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Development and numerical assessment of a regulation strategy for Sliding Rotary Vane Expander based on revolution speed variation

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Abstract. Sliding Rotary Vane Expanders (SVRE) are widely used in ORC-based power units for waste heat recovery in internal combustion engine (ICE) thanks to the capability to handle off-design conditions and their lower speed. In particular, SVRE revolution speed is usually varied together with the pump one to regulate the recovery unit. Nevertheless, this parameter affects SVRE performance and such effects should be taken into account. Thus, in order to reach this goal, in this paper a control strategy based on revolution speed variation was developed for SVRE. Its suitability and effects on expander performance were analyzed through a SVRE model developed in GT-Suite™ environment. The model was experimentally validated thanks to an extensive experimental campaign carried out on a 1.5 kW SVRE installed on an ORC-based power unit fed by the exhaust gases of a 3 liters supercharged Diesel engine. The results confirm the regulation strategy effectiveness as the maximum deviation between the intake-end pressure (object of regulation) and the set-point is 4% of its value for a wide range of operating conditions. Moreover, the numerical results show that the increase of revolution speed until a certain value leads to the expander global efficiency increase and mechanical power too.

1. Introduction

In recent years International attention has been moved towards the reduction of the atmospheric CO₂ concentration which has reached unprecedented peaks. Regarding this issue, a lot can be done in the road transportation sector which is responsible approximately for one fourth of greenhouse gases emissions in Europe [1]. In order to face this problem, European legislation has put more stringent limits related to fleet specific CO₂ emissions [2]. This latter aspect determined the need of finding new solutions with the aim of increasing the efficiency of internal combustion engines (ICE). For these systems, in fact, roughly one third of the chemical energy of the fuel is transferred as thermal energy in the exhaust gases and thus lost. Among the different technologies which are able to recover this waste heat, a promising solution is represented by Organic Rankine Cycle (ORC) systems. The use of organic fluids in power generation plants based on Hirn or Rankine cycles has reached through the years the right know how to become commercially available for stationary applications. However, there are several issues that limited their full potential in mobile applications as in vehicles propelled by Internal Combustion Engines, ICE. Some of these limits are related to ICE efficiency reduction due to backpressure effect produced by the Heat Recovery Vapor Generator (HRVG), space availability [3],



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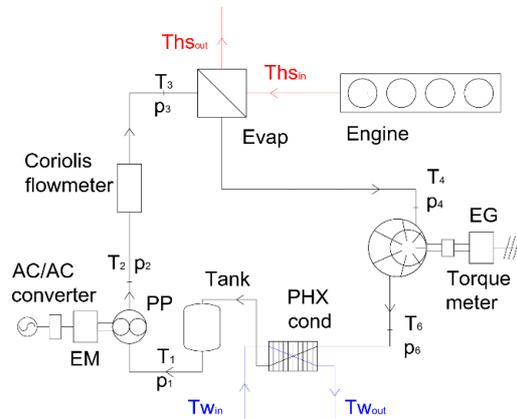
working fluid thermal degradation [4], safety, reliability and cost which have to be carefully evaluated in order to design the appropriate components [5]. Moreover, in this kind of applications, the power recovery units based on ORC system has to deal with a hot source which varies in terms of temperature and flowrates according to the engine operating points and these aspects lead frequently the unit in off design conditions with a very low recovery capability. For this reason, recently, many scientific studies on this matter have focused on the characterization of the off-design behavior of the ORC-based power units with the aim of defining a control strategy ([6],[7]). In [8] the authors studied the different behavior between ideal transformations and real operating conditions during the expansion in a radial turbine highlighting strong differences. Same considerations apply for volumetric machines, of a Sliding Vane Rotary type, [9]. In [10] and [11] the authors made use of semi-empirical models in order to define the expander and overall performances of the unit highlighting the need of a well-defined experimental data range to accurately extrapolate results outside the range itself. A novel layout characterized by two expanders in series has been evaluated in [12]. In this case a further degree of freedom is added to the system with the possibility of making profit of it in off-design situations. However, the greater complexity could not justify the increase in the efficiency in small size applications. HRVG's characteristics are also important in off-design situations [13] having the capability of dumping thermal fluctuations of the hot source and of reducing the overall weight of the unit and volume if the corresponding thermal inertia is correctly designed, [14]. Different control strategies have been investigated in literature both experimentally or theoretically studied through ORC unit off design models. In [15] three control strategies have been compared with the best solution represented by a combination of sliding pressure & sliding velocity, varying both expander and pump speed according to the best efficiency point for each engine thermal load evaluated at steady state conditions. On the other hand, the analysis reported in [16] shows that best performances can be obtained varying pump speed using superheating degree as a control parameter, at least for small sized ORC unit. However, the variation of the pump revolution speed and expander affects the expander intake pressure [17] which set the maximum pressure of the plant: this appears quite important for the recovery unit, influencing both the evaporator and the expander behavior. Therefore, SVRE regulation strategy based on these parameters should be furtherly analysed.

In order to achieve this goal, in this paper a regulation strategy of the expander intake-end pressure was developed and the effects of speed variation on SVRE performance were numerically investigated through a GT-Suite™ experimentally validated model developed by the author in [9]. The regulation strategy proposed is a simple iterative procedure easily implementable in an ECU based on a theoretical relation between the intake-end pressure and mass flow rate and expander revolution speed [17]. Thanks to the numerical model the regulation strategy suitability to follow the set point in terms of intake-end pressure was assessed and the effects of revolution speed variation on expander performance were predicted.

2. Experimental layout

The numerical model developed and used in this paper in order to evaluate the performances of the volumetric expander at different rotational speeds has been calibrated through experimental data collected in a wide range of operating conditions. The experimental setup shown in figure 1 consisted of: a volumetric gear pump controlled in speed by mean of an electric motor and an inverter; a plate and fin evaporator fed by the exhaust gases of a Diesel engine; a 1.5 kW Sliding Vane Rotary Expander SVRE connected to an electric generator which is linked to the electric network so the SVRE is constrained to rotate at 1500 RPM; a plate heat exchanger cooled by tap water as condenser; a 3 L tank upstream the pump to dump the mass flow rate fluctuations.

In addition to these, piezoresistive pressure transducers were mounted on the cover of the expander in order to analyse the indicated cycle and a torque meter between the expander and the electric generator. In this way the evaluation of the mechanical power together with the mechanical efficiency of the expander was possible. More details on the experimental campaign and on experimental set-up can be found in [9].



Legend

- EM- electric motor
- EG- electric generator
- Evap- Evaporator
- PHX- Plate Heat Exchanger
- PP- Gear rotor pump
- ThS_{in}- Exhaust inlet gas temperature
- ThS_{out}- Exhaust outlet gas temperature
- TW_{in}- Cooling water inlet temperature
- TW_{out}- Cooling water outlet temperature

Figure 1. ORC experimental layout

3. SVRE GT-Suite™ model

In order to develop expander speed control strategy, it is fundamental the assessment of its performance when the revolution speed varies. To do so a numerical model developed and validated by the Authors in previous work [9] was used. As the experimental campaign was conducted keeping constant the expander revolution speed, the model acts as a software platform and allows to extend the analysis of expander behavior outside the operating conditions experimentally observed, in particular, when the revolution speed changes. The model was developed in GT-Suite™ environment and combines a mono (1-D) and zero-dimensional (0-D) thermo-fluid-dynamic analysis. In particular, the 1-D analysis involves the discretization of the fluid domain in multiple sub-volumes and for each sub-element the Navier-Stokes equations expressing the mass, momentum and energy conservation are solved. This procedure was applied to reproduce the fluid behavior in correspondence of intake and exhaust pipes as these phases are characterized by a high transient rate. Otherwise, the 0-D approach was employed to treat the expander vanes which are modeled as phased volumes (in the sense of revolution speed) which vary according to the rotation angle. The 0-D approach was used to model also the volumetric losses and three major leakages path have been considered: the leakages through the gap between the rotor face and the casing; the leakages through the gap between blade tip and stator; the leakages through the gap between the blade side and rotor slots. The model allows the evaluation also of the power losses P_{loss} due to dry and viscous friction effects. The quite total amount of mechanical power is lost due to the dry friction between blade and stator inner surface. This contributes is represented by equation (1):

$$P_{loss} = C_{tip} N_v F_N r_v \omega \quad (1)$$

Where C_{tip} is the friction factor, N_v is the blade number, r_v is the actual distance between blade tip and rotor center and F_N is the force acting on the blade due two contributes: the first one is due to the action at blades bottom due to the fluid which fill the space between the rotor slots and the blades pushing them against the stator inner surface; the second one is the centrifugal force applied on the blades center which contributes to push the blades against the stator. Subtracting P_{loss} to the indicated power, evaluated in equation (2) (where p_i and V_i is respectively the actual pressure and volume in the i -Vane during rotation while t_{cycle} is the time cycle), the mechanical power produced by the expander can be obtained as in equation (3). More information about the model can be found in [9].

$$P_{ind} = \frac{\oint \sum_{i=1}^{N_v} p_i dV_i}{t_{cycle}} \quad (2)$$

$$P_{mech} = P_{ind} - P_{loss} \quad (3)$$

The boundary conditions of the model are the mass flow rate elaborated by the machine \dot{m}_{WF} , the intake end pressure $p_{in,end}$ and temperature T_{in} , the pressure exerted by the circuit at expander outlet p_{exh} and the revolution speed kept constant to 1500 RPM. The model has been validated in the authors previous work, [9] and it allows to reproduce with good accuracy the real expander behavior. Indeed, the Root Mean Square Errors RMSE in terms of intake pressure, mechanical power, volumetric (equation (4) - where $\rho_{in,end}$ and $V_{in,end}$ are respectively the density and volume at intake end) and global efficiency (equation (5) where h_{in} and $h_{out,is}$ are respectively the specific enthalpy at intake and outlet side in isentropic condition) are 2%, 7%, 2% and 5% respectively.

$$\eta_{vol} = \frac{\rho_{in,end} V_{in,end} N_v \omega}{\dot{m}_{WF}} \quad (4)$$

$$\eta_{exp} = \frac{P_{mec}}{\dot{m}_{WF}(h_{in} - h_{out,is})} \quad (5)$$

4. Results

The variation of ω has a great influence on $p_{in,end}$ value as demonstrated through the theoretical analysis performed by the authors in [17]. Indeed, in [17] it was found that the main driver of the expander intake pressure variation is ω and \dot{m}_{WF} . This can be observed introducing $\rho_{in,end}$ (eq.6) in the ideal gas equation corrected by the compressibility factor Z (equation (7)), obtaining equation (8) describing $p_{in,end}$ variation as function of \dot{m}_{WF} , ω and η_{vol} . R is the particular gas constant and T_{in} intake end fluid temperature.

$$\rho_{in,end} = \frac{\eta_{vol} \dot{m}_{WF}}{V_{in,end} N_v \omega} \quad (6)$$

$$p_{in,end} = ZRT_{in} \rho_{in,end} \quad (7)$$

$$p_{in,end} = ZRT_{in} \frac{\eta_{vol}(\omega, \dot{m}_{WF}) \dot{m}_{WF}}{V_{in,end} N_v \omega} \quad (8)$$

It is worth to notice that equation (8) was developed to control $p_{in,end}$, however, if Δp_{loss} is introduced in equation (8) to take into account the pressure losses between the start and the end of intake phase the control of the pressure at expander inlet (and that of the HRVG operation) would result more reliable. It is evident how \dot{m}_{WF} and ω are respectively directly and indirectly proportional to $p_{in,end}$. Thus, $p_{in,end}$ raises with \dot{m}_{WF} increase and ω decrease. On the other hand, the dependence on η_{vol} is not linear and should be carefully analyzed. Indeed, η_{vol} depends on ω so equation (8) cannot be used as a linear relation unless some assumption on relationship between η_{vol} and ω are made. So, in order to find this dependence, the numerical model was used. In particular, η_{vol} of the expander was evaluated varying \dot{m}_{WF} and ω and the results were fitted in a mathematical form for control purposes (equation (9)).

$$\eta_{vol} = 0.076 + 3.7 \cdot 10^{-4} \omega - 0.25 \dot{m}_{WF} - 3.9 \cdot 10^{-8} \omega^2 - 4.5 \cdot 10^{-4} \omega \dot{m}_{WF} + 3.9 \dot{m}_{WF}^2 \quad (9)$$

Once this relation is known, equation (10) can be obtained rearranging equation (8) in terms of ω . It can be used for regulation purpose when \dot{m}_{WF} is known and $p_{in,end}$ is desired:

$$\omega = ZRT_{in} \frac{\eta_{vol}(\omega, \dot{m}_{WF}) \dot{m}_{WF}}{V_{in,end} N_v p_{in,end}} \quad (10)$$

In fact, equation (10) allows to predict the expander revolution speed needed to ensure a certain $p_{in,end}$ for a given \dot{m}_{WF} entering the machine known when the pump speed is fixed. However, equation (10) cannot be solved in closed form being not linear in ω . To overcome this issue, the iterative procedure reported in figure 2 was proposed. The following steps apply:

- 1) Definition of the two set points $p_{in,end}$ and T_{in} ; acquisition of the mass flow rate circulating inside the plant by the knowledge of the pump revolution speed. Another input is the initial value of volumetric efficiency to start the iterative procedure;
- 2) Evaluation of ω through equation (9) to achieve the desired set-points for the mass flow rate sent by the pump;
- 3) Evaluation of the volumetric efficiency through equation (9);
- 4) Evaluation of $p_{in,end}$ with the value of η_{vol} obtained at point 3;
- 5) Comparison of the calculated working fluid density with the values obtained when the pressure and the temperature are equal to the set-point values.
- 6) If the condition 5 is achieved the procedure ends and the expander revolution speed was defined; if the condition 5 is not respected, the procedure was repeated introducing in the calculation the volumetric efficiency value obtained at step 3. The procedure was repeated until the condition 5 is satisfied.

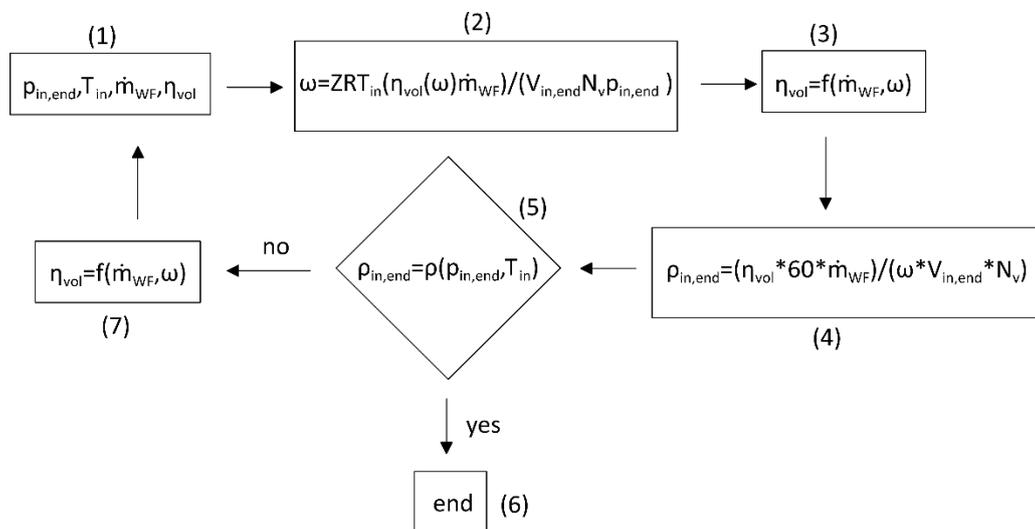


Figure 2: Scheme of expander speed regulation strategy.

In order to check the validity of the regulation strategy proposed (control of the intake-end pressure acting on ω when \dot{m}_{WF} varies), a theoretical procedure was run. For each \dot{m}_{WF} considered (in a range between 0.05 kg/s and 0.25 kg/s), the expander is rotated at the ω set by equation (10). The set-points in terms of pressure was 12 bar while in terms of temperature 361 K which allows to achieve 10 K of superheating degree. As reported in figure 3(a) the model is able to keep the $p_{in,end}$ close to the set point for a wide range of \dot{m}_{WF} . Indeed, if \dot{m}_{WF} is comprised between 0.1 kg/s and 0.225 kg/s the maximum relative error between $p_{in,end}$ and the set-point is equal to 4% of the set point. This value represent a good result considering the simplicity of the strategy proposed and its ability to maintain $p_{in,end}$ close to the set point, even for high excursion of \dot{m}_{WF} . The design value for the working fluid mass flow rate was equal to 0.12 kg/s. On the other hands, the regulation strategy does not allow to keep $p_{in,end}$ equal to the set point chosen when \dot{m}_{WF} is outside the range (0.1 kg/s - 0.225 kg/s). Indeed, for \dot{m}_{WF} comprised between 0.05 kg/s and 0.1 kg/s and for \dot{m}_{WF} higher than 0.225 kg/s the maximum error between the controlled $p_{in,end}$ and the set point is unacceptable and, respectively, equal to 40% and 12.3%. The first situation is the worst that can happen because \dot{m}_{WF} are too low for the volumes inside the machine defined when it was designed. In order to match the desired $p_{in,end}$ the procedure decreases ω till to values lower than 300 RPM (according equation (10)), very far from the design revolution speed of the machine (1500 RPM). This causes a severe reduction of η_{vol} lower than 20% (figure 3(b)) and this prevents the achievement of the desired $p_{in,end}$. At so low revolution speed, the stability of the expander is not insured too. The situation is less critical for \dot{m}_{WF} higher than 0.225 kg/s. In this condition, ω increases till to a value of 3450 RPM (to elaborate the higher mass flow rate entering the machine) thus avoiding that $p_{in,end}$

increases too much from the desired set point value: in any case a difference close to 13 % is still present with respect to the desired value. In this situation η_{vol} decreases up to 60% (figure 3(c)) while in the mass flow range between 0.05 kg/s to 0.225 kg/s the maximum value was 80 % when the flow rate was equal to 0.225 kg/s.

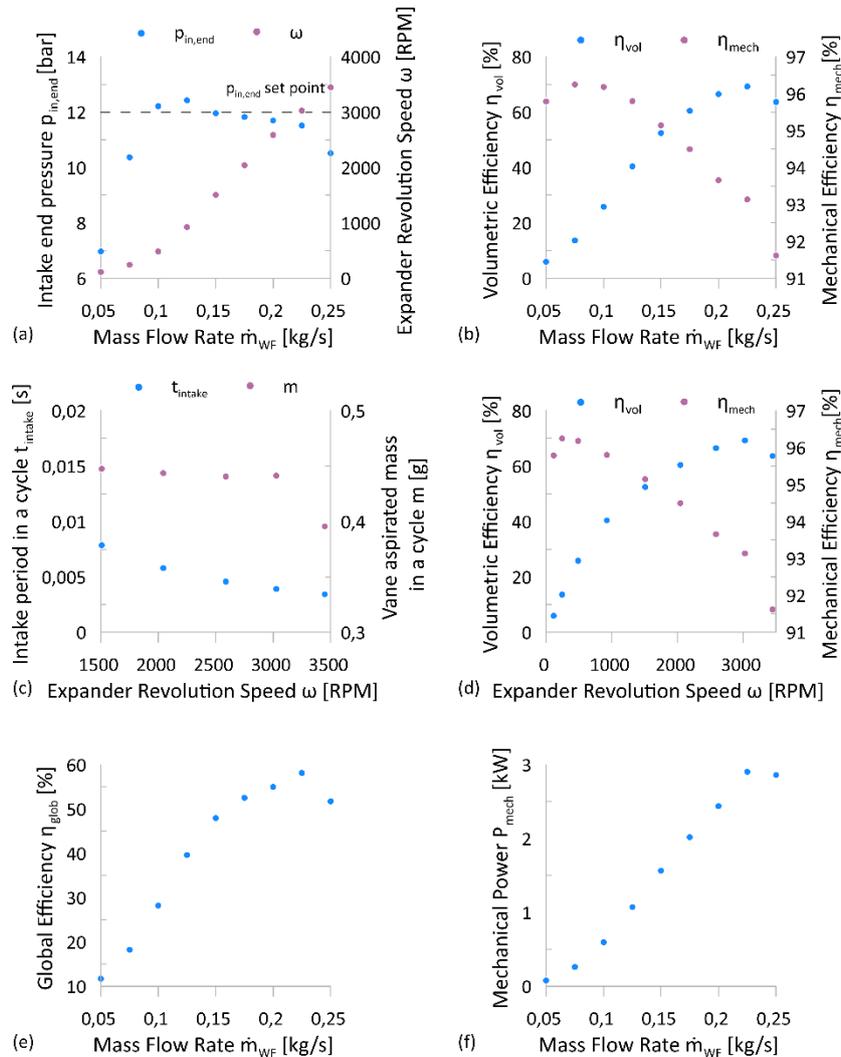


Figure 3: Effect of regulation strategy on intake-end pressure and expander revolution speed (a) and Mechanical and Volumetric Efficiency (b); Effect of revolution speed variation on intake period and mass aspirated by a vane in a cycle (c) and on volumetric and mechanical efficiency (d); Effect of regulation strategy on Global efficiency (e) and Mechanical Power (f).

This is due to the fact that at a so high revolution speed (3450 RPM) the time available to fill the vanes was not sufficient. Indeed, it decreased too much in correspondence of this ω (figure 3(c)), negatively affecting the mass aspirated by a vane in a cycle (figure 3(c)) and the volumetric efficiency (figure 3(d)). This means that the increase of ω has a global positive effect on η_{vol} due to the lower pressure value inside the vanes (which favor the filling process) and the enhancement of the sealing action at tip blades which reduces internal leakages (figure 3(d)). On the other hand, if ω is too high the time during which the intake port is reduced decreases and the filling action can't happen completely, so η_{vol} gets worse. Concerning the mechanical efficiency η_{mech} , it decreases with ω from 96% to 91% due to the enhancement of the centrifugal force acting on the blades which leads to a higher power lost by friction between the blades themselves (at tip) and the stator inner surface (figure 3(b) and figure 3(c)).

Thus, η_{vol} and η_{mech} show an opposite behavior when ω varies. Nevertheless, the effects of ω variation are higher on η_{vol} with respect η_{mech} as it can be seen in figure 3(d). Therefore, the increase of ω has a positive overall effect on expander performance as shown by the global efficiency η_{glob} in figure 3(e). Observing the results, it can be concluded that the expander suffers more the volumetric losses with respect the mechanical ones. When ω increases, P_{mech} grows as it can be seen in figure 3(f). Indeed, P_{ind} enhances as the numerator of eq.2 remains quite constant because the intake pressure was regulated while the denominator decreases as the machine performs more cycles per unit time. On the other side, the power lost by friction shows a weak growth with ω (as shown by the trend of η_{mech} in figure 3(d)). P_{mech} globally increases from 0.08 kW to 2.9 kW when ω passes from 115 RPM to 3025 RPM.

5. Conclusions

In this paper a regulation strategy based on the expander revolution speed variation was proposed in order to keep the intake-end pressure close to a certain set-point value when the working fluid mass flow rate entering the machine varies. The regulation strategy is based on an iterative procedure which, with few iterations, allows to define the ω at which the expander must rotate to ensure the desired intake-end pressure. The procedure was based on the following passages:

- 1) Definition of intake-end pressure and temperature set points, the mass flow rate entering the machine (known from the revolution speed of the pump) and an initial value of volumetric efficiency;
- 2) Evaluation of the expander revolution speed and volumetric efficiency through a law derived by a fitting of numerical data of η_{vol} as function of \dot{m}_{WF} and ω ;
- 3) Evaluation of intake-end density and comparison with the one corresponding to the pressure and temperature defined by the set points: if the two values are equal (under a certain tolerance), the calculation ends and ω can be defined, otherwise, the procedure was repeated updating the volumetric efficiency of the previous step until the stop condition was satisfied.

The suitability of the proposed regulation strategy was tested through a numerical model developed in GT-Suite™ environment by the authors and validated on the results of a wide experimental activity. The expander was a 1.5 kW SVRE used in an ORC based power unit tested when it was fed by the exhaust gas of a 3 L supercharged Diesel engine. The numerical analysis shows:

1. The regulation strategy allows to keep the intake-end pressure close to the set point value for a range of \dot{m}_{WF} comprised between 0.1 and 0.225 kg/s. For mass flow rate lower than the minor limit of this range the machine cannot be regulated due to the too low revolution speed to which correspond a volumetric efficiency comprised between 10 and 20%;
2. For mass flow rate higher than the upper limit of the range the maximum deviation is lower with the previous case (13%) and it is due to the reduction of intake phase time.

The model allows to evaluate the following effects of revolution speed variation on the expander performances:

1. η_{vol} and η_{mech} shows an opposite trend with ω variation. In particular η_{vol} increases and η_{mech} decreases.
2. The effect of ω is higher on η_{vol} which shows a higher variation with respect η_{mech} . This was confirmed by the global efficiency curve η_{glob} which reproduce the η_{vol} trend as function of ω ;
3. The mechanical power P_{mech} grows with ω .

It can be concluded as the proposed regulation strategy allows to maintain the intake-end pressure close to the set point value with a maximum deviation of 4% of the set point. This is a good result considering the simplicity of the regulation and the capacity to regulate the machine even when the operating conditions are very far from the ones considered during its design.

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