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Organic Rankine cycle design and performance comparison based on experimental database

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HIGHLIGHTS

- An open-access database of ORC experiments is presented and released.
- ORC main parameters are discriminated and harmonized in the database frame.
- A selection of simplified ORC thermodynamic performance criteria is proposed.
- Database analysis shows actual Organic Rankine Cycle state-of-the-art performances.
- A statistical method for ORC performances analysis and comparison is introduced.

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ABSTRACT

The Organic Rankine Cycle (ORC) is a technology commonly used for low-grade thermal energy conversion in electricity. This technology is mature for large power scale and last research focused on small scale units for domestic or onboard applications. This paper presents an extensive open-access database of more than 100 ORC experiments collected from about 175 scientific literature references. Data harmonization and database frame are presented. Clear and consistent components and ORC performance criteria are proposed and applied to the data set of various ORC. An overview of the ORC experimental state-of-the-art is displayed and major trends are drawn. Efficiency of key components such as expanders and pumps are analyzed and used for ORC parametric optimization case study. Correlations of some parameters with ORC performances are statistically investigated, performance improvement of novel fluid or cycles is evaluated.

1. Introduction

The energy sector is facing major challenges in the upcoming century as energy demand, driven by the world population and economic growth, is rising and its major impact on the global warming issue needs to be addressed. Among the solutions to overcome these challenges, renewable energies and process energy efficiency could be partially fulfill by the use of the Organic Rankine Cycle (ORC) technology.

The organic Rankine cycle is a heat to power conversion technology used since the 19th century to transform energy from a variety of sources such as geothermal, solar, biomass or waste heat recovered (WHR) from the industrial process or internal combustion engines (ICE). Current range of commercial ORC goes from 10 kWe to 10 MWe converting heat sources between 80 °C and 300 °C, but this range is extending as new application are developed such as ocean thermal energy conversion, micro-CHP (combined heat and power) or vehicle engine heat recovery [1–3].

The organic Rankine cycle is derived from classic Rankine cycle, the pressurized working fluid is heated and vaporized by the heating fluid, expands in an expander to produce mechanical work, condensates at low pressure by the cooling fluid and is pumped back to close the cycle. The major difference with the classic Rankine cycle is the use of organic fluids as working fluid instead of water, the working fluid can be selected according to the heat source and usage [4].

Researches on ORC increased in the last decades, focusing on design optimization, fluid selection, expander technologies or

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Nomenclature

c_p	specific heat capacity
E	exergy
e	specific exergy
h	specific enthalpy
I	electric current
m	mass flowrate
P	pressure
Q	heat power
s	specific entropy
T	temperature
U	electric tension
V	volume flowrate
v	specific volume
W	power

Greek symbols

Γ	torque
Δ	difference
ε	exergetic efficiency
η	energetic efficiency
ξ	fluid saturated vapor slope
ρ	Spearman's coefficient
Φ	dissipations
φ	electric phase
Ω	rotating speed

Subscripts

ad	adiabatic
aux	auxiliaries
el	electric
eva	evaporator
exp	expander
gen	generator
hf	hot fluid
hr	heat recovery
hy	hydraulic
in	inlet
ind	indicated
int	internal
is	isentropic
max	maximum
me	mechanic
out	outlet
pp	pump
rec	recovery
sup	supplied
wf	working fluid
0	reference
II	second law (efficiency)

Acronyms

AC	Alternative Current
BWR	Back Work Ratio
CHP	Combined heat and power
DC	Direct Current
HEx	Heat Exchanger
HF	Hot Fluid
ICE	Internal Combustion Engine
IHE	Internal Heat Exchanger
ORC	Organic Rankine Cycle
VSD	Variable Speed Drive
WHR	Waste Heat Recovery

dynamic control. To support those research, many experimental benches were built for validation or models data-feeding purposes. Colonna et al. [1], Quoilin et al. [2] and Tchanche et al. [5] presented general review of the Organic Rankine Cycle technology history and future, typical industrial applications and market trends as well as ORC key components' types and issues such as expander, working fluid, heat exchanger or pump. Rahbar et al. [6] proposed a similar review with a focus on small power scale (5 kW to 5 MW), while Vélez et al. [7] proposed a focus on the economic and market trends of the ORC technology. Other reviews focused on ORC working fluid characteristics and selection criteria [4,8,9], or expander technologies, performances and modeling [4,10,11]. Lecompte et al. [12] proposed an exhaustive review of ORC architectures and advanced cycles such as recuperated, regenerative, flash or multi-pressure. Previous reviews presented a qualitative state of the art around the ORC technology. However, they present a given time picture, with a limited number of experimental references.

In the present paper, an open-access and collaborative database of ORC experimental work is presented in Section 2. This database aims to be as exhaustive as possible and extensive as it can be continuously updated with new references. It allows an objective review of ORC experiments through a factual and quantitative survey. Each ORC bench is tested in a different environment (heat and sink sources), for different objectives and analyzed by different methods. It results an addition of measurements, methods and definitions uncertainties on the cycle efficiency while the relative performance difference between two fluids, expander or cycle architecture can be rather small. Therefore, one of the major challenges to perform an objective comparison is to propose a clear data discrimination and classification, as well as harmonized performances criteria for both components and cycle that might be applied to the present database. Lecompte et al. [12] already proposed a number of clear ORC performance criteria using both energetic and exergetic analysis for open and closed heat sources [13], while Branchini et al. [14] proposed advanced indexes such as expander volumetric expansion ratio or total heat exchangers surface as size/economic parameters.

ORC design & parametric optimization is used at the early stage of projects to evaluate the potential of the ORC implementation. Contrary to advanced ORC modeling [15], parametric study uses mostly constant isentropic efficiency for pump and expander performances, but provides few references to justify the selected efficiency, while it has a large impact on the optimization process [16]. In the literature, expander isentropic efficiency ranging from 70 to 85% are used and pump isentropic efficiency from 60 to 90% with heat exchanger pinch point from 5 to 10 K [17–20]. The present work and database provides a number of experimental references for components efficiency, especially at small-scale. Examples of parametric optimization will be performed and presented in Section 3.

Many research on ORC focus on working fluids, expander and cycle architecture. Out of the numerical modeling evaluation of performance improvement, experimental validation can be complex. A common way is to compare different cases with the same test bench. Some authors compared different expanders with the same fluid and set-up [21–25]. Other compared different fluids or mixture proportions on the same ORC [26–36]. Some studies compared simple configuration with recuperated configuration

[25,37–43] or subcritical with transcritical conditions [32,36,44–47]. The present database enables to identify same test benches and track modifications in order to compare different cases, even if data are collected from different references. However, a given test bench can be design and optimized for a specific fluid, expander or configuration and introducing a bias on the different cases comparison. Therefore, the present work introduces in Section 4 an alternative way for performance comparison through the use of statistical tools.

2. Organic Rankine cycle experiments database

2.1. Database construction

Data are collected from scientific journals or conference papers and PhD theses. Only complete Rankine units are considered. Therefore, prototypes without expander (simulated with a valve), expander test benches (gas cycle) and flash cycles are excluded. Since this study focuses on organic Rankine cycle, steam Rankine cycles are excluded. It is decided to exclude CO_2 cycles as it has specific properties and is not an organic compound.

In references referring to identical test benches are identified and grouped to avoid double counts. However, changes of expander, fluid or heat sources are tracked as they can provide useful information and significant changes on unit performances and behavior. By the time of paper writing (beginning of 2017), the present database includes 175 scientific references corresponding to 102 unique ORC prototypes. Table 1 summarizes references according to the target application and the type of expander.

Both qualitative and quantitative information on prototypes were collected. Table 2 summarizes the database layout and type of data collected. For cycle numerical parameters (temperature, pressure, power...), both minimum and maximum reached values were collected. In addition, a second database gathers information at a specific running point – maximum efficiency and/or maximum power – but for a limited number of parameters: working fluid temperatures and pressures, expander & pump powers, heat & cold source temperatures, powers and flow rate.

Data are manually extracted from text, tables and graphs. Non-available parameters are calculated, when possible, using the *Coolprop* library [48] or left empty. A data reduction is processed on qualitative data in order to provide a uniform nomenclature (*e.g.:* fluid names, expander and pump technologies).

2.2. Data discrimination

Colonna et al. [1] proposed a terminology for ORC power plants classification, but there are currently no standards for ORC power or efficiency nomenclature and definitions. Therefore, each author uses his own nomenclature and definitions resulting in confusion and complex comparisons. Those choices are based on the instrumentation available and personal choices. Table 3 summarizes the main type of power stated and used in the database references for pump and expander with their measurement methods.

The isentropic power is the ideal power for a reversible adiabatic machine. The adiabatic power is derived from energy conservation and fluid enthalpy assuming an adiabatic machine. The indicated power is the pressure forces work only. For a pump, as the fluid is considered incompressible, it denotes to the hydraulic power and equals the isentropic power. The mechanical power is a pure power, but measurement can be complex or even impossible for hermetic expanders. The electrical power is easier to measure and allows fair comparison; however some references use mechanical power only as the final output.

Figs. 1 and 2 show pump and expander power flow diagram, locating the different powers. Δe_c denotes to the kinetic energy and Φ to the mechanical losses and frictions. Potential energy is neglected. Note that a variable speed drive (VSD) might be installed or not. For the pump, Landelle et al. [194] have shown that adiabatic power is not sufficient as motor losses are neglected, and appropriate as it is very temperature sensitive. Since mechanical power is never measured, electrical power will be preferred.

For expander, the relation between the final output power and the adiabatic power depends on the expander type and the expander adiabatic assumption (heat dissipation negligible with respect to the output power). As Lemort et al. [127] have shown, volumetric machines cannot be considered adiabatic (low power/size ratio). Especially as most experimental volumetric expanders are modified compressors, and compressors are designed to dissipate heat in order to reduce the compression work. Large turbo-expander are often considered as adiabatic machines, however this assumption should be validated for micro-turbines.

Based on those power definitions, a combination of efficiencies can be derived for the pump and the expander. The most common and useful expander efficiencies are listed below:

- Isentropic efficiency: $\eta_{is} = W_{ad}/W_{is}$
- Mechanical efficiency: $\eta_{me} = W_{me}/W_{is}$
- Electrical efficiency: $\eta_{el} = W_{el}/W_{is}$
- Generator efficiency: $\eta_{gen} = W_{el}/W_{me}$
- Adiabatic efficiency: $\eta_{ad} = W_{me}/W_{ad}$ for open-type; $\eta_{ad} = W_{el}/W_{ad}$ for hermetic

The adiabatic efficiency represents the thermal insulation effectiveness of the expander. For pump, only the electrical efficiency is considered. An empirical correlation for generator efficiency, function of the mechanical power, is derived. This correlation is later used to estimate expander electric power from mechanical power when not available. Therefore, electric power is used as the reference power for both pumps and expanders.

2.3. ORC performance criteria

As for powers, different definitions are used to describe ORC system efficiency. Most ORC efficiencies definitions are based on

Table 1
Summary of experimental ORC database references.

Target\expander	Piston	Rotary	Screw	Scroll	Turbine	Other & n/a
Biomass		[49,50]		[51,52]	[53–56]	
Gas		[57]				
Geothermal			[58,59]	[34,60–62]	[36–39,63–68]	
Solar		[42,69,70]	[43,71]	[46,47,72–88]	[89]	[90]
Solar combined				[91–93]	[33]	[94]
WHR	[95]	[24,96–102]	[26,27,45,103–109]	[22–24,110–128]	[28,129–139]	[140–142]
WHR from ICE	[143–146]	[147–150]	[151–153]	[147,154–161]	[30,32,162–165]	[166]
Not indicated		[21]	[167–171]	[21,25,29,35,172–182]	[41,44,183–191]	[21,40,192,193,31]

Table 2
Experimental ORC database layout.

General info	Working fluid	Cycle conditions	Heat & cold source	Pump	Expander	Heat exchangers	Lubrication	Additional components
Country	Name	Expander inlet temperature	Energy source	Technology	Technology	Evaporator Type	Oil proportion	Subcooler
University	Category	Superheating	Temperature	Driver	Generator	Evaporator Area	Injection type	Filter
Target application	Critical temperature	Pump inlet temperature	Heat power	Control	Control	Condenser Type	Separator	Vapor Tank
Specificity	Critical pressure	Subcooling	Flow rate	Nominal power	Nominal power	Condenser Area	Pump	Liquid Tank
CHP	Saturation slope (ξ)	High pressure	Heat transfer fluid	Shaft speed	Built-in volume ratio	Internal HEx	Tank	Other
		Low pressure		Flow rate	Swept volume		Cooler	
		Net power		Powers	Shaft speed		Filter	
		Cycle efficiencies		Efficiencies	Pressure ratio			
					Powers			
					Efficiencies			

Table 3
Summary of power types.

Power name	Equation	Instrumentation required
Isentropic power	$W_{is} = \dot{m} \cdot [h(T;P)_{in} - h(s_{in};P_{out})]$	Pressure and temperature sensors, flow-meter
Adiabatic power	$W_{ad} = \dot{m} \cdot [h(T;P)_{in} - h(T;P)_{out}]$	Pressure and temperature sensors, flow-meter
Indicated power (exp.)	$W_{ind} = \dot{m} \cdot \int_{in}^{out} v \cdot dP$	Internal pressure sensors, tachometer
Hydraulic power (pp.)	$W_{hy} = \dot{V} \cdot \Delta P$	Pressure and temperature sensors, flow-meter
Mechanical power	$W_{me} = \Gamma \cdot \dot{\Omega}$	Torque-meter and tachometer
Electrical power	$W_{el} = U \cdot I \cdot \cos(\varphi)$	Power-meter/voltmeter

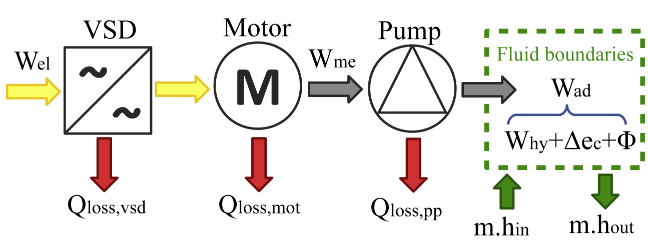

Fig. 1. Pump power flow diagram.

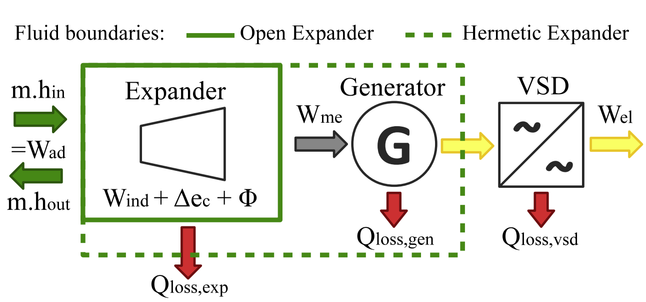

Fig. 2. Expander power flow diagram.

the same template as Eq. (1). The ORC net power output W_{ORC} is the difference between power produced by the expander and the power consumed to run the cycle. Components power type used to compute the net power must be consistent. Electrical power is preferred, for the reasons cited previously. In this work, the auxiliaries power W_{aux} (heat and cold fluid pumps, fans...) are not considered since they are not related to constrains and choices exterior to the ORC.

$$\eta_{ORC} = \frac{W_{ORC}}{Q_{heat}} = \frac{W_{exp} - W_{pp} - W_{aux}}{Q_{heat}} \tag{1}$$

In some references, the pump consumption W_{pp} is assumed small enough compared to the expander power W_{exp} to be neglected. Quoilin et al. [2] introduced the back work ratio (BWR) to compare pump with expander power and showed that pump cannot be neglected for ORC especially when working fluid critical temperature is low and ORC operates close to the critical pressure. Since only a few references provide the pump electrical consumption, the ORC gross electrical power $W_{ORC,gross} = W_{exp,el}$ is also used with the ORC net electrical power $W_{ORC,net} = W_{exp,el} - W_{pp,el}$ for the database analysis.

The heat power Q_{heat} is the system input power and depends on the system boundaries and input/output. ORC technology has different applications that can be divided in two categories: the closed heat sources (biomass, solar) and open heat sources (WHR, geothermal) [13]. Fig. 3 shows system scheme and T-Q diagram of closed and open heat sources.

In closed sources, the input is a heat flow Q_{sup} supplied to the ORC. The hot fluid (HF) inlet depends on the hot fluid outlet and the heat flow. In open sources, the input is a hot stream characterized by its mass flow rate (m_{HF}), specific heat (c_P) and temperature ($T_{HF,in}$). This input is not influenced by the hot fluid outlet, which is not recovered. Therefore, the supplied heat power Q_{sup} is lower than the maximum heat power Q_{max} that could be recovered if the stream was cool down to the reference temperature T_0.

In addition, a distinction can be made between the internal and external input [12]. For internal input, the heat power is the working fluid enthalpy variation between the evaporator inlet and outlet. For external input, this is the hot fluid enthalpy variation. For heat power, the difference between internal and external input is negligible and is only related to the evaporator thermal insulation. However, the difference can be significant when considering the exergy E_{heat} as the system input instead of Q_{heat}.

DiPippo [195] and Schuster [196] explained that exergy is more appropriate to evaluate thermodynamics performances. The specific exergy of a streaming fluid is defined in Eq. (2). The subscript 0 denotes the reference conditions, for ORC we will use the atmospheric pressure and the cold source inlet temperature as the reference. Table 4 sum-up the different types of system inputs and the related ORC efficiencies, assuming hot fluid constant specific heat.

$$e = h - h_0 - T_0 \cdot (s - s_0) \tag{2}$$

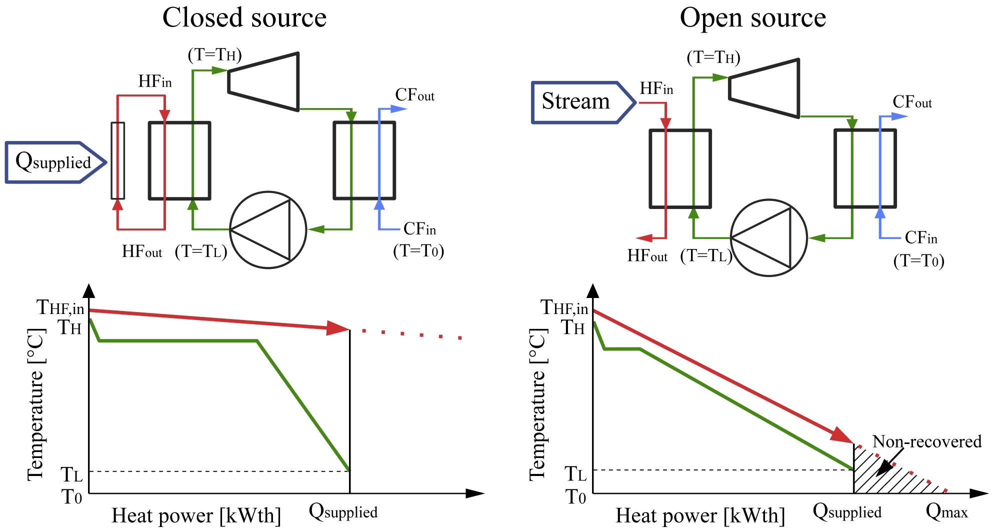

Fig. 3. Scheme and temperature-heat diagram for closed and open heat source types.

Table 4
Summary of ORC system inputs for efficiency definition.

	Energy	Exergy
Internal	$Q_{int} = \dot{m}_{wf} \cdot (h_{eva,out} - h_{eva,in})$ $\eta_{int} = W_{ORC}/Q_{int}$: ORC intern efficiency	$E_{int} = \dot{m}_{wf} \cdot (e_{eva,out} - e_{eva,in})$ $\varepsilon_{int} = W_{ORC}/E_{int}$: ORC intern exergetic efficiency
Closed source	$Q_{sup} = \dot{m}_{hf} \cdot c_{P,hf} \cdot (T_{HF,in} - T_{HF,out})$ $\eta_{th} = W_{ORC}/Q_{sup}$: ORC thermal efficiency	$E_{sup} = \dot{m}_{hf} \cdot (e_{HF,in} - e_{HF,out})$ $\varepsilon_{ORC} = W_{ORC}/E_{sup}$: ORC exergetic efficiency
Open source	$Q_{max} = \dot{m}_{hf} \cdot c_{P,hf} \cdot (T_{HF,in} - T_0)$ $\eta_{rec} = W_{ORC}/Q_{max}$: ORC recovery efficiency	$E_{max} = \dot{m}_{hf} \cdot (e_{HF,in} - e_{HF}(T = T_0))$ $\varepsilon_{rec} = W_{ORC}/E_{max}$: ORC exergetic recovery efficiency

The exergy requires precise data on the fluid properties to be calculated. We only consider the thermal exergy for the hot fluid. When assuming a constant specific heat, the hot fluid specific exergy is expressed as:

$$e_{hf} = \int_{T_0}^{T} c_P \left(1 - \frac{T_0}{T}\right) dT = c_P \left(T - T_0 - T_0 \cdot \ln\left(\frac{T}{T_0}\right)\right) \tag{3}$$

Therefore, the supplied exergy of the hot fluid is:

$$\begin{aligned} E_{sup} &= \dot{m}_{hf} \cdot c_P \left[(T_{HF,in} - T_{HF,out}) - T_0 \cdot \ln\left(\frac{T_{HF,in}}{T_{HF,out}}\right) \right] \\ &= Q_{sup} \left[1 - T_0 \cdot \frac{\ln(T_{HF,in}/T_{HF,out})}{T_{HF,in} - T_{HF,out}} \right] \end{aligned} \tag{4}$$

In the same way, we can derive the maximum exergy of the hot fluid:

$$E_{max} = Q_{sup} \left[\frac{T_{HF,in} - T_0(1 + \ln(T_{HF,in}/T_0))}{T_{HF,in} - T_{HF,out}} \right] \tag{5}$$

Additionally, we can introduce the heat recovery efficiency η_{hr} [20] and the hot exergy recovery efficiency ε_{hr}:

$$\eta_{hr} = \frac{Q_{sup}}{Q_{max}} = \frac{(T_{HF,in} - T_{HF,out})}{(T_{HF,in} - T_0)} \tag{6}$$

$$\varepsilon_{hr} = \frac{E_{sup}}{E_{max}} = 1 - \frac{T_{HF,out} - T_0(1 + \ln(T_{HF,out}/T_0))}{T_{HF,in} - T_0(1 + \ln(T_{HF,in}/T_0))} \tag{7}$$

With those assumptions, ORC exergetic efficiencies (ε_{ORC} & ε_{rec}) can be computed for a large number of database references, as it only requires heat power and inlet/outlet temperatures of the hot fluid. When the cold source inlet temperature is not available, 293 K is taken as the reference. The thermal loss of evaporators will be assumed negligible and therefore $Q_{int} \approx Q_{sup}$. The ORC internal exergetic efficiency ε_{int} is not considered, as it does not take into account evaporator exergy destruction.

For closed source applications, the target is to maximize the ORC thermal efficiency η_{th} with heat power as system input. An ideal closed source has an infinite heat capacity m.c$_P$ and the ideal cycle is the Carnot cycle. In this ideal case, the hot fluid temperature glide tends toward zero and the ORC exergetic efficiency ε_{ORC} tends toward the second law efficiency η_{II}, which is the ratio of the thermal efficiency η_{th} and the Carnot efficiency (Eq. (8)). The second law efficiency is useful to compare the degree of perfectness of the ORC without the influence of the heat source temperature. However, the thermal efficiency η_{th} is preferred when the heat source temperature is a system optimization parameter.

$$\lim_{T_{HF,out} \to T_{HF,in}} \left(\frac{W_{ORC}}{E_{sup}}\right) = \frac{W_{ORC}}{Q_{sup}(1 - T_0/T_{HF,in})} = \frac{\eta_{th}}{\eta_{Carnot}} = \eta_{II} \tag{8}$$

For open source applications, the target is to maximize the power extracted from the hot stream. The ORC exergetic recovery efficiency ε_{rec} is more appropriate than the ORC recovery efficiency η_{rec} to compare systems with different heat source temperature. DiPippo [197] shows that ideal cycle for open sources is a triangular cycle. Finally, we keep three types of efficiency for ORC comparison: η_{th}, ηII and ε_{rec}. To have a significant number of data, they are computed with the ORC gross power.

2.4. Database overview

Prototypes from the database cover a wide range of conditions, going from a few Watts to 1 MW of gross output power with hot source temperatures ranging from 60 to 675 °C. Fig. 4 shows a hot temperature – power map of the references sorted by target application. 23% of the ORCs don't have a specified application. Among the rest, 62% are dedicated to waste heat recovery, including 19% specifically for ICE WHR. 23% are for solar application, including 6% of solar combined with another heat source, 10% for geothermal, 3% for biomass and 1% for gas (see references in Table 1). 13% of ORCs are dedicated to combined heat & power (CHP) production, but they are mostly closed source applications: 100% of the biomass or gas and 30% of the solar targeted applications aim to produce CHP.

In practice, most references use controllable and flexible heat source: 48% use electric heaters and 20% oil or gas burner versus 14% for waste heat (including 11% of engine waste heat), 9% for solar field, 5% biomass, 2% for geothermal stream. To transfer the heat, water or glycol-water is widely used (54%, including 8% of steam) as well as thermal oil (27%) or exhaust gas/air (15%). In the same way, to condense the working fluid, 36% use cooling tower, 34% water flow (sea or tap) and 27% air chillers. 88% of the references use water or water-glycol mixture as a transfer fluid, but 11% performs direct condensation usually with air chillers.

Fig. 5 shows the evolution of refrigerant categories used as ORC working fluids over time. CFC quickly declined in the 80s due to Montreal protocol and mostly replaced by HCFC in the 2000 decade. HFC are now the dominant refrigerants even if HCFC are still used, mostly in Asia. Recently, HFO (R1233zdE) started to be tested on ORCs [29–31] and research on fluid mixtures increased [34,35,75,128]. Overall, 52% of the working fluid used are HFC, 20% HCFC, 7% Hydrocarbons, 6% HFE, 4% Mixtures, 2.5% PFC, 2.5% CFC, 2.5% HFO and 5% of others. There are over 30 different fluids, but only three fluids are used two times out of three: R245fa (38%), R123 (18%) and R134a (7%).

The number of experimental references can be a good track of R&D investments on the ORC technology. The 2011–2015 periods show a strong increase in the number of references, but a decline is observed in 2016. However, by the time of paper writing (beginning of 2017), not all 2016 references may be available. In addition, 18 references are extracted from the International Seminar on ORC Power Systems from 2015 and increase the 2015 peaks. A bibliometric analysis on the global ORC scientific literature shows a similar, but softer trend.

Fig. 6 shows the maximum operating condition of those working fluids in the cycle. In average, the reduced temperature – ratio over the critical temperature in Kelvin – is about 0.9 at the evaporator outlet, and the reduced pressure is about 0.4. Four ORCs running at supercritical conditions are reported [32,45,47,190].

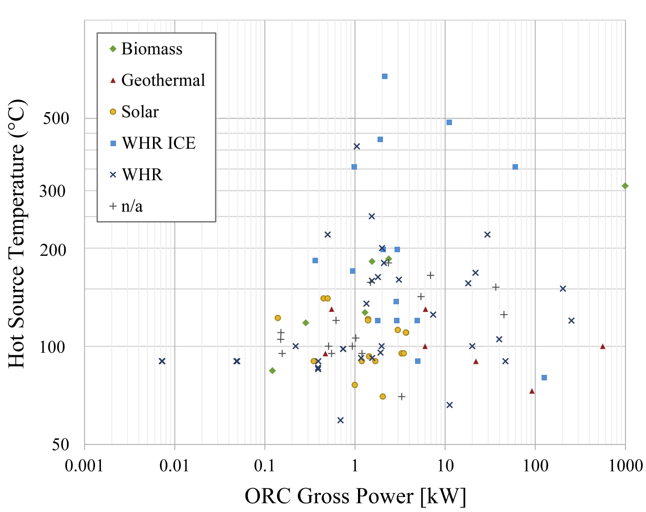

Fig. 4. Hot source temperature and ORC gross power map of the references.

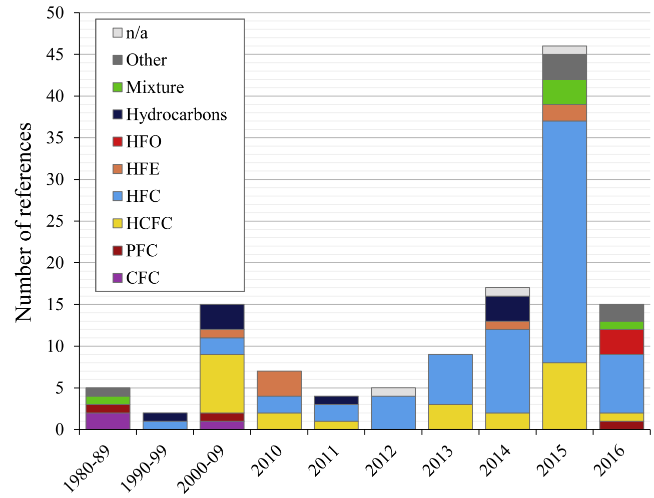

Fig. 5. Type of refrigerant used as working fluid over time in the referenced units.

3. Data for ORC design

3.1. Components performances and analysis

The expander is the most investigated ORC components and many different technologies are tested. A fourth of the references are turbo-expanders, for three quarters of them radial type. Others technologies are volumetric expanders. Scroll is the most used (45%), both hermetic and open-drive types, followed by screw and rotary expanders (12% each) plus a few references of piston or swash-plate piston expanders.

The adiabatic efficiency of expanders (see Section 2.2) has an upper quartile at 77% and a lower quartile at 35% for an average of 56%, whatever the expander technology. Therefore, isentropic efficiency and adiabatic power are not sufficient to evaluate and compare expanders.

As described in Section 2.2, an empirical correlation for generator efficiency is used to derive electrical power from mechanical power when it is not available (Fig. 7). When electric generator is used, 41% of them are alternative current (AC) synchronous generators, 35% AC asynchronous and 24% direct current (DC). Fig. 8 shows the expander electric power-efficiency map.

Expander efficiency is strongly correlated with its power scale. As expected by Quoilin et al. [198] there is an optimal technology depending on the power scale. Scroll expander seems more adapted below 5 kWe, screw expander for the 5–50 kWe scale and turboexpander above. Rotary technology is in the same power scale than scroll, but efficiency is lower.

Fig. 9 shows boxplots of expander pressure ratio and shaft speeds for the main technologies. Turboexpander covers the all range of pressure ratio; however its higher shaft speed requires high speed generator or speed reduction. If scroll, rotary and piston have the same power scale range, each one covers a different pressure ratio range. To control the expander speed, there are two main strategies. Two thirds use the expander load with electrical resistances or mechanical brakes, the other third use speed control with variable frequency drive or connection to the electrical grid and mainly AC asynchronous generators.

Different lubrication strategies are implemented depending on expander technology (Fig. 10). Turbines are mostly oil-free expanders, but bearings lubrication may be necessary. Some authors used working fluid liquid injection for bearings lubrication [30]. Some volumetric expanders, especially scrolls, run without lubrication. When oil is added, there are two main strategies. Create a

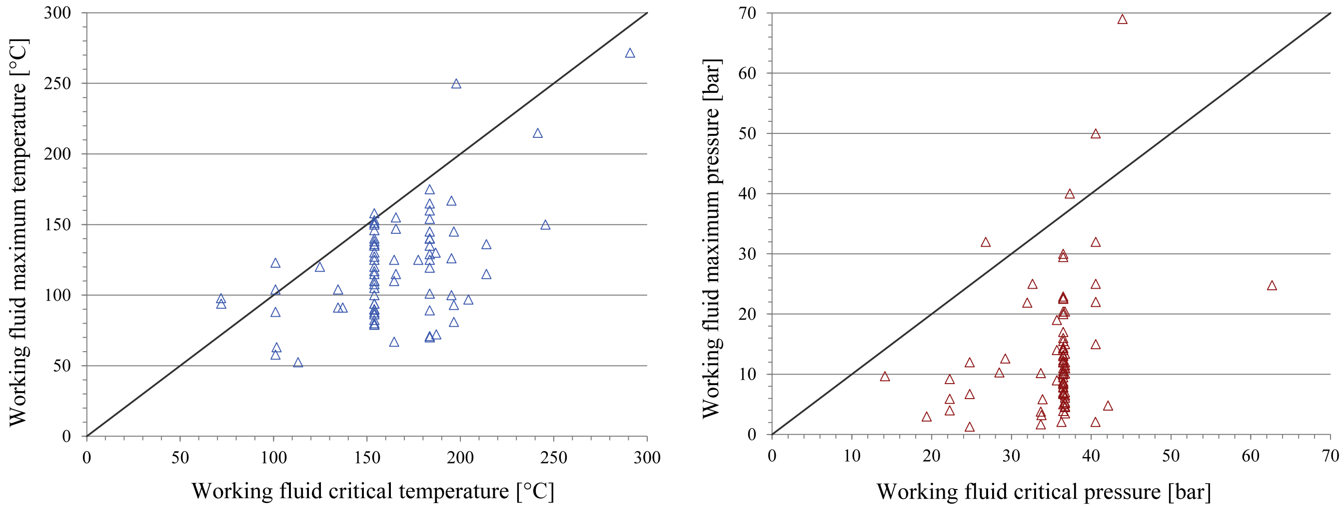

Fig. 6. Working fluids maximum operating conditions on the ORC.

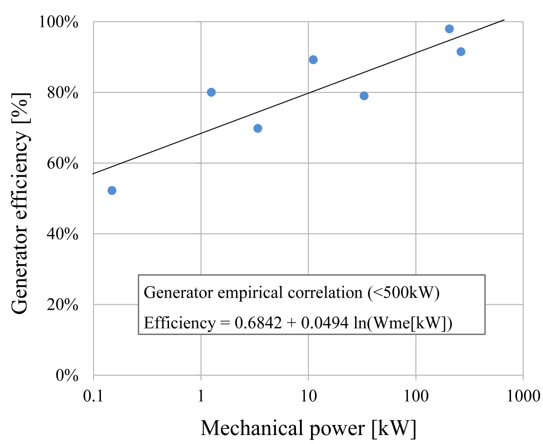

Fig. 7. Expander generator efficiency empirical correlation.

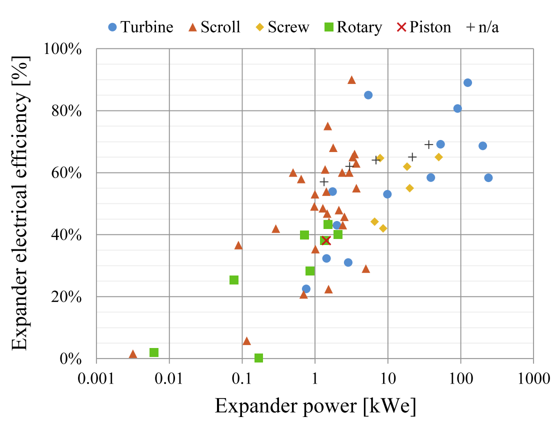

Fig. 8. Electric power and efficiency map of expander references.

mixture of working fluid and oil that circulates in the ORC – in average 5% of oil mass fraction. Use an additional oil-loop. This solution is more complex as it requires a secondary oil circuit to separate the oil from the fluid and pump it back to the expander inlet or bearings. However, it reduces the oil fraction circulation in the rest of the ORC. 43% of the lubricated ORCs (circulating mixture + additional oil-loop) added an oil pump, 39% an oil-fluid separator, 13% an oil tank, 8% a filter and 5% a cooler.

To study lubrication effect on expander efficiency, we select data of scroll, screw and rotary expanders in the 0.5–10 kWe scale. The average expander efficiency increases from 47% to 53% when using lubrication. But it is based on 32 data and therefore this hypothesis is not statistically strong.

Many different pump technologies can be used on ORC, we group them in 3 categories: reciprocating (56%) grouping diaphragm, piston and plunger pumps; rotary (11%) grouping gear, rotary piston and rotary vane pumps; and centrifugal (30%) mono or multistage pumps. A pumpless concept is experienced by Yamada et al. [121] or Gao et al. [181]. 95% of the pumps are driven by electric motor, Larjola and Turunen et al. [129,130] directly used the expander mechanical shaft to drive the pump.

Fig. 11 shows a pumps operating conditions map with the maximum mass flow rate and the maximum pressure lift. Fig. 12 shows a pumps performances map. Centrifugal pump performs better above 1 kW of hydraulic power and are the dominant technology for flow rates above 1 kg/s. They can handle high pressure increase of 20 bar, even at low flow rate. Reciprocating pump perform better below 1 kW of hydraulic power and can handle very high pressure lift. As for expanders, pump electric efficiency is correlated with its power scale. Pump low electrical efficiency could be explained by poor motor performances, especially if running at part-load and speed [194].

The ORC is usually composed by at least two heat exchangers: the evaporator and the condenser. Some authors added heat exchanger in series (pre-heater) or in parallel. A common supplementary heat exchanger is the internal heat exchanger (IHE) or regenerator. The IHE recovers expander exhaust superheat to pre-heat the pressurized liquid before entering the evaporator.

Fig. 13 shows the types of heat exchangers used for those 3 main ORC components. Plate heat exchanger is the dominant technology, almost 75% of evaporators and condensers. Then, comes shell & tubes and various types of tube exchangers. Only 23% of ORCs have an IHE. As expected by Bao et al. [4], the IHE usage increase with the organic fluid dryness, as they have more de-superheating potential from the expander outlet.

Auxiliaries are often added to the simple ORC configuration. 58% of the authors declare using a liquid tank at the condenser outlet to absorb the charge variation. 18% uses working fluid filter. 12% use vapor separator or vapor tank at the evaporator outlet to avoid liquid droplets at the expander inlet. To prevent cavitation 6% use a subcooler prior to the pump and 3% a pre-feed pump. A pulsation dumper can be added (3%) to reduce reciprocating pumps pressure and flow rate pulsations.

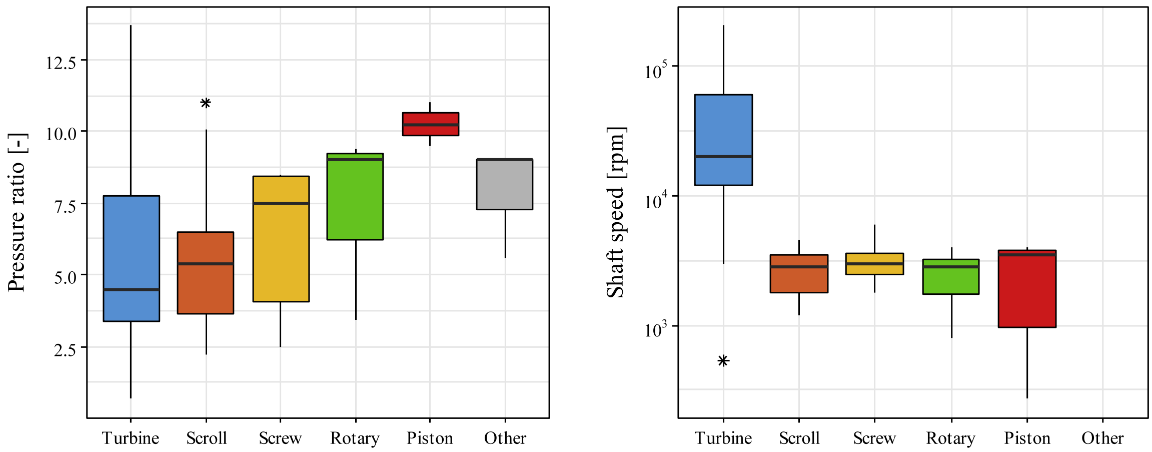

Fig. 9. Expander pressure ratio and shaft speed.

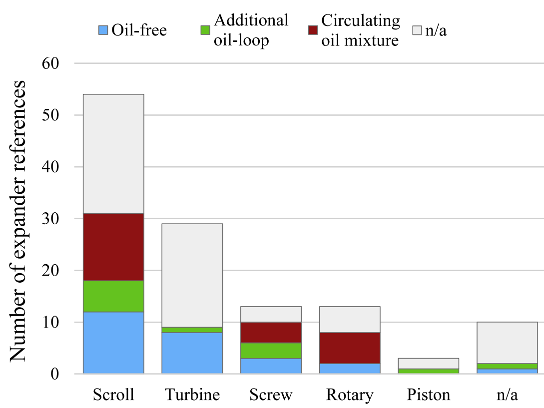

Fig. 10. Expander lubrication strategies.

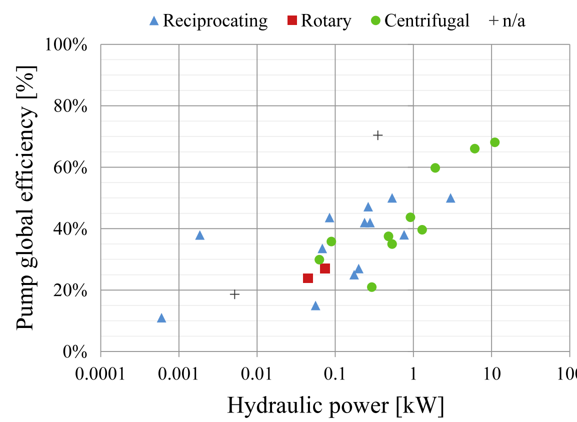

Fig. 12. Pumps performances map.

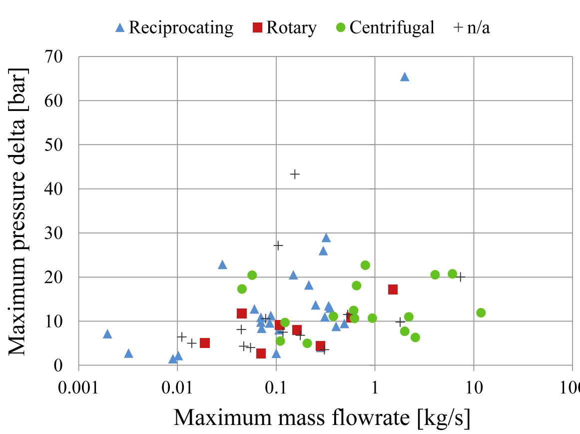

Fig. 11. Pumps operating conditions map.

3.2. ORC parametric optimization

Two different case study are presented and analyzed through a parametric optimization of a small scale ORC design. The first case is a closed heat source (e.g. thermal solar) able to deliver 50 kWth at 125 °C. The second case is an open heat source (e.g. waste heat) of 0.5 kg/s of pressurized hot water at 125 °C. The heat sink is supposed to be a quasi-isothermal source at 20 °C. The evaporator pinch is set to 10 K, the condenser pinch to 5 K. The regenerator configuration is also tested for the closed heat source, with a pinch set to 5 K. The subcooling is set to 10 K and the minimum superheating to 5 K. Since output power is expected to be in the kW range, the expander isentropic is set to 65% according to Fig. 8. For the pump efficiency, an empirical correlation is created from pump data (cf. Fig. 12): $\eta_{is,pp} = 0.55 + 0.15\,\log_{10}(W_{hy,pp})$ with pump hydraulic power in kW, valid for W_{hy} from 0.05 kW to 10 kW.

For closed heat source, the optimization criterion is the second law efficiency, while the evaporation pressure and working fluid flow rate are used as optimization variables. For open heat source, the evaporation pressure, expander inlet temperature and working fluid flow rate are the optimization variable and the exergetic recovery efficiency the optimization criteria. Six fluids are tested: R134a, R152a, R236fa, R245fa, R404a and SES36. Table 5 summarizes the optimization results for the different cases and fluids. It presents the relative pressure, working fluid flow rate and expander inlet superheating at the optimum as well as the net output power and related efficiency.

R245fa is found to be the optimum fluid for closed heat source, both with the simple or regenerated configuration, with a net second law efficiency of 33.8% and 39.6% respectively. In both cases, the expander pressure ratio is found to be around 7.5 for gross output power around 5 kW, therefore small-scale screw expander or a turbine could be found and used. The pressure difference is only 14 bar, therefore either reciprocating or centrifugal pump technology can be considered. When considering an open heat source, the optimum fluid is found to be R404a running at transcritical conditions, with a relative pressure of 1.35. In this case, the expander has a pressure ratio of 3 with a gross output power of 12 kW. There-

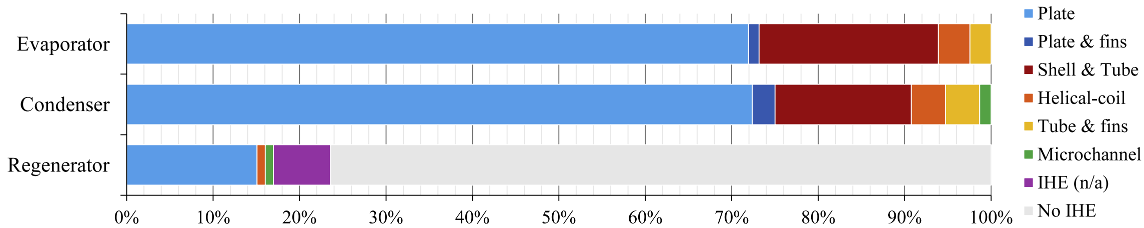

Fig. 13. Heat exchanger technologies.

Table 5
Parametric optimization results (most efficient case in bold).

	Closed source without regenerator	Closed source with regenerator	Open source
R134a	rP: 0.99 – m: 0.23 kg/s – T_{sh}: 14.4 K W_{net}: 3.5 kWe – η_{II}: 26.4%	rP: 1.02 – m: 0.23 kg/s – T_{sh}: 13.9 K W_{net}: 3.8 kWe – η_{II}: 29.0%	rP: 0.63 – m: 0.47 kg/s – T_{sh}: 36.0 K W_{net}: 6.9 kWe – ε_{rec}: 21.4%
R152a	rP: 0.85 – m: 0.16 kg/s – T_{sh}: 10 K W_{net}: 3.9 kWe – η_{II}: 29.4%	rP: 0.75 – m: 0.16 kg/s – T_{sh}: 16.5 K W_{net}: 4.1 kWe – η_{II}: 31.2%	rP: 0.48 – m: 0.34 kg/s – T_{sh}: 24.6 K W_{net}: 6.9 kWe – ε_{rec}: 21.5%
R236fa	rP: 0.74 – m: 0.26 kg/s – T_{sh}: 5 K W_{net}: 3.9 kWe – η_{II}: 29.8%	rP: 0.33 – m: 0.32 kg/s – T_{sh}: 41.9 K W_{net}: 3.6 kWe – η_{II}: 27.1%	rP: 0.37 – m: 0.69 kg/s – T_{sh}: 5 K W_{net}: 7.2 kWe – ε_{rec}: 22.4%
R245fa	rP: 0.43 – m: 0.20 kg/s – T_{sh}: 5 K W_{net}: 4.5 kWe – $\boldsymbol{\eta_{II}}$**: 33.8**%	rP: 0.43 – m: 0.23 kg/s – T_{sh}: 5 K W_{net}: 5.2 kWe – $\boldsymbol{\eta_{II}}$**: 39.6**%	rP: 0.20 – m: 0.50 kg/s – T_{sh}: 5.2 K W_{net}: 7.1 kWe – ε_{rec}: 22.0%
R404a	rP: 1.41 – m: 0.27 kg/s – T_{sh}: 42.9 K W_{net}: 2.6 kWe – η_{II}: 19.6%	rP: 1.66 – m: 0.33 kg/s – T_{sh}: 42.9 K W_{net}: 2.8 kWe – η_{II}: 21.0%	rP: 1.35 – m: 0.87 kg/s – T_{sh}: 34.2 K W_{net}: 7.9 kWe – $\boldsymbol{\varepsilon_{rec}}$**: 24.5**%
SES36	rP: 0.28 – m: 0.22 kg/s – T_{sh}: 5 K W_{net}: 4.1 kWe – η_{II}: 31.4%	rP: 0.27 – m: 0.27 kg/s – T_{sh}: 6.2 K W_{net}: 5.1 kWe – η_{II}: 38.7%	rP: 0.12 – m: 0.56 kg/s – T_{sh}: 6.9 K W_{net}: 6.4 kWe – ε_{rec}: 19.8%

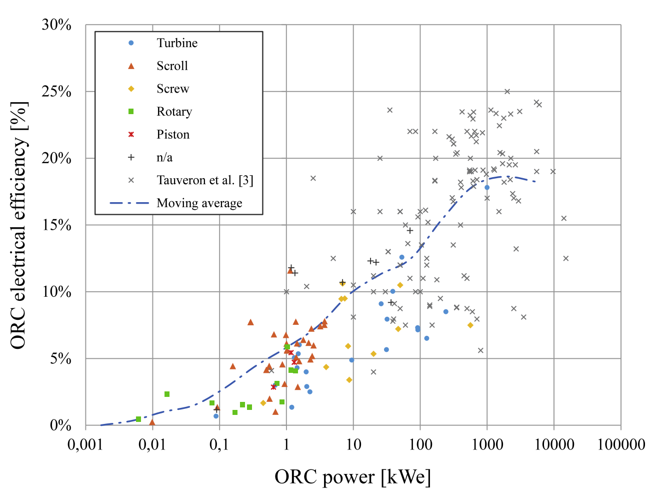

Fig. 14. ORC power and thermal efficiency map.

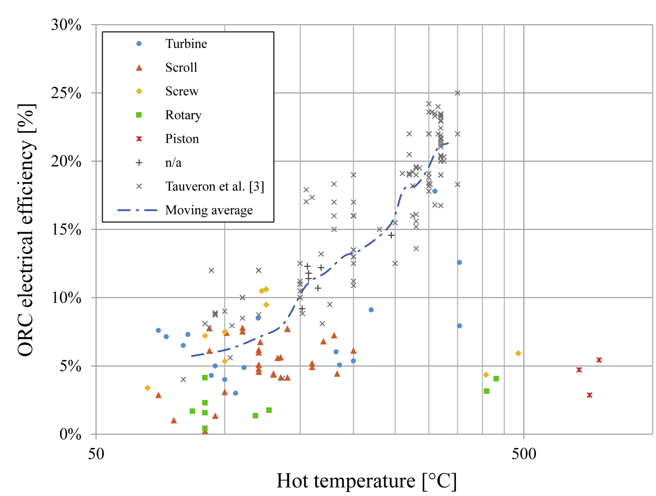

Fig. 15. Hot temperature and ORC thermal efficiency map.

fore, a large scroll expander, a screw expander or a turbine might be used. With a pressure difference up to 35 bar, reciprocating pump might be more appropriate.

4. Data for ORC performance comparison

4.1. General performances

The present database is merged with commercial ORC references from Tauveron et al. [3], without duplicates. Figs. 14and 15 show respectively the evolution of ORC thermal efficiency η_{th} with ORC power and hot temperature, distinction is made between the different expander technologies. A centered sliding average is plotted and provides the mean evolution. The ORC thermal efficiency is closely linked to its power scale, and as expected, to the hot source temperature.

Fig. 16 shows a map of ORC second law efficiency η_{II} with ORC power scale. The moving average has an inflection point for ORC power around 20–50 kWe. This inflection point could represent a technological maturity limit. But it is located at the transition zone between both databases and may be due to intrinsic methodology difference between the two databases.

Fig. 17 shows the ORC exergetic recovery efficiency ε_{rec}, as defined in Section 2.3. Fewer references are available as it requires more information to compute. This efficiency follows the same relations with the power scale. Closed source target applications (biomass, gas, solar) and open source target applications (geothermal, WHR) are differentiate as exergetic recovery efficiency might be more appropriate to evaluate performances of open heat sources.

All the previous efficiencies and power are gross. The ORC net efficiency is linked to the BWR and the gross efficiency by Eq. (9)

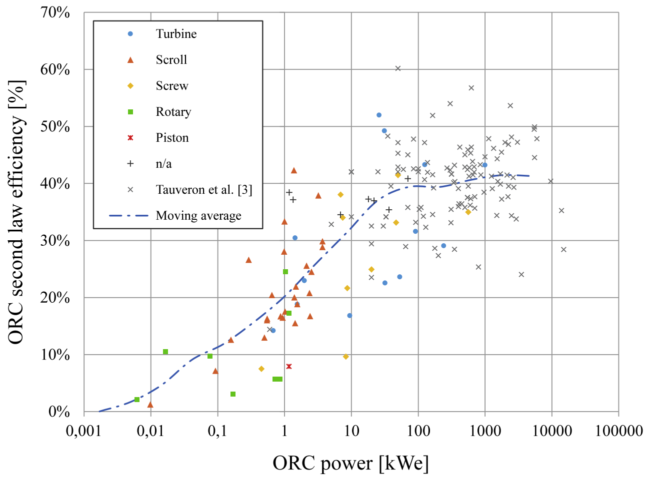

Fig. 16. ORC power and second law efficiency map.

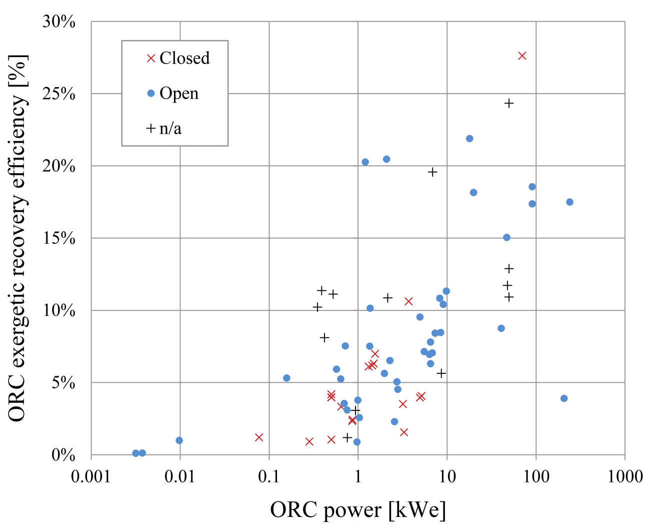

Fig. 17. ORC power and exergetic recovery efficiency map for different target application types.

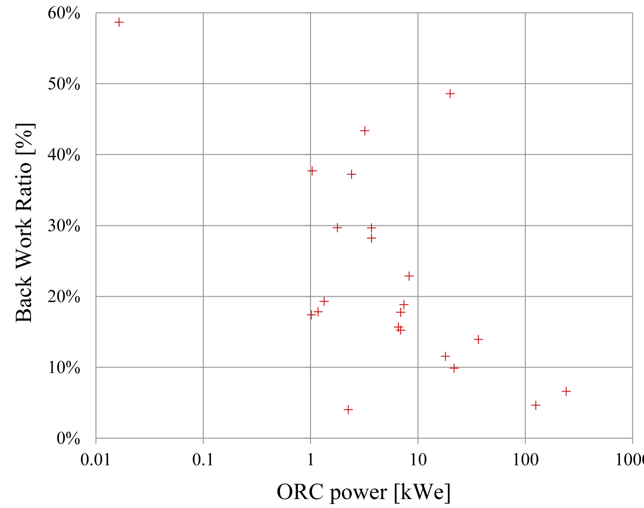

Fig. 18. ORC power and back work ratio map.

relation. Fig. 18 shows the back work ratio (BWR) in relation with the ORC gross power. The BWR as well is closely linked to the power scale. From around 5% at 100 kW scale, it reaches around 30% at 1 kW scale. This is due to the combined relation of expander and pump efficiencies with the power scale. We can introduce the back work ratio efficiency (Eq. (10)) to compare actual BWR with the minimum ideal BWR and establish the relation with pump and expander efficiencies. This efficiency reaches a maximum of 40% for a 100 kW scale ORC and drops around 15% for 1 kW scale ORC.

$$\eta_{net} = \eta_{gross} \cdot (1 - BWR) \tag{9}$$

$$\eta_{BWR} = \frac{BWR_{is}}{BWR} = \frac{W_{pp,is}/W_{exp,is}}{W_{pp,el}/W_{exp,el}} = \eta_{pp} \cdot \eta_{exp} \tag{10}$$

4.2. Statistical analysis of ORC global performances

With 175 references of more than a hundred unique prototypes, the database has a sufficient size to perform some statistical analysis. The Spearman's rank correlation is used to evaluate the influence of some parameters over the ORC efficiency. Spearman's correlation measures the monotonic relationship between two variables [186]. The sign of the correlation coefficient ρ shows if variables have similar (positive) or opposed (negative) trends, its value reaches zero when there is no correlation and +/−1 for a strictly monotone correlation. The p-value shows the statistical significance of the correlation. Below 0.05, the test result is considered as significant.

For this statistical analysis, duplicates are removed except change of expander or fluid. Table 6 shows the main correlation results for ORC thermal and second law efficiency in relation with the power scale, the hot temperature, the expander efficiency, the use of regenerator (IHE) or the use of lubricant. ORC thermal efficiency is strongly and significantly related to power scale and expander efficiency, and more than the hot temperature. The use of regenerator also increase the efficiency, but in a smaller proportion. Lubrication effect is not significant. ORC second law efficiency is also strongly related to power scale and expander efficiency. But, as theoretically expected, there is no more influence of the hot temperature on this efficiency. Regenerator influence on second law efficiency is weak, as well as lubrication.

4.3. Evaluation of specific innovative ORC

Exergetic efficiency criteria are not affected by the heat source temperature, and therefore, are useful for different ORC comparison. However, the ORC power scale has a major impact on the achievable efficiency, therefore, a specific ORC must be compared with same scale ORC bench. Different test bench using innovative ORC solutions are evaluated and compared with other ORC. Fig. 19 compare specific ORC with similar power scale ORC based on second law or exergetic recovery efficiency. Every ORC reference in the selected power range is sorted by increasing efficiency. Based on its efficiency, the relative ranking of a specific ORC can be identified.

Two transcritical ORC performances are evaluated. The first transcritical ORC tested by Kosmadakis et al. [47] is dedicated to a closed source application (solar energy) and reached a power of 3.3 kW. Therefore, it is compared with ORC in the 1–10 kW range based on second law efficiency in Fig. 19(a). With a 38.8% efficiency, it ranked in the first fifth of same-scale ORC with a relative ranking of 0.84. The second transcritical ORC tested by Hsieh et al. [45] is dedicated to an open source application (waste heat recovery) and reached a maximum power of 20 kW. With 18.1% of exer-

Table 6
Spearman's correlations for ORC thermal and second law efficiency.

	ORC thermal efficiency		ORC second law efficiency	
	ρ	p-value	ρ	p-value
ORC power	0.6805	1.26 e−10	0.7197	4.38 e−11
Hot temperature	0.4363	5.51 e−4	−0.0558	0.67
Expander efficiency	0.7308	5.81 e−8	0.7655	2.16 e−8
IHE	0.2951	0.0125	0.1925	0.128
Lubrication	0.1942	0.250	0.1523	0.390

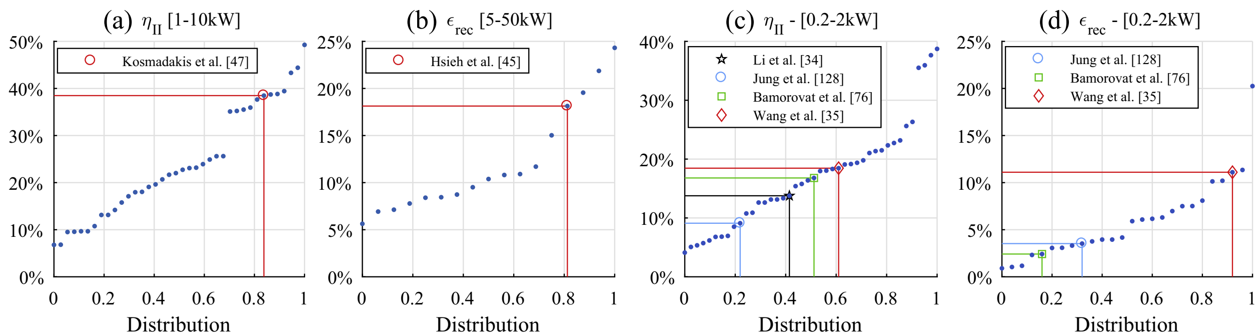

Fig. 19. Comparison of specific ORC benches with same scale ORC: (a) transcritical ORC for solar application – (b) transcritical ORC for WHR application – (c) & (d) ORC with mixture fluid.

Table 7
Comparison of HFO R1233zdE with R245fa and same scale benches {Citation}.

	Power Scale	ORC second law efficiency		ORC exergetic recovery efficiency	
		R245fa	R1233zdE	R245fa	R1233zdE
Eyerer et al. [29]	0.1–1.2 kW	η_{II}: 19.2% Rel. Rank: 0.73	η_{II}: 19.4% Rel. Rank: 0.80	ε_{rec}: 11.3% Rel. Rank: 0.95	ε_{rec}: 10.2% Rel. Rank: 0.89
Guillaume et al. [30]	0.9–8.9 kW	η_{II}: 9.6% Rel. Rank: 0.10	η_{II}: 9.5% Rel. Rank: 0.08	ε_{rec}: 5.0% Rel. Rank: 0.27	ε_{rec}: 4.5% Rel. Rank: 0.23
Moles et al. [31]	0.4–4.2 kW	η_{II}: 37.7% Rel. Rank: 0.93	η_{II}: 38.7% Rel. Rank: 0.98	n/a	n/a

getic recovery efficiency, it has a relative ranks of 0.81 compared to ORC in the 5–50 kW range (Fig. 19(b)). However, transcritical cycle has a large BWR, so similar evaluation based on net efficiency could led to less optimistic results for transcritical ORC.

Four ORC using fluid mixture are evaluated. The four benches reached a maximum power of 0.5–0.9 kW, and therefore are compared with ORC in the 0.2–2 kW power range. Li et al. [34] used a mixture of R245fa and R601 with a fixed mixing ratio of 0.72/0.28 and proposed a comparison with pure R245fa. Jung et al. [128] used a mixture of R245fa and R365mfc with a mixing ratio of 0.485/0.515 for exhaust gas heat recovery application. Bamorovat et al. [75] tested a mixture of R245fa and R134a with a ratio of 0.6/0.4, while Wang et al. [35] tested mixture of R601a and R600a with different fluid ratio and achieved the highest efficiency and output power for a mixing ratio of 0.6/0.4. Based on the second law efficiency (Fig. 19(c)), ORC using fluid mixture does not appear to have significantly higher potential for heat conversion since they are in the mean of same-scale ORC. For recovery potential, Wang et al. [35] performes very well, but others do not. However, the average relative ranking of ORC using fluid mixture is higher based on exergetic recovery efficiency (0.47) than based on second law efficiency (0.44). Therefore, fluid mixture may have a good potential for open heat source applications and fluctuating heat sources [75].

In the same way, three ORC using last generation of organic fluids, the HFO R1233zdE are evaluated. The three reference tested their ORC both with R245fa and R1233zdE as working fluid. Table 7 summarize for each bench its power scale for statistical comparison, performance and relative ranking base on both second law and exergetic recovery efficiency for both R245fa and R1233zdE. Based on second law efficiency, R1233zdE seems to have slightly higher efficiency – 1% efficiency increase in average. However, based on exergetic recovery efficiency, R1233zdE has efficiency decreased by 10%. If R1233zdE seems to be a great R245fa replacement fluid for closed source applications, a deeper evaluation must be performed for open source applications.

5. Conclusion

In this paper, an open-access database of Organic Rankine Cycle experiments is introduced. The database architecture and construction is described. ORC and component main parameters are recorded, discriminated and harmonized among the different scientific references.

The wide disparity of power types and definitions used for ORC power output is highlighted. A clarification of ORC performance criteria is proposed, based on heat source type and energy/exergy distinction. Three main criteria are selected for ORC comparison and analysis: thermal efficiency, second law efficiency and exergetic recovery efficiency.

The 175 references database is presented. Power scale ranges from a few Watts to a MW with hot source temperature between

60 °C and 600 °C. Most ORCs experiments run with R245fa as working fluids and are dedicated to waste heat recovery applications.

Then, a focus is made on ORCs main components such as the expander or the pump. A strong correlation between component efficiency and power scale is highlighted. Optimal expander technology is identified based on power scale and pressure ratio. Based on reported components performance, a parametric optimization of an ORC is proposed, considering two case study. The closed source case would achieve the highest second law efficiency using a subcritical regenerative ORC with R245fa as working fluid, while the open source case would achieve the highest exergetic recovery efficiency using a transcritical ORC with R404a.

The experiments database is merged with a commercial unit database and the three proposed ORCs gross performance criteria are presented as a function of ORC power scale or hot temperature. Influence of five parameters on ORC performance are statistically tested. Expander efficiency has the highest correlation on the thermal efficiency, closely followed by the power scale, then the hot temperature and the use of regenerator. The lubrication has no significant influence.

Back work ratio is found to be strongly related to the power scale due to its relation with pump and expander efficiency. For 1 kW scale, the ORC net efficiency can be 20–40% lower than gross efficiency due to pump consumption.

Performance of innovative ORC experiments like transcritical cycle, zeotropic fluid mixtures or environmentally friendly HFO fluids are highlighted and each specific bench is compared with same power scale ORC to evaluate their potential.

Database linking

The database "Experimental ORC database – v2016.12" is licensed under a Creative Commons Attribution 4.0 International License and available via the Zenodo.org repository at http://dx.doi.org/10.5281/zenodo.400556. Collaborative and updated database is available via Github.com or Zenodo.org.

Acknowledgements

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