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Experimental and numerical analyses on a plate heat exchanger with phase change for waste heat recovery at off-design conditions

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Abstract. This paper analyzes the performances of an evaporator for small scale waste heat recovery applications based on bottoming Organic Rankine Cycles with net output power in the range 2-5 kW. The heat recovery steam generator is a plate heat exchanger with oil as hot stream and an organic fluid on the cold side. An experimental characterization of the heat exchanger was carried out at different operating points measuring temperatures, pressures and flow rates on both sides. The measurement data further allowed to validate a numerical model of the evaporator whereas heat transfer coefficients were evaluated comparing several literature correlations, especially for the phase-change of the organic fluid. With reference to a waste heat recovery application in industrial compressed air systems, multiple off-design conditions were simulated considering the effects of oil mass flow rate and temperature on the superheating of the organic fluid, a key parameter to ensure a proper operation of the expansion machine, thus of the energy recovery process.

1. Introduction

Medium and low grade energy recovery might be accomplished in multiple contexts, such as cement and glass industry, heavy duty internal combustion engines as well as solar and geothermal power plants [1]. Among the industrial situations, compressed air systems might highly benefit from an energy recovery system in mechanical or electrical form to lower package specific consumptions of compressors since they typically waste almost 80 % of the electrical energy supplied to the machine through the oil radiator [2]. Indeed, literature studies estimate the saving potential from waste heat recovery as 10 % of the overall energy consumptions [3]. To date, waste heat recovery in oil flooded compressors usually occurs through a thermal approach that uses the hot temperature oil stream (90-120 °C) to heat up water in a heat exchanger and partially fulfill heating or hot water needs of the industrial site. On the other hand, a universally referenced technology which converts thermal energy into mechanical form is based on an Organic Rankine Cycle (ORC).

The evaporator is a key component of the ORC-based energy recovery system since it has to guarantee a correct heat transfer during the pre-heating, vaporization and superheating of the high pressure organic fluid. Furthermore, it should have high thermal performances to maximize the heat recovery as well as low pressure drops on both sides to prevent re-condensation of the organic fluid or excessive pressure loss for the external heat source. Compared with other technologies, plate heat exchangers (PHEs), thanks to the high degree of turbulence established at the flow channels, require lower surface and volume than those needed by shell and tube devices to exchange the same amount of power. Moreover, PHEs can be simply maintained, cleaned and inspected. Their high effectiveness and low cost led to a large use of these devices as evaporators and condensers in industrial applications [4, 5]. Another important consequence of their compactness is the low refrigerant charge [6]. On the other hand, drawbacks of PHEs are large pressure drops due to the restricted flow paths. However, this issue could be overcome using the manifold-multichannel technology [7], which allows to reduce single-phase flow pressure drops by a factor of n^2, where n represents the number of channel divisions [8].

A major concern in plate heat exchangers is the difficulty to understand flow boiling mechanisms due the complex nature of two-phase flow along the channels [9]. Consequently, quantitative predictions of the flow boiling heat transfer coefficient were obtained through experimental investigations and expressed by empirical correlations such as the ones of Yan Lin for R141a [10], of Ayub [6] for ammonia and R22, of Sterner and Sunden for ammonia [11] and the correlation of Han at al. for R22 and R422A [12]. Research carried out in [13] investigated the effect of refrigerant mass flow rate , saturation temperature (pressure) and fluid properties on pressure drops and heat transfer during the saturated vapor condensation of R236fa, R134a and R410A for a brazed plate heat exchanger that used water as hot stream. The analysis resulted in a great influence of fluid properties and refrigerant mass flow rate and a feeble effect of saturation temperature on the heat transfer coefficient. As concerns pressure losses, it was observed a linear dependence on the kinetic energy per unit volume of refrigerant flow and, consequently, a quadratic dependence on the refrigerant mass flow. According to this study, R410A and R134a have similar heat transfer coefficient, whose values are 10% higher than R236fa for the same device and operating conditions. Moreover, pressure drops for R410A are 50-60% lower than R236fa and 40-50% lower than R134a.

In this paper, a plate heat exchanger used as evaporator of an ORC system for waste heat recovery on an industrial air compressor was tested at different operating conditions using oil as hot stream and R236fa as cold stream. The experimental data acted as a validation baseline for a one-dimensional model of the plate heat exchanger whereas multiple semi-empirical correlations were compared. Off-design conditions of the evaporator were eventually simulated considering the most common regulation strategies in industrial compressors, namely variable speed and throttling of the suction process. The off-design analysis aimed at assessing the influence of oil mass flow rate and temperature on the superheating of the organic working fluid which is a feature strictly required by expansion machines of ORC systems to accomplish a proper conversion of the thermal power recovered into mechanical one.

2. Experimental campaign

The test campaign used as reference for the current study is the one performed on a waste heat recovery application in compressed air industry [14, 15]. In that case, the hot source of the energy recovery system based on Organic Rankine Cycle was the oil of an industrial air compressor that usually heats up during the compression phase because of friction between the machine components it lubricates and the convective heat exchange with air along the compression and discharge processes [16, 17]. The evaporator of the energy recovery system was a off the shelf oil-water plate heat exchanger (PHE) whose features are listed in Table 1. On the cold side of the PHE, a R236fa organic fluid recuperated the thermal energy of the lubricant, that otherwise would be usually dissipated at the compressor radiator, to economize, vaporize and eventually reach a superheated state. Part of the enthalpy of the superheated steam was then converted to mechanical power through a sliding vane rotary expander; the remaining share was rejected to the environment through a water cooled condenser.

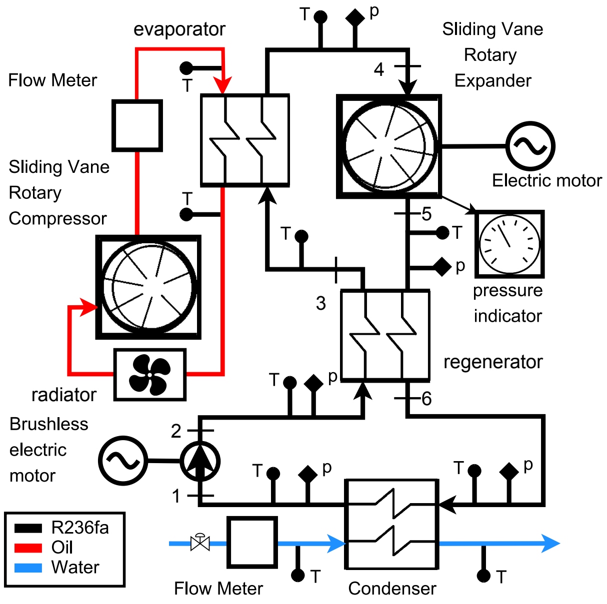

Figure 1: ORC system test bench with sensors layout

plate length	286	mm
plate width	117	mm
number of plates	41	–
nominal heat transfer area	1.2	m^2
plate thickness	1	mm
corrugation amplitude	1.15	mm
plate roughness	1	μm
hydraulic diameter	4.5	mm
weight	6.9	kg
material	copper	

Table 1: Evaporator specifics

temperature	0.1	°C
R236fa mass flow rate	3.57%	
pressure transducers	0.08	bar
thermal power exchanged	2.90%	
oil flow rate	0.6	L/min

Table 2: Direct and derived measurement uncertainties

As reported in Figure 1, the evaporator performances were assessed through measurements across the inlet and outlet ports of both sides of the PHE. In particular, fluids temperatures were measured with T-type thermocouples while a gear flow meter was used to retrieve the oil flow rate. Unlike the oil that stayed in liquid phase while travelling along the evaporator, pressure measurements were additionally needed across the cold side of the heat exchanger to monitor the phase change of the organic fluid. In order to maximize the heat exchanger efficiency, the device was thermally insulated such that the R236fa mass flow rate was calculated from an energy balance. This indirect measurement led to a higher uncertainty on this quantity (Table 2). However, this issue can be overcome using a Coriolis mass flow meter, as did in [2, 18].

Table 3 presents a summary of the experimental campaign performed at working points. For a proper air compressor operation, the oil mass flow rate was kept constant at 1.25 kg/s in all the experimental cases, except for # 13. Furthermore, a by-pass of the oil radiator allowed an overheating of the lubricant up to 120 °C with respect to a conventional operating point at 80-90 °C (# 12). Since mass flow rate of the organic fluid could be varied acting on the revolution speed of the sliding vane rotary pump in the ORC system, this parameter assumes a significant spread and affects the operating pressure on the cold side of the plate heat exchanger. Indeed, high mass flow rates led to higher pressure drops along the circuit and allowed greater upper cycle pressures. On the contrary, the constancy of R236fa inlet temperature is due to fixed operating conditions at the condenser. All these remarks led to a thermal power recovery from 5.6 kW (# 11) up to 19.5 kW (# 7). In this case, the thermal power exchanged at the evaporator was equal the design one with water (20 kW), whose specific heat is almost 3 times than that of liquid R236fa at the inlet conditions of test # 7.

3. Modeling activity

The plate heat exchanger previously tested was further modeled using a steady one-dimensional approach to characterize heat transfer and pressure loss phenomena along fluid passages. In particular, pressures, mass flow rates and inlet temperature of both fluids are the model inputs while outlet temperatures, cold side pressure losses and overall thermal power exchanged are the calculated variables.

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case ♯	$T_{hot,in}$ °C	$T_{hot,out}$ °C	m_{hot} kg/s	$p_{cold,in}$ bar	$p_{cold,out}$ bar	$T_{cold,in}$ °C	$T_{cold,out}$ °C	m_{cold} kg/s	Q kW
1	114.8	107.7	1.34	12.4	12.1	42.9	90.6	0.111	18.8
2	118.1	110.9	1.25	12.7	12.4	45.0	96.1	0.111	19.1
3	117.3	110.5	1.25	12.1	11.9	45.1	98.3	0.103	18.0
4	117.1	110.7	1.25	11.4	11.2	44.7	100.1	0.095	16.9
5	121.4	115.4	1.25	10.8	10.6	45.0	106.4	0.086	16.0
6	116.9	111.1	1.25	10.6	10.4	46.6	101.6	0.086	15.3
7	119.7	112.3	1.25	13.4	13.2	48.1	98.3	0.115	19.5
8	118.3	113.4	1.25	9.9	9.8	46.5	104.5	0.071	13.0
9	119.1	115.5	1.25	8.8	8.6	46.5	107.6	0.052	9.80
10	120.6	117.7	1.25	7.9	7.8	46.0	110.8	0.039	7.5
11	120.4	118.2	1.25	6.7	6.5	45.9	112.4	0.029	5.6
12	84.3	80.5	1.28	7.1	7.1	36.0	71.7	0.059	9.9
13	118.9	109.2	0.94	12.4	12.1	48.2	90.5	0.110	19.3

Table 3: Summary of the experimental campaign

3.1. Thermal model

The model is based on the ϵ-NTU method and starts from the geometrical features of the heat exchanger. It follows a 1D approach in which flow channels defined by consecutive plates are divided in a sequence of elementary ducts. Therefore, for each j-th sub-volume, thermal power exchanged Q_j is represented by effectiveness ϵ and NTU values that can be calculated according to Eqns. 1 and 2 respectively.

$$\varepsilon_j = \frac{Q_j}{C_{\min,j}(T_{hot,in,j} - T_{cold,in,j})} = f\left(NTU_j, \frac{C_{\min,j}}{C_{\max,j}}\right) \tag{1}$$

$$NTU_j = \frac{U_j \cdot A_j}{C_{\min,j}} \tag{2}$$

where the j-th thermal exchange area A_j is the overall heat transfer area divided by the number of space discretization steps chosen and the minimum heat capacity rate as in Eqn. 3.

$$C_{\min,j} = \min(C_{hot,j}, C_{cold,j}) = \min(m_{hot}c_{hot,j}, m_{cold}c_{cold,j}) \tag{3}$$

The global heat transfer coefficient U_j is calculated by conductive and convective thermal resistances, according to Eqn. 4.

$$\frac{1}{U_j} = \frac{1}{h_{cold,j}} + \frac{t_{wall}}{k_{wall}} + \frac{1}{h_{hot,j}} \tag{4}$$

where t_{wall} and k_{wall} are thickness and thermal conductivity of the PHE's walls. On the other hand, convective heat transfer coefficients h are calculated relying on empirical correlations.

In this regard, considering that heat transfer in PHEs is very often enhanced by turbulent flow [19, 20], for single phase fluids most of the semi-empirical correlations that have been proposed in literature are based on the Dittus-Boelter one [21, 22]. Furthermore, since in the current application flow Reynolds number ($Re \simeq 10^6$) falls in the same range of the above mentioned correlation ($Re \geq 10^4$), convective heat transfer coefficient is calculated according to Eqn. 5.

$$h = 0.023\, Re^{0.8}\, Pr^n\, k_{fluid}/D_h \tag{5}$$

where the Prandtl number exponent n is equal to 0.4 for the cold fluid and to 0.3 for the hot fluid.

As concerns the phase change of the organic fluid, different correlations were tested in order to evaluate the best agreement with experimental data [23, 24].
The first correlation, reported in Eqn. 6, is the Kandlikar one.

$$h = (1.1837\mathrm{Co}^{-0.3} + 225.55\mathrm{Bo}^{2.8})(1-x)^{0.003}h_l \tag{6}$$

where Convection number Co and Boiling number Bo are defined in Eqn. 7.

$$Co = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_{sv}}{\rho_{sl}}\right)^{0.5} \qquad Bo = \frac{Q_j}{(H_{sv}-H_{sl})} \tag{7}$$

Since this correlation tends to under predict the heat transfer coefficient due to high values of the Boiling number exponent, a modified Kandlikar correlation [25] was additionally taken into account (Eqn. 8).

$$h = (1.1837\mathrm{Co}^{-0.3} + 225.55\mathrm{Bo}^{0.7})(1-x)^{0.003}h_l \tag{8}$$

h_l is the liquid heat transfer coefficient evaluated according to Eqn. 5.

Another suitable correlation for plate heat exchangers is the Yan-Lin [10] one that it is reported in Eqn. 9.

$$h = 1.1926\, {Bo_{eq}}^{-0.3}\, {Re_{eq}}^{0.5}\, {Pr_l}^{0.33}\left(1 - x + \sqrt{\frac{\rho_{sl}}{\rho_{sv}}}\right)\frac{k_{fluid}}{2b} \tag{9}$$

where equivalent Boiling and Reynolds numbers are evaluated according to Eqn. 10.

$$\mathrm{Re}_{eq} = \frac{m\left(1-x+x\sqrt{\frac{\rho_{sl}}{\rho_{sv}}}\right)2b}{A_s\mu_l} \qquad \mathrm{Bo}_{eq} = \frac{Q_j}{m\left(1-x+x\sqrt{\frac{\rho_{sl}}{\rho_{sv}}}\right)(H_{sv}-H_{sl})} \tag{10}$$

Moreover, two more general heat transfer correlations were considered. Among them, Klimenko's [26] is the unique correlation that accounts for material properties (Eqn. 11).

$$h = 0.087Re^{0.8}\,Pr^{1/6}\left(\frac{\rho_{sv}}{\rho_{sl}}\right)^{0.2}\left(\frac{k_{wall}}{k_{sl}}\right)^{0.09}\frac{k_{sl}}{2b} \tag{11}$$

while Shah and Thome correlation [27] has a better behaviour at lower liquid Reynolds number. In particular, it considers the largest value of Equations 12.

$$\begin{aligned} h &= 230\mathrm{Bo}^{0.5}h_l & h &= 1.8\left[\mathrm{Co}\left(0.38\mathrm{Fr}_l^{-0.3}\right)^{\nu}\right]^{-0.8}h_l \\ h &= F\cdot e^{2.47\left[\mathrm{Co}\left(0.38\mathrm{Fr}_l^{-0.3}\right)^{\nu}\right]^{-0.15}}h_l & h &= F\cdot e^{2.74\left[\mathrm{Co}\left(0.38\mathrm{Fr}_l^{-0.3}\right)^{\nu}\right]^{-0.1}}h_l \end{aligned} \tag{12}$$

where F is the Shah coefficient that it is equal to 0.064 for Bo>0.0011 and 0.067 otherwise. υ is equal to 1 for low liquid Froude number ($Fr_l < 0.04$) and to 0 elsewhere. Liquid Froude number is evaluated according to Eqn. 13.

$$\mathrm{Fr}_l = \frac{m^2}{2b\,g\,A_s^2\,\rho_l^2} \tag{13}$$

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3.2. Pressure drops

Measured pressure drops of the organic fluid were correlated to specific kinetic energy according to Eqn. 14.

$$\Delta p_{cold} = f \frac{2\, m_{cold}{}^2}{A_s{}^2\, \rho_m} \frac{L}{D_h} \tag{14}$$

where f is the Fanning factor.

4. Model validation

This section presents a comparison between modeling results and experimental measurements on the evaporator at the operating conditions listed in Table 3. In all the simulations, Dittus-Boelter correlation was used for single phase heat exchanges. On the other hand, the literature correlations previously proposed were considered for the phase change of the organic fluid [28, 29]. The goodness of numerical data was evaluated with reference to outlet hot and cold fluids temperatures, which are quantities measured directly as well as output data of the model. Results are presented in Figure 2 for the hot and cold fluids while Figure 3 presents the model validation in terms of thermal power exchanged.

Figure 2a shows a suitable accuracy of all the correlations except for the standard Kandlikar one that underestimates the thermal power exchanged in all cases. Klimenkos formulaiton seems to be quite far from measured values while other correlations have a similar behaviour, with an error always lower than 6 °C.

Thermal powers exchanged (Figure 3) confirmed the same behaviour of Figure 2b. Best fitting of measured values is represented by modified Kandlikar correlation, but also Yan-Lin and Shah-Thome ones have good results. Their mean error is about 3.5 %, with a maximum one of 10 %.

As concerns pressure drops at the cold side of the evaporator, experimental data were used to calculate the Fanning Factor as in Eqn. 14. Figure 4 shows the Fanning factor evaluated with respect to different mass flow rates of the cold fluid; indeed, it represents the link between specific kinetic energy and pressure drops occurred in heat exchanger. The fitting follows a hyperbolic law, confirming a literature result [30].

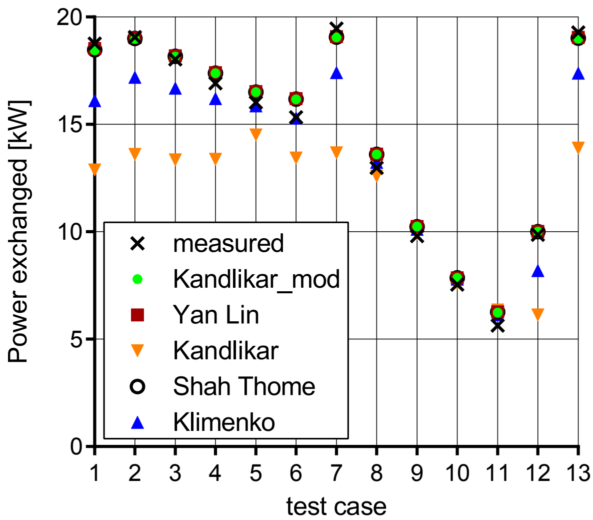


Figure 3: Model validation on thermal power exchanged

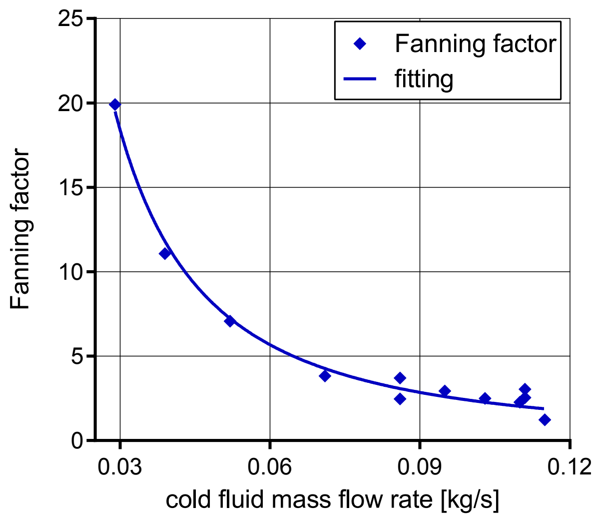
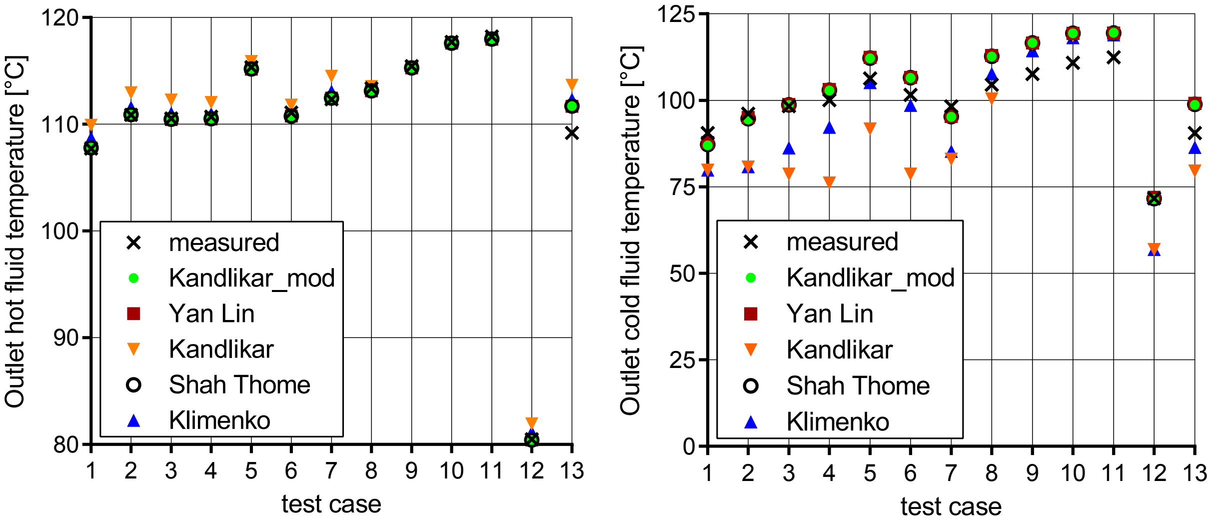


Figure 4: Fanning factor calculated in test cases



(a) hot side (b) cold side

Figure 2: Comparison between measured outlet temperatures with model results obtained using different correlations for the phase change of the organic fluid

5. Off-design analysis

Even though the modeling methodology developed for plate heat exchangers can be applied to any waste heat recovery application that uses oil and R236fa with phase change, the current study focused on the occurrence of off-design conditions due to an unconventional operation of the sliding vane air compressor whose lubricant was used as upper thermal source of the ORC-based energy recovery system displayed in Figure 1. In an industrial site, compressed air needs are usually variable according to production level. For this reason, the air compressor should always guarantee that air is available at almost the delivery pressure value. Therefore, the most common strategies to vary compressed air supply to match the demand are a regulation on the revolution speed of the compressor through an inverter or throttling the suction valve to reduce the air flow rate.

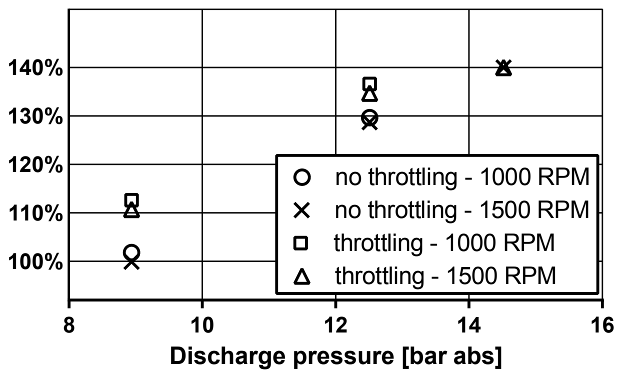


Figure 5: Oil flow rate at off design with respect to test # 7 of Table 3 (8.5 bar at 1500 RPM)

	min	max	
$T_{hot,in}$	100.0	120.0	°C
m_{hot}	1.00	1.50	kg/s
m_{cold}	0.055	0.115	kg/s
p_{cold}	10.0	13.4	bar$_a$

Table 4: Parameters and simulation ranges for the off-design analysis

As reported in Figure 5, in sliding vane compressors only the latter approach leads to a change in the oil flow rate. Moreover, if flow rate changes, oil temperature at the compressor outlet is also affected. Since both quantities would influence the recovery plate heat exchanger behavior,

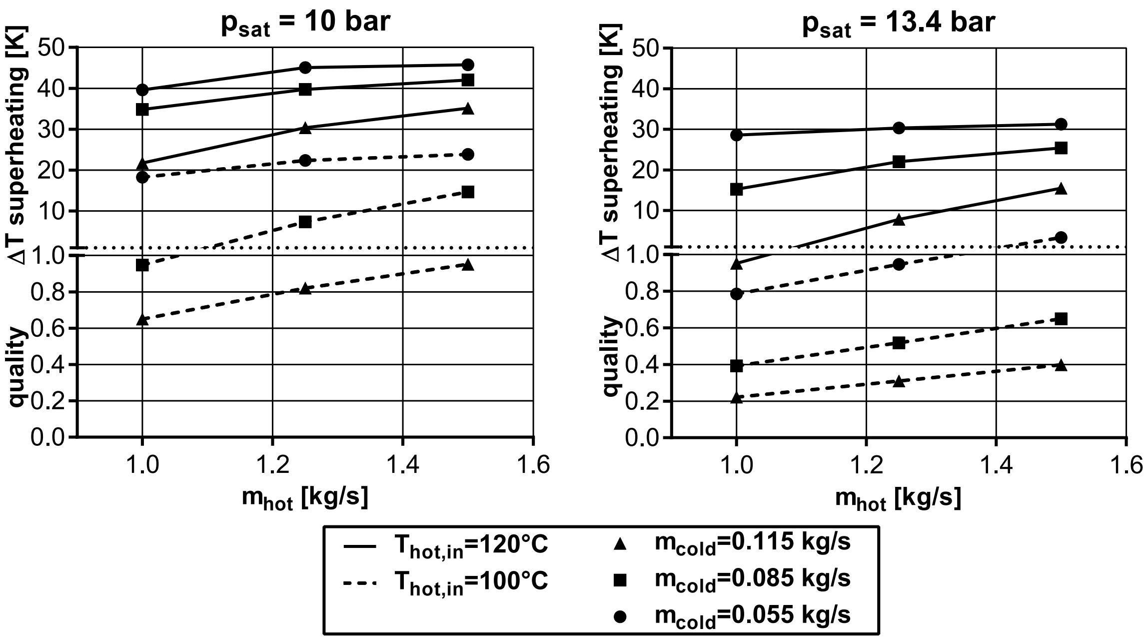


Figure 6: Effects of off-design operating conditions of a plate heat exchanger for waste heat recovery in terms of quality and superheating or subcooling of the organic fluid at the heat exchanger outlet

the off design analysis explored different operating conditions taking into account a variability of organic working fluid mass flow rate and vaporization pressure according to the ranges reported in Table 4.

Figure 6 reports a summary of the off-design analysis carried out at different oil mass flow rate and inlet temperature values. The comparison is made with respect two thermodynamic quantities of the organic fluid, namely quality and degree of superheating (quality>1) or subcooling (quality<0). The choice on these parameters is motivated on the need to ensure a correct operation of the ORC-based power unit even when the compressor operates far from the nominal conditions. In particular, multiple studies on ORC systems confirm that for a proper energy recovery, outlet conditions at the evaporator should consider a 5-10 K of superheating of the working fluid since wet expansion is an unfavourable situation for expansion machines, especially for turbines [31]. Simulations at all pressure levels show that inlet oil temperature highly affect the outlet conditions of R236fa. Indeed, if the thermal energy recovery took place at low temperatures ($T_{hot,in} = 100°C$), the organic fluid would never reach a complete vaporization, except for a particular case ($m_{cold} = 0.085\,kg/s$, $p_{cold} = 10\,bar$). Moreover, for a given inlet temperature of the cold fluid, vaporization pressure affects the shares of thermal powers needed for pre-heating, vaporization and superheating. For this reason, Figure 6 shows that the higher is the saturation pressure, the lower is the quality of the organic fluid at the evaporator outlet.

According to the trends reported in Figure 6, two control strategies can be outlined to adapt the ORC system at off-design operating conditions imposed by the upper thermal source:

- **variable oil mass flow rate**: if one considers a given inlet temperature of the hot stream ($T_{hot,in}$) and fixed operating conditions of the organic flow circuit (m_{cold}, p_{cold}), a by-pass of the oil mass flow rate at the evaporator could tune the heat gain of the organic fluid to keep it superheated.
- **variable organic fluid mass flow rate**: for given conditions at the hot side of the evaporator ($m_{hot}, T_{hot,in}$), organic fluid mass flow rate might be adjusted to ensure the

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superheating. This strategy would be implemented through a feed-forward control on the pump revolution speed of the ORC system. However, this approach should also consider the additional influence of mass flow rate on pressure losses of the organic fluid circuit. Indeed, the higher is the mass flow rate, the higher would be pressure losses that the pump should overcome and, in turn, the vaporization pressure of the organic fluid.

The first strategy is easier to implement (e.g. through a calibrated by-pass valve) but would reduce the energy recovered. On the other hand, in view of a greater complexity, the latter approach would lead the recovery power unit to suitably perform a mechanical conversion of the low grade thermal waste. The optimal regulation technology will definitively depend on economical considerations that can not be forecasted at this development stage of the technology.

6. Conclusions

The heat recovery steam generator is a fundamental component of waste heat recovery systems which are based on an organic Rankine cycle since it has to accomplish an efficient energy recovery with multiple factors that constraint its operation such as space, weight, operating pressures etc.. An even more challenging situation occurs whether the energy recovery system worked at off-design conditions since vaporization and slight superheating of the organic working fluid must be anyway guaranteed for a proper mechanical energy recovery through the expansion machine. To address these issues, this research work analyzed the performances of a 20 kW plate heat exchanger both with numerical and experimental approaches with reference to a waste heat recovery application on industrial air compressors. Experimental data allowed to validate the heat exchanger model in several operating points, whereas the modified Kandlikar semi-empirical correlation turned out to be the most appropriate formulation to represent the phase change of the working fluid R236fa. The off-design analysis that was eventually carried out on the PHE showed that in order to ensure a slight (5-10 K) superheating of the organic working fluid at the heat exchanger outlet, oil mass flow rate might be by-passed or organic fluid mass flow rate could be tuned on the thermal power available at the evaporator. The first control strategy is easier to implement but reduces the energy recovery. On the other hand, a control of the organic fluid mass flow rate acting on the pump revolution speed of the ORC system would maximize the energy recovered but it might be more complex and costly.

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Nomenclature

A	heat transfer area	m^2	t	plate thickness	mm
A_N	PHE nominal heat tr. area	m^2	T	temperature	K
A_s	flow cross sectional area	m^2	U	overall heat tr. coeff.	$W/(m^2\,K)$
b	corrugation amplitude	mm	W	plate width	mm
Bo	boiling number	$-$	x	vapor quality	$-$
c	specific heat at const. pressure	$J/kg/K$	Δp	pressure drop	Pa
C	heat capacity rate	W/K	ϵ	efficiency	$-$
Co	convection number	$-$	μ	dynamic viscosity	$Pa\,s$
D_h	hydraulic diameter	m	ρ	density	kg/m^3
F	Shah coefficient	$-$		**subscripts**	
f	Fanning factor	$-$	hot	hot fluid	
Fr	Froude number	$-$	cold	cold fluid	
g	gravitational acceleration	m/s^2	in	inlet	
h	convective heat tr. coeff.	$W/(m^2\,K)$	out	outlet	
H	specific enthalpy	J/kg	sat	saturation	
k	thermal conductivity	$W/(m\,K)$	j	discretization index	
L	plate length	mm	max	maximum	
m	mass flow rate	kg/s	min	minimum	
N	number of plates	$-$	wall	heat exchanger wall	
n	Prandtl number exponent	$-$	fluid	generic fluid	
NTU	number of thermal units	$-$	sl	saturated liquid	
p	pressure	Pa	sv	saturated vapor	
Pr	Prandtl number	$-$	l	liquid	
Q	thermal power	W	eq	equivalent	
Re	Reynolds number	$-$	m	mean	