MODEL BASED DESIGN OF A DUAL INTAKE PORT SVRE FOR SMALL SCALE ORC-BASED POWER UNITS

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ABSTRACT

Volumetric expanders are among the most suitable technological solutions for small scale ORC-based power units, thanks to their compactness, flexibility, capacity to manage two-phase working fluid and low operating revolution speed. Moreover, sliding vane rotary expanders (SVRE) seems to have also reliability and cheapness characteristics. Nevertheless, volumetric and friction losses affect their performances more than in other technologies. A way to improve the performances of SVRE is represented by the dual-intake port technology, which involves a further intake phase after the end of the main one. Indeed, keeping constant the mass flow rate provided by the pump, the dual-intake expander produces a reduction of the intake pressure with a mechanical power similar to the single intake machine. The machine, in fact, is more permeable and this reduces, for a given flow rate, the upstream expander pressure with a beneficial effect in terms of operating conditions. This aspect can enhance the expander operability in off design conditions, which is particularly interesting when the hot source is represented by the exhaust gases of an Internal Combustion Engine (ICE). In fact, when the ICE works at full load the mass flow rate circulating inside the plants should increase, in order to recover more thermal power. If the dual intake expander is used, a higher mass flow rate can be aspirated with respect to the single-intake port, without significantly increasing the intake pressure. In this way, the sealing components integrity and machine reliability can be preserved. In this paper, referring to the ICE operating points of the ESC-13-mode steady state procedure, the benefits introduced by this novel design option on expander operability was analyzed through a 1-D CFD model, validated on experimental data collected on an ORC-based power unit operating in a test bench. The analysis opened the way to a model based design approach oriented to a new technological concept of SVRE.

1. INTRODUCTION

Waste heat recovery (WHR) via thermodynamic cycles in ICE has been studied from several decades (Sprouse and Depcik, 2013) and it appears as one of the most significant technologies for fuel saving and CO_2 emission reduction in the transportation sector. ORC systems are a promising solution for WHR applications: they are commercially available for large plants, but they still meet several difficulties in small scale units. Moreover, they are applied to a vehicle, several factors should be considered: weight and encumbrances associated to the recovery unit limit their application to heavy-duty transportation means (Pili *et al.*, 2017), engine backpressure introduced by the heat exchanger in the exhaust line bring to an overconsumption that should accounted for (Di Battista *et al.*, 2015), the intrinsic unsteady behavior of the engine requires the optimization of the components sizing (Braimakis and Karellas, 2017), the cooling medium of the condenser affects significantly the design and layout of the unit on-board (Di Battista *et al.*, 2018). Expander technology is one of the most discussed aspects. Axial and radial turbines have been widely studied (Klonowicz *et al*., 2014), but they present some issues related to the management of off-design conditions (Cipollone *et al.*,2016) and high revolution speed (Li and Ren., 2016 and Fiaschi *et al.*, 2016). Volumetric machines, such as scroll (Song *et al.*, 2015) and screw (Zhang *et al.*, 2014) expanders have been reported as the most suitable solutions in micro and small scale applications for their compactness, ease of maintenance and off-design management. More recently, swash-plate expanders were introduced but they showed

low isentropic and volumetric efficiencies (Galindo *et al.*, 2015). Reciprocating expanders could be considered too (Imran *et al.*, 2016, and Gao *et al.*, 2016), but they do not seem to add any further advantages and have several drawbacks such as vibrations, dimensions, limited flow rates, pulsations, etc. Sliding Vane Rotary Expanders (SVREs) also have good potential, in particular, for ease of operation, maintenance, compactness and cost (Gnutek and Kolasiński, 2013). They are less sensible to the inlet thermodynamic conditions of the working fluid, making easier the off-design and transient management, (Cipollone *et al.*, 2015). Main disadvantages are represented by low capacity and lubrication requirements for sealing and wearing issues (Bao and Zao, 2013). A novel technology which ensures to improve the SVRE performance is the dual intake port (Fatigati *et al.*, 2019). This consists of the introduction of an additional intake port in order to continuously feeding the expander after the closing of the main intake one (during the expansion phase). In this way, the pressure is sustained during the expansion phase, due to the further mass flow rate aspirated, and, consequently, the indicated power grows (Fatigati *et al.*, 2018). The dual-intake port increases also the permeability of the machine, allowing a greater energy recovery when the ICE works at highest loads, avoiding the eventual by-pass of the exhaust gases, so giving the possibility to increase the fluid fow rate. A higher working fluid mass flowrate can be elaborated without increasing the intake pressure, as in conventional machines, where this situation would lead to a reduction of the volumetric and mechanical efficiency. Hence, the operability of the expander results to be widened. In this paper, an assessment of the operability improvement of the unit due to the introduction of the dual-intake expander was carried out. The ICE operating points of the ESC-13-mode steady state procedure were tested on a turbocharged 3 liters diesel engine (IVECO F1C). The experimental data allows to perform a validation of an expander model developed by the author in previous works, (Fatigati *et al.*, 2018), allowing to evaluate the expander permeability and its mechanical power produced. The results allowed to refine an overall off-design model of the ORC-based power unit when single- and dual-intake port expander technology are used, being the ICE working conditions far from the design case which originally defined the design of the components.

2. MATERIALS AND METHODS

2.1 Experimental Activity

In order to assess the benefits introduced by the dual-intake port technology with respect to the conventional machine (single intake), an experimental comparison between the two machines was performed keeping constant the mass flow rate provided by the pump. The working fluid adopted is R236fa for the sake of continuity of the previous experimental campaigns. The experimental analysis was carried out on full instrumented ORC-based power unit fed by the exhaust gases of a 3 liters supercharged diesel engine. The main components of the ORC-based power unit are:

1) A volumetric pump;
2) A fins and tubes evaporator, properly designed to reduce engine back-pressure;
3) A sliding vane rotary expander, whose geometric properties are reported in Table 1 and Figure 1. The dual intake expander was obtained introducing on the OEM expander a further intake port; thus the two machines have the same dimensions. The expander moves an electric generator connected to the network so it is constrained to rotate at 1500 RPM. After the characterization of the OEM, the second port was introduced on the machine and it was tested.
4) A plate heat exchanger, cooled by tap water;
5) A plenum upstream the pump in order to dump the pulsation of the working fluid;

Pressure and temperature transducers are upstream and downstream to each component. The mass flow rate of the working fluid is measured thanks to a Coriolis flow meter while that of the coolant through a magnetic flow rate. The torque and revolution speed of the expander and pump were measured through two dedicated torque meters. A peculiarity of this experimental campaign is the measurement of the indicated cycle performed as in (Fatigati *et al.*, 2018 and Fatigati *et al.*, 2019), through the installation of three piezo-resistive sensors angularly spaced in order to reconstruct the p-V cycle.

Table 1. Expander Geometry

Number of Chambers	7
Stator Inner Diameter	75.9 mm
Rotor Outer Diameter	65.0 mm
Eccentricity	5.45 mm
Chamber Width	60.0 mm
Intake port start $\theta_{main,int,start}$	4.4°
Intake port close $\theta_{main,int,start}$	48°
Dual intake port open $\theta_{main,int,start}$	90°
Dual intake port close $\theta_{main,int,start}$	103.7°
Exhaust port open $\theta_{main,int,start}$	180°
Exhaust port close $\theta_{main,int,start}$	320°
Dual intake port position $\Phi_{dual\ int}$	42°

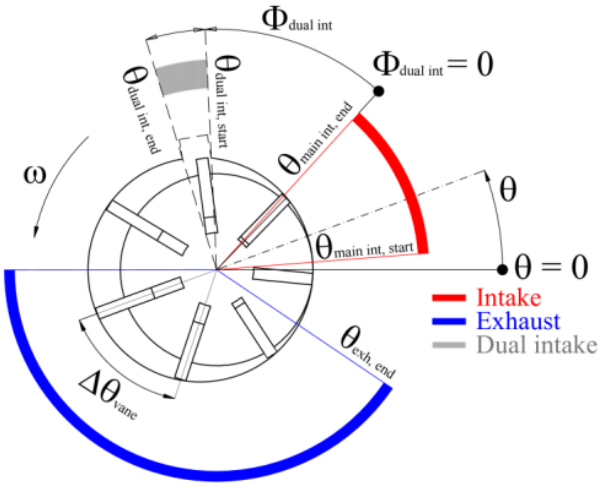

Figure 1. Dual intake and OEM expander

2.2 Expander numerical model

The expander numerical model was developed by the authors in GT-Suite™ environment in previous works (Fatigati *et al.*, 2018 and Fatigati *et al.*, 2019). This was done following an integrated approach between a mono-dimensional (1-D) and zero-dimensional (0-D) thermodynamic and fluid dynamic analysis of the working fluid. The 1-D analysis was adopted to model the filling and emptying phase of the expander which are intrinsically unsteady processes. It involves the subdivision of the intake and exhaust pipe discretization into a multiple sub-volumes for which the Navier-Stokes equations (mass, momentum and energy conservation) are solved. The 0-D approach was adopted to model the expander vanes, which are treated as angularly phased lumped capacity, and the volumetric losses. Therefore, the model is able to predict the effective mass of working fluid inside the vanes during the expander rotation. So, the volumetric efficiency η_{vol} of the machine can be valuated as of the ratio between the mass inside the vane at the end of the intake phase and that provided by the pump (1). The volumetric efficiency expressed in (1) is the reciprocal of FF factor generally considered in literature. However, both η_{vol} and FF indicate that the difference between the real mass flow rate aspirated by the expander and the theoretical one is due to the leakage.

$$\eta_{vol} = \frac{\rho_{int} V_{int} N_v \omega}{m_{WF}} \tag{1}$$

The model allows evaluating also the pressure inside the vanes in correspondence of each shaft rotational angle. Therefore, the prediction of the indicated power can be carried out according to (2):

$$P_{ind} = \frac{\sum_{i=1}^{N_v} \oint_V p_i dV_i}{t_{cycle}} \tag{2}$$

Once the indicated power is known, subtracting to it the mechanical power lost evaluated by the model, the mechanical power can be assessed (3):

$$P_{mech} = P_{ind} - P_{mech,lost} \tag{3}$$

2.3 Off-design model of the ORC-based power unit

In order to define the expander inlet conditions in different operating points of the internal combustion engine a simplified off design model of the unit has been adopted. In particular, the behavior of the

evaporator when the characteristics of the hot source are different from the design ones is particularly important to calculate outlet temperatures and pressures. For this purpose, the approach used in (Imran *et al*., 2018) has been followed. The product of the overall heat transfer coefficient and the heat transfer area has been defined as in (4):

$$UA_{od} = UA_d \left(\frac{mhs_{od}}{mhs_d}\right)^{0.8} \tag{4}$$

Where m_{hs} is the exhaust gas flowrate and the subscripts "*d*" and "*od*" stand for design and off design conditions respectively. In a similar manner the evaporator pressure drop has been calculated as follows (5):

$$DP_{od} = DP_d \left(\frac{m_{wf,od}}{m_{wf,d}}\right)^2 \left(\frac{p_{evap,od}}{p_{evapf,d}}\right)^{-1} \tag{5}$$

where m_{wf} is the working fluid mass flowrate and p_{evap} is the evaporating pressure. According to these assumptions, the thermodynamic conditions of the fluids entering and leaving the evaporator have been calculated, as well as the pressure drops. A similar approach was used for the condenser, focusing more the attention on the working fluid.

3. RESULTS AND DISCUSSIONS

3.1 Experimental and numerical comparison between single and dual intake expander

The experimental campaign allowed to characterize both the single and dual intake expander, in terms of permeability and mechanical power produced. The permeability is expressed as the trend of the intake pressure as a function of the mass flow rate aspirated by the machine. So, the lower is the permeability, the higher is the intake pressure, for a given mass flow rate entering the expander. In Table 2 the results of the experimental characterization of the two machines are reported: the experimental cases 1-4 refer to single intake expander, while the others (5-8) are related to dual intake machine. In Table 2, the intake pressure, the power produced and the performance of the single and dual intake expander are highlighted for different mass flow rate entering the machine.

For the range of mass flow rates considered (0.068-0.140 kg/s), the intake pressure varies from 0.70 to 1.10 MPa for single intake expander and from 0.38 MPa to 0.70 MPa for dual intake port machine: this demonstrates the higher permeability of the dual-port expander, which has lower inlet pressure with the same mass flow rate. On the other hand, keeping constant the intake pressure, the dual intake expander allows to aspirate a greater mass flow rate. This can be noticed comparing case 1 (single intake port) with case 8 (dual intake port), which have a similar intake pressure equals to 0.70 MPa. The mass flow rate aspirated by the dual intake expander (case 5) is up to 50% higher than the single intake one, leading to an increase of 52% of the mechanical power produced. Thus, the dual intake port ensures to aspirate a greater mass flow rate and, consequently, it allows to recover a greater thermal power in similar thermodynamic conditions (i.e. expander inlet pressure).

Table 2: Experimental comparison between single and dual intake expander.

	Single Intake Expander				Dual Intake Expander			
Case	1	2	3	4	5	6	7	8
m_{WF} ±0.15% [kg/s]	0.068	0.079	0.104	0.117	0.071	0.094	0.13	0.14
p_{in} ±0.03 [MPa]	0.70	0.81	1.00	1.10	0.38	0.48	0.62	0.70
T_{in} ±0.3 [K]	356.4	345.5	360.9	362.3	330.3	343.2	344.8	363.4
p_{out} ±0.03 [MPa]	0.34	0.39	0.44	0.49	0.2	0.23	0.27	0.31
P_{mech} ±0.8% [W]	281.0	320.3	558.1	582.4	198.9	320	493.9	590.4
η_{vol} ±2.3%	0.49	0.49	0.47	0.47	0.61	0.59	0.58	0.58
η_{exp} ±1%	0.33	0.34	0.38	0.37	0.26	0.27	0.27	0.29

This allows to preserve the integrity of the sealing components and to reduce the mechanical and volumetric losses, which depend on the vane pressure. In fact, the volumetric efficiency evaluated according (1) shows an increase of 18% with the introduction of the dual intake port. It is worth to notice how in the evaluation of η_{vol}, V_{int} represents the volume of the vane at the end of the intake phase. For the dual intake expander as V_{int} was considered the volume of the vane at the end of the dual intake phase. Concerning the global efficiency, that of the single-intake port expander is slightly lower than that of the dual-intake port, meaning that the introduction of this technology does not worsen the expander performance. The numerical model was used in order to extend the comparison between the two machines outside the experimental operating condition. Table 3 shows the good agreement of the numerical and experimental data in terms of mass flow rate for a given pressure difference between intake and exhaust and the mechanical power produced by the expander.

Table 3: Relative percentage error between numerical and experimental data.

	Single Intake Expander				Dual Intake Expander			
Case	1	2	3	4	5	6	7	8
m_{WF}	3.5%	7.0%	1.2%	-2.0%	3.9%	1.8%	4.2%	4.0%
P_{mech}	-4.4%	1.2%	-3.6%	-6.2%	-1.5%	0.2%	4.2%	-6.3%

Figure 2 shows the different slope of the intake pressure vs. mass flow rate of the two machines: this slope represents exactly the permeability of the expander. The higher permeability of the dual intake expander ensures to aspirate a higher mass flow rate for the same intake pressure producing a greater mechanical power. In fact, for an intake pressure equals to 1 MPa the mass flow rate entering the dual intake expanders is equals to 0.25 kg/s while the one aspirated by the single intake machine is 0.1 kg/s (Figure 2 (a)). Thus, the dual intake expander is able to produce a mechanical power of 1000 W while the single intake intake machine 560 W (Figure 2(b)).

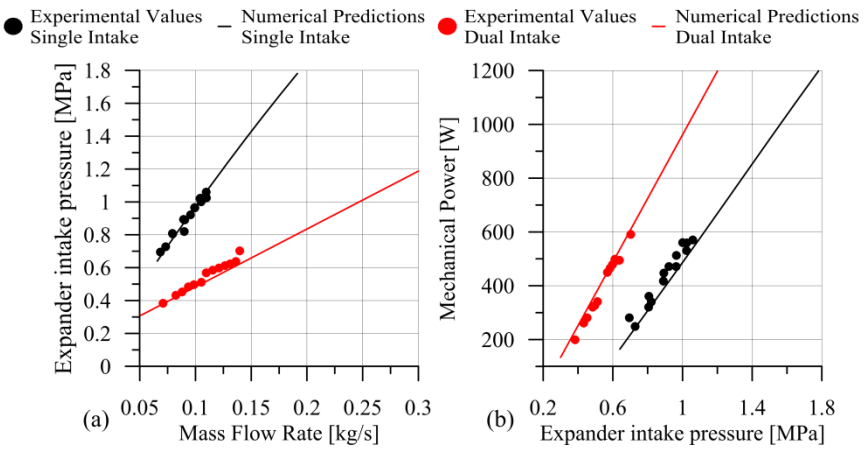

Figure 2: Experimental and numerical values of intake pressure (a) and mechanical power (b) as a function of mass flow rate and intake pressure for single and dual intake expander.

This means that the dual intake technology ensures to enhance the flexibility of the machine without modify its size. So, for the dual intake machine, the ratio between the power produced and expander weight and the power produced and the machine volume is 50 % greater with respect to the single intake port expander. In fact, the expander weight and volume are the same and the power increase in the order of 50% keeping constant the intake pressure. On the other hand, when the comparison was carried out for the same mass flow rate aspirated, the mechanical power of the dual intake machine is higher than that of dual intake machine. In this case the ratio power/weight and power/volume is higher for single intake machine. Nevertheless, when the two machine operate with the same mass flow rate, the dual intake expander produce a lower (but comparable) mechanical power with a lower intake pressure. This means that the machine components can be preserved and its weight contained

as the components deal with a lower operating pressure. Moreover, benefits on mechanical and volumetric power can be achieved as they are proportional to intake pressure.

3.2 Comparison between single and dual intake expander in off-design condition

The dual-intake port technology allows widening the operability of the expander, in particular when the mass flow rate circulating inside the plant increases: this is the case when higher thermal power is to be recovered. Through the combination of the off design model of the whole ORC-based power unit and that of the expander, the comparison between the single-intake port and the dual-intake port of the expander was outlined for all the ICE operating points of the ESC-13-mode steady state procedure (Figure 3).

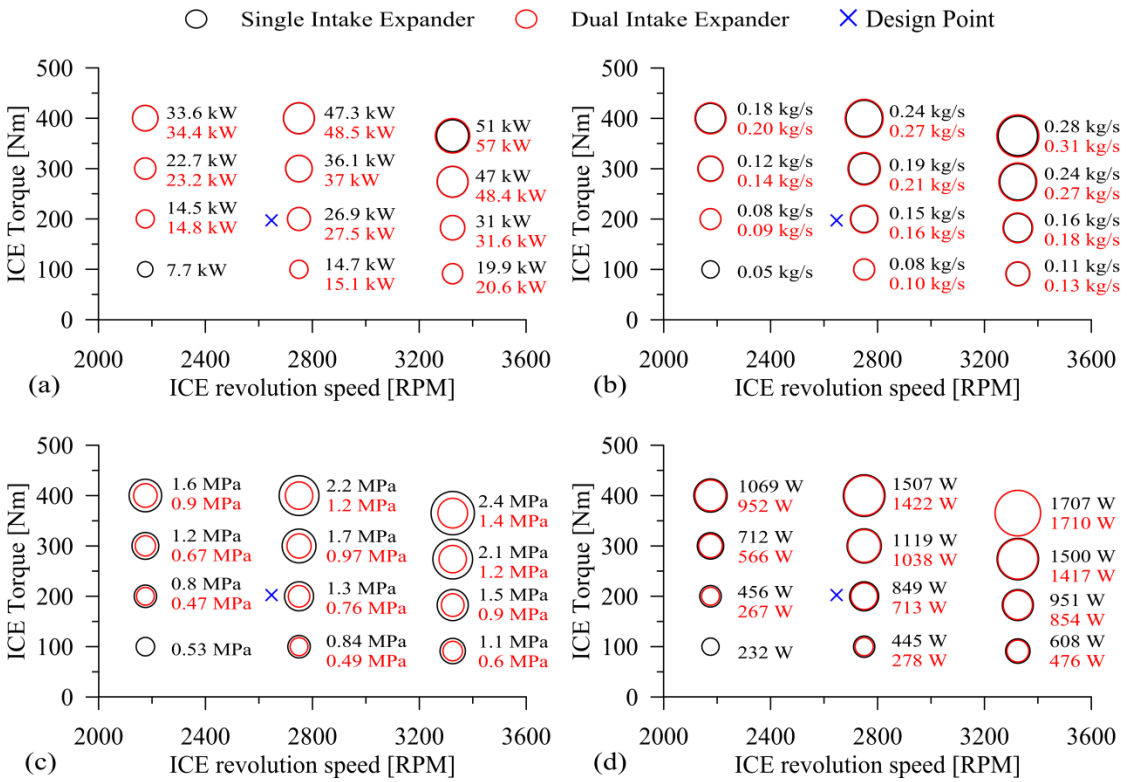

Figure 3: ESC-13-mode steady state operating point, (a) thermal power recovered by the single and dual intake expander; (b) mass flow rate entering the single and dual intake expander; (c) Intake pressure of single and dual intake expander; (d) Mechanical power produced by single and dual intake expander.

Figure 3(a) shows the thermal power recovered by the units: with a dual-intake port, it is always higher than the one of the single-port expander. Indeed, for each ICE operating point, the mass flow rate of the working fluid circulating inside the plant is always higher than that of the original case (single intake), as it can be noticed in Figure 3(b). The higher permeability of the dual-intake port expander ensures that, for corresponding points, the intake pressure is lower in comparison of that reached in the single-intake case. In fact, Figure 3(c) shows that, for each point, the intake pressure of the dual intake expander is 50% lower than the single-intake port one. Nevertheless, despite the intake pressure is lower for the dual intake machine, the mechanical power produced by the expander is comparable in the whole operating range of the ICE, as Figure 3(d) reports.

It is interesting, for instance, the case in which the ICE moves from the design point (on which the ORC-based power unit was designed) characterized by a torque equal to 200 Nm and revolution speed equal to 2800 RPM to another working condition where the torque and revolution speed are 400 Nm and 2800 RPM, respectively. In order to recover a thermal power equal to 47.3 kW and produce a

mechanical power of 1507 W, the mass flow rate entering the single-intake port machine should be equals to 0.24 kg/s. Nevertheless, in this condition the intake pressure at the expander would be equal to 2.2 MPa which can compromise the sealing components integrity and requires greater dimension of the expander parts to stant with higher pressure. Adopting the dual-intake port technology, if the mass flow rate is increased till to 0.27 kg/s, thermal power of 48.5 kW is recovered producing a mechanical power at the expander equals to 1422 W with an intake pressure of 1.2 MPa. Thus, the expander can work with an acceptable pressure value, without worsening mechanical power production.

4. CONCLUSIONS

In the present work, the operability gain due to the introduction of the dual-intake port technology in a SVRE was assessed. After an experimental characterization of the dual intake expander, a numerical model was validated and used as virtual software platform to study the expander performance of the single- and of the dual-intake port in off-design condition. Combining the model of the expander with that of the whole ORC-based power unit, the comparison between the two machines was extended for all the operating conditions of the engine test ESC-13-mode steady state procedure. Off design behaviors of the evaporator and of the condenser were considered. The results showed that when the ICE works at full load and the mass flow rate circulating inside the plant increases, the dual intake expander allows to produce a comparable mechanical power than that obtained by the single-intake port expander, but with an intake pressure 50% lower, improving reliability and machine integrity.

NOMENCLATURE

		Subscript	
A	Heat transfer area (m^2)		
m	Mass flow rate (kg/s)	d	design
P	Power (W)	od	off-design
p	Pressure (MPa)	hs	exhaust gases
V	Volume (m^3)	ind	indicated
U	Heat transfer coefficienct (W/m^2K)	int	intake phase end
ρ	Density (kg/m^3)	mech	mechanical
N_v	Vane Number	wf	working fluid

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