STEADY AND TRANSIENT OPERATIONS OF SMALL-SCALE ORC-BASED POWER UNIT FOR LOW-GRADE THERMAL SOURCES

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ABSTRACT

Waste Heat Recovery (WHR) from exhaust gases of Internal Combustion Engines (ICE) is one of the most significant opportunities to reduce carbon dioxide emissions in the transportation sector.
This is particularly useful because the category, at European defined a fine for any gram of CO_2 which exceed the specified limits, redefining in this way the value of a low-carbon technology.
In ICE applications, the main critical aspect is the intrinsic time-varying nature of the upper thermal source, which forces the ORC-based units to work frequently in off-design condition. This requires the recourse of reliable control systems and a careful choice of the design specifications of components. A key role is played by the expander which it must ensure to deal with highly variable flow rates and inlet thermodynamic conditions with the eventuality to receive two-phases flow when the high temperature thermal source is not enough to guarantee a full vaporization. Hence, to address all these issues, positive displacement machines are generally preferred to the dynamic types (radial or axial turbines). Scroll expanders among the other types appear to be the most suitable candidates, thanks their reliability, compactness, high efficiency and ease of starting.
In this paper a scroll expander has been tested inside a test rig which allows to evaluate the performance of a 1.5 kW mechanical ORC-based power unit fed by the exhaust gases of a 3L turbocharged diesel internal combustion engine represented by an Iveco F1C. As working fluid, R245fa was selected. The results are presented for different steady working condition observing a maximum plant efficiency and power respectively equal to 4% and 500 W. Also transient analysis were performed observing that the ORC unit presents a relatively low start up time (370 s) thanks to the capability of scroll machine to work with two phase working fluid.

1 INTRODUCTION

The improvement of the internal combustion engines (ICEs) will play a key role in the transition toward a more sustainable transportation sector. Indeed, waiting for a green electricity and for a significant cost reduction of full electric vehicles, it will be fundamental to increase the efficiency of ICEs to reduce their environmental impact. In fact, focusing the attention on CO_2, to date the on-the road transportation sector is responsible of the 21% of the whole total emissions. Thus, the Governments set strict regulation in terms of CO_2 emissions, and for any grams of CO_2 exciding the limits severe fees must be paid by the customer.
Waste Heat Recovery of ICEs through ORC based power unit is one of the most promising solution to increase the engine efficiency thus reducing the pollutant emissions. *Di Cairano et al.* (2020) developed a compact system called ReverCycle operating in two different ways. The systems indeed can work as standard mobile air conditioning if cabin cooling is needed or as ORC plant to recover the waste heat of the engine cooling system. Thanks to this system a fuel consumption reduction of 1.3 and 2% are respectively observed with cold and hot start condition.

Concerning the waste heat of the exhaust gases it represents roughly the 30-40% of the input energy is (*Wang et al., 2019*) thus its recovery allows to achieve significant benefits in terms of engine efficiency (*Song et. al, 2020*).
Nevertheless, several issues concerning the intrinsic transient nature of the heat source, the engine back-pressure, the vehicle weight increase and the reduction of space on board should be solved prior to enable a fully development on the market, (*Di Battista et al, 2018*). In particular, the high variation to which the exhaust gases are subjected leads the unit to work often far from the design condition, hence, the maximization of the ORC performance in off-design operation is fundamental to widen the adoption of this technology (*Chatzopoulou et al., 2019*). For this reason, besides the adoption of suitable control system (*Marchionni et al., 2018*) a proper selection and design of ORC components is required (*Chatzopoulou et al., 2019*).
The expander is commonly retained the key components of the whole unit since its behavior sensibly affect the whole plant performance (*Moradi et al., 2021*). For ORC based power unit volumetric machine are generally preferred to work as expander with respect to the dynamic one due to the lower revolution speed and the capability to deal with off-design conditions and to tolerate two-phase working fluid (*Pantano e Capata 2017*).
Among the volumetric expanders, scroll machines are one of the most reliable candidates due to the high isentropic efficiency (*Dumont et al., 2018*), compactness and reliability (*Imram et al., 2016*). Due to these important properties, scroll expanders were widely characterized in literature for different applications, (*Song et al., 2015*). *Declaye et al. (2013),* experimentally analyzed an open-drive scroll integrated into an ORC plant with R245fa as working fluid achieving a maximum isentropic efficiency and power equal to 75.5% and 2.1 kW respectively. *Moradi et al., (2021)* tested a scroll for a small-sale ORC unit using R123a as working fluid finding that the maximum expander power is achieved at a pressure ratio higher than the ones which guarantee the highest isentropic efficiency. *Ziviani et al.* (2018), experimentally characterized a 5 kW oil free open-drive scroll expander with R245fa as working fluid, observing a maximum efficiency of 58% for a hot source inlet temperature equal to 110 °C and a revolution speed of 1600 rpm. (*Manolakos et al.*,2020) investigated the ORC plant performance when a series of two scroll expanders is installed. The results show that a maximum thermal efficiency equal to 10% is achieved while the expanders present an isentropic efficiency varying between 57% and 68%. *Suman et al.* (2017) tested a 5 kW oil free scroll expander in a revolution speed range equal to 400-2400 RPM, observing a maximum torque equal to 2 Nm when the machine rotate at 2400 RPM. ORC performances with two scroll expanders running in parallel are evaluated by *Kaczmarczyk et al.* (2020), who observed a maximum isentropic efficiency equal to 83%. *Liu et al. (2019).* experimentally analyzed a micro-ORC system based on scroll operating with R123. They observed the dynamic response of the ORC system operated by low-grade source at different load. The results show that the expander isentropic efficiency and the power output growth rate decreased with the load enhancement. *Campana et al.* 2019, tested a small-scale scroll expander operating in an ORC plant for low-temperature waste heat recovery, noticing that the highest performances are achieved by the unit when the superheating degree is between 10 K and 30 K.
Despite the huge effort made on the scroll characterization, there is a lack of analyses to the author best knowledge, when the hot source is represented by the exhaust gases of ICE. For this reason, an ORC plant equipped with a 1.5 kW hermetic scroll expander is developed to recover the thermal energy of the exhaust gases of a 3L Diesel engine. In these situations, in fact, the components of the recovery unit are forced to work in off design conditions if not in transient conditions, due to the high time variations of the enthalpy of the exhaust gas. The results allowed to assess the plant and expander performances as a function of operating parameters such as mass flow rate, expander intake temperature and pressure. Moreover, it is assessed the start-up time of the machine when the plant is run starting from ambient conditions.

2 EXPERIMENTAL TEST BENCH

The ORC-based power unit (Fig.1(a)) is developed and fully instrumented to recover the thermal power of the exhaust gases of an IVECO F1C Diesel engine (Fig.1(b)). R245fa is selected as working fluid of the ORC plant with a fluid charge equal to 6 kg. An ISO VG 68 POE oil is added (8% of the charge of

R245 fa) to lubricate the rotating machine. In Figure 1(c) the scheme of the ORC power unit is reported where the main ORC components can be recognized:

1) A gear pump with a volume capability of 22 L/min which elaborates the working fluid;
2) A tube and fins Heat Recovery Vapor Generator (HRVG), designed to reduce the gas side pressure drop. The heat exchange surface is equal to 0.9 m^2 from the R245fa side while from the exhaust gases one the area is 4 m^2. The heat recovery vapor generator is based on fin and tube technology. The working fluid moves inside the tubes whereas the exhaust gases flow across the finned channels. Thus, the presence of fins on the exhaust gas side leads to a higher heat exchange area with respect to the one corresponding to working fluid side.
3) A 1.5 kW hermetic scroll expander integrated with an electrical conversion;
4) A plate heat exchanger cooled by water as condenser; The heat transfer surface is equal to 4.46 m^2.
5) A 3 liters tank, employed to dampen the mass flow rate fluctuations.

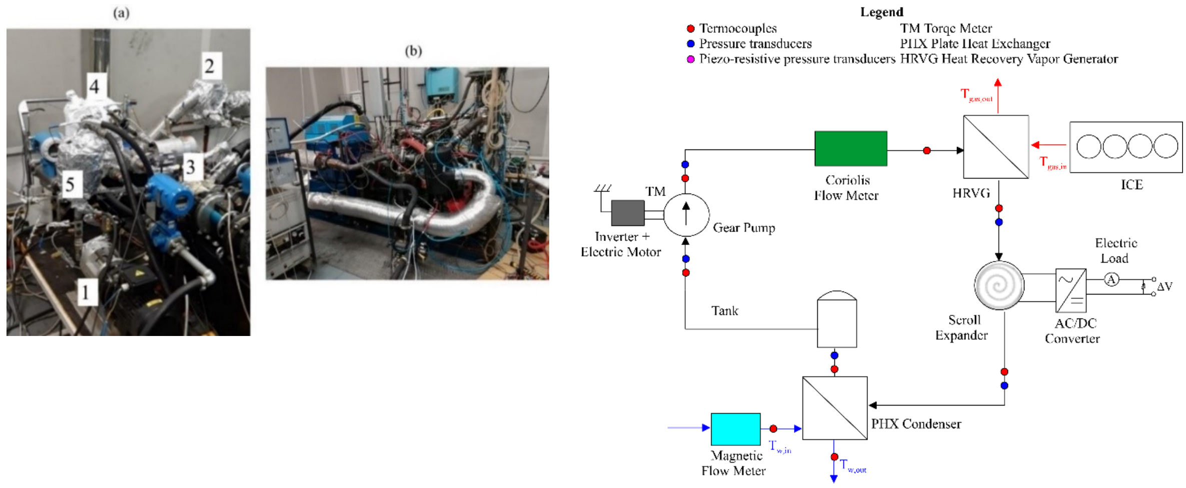

Figure 1: ORC test bench (a); IVECO F1C Diesel Engine (b), Scheme of ORC-based power unit.

Concerning the experimental apparatus, a Coriolis mass flow meter is adopted to measure the working fluid flow rate and a magnetic one for cooling water. Downstream and upstream each component, pressure sensors and thermocouples are located to evaluate their performance. A Torque meter is mounted between the electric motor and the pump to measure the whole mechanical power required. Concerning the expander, it shares with the electric generator the same shaft, being integrated into a hermetic case. The generator produces a three-phases voltage, converted in a DC voltage by a power conditioning system. It is dissipated through a variable electronic load able to measure the electric DC power produced. The uncertainties of measure of all the devices are reported in Table 1.

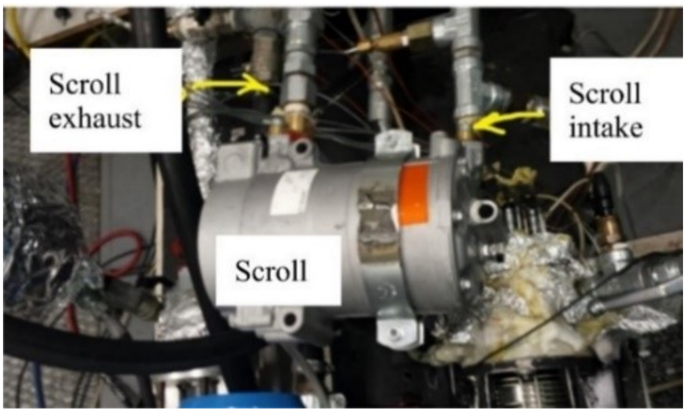

Figure 2: Detail of the Scroll Expander

Table 1: Uncertainties of Measurement.

Sensors	Quantity measured	Uncertainty
Coriolis Mass-flow rate	Working fluid mass flow rate	±0.15% of measured value
Magnetic mass flow rate	Water mass flow rate	±0.5% of measured value
Pump torque meter	Torque	0.02 Nm
	Revolution speed	1 RPM
T-Thermocouple	Working fluid temperature	0.3 °C
K-Thermocouple	Exhaust gases temperature	2.2 °C
Pressure sensor	Working fluid pressure	0.3 bar
Current sensor	DC current (electric load)	1% of FS-(0.1 A)
Voltage sensor	DC voltage (electric load)	0.6% of FS, 4.2 Volt
Frequency sensor	Electric signal frequencies f	Of reading 0.01%

3 EXPERIMENTAL RESULTS

The scroll expander and the whole plant were experimentally characterized for a wide range of mass flow rate (0.03 kg/s-0.09 kg/s) which was varied to follow the thermal power available at the HRVG.
For the experimental test, the mass flow rate of the exhaust gases varies between 0.08 kg/s and 0.11 kg/s while the exhaust gases temperature is in a range comprised between 240°C and 365°C. Based on these values the heat available from the exhaust gases varies between 19.2 kW and 40.8 kW. These quantities are reported in Table 2 together with the corresponding torque and revolution speed values. Concerning the heat sink, it is constituted by tap water whose temperature varies between 14°C and 20°C while the mass flow rate is comprised between 1.4 kg/s and 1.5 kg/s.
Thanks to the low scroll permeability, represented by the ratio between the mass flow rate sent by the pump and the pressure difference between scroll intake and exhaust side, the machine allows to ensure a proper plant pressurization also for small mass flow rates. Indeed, permeability represents the attitude of the expander (and hence of the plant) to be crossed by the working fluid and it depends on volumetric efficiency, intake volume and revolution speed of the expander (Fatigati et al., 2019). The permeability, it is not defined only by volumetric efficiency but also on the accordance between volumetric pump and expander revolution speed (Fatigati et al., 2021). The pump speed is related to the flow rate circulating inside the plant whereas the expander speed defines the expander intake pressure. Indeed, a volumetric expander presents a periodic behavior so the continuity of the flux is broken and the machine can be seen as a revolving valve. If the revolution speed raises the intake pressure diminishes whereas a pressure raises is observed when the revolution speed diminishes. Hence, besides the volumetric efficiency also the revolution speed plays a fundamental role on permeability and hence on plant pressurization (Fatigati et al., 2021).

Table 2: Operating range of heat source and Heat sink properties

Heat source properties		
Engine Torque [Nm]	60.5	100
Engine speed [rpm]	2609	3300
Exhaust gases mass flow rate [kg/s]	0.08	0.11
Exhaust gases temperature [kg/s]	243	362.4
Heat released by the exhaust gases [kW]	19.2	40.8
Heat sink properties		
Water temperature at condenser inlet [°C]	14	28
Water mass flow rate [kg/s]	1.4	1.6

Thus, if a volumetric expander is employed, the maximum pressure of the ORC unit, and consequently the evaporating pressure, is defined by the expander permeability (*Fatigati et al.,* 2019). This concept still applies also when a scroll machine is adopted, as it can be observed from Figure 3(a), which shows how the pressure at the HRVG quite coincides with the one at the scroll intake side, showing a growth with mass flow rate. The relationship between mass flow rate circulating inside the unit and pressure at the inlet of the expander must be taken into consideration for safety reasons and thermodynamic aspects: when the pressure changes, in fact, the thermodynamic characteristics of the cycle change with respect to a previous design. Concerning the pressure at the condenser inlet, representing also the pressure exerted by the circuit at expander outlet, it can be seen as it is independent from the mass flow rate, being basically defined by the condensing temperature reported in Figure 3(b), where also the values observed at expander inlet and outlet are reported. Observing Figure 3, it can be concluded that the low expander permeability ensures a proper pressure difference in every working condition. This involves (Figure 4(a)) that the expander can provide a noticeable power (400 W) also when the mass flow rate assumes the minimum value (0.03 kg/s). As Figure 4(a) shows, the expander power grows from 400 W to 750 W when the mass flow rate increases from 0.03 kg/s to 0.09 kg/s to follow the available thermal power at HRVG (Figure 4(b)).
This feature also appears particularly interesting to understand the plant behavior: when the thermal power available on the exhaust gases increases, the only way to limit the superheating degree or to favor a greater recovery is to increase the working fluid mass flow rates. When this happens, usually increasing the speed of rotation of the pump, the circuit reacts increasing the pressure of the circuit at the expander inlet (or at the HRVG) according to its permeability. So, the lower is the permeability the higher is the pressure difference at expander side for a certain mass flow rate entering the expander.
Expander speed is assessed through the measurement of the voltage frequency thanks to an oscilloscope. The expander rotates between 5100 and 10500 RPM and the driver of the speed variation is the pressure difference between the intake and exhaust side. The revolution speed is not constant due to the electric generator is not connected with an electric grid which involves that the expander rotates at fixed speed. The tested Scroll Machine rotates at high speed (5000-10000) because the electric load was set close to its minimum value. This was done with the aim to increase the permeability of the expander consequently avoiding that the expander intake pressure reaches too high values. In fact, the scroll machine presents a significant low permeability which is a positive factor in terms of plant performance. Nevertheless, if the machine rotates at low speed the pressure could reach too high value due the machine low permeability. For this reason, the electric load was reduced to provide an increase of the expander revolution speed. In the revolution speed interval between 6000-10000 RPM the power produced by the machine grows from 400 W to 824 W as the machine performs more cycle per second. Concerning the expander isentropic efficiency, it is evaluated as in (1) and it varies between 50% to 25 % when flow rate increases from 0.03 to 0.09 kg/s.

$$\eta_{exp} = \frac{P_{exp}}{\dot{m}_{WF}(h_{in} - h_{out,is})} \tag{1}$$

Differently from the power/mass flowrate relationships, the expander efficiency decreases with increasing mass flow rates. This means that when the mass flow rate rises, the expander intake pressure grows while the expander outlet pressure is the same (Figure 3(a)).
As a consequence, under-expansion phenomena occur, which involve a reduction of the isentropic efficiency. Indeed, the isentropic efficiency considers an isentropic expansion between the intake pressure and the exhaust pressure. The exhaust pressure is intended as the pressure measured at the expander outlet which hence represents the pressure which the circuit explain at the expander outlet. If the pressure inside the chamber when the exhaust ports open is different from the exhaust one two cases can take place:

1. If the pressure is higher than the exhaust one an isochoric expansion takes place thus the machine performs an under-expansion;
2. If the pressure is lower than the exhaust one an isochoric compression happens so the machine performs an over-expansion;

In both cases the isentropic efficiency diminishes but the case of under-expansion does not involve a power penalty; This aspect demonstrates as a more suitable reference transformation should be considered to represent the volumetric expander efficiency. Literature, in fact, universally considers the adiabatic-isentropic efficiency as reference for the expanders but, in reality, this reference does not match the ideal expansion which is adiabatic-isentropic till to a given pressure level and after this is ideally isochoric.

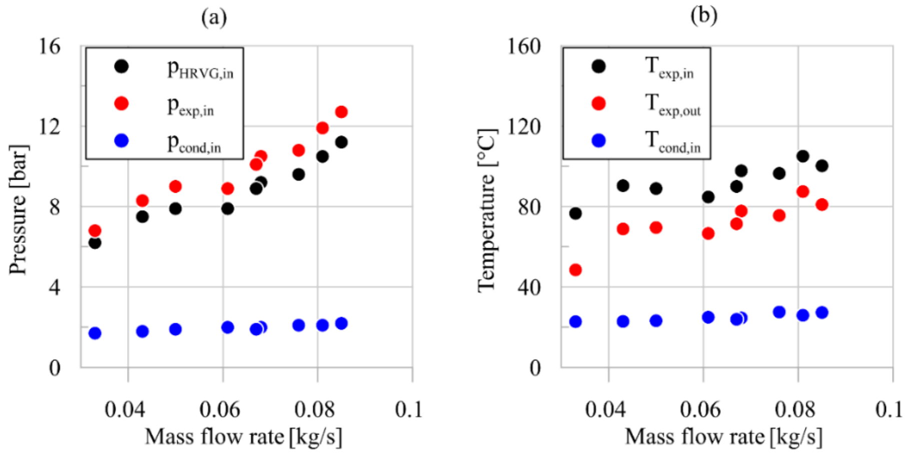

Figure 3: Pressure (a) and temperature (b) of ORC components as function of mass flow rate.

Concerning the whole ORC plant performance, the net power (P_{ORC}) and efficiency are reported in Figure 5(a) and (b) respectively. The net power is evaluated as the difference between the expander and pump power (2).

$$P_{ORC} = P_{exp} - P_{pump} \tag{2}$$

It can be seen from Figure 5(a) that P_{ORC} does not follow the same trend of P_{exp} but shows a flat region (0.05-0.08) kg/s where a maximum value of (400-500) W is reached. Indeed, the absence of a stable growth of P_{ORC} with $\dot{m}_{WF}$ is due to the increase of the Backwork Ratio BWR (3), which represents the quota of P_{exp} eroded by the pump:

$$BWR = \frac{P_{pump}}{P_{exp}} \tag{3}$$

As Figure 5(b) shows, the BWR varies from 25% to 50% when mass flow rate grows from 0.03 kg/s to 0.09 kg/s, reinforcing the concept that the pump has a significant effect on the unit performance. Indeed, Figure 5(b) shows as the pump power assumes non negligible power growing in the considered mass flow rate interval from 117 to 371 W.

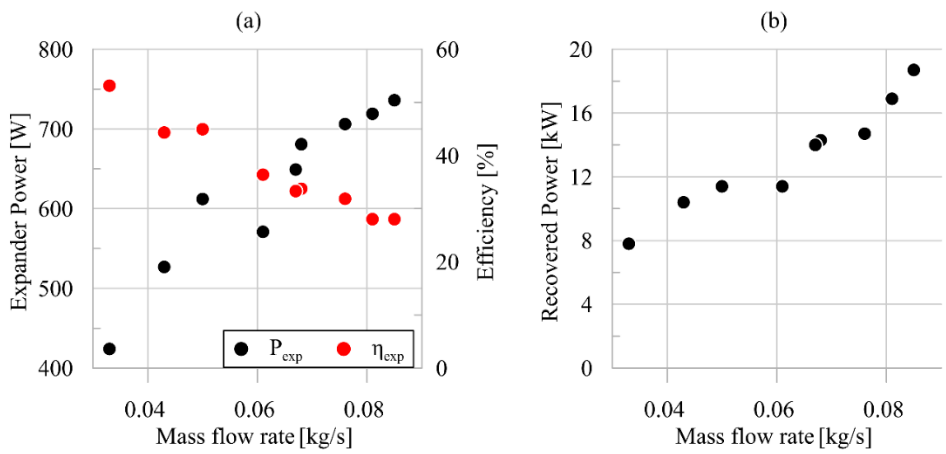

Figure 4: Expander power and efficiency (a) and Recovered power (b) variation as function of mass flow rate.

In Figure 5(a) the efficiency (Equation 4) is reported in the considered mass flow rate range, which varies between 4% and 2%. The decrease of η_{ORC} with mass flow rate is hence due to the behavior of P_{ORC}, which shows a decrease trend when $\dot{m}_{WF}$ is in between 0.07 kg/s and 0.09 kg/s.

$$\eta_{ORC} = \frac{P_{ORC}}{P_{rec}} \tag{4}$$

This is not due to the expander, whose power grows with $\dot{m}_{WF}$, but to the power erosion caused by the pump. Therefore, the effort performed to improve the ORC performance should be addressed to optimize the pump efficiency as it was outlined by *Bianchi et al.* (2017).

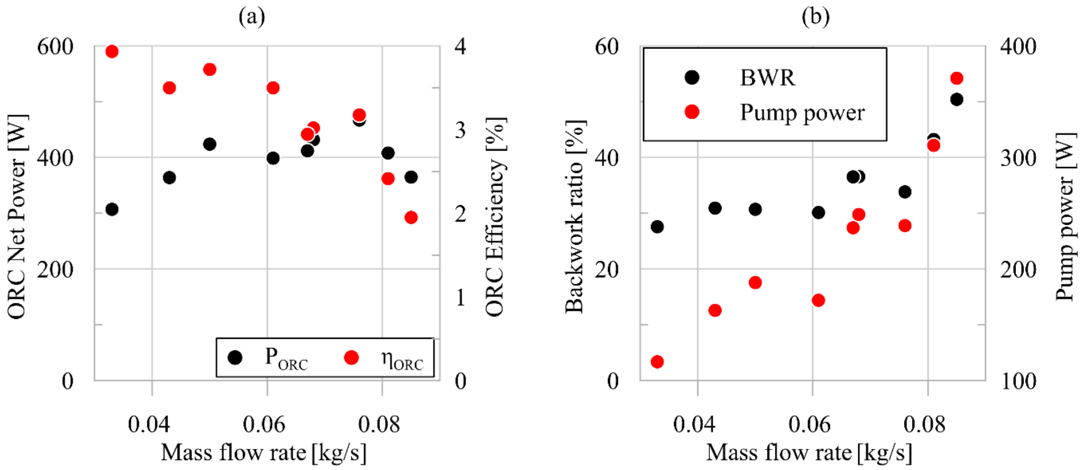

Figure 5: ORC net power and efficiency (a) and Backwork ratio (b) variation as function of mass flow rate.

Table 3. Measurement uncertainties of the considered quantities

Measured quantities	uncertainty
Inlet pump temperature	±0.3 °C
Inlet pump pressure	±0.3 bar
Outlet pump temperature	±0.3 °C
Outlet pump pressure	±0.3 bar
Inlet expander temperature	±0.3 °C
Inlet expander pressure	±0.3 bar
Outlet expander temperature	±0.3 °C
Outlet expander pressure	±0.3 bar
R245fa mass flow rate	± 0.15 % [kg/s]
Recovered thermal power	±1 % [kW]
Expander power	±5 % [kW]
Pump power	±1 % [kW]

In Table 3 the measurement uncertainties for the considered quantities are reported. The experimental results are in line with the one observed in literature for this application. *Alshammari and Pesyridis (2019)* observed an efficiency of 4% for the plant and 35% for a radial turbine employed as expander. *Di Battista and Cipollone (2017)* reports an ORC plant efficiency equal to 2.8 % adopting a 2.5 kW axial turbine. *Galindo et al. (2015)* sees a maximum efficiency of 6% for the ORC unit and of 38% for the expander. *Zhang et al. (2014)* notice a plant efficiency of 6.5% with a single screw expander with 57.8% of efficiency. *Cipollone et al. (2015)* experienced an efficiency of the ORC power unit between 3.8 and 4.8 % employing a sliding vane rotary expander whose power is comprised between 700 W and 2 kW.

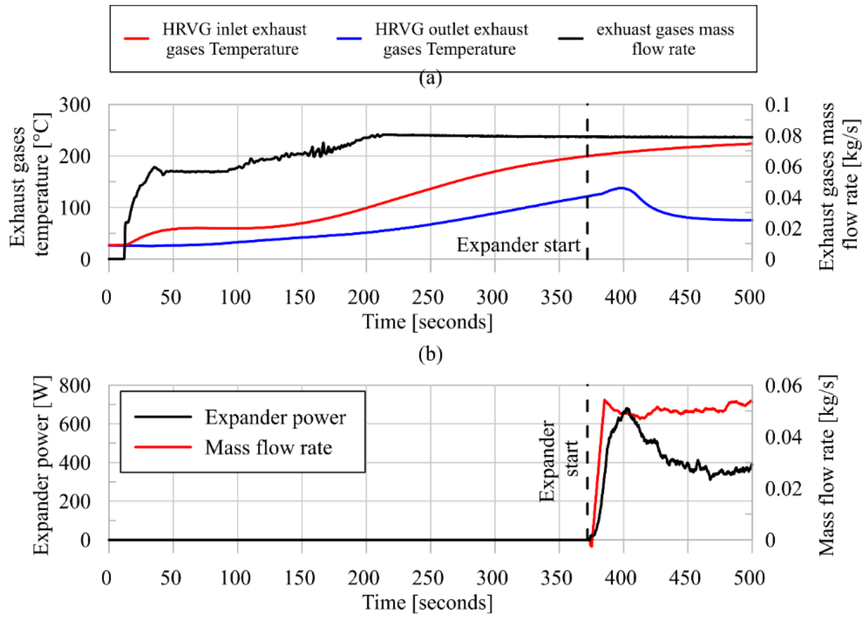

Figure 6: Analysis of Scroll expander activation.

The above reported results show that the scroll expander allows to achieve a high power for relatively small mass flow rates. Moreover, a further benefit presented by this machine is fast start-up when the plant is in initial cold condition. This can be seen in Figure 6(a) where the mass flow rate of the exhaust gases and the temperature at the inlet and outlet of HRVG is reported. It can be seen as the mass flow rate shows a sudden increase from 0 to 0.03 kg/s and after a gradual increase until to 0.08 kg/s when the steady state conditions are reached. This trend represents as the ICE starting from cold conditions reaches the operating point. In this sense can be observed the increase of the exhaust gases temperature entering the HRVG. In fact, after 500 s from the ICE ignition the regime value (220°C) is achieved for this operating point. Therefore, exhaust gases temperature at HRVG inlet is characterized by a first order dynamic behavior in accordance with the theoretical expectation. Concerning the exhaust gases temperature at the HRVG outlet, its trend is basically due to the thermal exchange at the HRVG. Indeed a sudden decrease can be seen when the pump start to provide the mass flow rate of R245fa (Figure 6(b)). Before the instant in which pump start to elaborate R245fa mass flow rate the difference between the exhaust gases inlet and outlet temperature is due to the accumulated heat by the HRVG metallic masses. Figure 6(b) provides another important information as it shows that after a warm-up period equal to 370 s during which the exhaust gases are flowing inside the evaporator, when the pump starts to provide the mass flow rate, the expander suddenly begins to rotate producing power (Figure 6(b)). This means that the expander starts to rotate prior that the ICE reaches the stationary condition for the considered operating point.
This is basically due to the high-pressure difference among the expander sides, which takes place as soon as the pump starts. Moreover, a further reason of this fast plant initialization is the capability of the machine to elaborate a two-phase working fluid. In fact, it is observed that in the initialization period the working fluid entering the machine is not completely vaporized. This property is particularly important for the current application: indeed, high initialization time could cause that the unit never comes into operation if the travel times of the vehicles are not long enough.

4 CONCLUSIONS

In the present paper, an experimental characterization of an ORC-based power unit fed by the exhaust gases of an internal combustion engine is performed. The attention is focused on the scroll expander present in the unit, and the main experimental results are the following:

1. the scroll machine shows a low permeability that ensures to achieve higher pressure difference, and consequently high power, also for small mass flow rate provided by the pump;

2. the low permeability of the scroll ensures to achieve proper pressure difference at expander sides also for low mass flow rates provided by the pump (0.03-0.09 kg/s) thus producing power up to 750 W;
3. the scroll efficiency reaches a maximum value of 50%; it decreases for the effect of the under-expansion when the mass flow rate grows;
4. the net power recovered and the efficiency decrease when mass flow rate rises due to the increase of the power required by the pump which demonstrates to be a critical component on which a further attention should be deserved; the maximum ORC efficiency is equal to 4%, when the mass flow rate is 0.03 kg/s. On the other hand, the maximum power (500 W) is obtained for a mass flow rate equal to 0.06 kg/s. These results lead to a trade-off optimization problem;
5. the scroll machine presents a short start-up phase: indeed, after a warm-up period of about 370 seconds, if the pump starts to elaborate mass flow rate, the expander begins immediately to rotate and produce power.

Therefore, the present work confirms how the scroll expander is a real candidate for this application. Moreover, a proper design of this machine would bring to higher performances, particularly related to fluid dynamic behavior and energy efficiency in the conversion into electrical form, considering that often these machines are not specifically designed to behave as expanders but simply reversed from the more common scroll compressors.

NOMENCLATURE

BWR	backwork ratio	out	outlet/exhaust
cond	condenser	$\dot{m}$	mas flow rate [kg/s]
exp	expander	ORC	Organic Rankine Cycle
HRVG	Heat Recovery Vapor Generator	P	power [W]
h	Enthalpy [J/kg]	rec	recovered
in	inlet	T	temperature [K]
is	isentropic condition	η	efficiency

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ACKNOWLEDGEMENT

The authors are fully grateful to Sanden S.p.A. for the technical support during this research activity. The research was developed in the framework of the European Project Development of efficient and environmental friendly LONG distance powertrain for heavy dUty trucks aNd coaches- LONGRUN.