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Assessing the performance of a scroll expander with a selection of fluids suitable for low-temperature applications

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Abstract

This paper presents calculations of micro organic Rankine cycles equipped with a scroll expander whose performance is simulated by a model reported in literature. In particular, the original working fluid (R245fa) is replaced with selected hydrofluorocarbons and hydrocarbons for assessing the performance of the power cycle as a function of the specific working fluid. After setting the power output from the expander at 2 kW, the relationship between cycle efficiency and maximum cycle temperature is presented for each fluid. Thus, the possibility of matching the operation of the organic Rankine cycle to heat sources at lower temperature is investigated.

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Keywords: hydrocarbons; hydrofluorocarbons; organic Rankine cycle; scroll expander; semi-empirical model.

1. Introduction

The exploitation of low-temperature heat sources for power generation is a more and more investigated field in the recent years and the development of suitable technologies for the energy conversion from solar and geothermal sources or simply of the waste heat of industrial processes becomes a crucial point in the scenario of power generation. Among the well-proven technologies, the organic Rankine cycle (ORC) is one of the most promising ways in low- to medium-temperature applications [1].

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The ORC layout is somewhat simpler than the one of the steam Rankine cycle: there is no water-steam drum connected to the boiler and one single heat exchanger can be used to perform fluid preheating, vaporization and superheating. The variations of the cycle architecture are also more limited: reheating and expander bleedings are generally not suitable for the thermodynamic cycle, even though a recuperator can be installed as liquid preheater downstream of the pump. These features make ORC systems attractive in the scenario of distributed power generation, when low-temperature thermal energy can be exploited. In particular, the development of micro ORC technologies, suitable for residential applications (a few kW), has received significant attention over the last years, even though a satisfactory efficiency is still a challenge [2-8]. As a key component of ORC systems, the expander plays a vital role in the system performance. Both turbines and volumetric machines could be available according to the system scale. Currently, expanders with kW scales for organic working fluids are still under development and demonstration, but positive displacement machines are suitable for small size plants [9,10]. As a matter of fact, several types of commercial scroll machines can be modified into expanders, which can be integrated into ORC systems for low grade heat recovery. Numerous researchers [11] have investigated the performance of expanders resulting from original hermetic refrigeration scroll compressors, automotive air-conditioning compressors and open-drive scroll air compressors.

Although R123, R134a and R245fa are the most investigated fluids in ORC systems with scroll expanders [11], the selection of the working fluid is a crucial issue. Among the main principles of the selection there are [12,13]:

- chemical stability in the operating temperature range;
- compatibility with materials and lubricating oil;
- non-toxicity, non-flammability, non-explosibility, non-radioactivity and industrial acceptability;
- low ozone depletion potential (ODP), global warming potential (GWP) and atmospheric lifetime (ALT);
- availability and low cost.

There are several studies in technical literature oriented to the optimal selection of working fluids for ORC applications. In particular, Wang et al. [14] proposed a thermal efficiency model based on an ideal ORC in order to analyze the influence of working fluid properties on the thermal efficiency and the optimal operation condition in the low heat source temperature. According to the heat source temperature, Wang et al. [14] classified a number of working fluids into six categories, as shown in Fig. 1. Thus, the selection of the working fluid reasonably based on an increasing temperature as in Fig. 1 will help in performance optimization during the ORC design. Ultimately, there is not a working fluid suitable for any organic Rankine cycle system. At the same time, the selection of the working fluid should not only consider the thermodynamic performance (first law of thermodynamics, output power, etc.) and the system economy, but also other factors, such as the maximum and minimum bearable temperature and system pressures, the design of expanders or turbines as well as environmental and safety issues.

This paper aims at simulating the performance of a 2 kW hermetic scroll expander for use in a micro-scale organic Rankine cycle when replacing the original working fluid (R245fa) with selected substances among the ones in Fig. 1. In particular, hydrofluoroclorocarbons are not included in the analysis because of their non-zero ODP. The following sections focus on the simulation model adopted to calculate the expander performance and on the main results of the micro ORC systems equipped with such an expander. In detail, the simulations suggest the relationship between cycle efficiency and maximum cycle temperature for each fluid. Thus, the possibility of better matching the ORC operation to heat sources at lower temperature is presented.

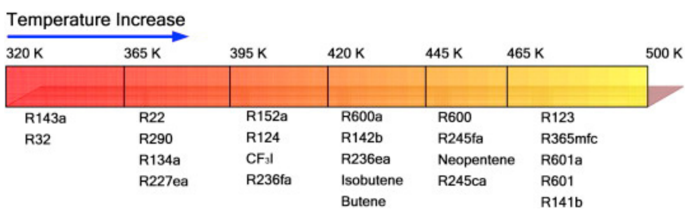

Fig. 1. Optimal selection of working fluids corresponding to the heat source temperature level [14].

Nomenclature

ALT	atmospheric lifetime	P	power, W
AU	heat transfer coefficient, $W \cdot K^{-1}$	Pr	Prandtl number
BVR	built-in volume ratio	PR	pressure ratio
c_p	specific heat, $J \cdot kg^{-1} \cdot K^{-1}$	Q	volumetric flow rate, $m^3 \cdot s^{-1}$
GWP	global warming potential	$\dot{Q}$	heat transfer rate, W
h	enthalpy, $J \cdot kg^{-1}$	Re	Reynolds number
L	specific length, m	s	entropy, $J \cdot kg^{-1} \cdot K^{-1}$
$\dot{m}$	mass flow rate, $kg \cdot s^{-1}$	T	temperature, K
N	rotational speed, rpm	TPI	thermal power input, W
Nu	Nusselt number	v	specific volume, $m^3 \cdot kg^{-1}$
ODP	ozone depletion potential	V	volume, m^3
ORC	organic Rankine cycle	η	efficiency
p	pressure, Pa	λ	thermal conductivity, $W \cdot m^{-1} \cdot K^{-1}$

2. Simulation of the scroll expander

In order to simulate the performance of the hermetic scroll machine, originally investigated by Lemort et al. [15] in case of R245fa as the working fluid, the semi-empirical model of a scroll expander previously proposed by the same authors [16] was used. For the sake of brevity, only the main features of the model are recalled in this paper.

The evolution of the fluid through the expander is decomposed into a number of steps as schematically shown in Fig. 1:

- an adiabatic pressure drop (su → su,1) and an isobaric cooling down by contact with the metal mass of the machine (su,1 → su,2) during the intake process;
- an isentropic expansion to the adapted pressure imposed by the built-in volume ratio (BVR) of the machine (su,2 → ad) and an adiabatic expansion at constant machine volume (ad → ex,2);
- an adiabatic mixing between internal and leakage flows (ex,2 → ex,1), $\dot{m}_{in}$ and $\dot{m}_{leak}$ respectively;
- an isobaric exhaust cooling down or heating up (ex,1 → ex).

The mass flow rate through the expander is a function of expander swept volume V_s, rotational speed N as well as leakages:

$$\dot{m} = \dot{m}_{in} + \dot{m}_{leak} = \frac{V_s \cdot N}{v_{su,2}} + \dot{m}_{leak} \tag{1}$$

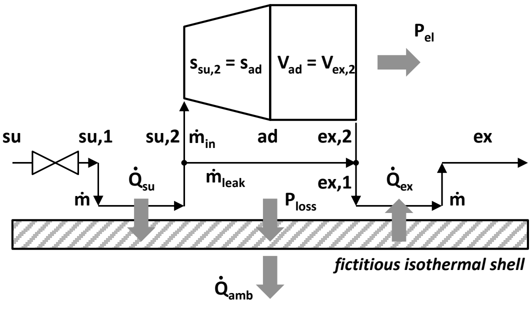

Fig. 2. Schematic representation of the expander simulation model.

All the leakage paths are lumped into one unique fictitious path connecting the expander intake and exhaust. Both the leakage flow rate and the pressure drop (su → su,1) are computed by reference to the isentropic flows through simply convergent nozzles of fixed cross-sectional areas [16].

Under- and over-expansion losses are modelled by splitting the expansion process into an isentropic and a constant-volume evolution, which yields the following expression for the internal expansion power

$$P_{in} = \dot{m}_{in} \cdot \left(h_{su,2} - h_{ad}\right) + \dot{m}_{in} \cdot v_{ad} \cdot \left(p_{ad} - p_{ex,2}\right) \tag{2}$$

where the adapted condition is calculated by $v_{ad} = BVR \cdot v_{su,2}$ and $s_{ad} = s_{su,2}$.

The modelling takes into account both mechanical losses ($P_{loss,1}$) due to friction between moving elements (scrolls, journal bearings, Oldham coupling, thrust surface) and electromechanical losses ($P_{loss,2}$) in the asynchronous machine:

$$P = P_{in} - P_{loss,1} - P_{loss,2} \tag{3}$$

Mechanical losses are computed on the basis of the mechanical efficiency:

$$P_{loss,1} = \left(1 - \eta_{mecc}\right) \cdot P_{in} \tag{4}$$

Electromechanical losses are evaluated by referring to the performance of the asynchronous machine in motor mode given by the manufacturer [15].

Internal heat transfers are lumped into equivalent supply and exhaust heat rates, $\dot{Q}_{su}$ and $\dot{Q}_{ex}$ respectively, between the fluid and a fictitious shell at uniform temperature (T_{shell}), on the basis of overall heat transfer coefficients AU_{su} and AU_{ex}. In detail, the supply heat transfer rate is calculated as

$$\dot{Q}_{su} = \dot{m} \cdot \left(h_{su,1} - h_{su,2}\right) = \left(1 - e^{\frac{-AU_{su}}{\dot{m} \cdot c_p}}\right) \cdot \dot{m} \cdot c_p \cdot \left(T_{su,1} - T_{shell}\right) \tag{5}$$

A similar formulation is adopted for the exhaust heat transfer rate, whereas the external heat loss $\dot{Q}_{amb}$ is calculated by an overall heat transfer coefficient AU_{amb}. In steady state regime, the shell temperature is given by

$$\dot{Q}_{su} - \dot{Q}_{ex} - AU_{amb} \cdot \left(T_{shell} - T_{amb}\right) + P_{loss,1} + P_{loss,2} = 0 \tag{6}$$

Although a more physically sound modelling can be proposed as regards mechanical and ambient heat losses [17,18], the model adopted by Lemort et al. [15] for the simulation of the hermetic scroll expander is

- reliable with a maximum deviation between the prediction and the measurements of 2% for the mass flow rate, 6% for the power output, and 2 K for the exhaust temperature;
- applicable in wide ranges of fluid pressure at the expander inlet (from 2 to 35 bar) and of pressure ratio (from 2 to 20).

A reasonable assessment of the performance of the expander in case of other working fluids is possible based on the procedure proposed in a former work [19]. In detail, only heat transfer coefficients should be varied when changing the working fluid, since the geometry of the expander is definitively fixed.

In order to simulate supply and exhaust heat transfer rates as in Eq. (5), Lemort et al. [15] proposed the following formula for calculating the heat transfer coefficient:

$$AU = 30 \cdot \left(\frac{\dot{m}}{0.1} \right)^{0.6} \tag{7}$$

where $\dot{m}$ and 0.1 (kg/s) are the mass flow rate and a reference value for normalization, respectively. Based on the consideration that orifices, flow passage areas and the general geometry of the scroll expander do not vary when changing the working fluid:

$$AU \propto \frac{Nu \cdot \lambda}{L} \propto Re^{x} \cdot Pr^{y} \cdot \lambda \tag{8}$$

where $Re^{x} \cdot Pr^{y}$ is introduced in place of the Nusselt number. Thus, the heat transfer coefficients are reasonably revised according to the specific working fluid [19], by leaving intact the original structure modelling the expander.

Ultimately, the scroll expander model has been implemented in the MATLAB® platform and REFPROP 9.1 [20] has been used to include the thermodynamic and transport properties of the working fluid in the calculation environment.

3. Results

The results of simulations of micro ORC systems powered by the scroll expander are presented in this section. As the main cycle calculation assumptions, (i) the overall efficiency of the pump is set at 0.5, (ii) the pressure losses in both evaporator and condenser are neglected and (iii) the fluid condensation temperature is set at 40°C, without liquid subcooling.

The input variables of the expander simulation model are the supply temperature and the supply and exhaust pressures, along with eight specific parameters [15]. The main output variables are the mass flow rate displaced by the expander, its electrical power production and the fluid temperature at the expander exhaust. In detail, the model adopted in this paper was validated for a certain degree of fluid superheating at the expander inlet [15]. As a preliminary result, based on the above-mentioned assumptions, Fig. 3 reports the cycle efficiency in case of electric power output from 1 to 2 kW for three degrees of R245fa superheating at the expander inlet. Based on the calculation assumptions, higher pressure ratio, reflecting on higher fluid temperature at the expander inlet, is required for higher power output from the expander. Fig. 3 shows that the cycle efficiency is shifted to the right for higher superheating degrees without significant variations. According to this first result, 5 K may be selected as the fluid superheating in order to use a heat source with a temperature as low as possible. In particular, Fig. 4 shows the T-s diagram for the case corresponding to 2 kW of electric power output from the expander.

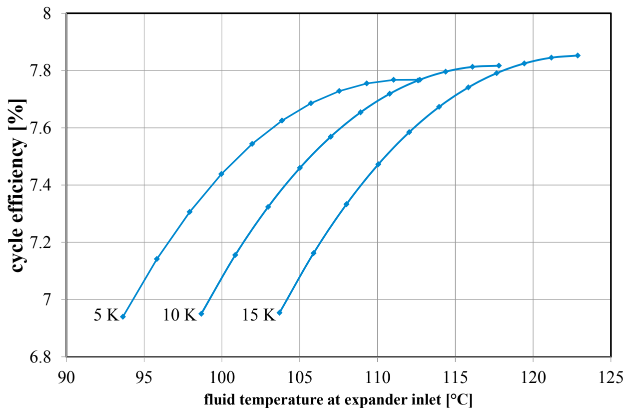

Fig. 3. Cycle efficiency as a function of R245fa temperature at the expander inlet for an electric power output of the expander ranging from 1 to 2 kW and for three degrees of fluid superheating.

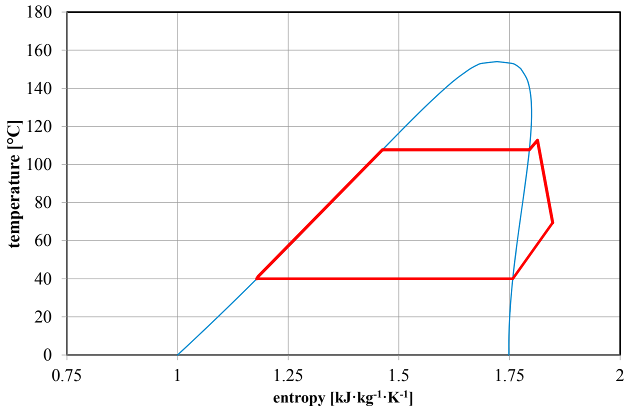

Fig. 4. T-s diagrams for the case with R245fa, corresponding to 2 kW of electric power output from the expander.

After setting the fluid superheating at 5 K, a performance analysis of the scroll machine as the expander of a micro ORC system is proposed for a number of working fluids, as anticipated in Fig.1. The results of ORC calculations are reported hereafter, for a net power output from the expander set at 2 kW. Thus, in order to simulate the scroll expander performance, the calculation for each fluid will result in just the fluid evaporation pressure.

Fig. 5 shows the trend of the cycle efficiency calculated for the investigated fluids, based on the pressure and the temperature of the fluid at the expander inlet for the selected power output. The fluids in the x-axis are listed according to an increasing temperature. In general, the higher the fluid temperature at the expander inlet, the higher the cycle efficiency. Starting from the reference case with R245fa, a fluid evaporation temperature of 107.7°C has been calculated with a cycle efficiency of ~7.8%. Looking at Fig. 5, fluids on the right of R245fa seem to allow for cycle efficiency improvements, except for neopentane and R365mfc, which result in cycle efficiency of ~7.7%. Nevertheless, in order to better matching the ORC operation to a lower temperature heat source, fluids on the left of R245fa should be considered. In detail, a cycle efficiency down to ~6.7% has been calculated in case of R13I1 with an evaporation temperature of 92.7°C or in case of R236fa with an evaporation temperature of 95.5°C. Even lower evaporation temperatures can be considered as reported in Fig. 5, but cycle efficiency results ~5.3% (in case of R152a and R227ea) or even lower.

Further simulation results which are worthy of attention are detailed in Table 1.

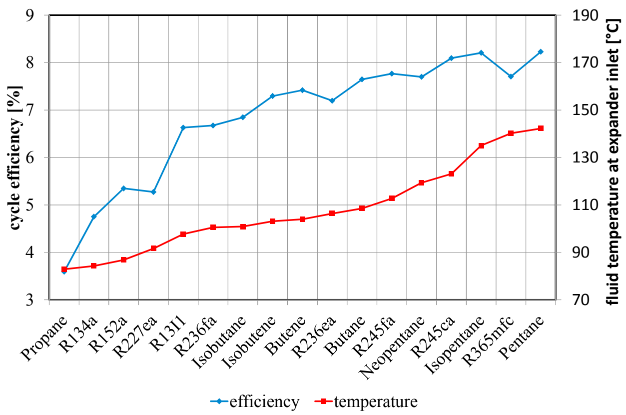

Fig. 5. Cycle efficiency and corresponding fluid temperature at the expander inlet for a power output from the expander set at 2 kW

Table 1. Main results of the ORC simulations.

	T_{crit}, °C	T_{su}, °C	p_{su}, bar	p_{ex}, bar	PR	$\dot{m}$, g/s	Q_{su}, dm^3/s	Q_{pump}, cm^3/s	P_{pump}, W	TPI, kW
Propane	96.7	82.9	30.1	13.7	2.2	105.2	1.45	225	736	35.2
R134a	101.1	84.3	25.9	10.2	2.6	178.1	1.27	155	488	31.8
R152a	113.3	86.8	24.3	9.1	2.7	108.4	1.35	126	383	30.2
R227ea	101.8	91.6	21.4	7.0	3.0	223.6	1.14	169	485	28.7
R13I1	123.3	97.7	23.0	7.3	3.1	256.4	1.17	131	410	24.0
R236fa	124.9	100.5	17.6	4.4	4.0	148.9	1.17	114	301	25.5
Isobutane	134.7	100.9	18.4	5.3	3.5	63.8	1.30	120	313	24.6
Isobutene	144.9	103.1	17.6	4.7	3.8	57.5	1.31	101	261	23.8
Butene	146.1	104.0	17.5	4.6	3.8	56.8	1.31	100	258	23.5
R236ea	139.3	106.4	16.2	3.4	4.8	128.5	1.17	93	239	24.5
Butane	152.0	108.5	16.3	3.8	4.3	52.7	1.29	95	238	23.0
R245fa	154.0	112.7	15.0	2.5	6.0	100.7	1.19	78	193	23.3
Neopentane	160.6	119.4	14.7	2.7	5.5	56.0	1.24	99	237	22.9
R245ca	174.4	123.1	13.8	1.7	8.0	87.5	1.19	65	157	22.8
Isopentane	187.2	135.0	13.1	1.5	8.7	46.1	1.24	77	179	22.2
R365mfc	186.9	140.2	12.7	1.0	12.6	85.6	1.17	70	163	23.8
Pentane	196.6	142.2	12.6	1.2	10.9	43.2	1.23	71	164	22.3

- The higher the critical temperature (T_{crit}) of the considered fluid, the higher the fluid temperature and the lower the fluid pressure at the expander inlet, T_{su} and p_{su} respectively, with few exceptions.
- The maximum cycle pressure, i.e. the fluid pressure at the expander inlet (p_{su}), is less than 35 bar[1] and the pressure ratio ($PR = p_{su}/p_{ex}$) is always within the applicability limit of the model [15].
- A condensation pressure (100.6 kPa) less than the atmospheric level has been calculated in the only case of R365mfc.
- Various values of the mass flow rate ($\dot{m}$) have been calculated, depending on the specific working fluid, but the volumetric flow rate at the expander inlet is not strictly constant, in spite of the constant expander rotational speed. This result can be justified by the different values of the leakage flow rate, according to Eq. (1). As a matter of fact, the model adopted for leakage simulation mainly depends on the fluid thermodynamic properties at the station su,2 as in Fig. 2, just downstream of the expander inlet [16].
- Two results are included in Table 1 as regards the pump, namely the volumetric flow rate (Q_{pump}) and the power consumption (P_{pump}). The first result gives an idea of the size of the pump, in case of adopting a positive-displacement machine and fixed driving speed. The results for the power consumption are strictly related to the volumetric flow rate delivered by the pump. Moreover, there is a direct relationship also with the pressure rise (p_{su} - p_{ex}) across the pump, which is higher for those fluids with a lower critical temperature. The result of power consumption finally depends on the efficiency set at 0.5, as previously anticipated. In general, a decreasing trend of both the volumetric flow rate and the power consumption is evident with respect to the temperature T_{su}, i.e. in case of selecting a fluid with higher critical temperature.
- The thermal power input (TPI) to the cycle is also reported in Table 1. Once again, a trend similar to the one characteristic of the volumetric flow rate and of the power consumption calculated for the pump can be appreciated for variations of the temperature T_{su}.

[1] Although included in Fig. 1, R143a and R32 cannot be used based on the calculation assumptions, since the simulations have resulted in fluid pressures at the expander inlet exceeding 35 bar.

As suitable fluids in organic Rankine cycles for low temperature heat utilization, along with the ones included in Fig. 1, there is current interest in hydrofluoroolefins because of their strongly lower GWP than hydrofluorocarbons. Referring to these new fluids, databases for thermodynamic and transport properties calculations are limited to few fluids, so a less comprehensive analysis than the one in Table 1 can be proposed. Nevertheless, as reported in a companion paper, R1234ze(Z) and R1233zd(E) seem to be interesting drop-in replacements for R-245fa [21].

4. Conclusions

A literature model [15] proposed for the simulation of the performance of an hermetic scroll expander has been revised and used to study the behavior of the machine in case of replacing the original working fluid (R245fa) with selected hydrofluorocarbons and hydrocarbons as suggested by Wang et al. [14]. Therefore calculations of micro scale ORC systems with such an expander have been presented and discussed for a number of working fluids.

In general, for a fixed expander power output set at 2 kW, the higher the fluid temperature at the expander inlet, the higher the cycle efficiency. On the other hand, a decreasing trend of the volumetric flow rate delivered by the pump, of the pump power consumption as well as of the thermal power input to the cycle is evident with increasing fluid temperature at the expander inlet.

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