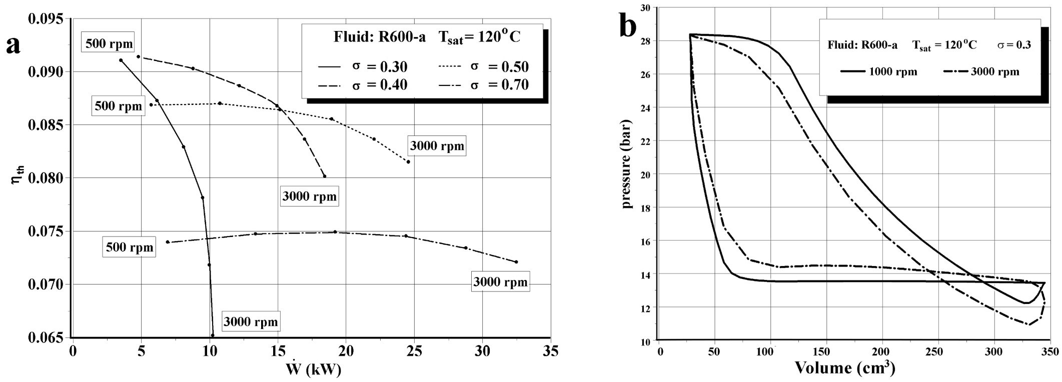

Fig. 11. Delivered power as a function of both the admission grade σ and the rotating speed N (a) and effect of the increase of the rotating speed on the indicated cycle (b)

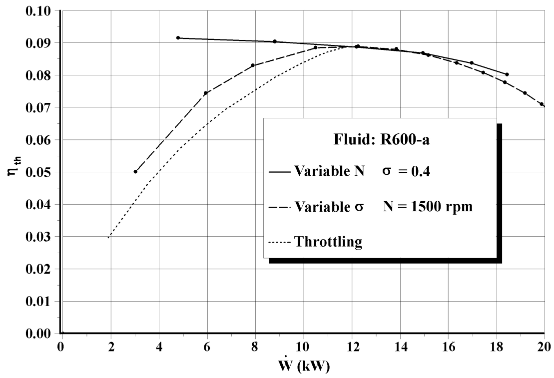

Fig. 12. Comparison between the various regulation strategies

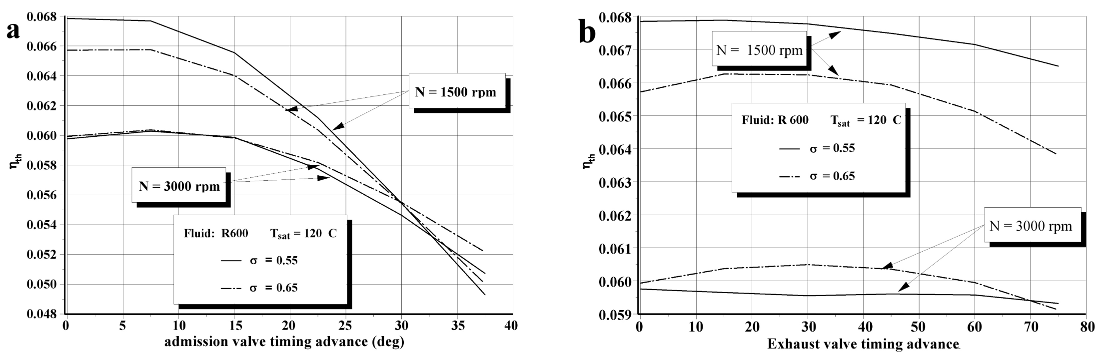

Fig. 13. Effects of timing advance of the intake (a) and the exhaust valve (b) on the thermal efficiency of the cycle using R600-a

valve values up to 25-35° may be needed depending upon the admission grade σ. The variation of the efficiency is anyway quite reduced, so that advances near to zero may be adopted in almost every case.

5. Conclusions

In this paper the performances of a rotary expansion device based on the Wankel mechanism were evaluated as a function of the operating parameters. More in detail the influence of the type of the working fluid, the rotating speed, the admission and recompression grades were evaluated. The analysis took also into account the valves opening advances. These investigations were performed by means of a numerical model that was validated using the experimental data collected at the engine test bench on a prototype operated with compressed air.

The analysis pointed out that the maximum isentropic efficiency is not reached with the as lowest as possible admission grades, but on the contrary a value of σ giving the best efficiency may be found and its value depends on the pressure ratio between admission and exhaust. This is due to the possibility of overexpanding the working fluid, with a consequent loss due to the inversion of a part of the cycle.

When the regulation is performed by means of the rotating speed, the cycle of the engine may be overexpanded if the admission grade is too small; in this case it is preferrable to choose an admission grade greater than the value giving the maximum efficiency since this analysis proved that in this case the device is less sensitive to the change of rotating speed.

As regard of the influence of the fluid, for the analysed cases the maximum efficiency is reached for the same value of σ whether is the fluid and a greater influence is assumed by the temperature drop.

The numerical model employed proved to be a very useful tool for the preliminary analysis of the expansion device, providing interesting informations as regard of the behavior of this component respect to its main operating parameters and the controlling strategies.

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