Effects of void fraction and heat transfer correlations in a charge-sensitive ORC model – a comparison with experimental data

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Abstract:
In order to properly evaluate the off-design performance of an ORC unit, it is important to use simulation tools that minimize the number of assumptions regarding the system state. To avoid imposing the condenser subcooling (or any other equivalent state variable), the ORC model should account for the mass repartition of working fluid through the unit in function of the operating conditions. Among the various components constituting ORC power systems, the proper modelling of the mass of working fluid enclosed in the heat exchangers is of primary importance. The goal of this work is to develop such a reliable charge-sensitive ORC model. To this end, a 2kWe recuperative ORC system is used as case study and experimental measurements are used as reference dataset. The ORC system features two brazed plate heat exchangers and one fin coil condenser. For these three heat exchangers, a large set of empirical correlations is investigated in order to evaluate both the fluids void fractions and their convective heat transfer coefficients. By comparing the models predictions with the experimental data, the study highlights the limitations of existing correlations and investigates three different correction methods to improve them. Ultimately, the analysis compares nine models of brazed plate heat exchangers, two models of condenser, three correction methods for improving the heat transfer correlations and four void fraction methods. Accounting for every combination possible, 288 different models of the ORC are compared for predicting both the heat exchangers thermal performance and the total mass enclosed in the ORC unit. Out of this study, the best modelling approach is identified and details of its charge inventory predictions are presented.

Keywords:
ORC, off-design, charge-sensitive, heat transfer, void fraction, correlations

1. Introduction

A common aspect of ORC power systems is the versatile nature of their operating conditions. Whether for solar, geothermal or waste heat recovery applications, the heat source (and even sometimes the heat sink) conditions may fluctuate in time, which imposes the system to adapt its working regime for performance or safety reasons. Therefore, once sized and built, an ORC engine is often led to operate in off-design conditions. To properly evaluate the performance of such ORC systems in off-design conditions, reliable and validated simulation tools must be used. In order to best replicate the physics of the engine, these models should avoid any user-defined assumption in the system operation (e.g. an imposed sub-cooling, super-heating, condensing pressure, etc.) and the ORC performance should be derived from its boundary conditions only (i.e. the heat source and heat sink supply conditions, the mechanical components rotational speed, the heat exchanger surface area, etc.). In order to reach such modelling level, the ORC model must be charge-sensitive i.e. to account for the mass repartition of working fluid through the system in function of the operating conditions [1]. Among the various components composing an ORC system, it is crucial to properly estimate the mass enclosed in the heat exchangers because of their relatively high internal volume. Unlike in steam power plants or higher-capacity systems, small-scale ORC units often use a single once-through heat exchanger to perform the complete heating (or cooling) , including phase change, of the working fluid. Consequently, several phases of fluid coexist in a same component (i.e. liquid, two-phase and vapour) and the heat exchanger is divided in multiple zones. The mass of working fluid enclosed in a heat exchanger of volume V_{HEX} can be computed as the sum of the masses included in each of the *N* sub-zones, i.e.

$$M_{HEX} = \sum_{i=1}^{N} \overline{\rho}_i \, (\omega_i \, V_{HEX}) \qquad (1)$$

where ω_i is the volume fraction occupied by the ith zone in the heat exchanger and $\overline{\rho}_i$ is the corresponding mean density of the fluid. Based on this equation, it appears that a reliable charge-sensitive ORC model implies two main conditions:

1) A proper knowledge of ω_i, the volume fraction occupied by each zone of fluid in the different heat exchangers. This spatial division is directly affected by the *convective heat transfer coefficients* used to simulate the heat exchangers.

2) A proper evaluation of the fluid density along the heat exchangers. In the case of single-phase zones, the density calculation is straightforward and can be determined from the fluid thermodynamic state. In the case of a two-phase flow, however, the density is not only function of the fluid pressure, temperature and quality, but it also depends on the void fraction characterizing the flow. Therefore, the density calculation is directly impacted by the *void fraction model* used to characterize the boiling or condensing processes.

As demonstrated latter in the text, many different correlations may be found in the scientific literature to evaluate these void fractions and convective heat transfer coefficients. However, most of these correlations are derived empirically, based on limited experimental data and their reliability for extrapolated conditions is not ensured.

The goal of this work is to investigate different correlations of heat transfer coefficient and void fraction to model the heat exchangers in a charge-sensitive ORC model. To this end, a 2kWe ORC unit is used as case study. The reliability of the correlations is evaluated by confronting the simulations results to experimental measurements, both in terms of thermal performance and charge inventory predictions. The steady-state simulations are performed in Matlab and the heat exchangers are modelled so as to compute the heat transfer based on the fluids supply conditions only. The model is of type “moving-boundary” (see [[2], [3]] for further information) and the effective heat transfer in the heat exchanger is calculated to match the total surface area occupied by the *N* different zones with the geometrical surface area of the component A_{HEX} i.e.

$$A_{HEX} = \sum_{i=1}^{N} A_i \qquad (2)$$

The heat exchanger model developed for this work is open-source and can be found in the ORCmKit library [4]. The paper is structured as follows: firstly, in section 2, the test rig used as case study and the reference dataset is described. In section 3, the thermal performance obtained with the various heat transfer correlations is assessed. This analysis highlights the limitations of the original correlations and three correction methods are proposed for improving them. Finally, in section 4, the effects of the void fraction models are evaluated when computing the total mass enclosed in the ORC system. In total, 288 different simulations are performed. Out of this analysis, the best modelling approach is identified and details of its charge inventory predictions are presented. It is worthwhile to note that similar analyses exist for HVAC systems ([5]–[7]), however, to the best of the authors knowledge, such study has never been presented for a ORC power unit.

2. Experimental set up and dataset

The system investigated as case study is the Sun2Power ORC unit developed by the University of Liège [8]. It is a 2kWe recuperative ORC system using R245fa as working fluid and built to be operated with solar collectors (although the experimental data presented in this paper are gathered using an electrical boiler as heat source). The system consists of three heat exchangers, a diaphragm pump, a scroll expander and a liquid receiver. Both the recuperator and the evaporator are thermally-insulated brazed plate heat exchangers while an air-cooled fin coil heat exchanger is used for the condenser. The system features three control variables: the pump and the fan condenser rotational speeds (controlled with a variable frequency drive) and the expander rotational speed (controlled by adjusting the electrical load). As depicted in Figure 1, the system is also instrumented with thermocouples, flow meters, power meters and pressure sensors in order to properly measure the performance of the complete test rig and of each subcomponent individually. Although it would have been interesting to measure the working fluid mass repartition through the system, the ORC test rig does not feature individual scales under each component. However, the total charge of R245fa enclosed in the system is known to be 26kg (± 0.5 kg).

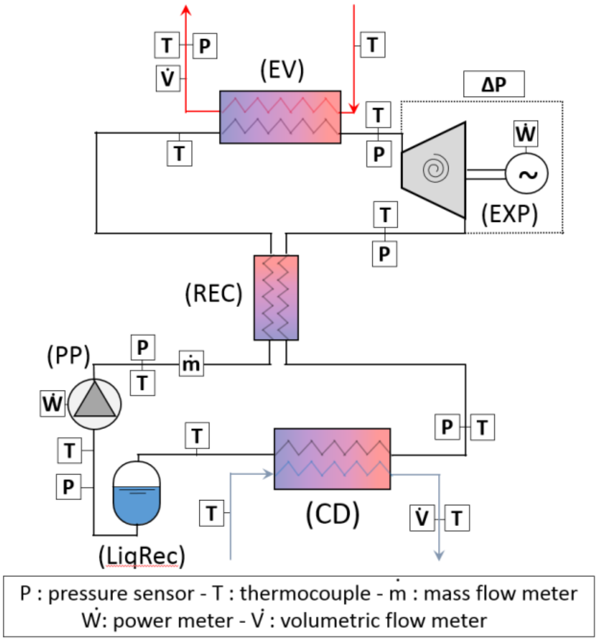


Figure 1. Scheme of the Sun2Power ORC test rig

Using this test rig, an experimental campaign is performed and a set of 40 steady-state points are gathered as reference dataset. These reference points are obtained by averaging the experimental measurements over 5-minute periods in stabilized regimes and by the application of a data post-treatment. For further details regarding this reference dataset and the reconciliation method applied to the raw measurements, please refer to [1]. As a summary, the ranges of operating conditions covered by the reference dataset are given in Table 1.

Table 1. Operating ranges of the reference dataset

Variable	***Min***	***Max***	***Unit***
Working fluid mass flow rate	16	70	*g/s*
Heat source mass flow rate	910	990	*g/s*
Heat source supply temperature	88	120	*°C*
Heat sink mass flow rate	0.149	1.45	*kg/s*
Heat sink supply temperature	17	25	*°C*
Evaporating pressure	6.5	14.4	*bar abs.*
Condensing pressure	1.6	6.6	*bar abs.*
Net power output	23	1170	*W*
Net thermal efficiency	0.5	8	*%*

3. Heat transfer correlations analysis

As presented in the previous section, the ORC system features two types of heat exchangers, namely an air-cooled fin coil condenser and two brazed plate heat exchangers (BPHEXs). In order to simulate their thermal performance and to properly evaluate the volume occupied by each zone of fluid, it is important to use reliable correlations for calculating the fluids convective heat transfer coefficients. For both technologies, many different correlations may be found in the scientific literature (e.g. see [9]–[11] for the BPHEXs and [12] for the fin coil condenser). These correlations generally evaluate the Nusselt number as a function of the flow conditions, the fluids properties and some geometrical parameters of the heat exchanger. In most cases, these correlations are purely empirical and calibrated to fit some experimental data. Because there is not one unique reference correlation for characterizing these heat exchangers, several candidates (among the most common employed nowadays) are tested for modelling both types of technologies. More specifically, nine different correlations are investigated to characterize the brazed plate heat exchangers, i.e.:

- 3 correlations for single-phase flows in both the recuperator and the evaporator, respectively the correlations proposed by Martin [13], Wanniarachchi [14] and Thonon [9]. It must be noted that the same single-phase correlation is applied for both the hot and cold fluid.

- 3 correlations for the boiling flow in the evaporator, i.e. the correlations proposed by Amalfi et al. [15], Han et al. [16] and Cooper [17].

- 3 correlations for the condensing flow in the recuperator, namely the correlations proposed by Longo et al. [18], Han et al. [19] and Shah [20].

Regarding the air-cooled condenser, the following correlations are tested:

- 1 general correlation proposed by Gnielinski [21] for the single-phase flow of working fluid in horizontal tubes.

- 1 general correlation proposed by Cavallini et al. [22] for the condensing flow of working fluid in the condenser tubes.

- 2 correlations for the air flow across the coil, i.e. the empirical laws proposed in [21] and the one proposed by Wang et al. [23].

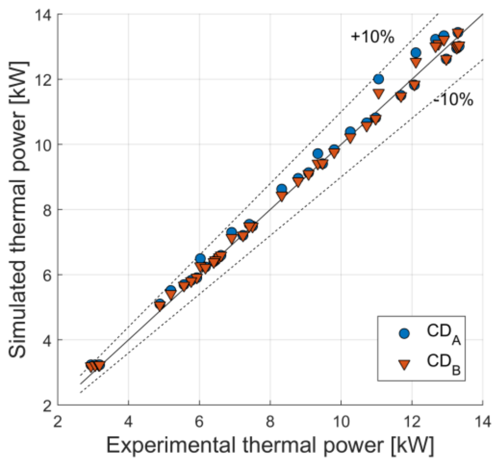
Fewer correlations are tested for the condenser because of the better characterized flow occurring in the horizontal tubes. For the sake of conciseness, the equations constituting the heat transfer correlations are not presented in the current paper, but they may be found in the corresponding references. Because of the multiple zones coexisting in the heat exchangers, correlations must be coupled to simulate the components. Since the single-phase and two-phase correlations are, *a priori*, independent to each other, any combination between them is possible. As shown in Table 2, it results nine different models for the brazed plate heat exchangers (i.e. the evaporator and the recuperator), and two models for the condenser. In order to assess their reliability, the different models presented in Table 2 are evaluated in the exact same conditions as in the reference dataset and the heat transfer predicted by each model is confronted to the experimental observations.

In the case of the condenser, the two models investigated (i.e. the models CD_A and CD_B in Table 2) demonstrate fairly good agreement with the experimental data. As shown in Figure 2a, both models properly predict the heat transfer and the modelling deviations are smaller than 10% (on average, the relative errors are 2.6% and 1.9% for the models CD_A and CD_B respectively). Regarding the brazed plate heat exchangers, the modelling performance of the different correlations is much poorer. For both the recuperator and the evaporator, it appears that *all* the models, in every case, highly overpredict the heat transfer in the BPHEXs. In the case of the recuperator, as depicted in Figure 2b, the heat transfer is overpredicted by 40% on average and similar results are observed for the evaporator.

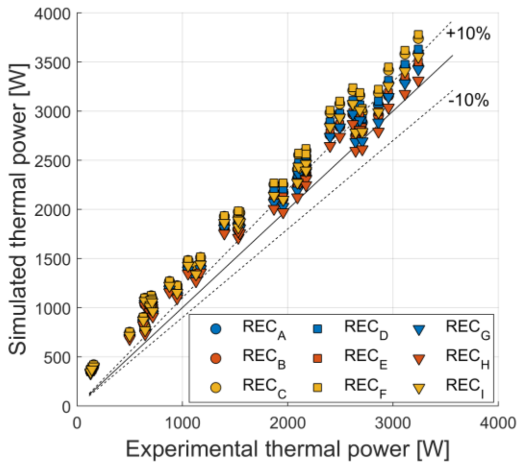
Table 2: Heat exchangers models investigated for the evaporator, the recuperator and the condenser (NB: [1]Han's boiling correlation [16]*; [2]Han's condensation correlation* [19]*)*

Evaporator models (correlations for single-phase & boiling flows in BPHEX)		
EV_A : Martin + Almalfi	EV_D : Martin + Han (1)	EV_G : Martin + Cooper
EV_B : Wanniarachchi + Almalfi	EV_E : Wanniarachchi + Han (1)	EV_H : Wanniarachchi + Cooper
EV_C : Thonon + Almalfi	EV_F : Thonon + Han (1)	EV_I : Thonon + Cooper
Recuperator models (correlations for single-phase & condensing flows in BPHEX)		
REC_A : Martin + Longo	REC_D : Martin + Han (2)	REC_G : Martin + Shah
REC_B : Wanniarachchi + Longo	REC_E : Wanniarachchi + Han (2)	REC_H : Wanniarachchi + Shah
REC_C : Thonon + Longo	REC_F : Thonon + Han (2)	REC_I : Thonon + Shah
Condenser models (correlations for single-phase & condensing flows in tubes + air flow through the coil)		
CD_A: Gnielinski + Cavallini + VDI	CD_B: Gnielinski + Cavallini + Wang	

These large deviations may be explained by different reasons. Firstly, the single-phase correlations are all empirically determined using water as heat transfer fluid and their extrapolability to organic fluids is not verified. Secondly, even though the two-phase correlations are developed with refrigerants (or other organic fluids), the operating conditions used to calibrated them are more typical of refrigeration systems than those encountered in ORC systems. In the case of condensing flows, the saturation temperatures in the two technologies (ORC and HVAC) are of the same order (20°C to 50°C). However, the evaporating temperature in HVAC systems is much smaller than in ORC power units (-10°C/ 20°C vs. 80°C/150°C) and the validity of the correlations to characterize boiling flows in ORC systems is not guaranteed. Finally, the correlations employed require a good knowledge of the BPHEX geometry (i.e. the chevron angle, the plate thickness, the enlargement factor, the corrugation pattern, etc.). Although some characteristics may be retrieved indirectly from the heat exchanger datasheet, uncertainties remain and these may alter the proper predictions of the correlations.



(a)



(b)

Figure 2. Heat transfer predictions (a) in the condenser (b) in the recuperator

Based on these results, it appears that the correlations investigated in this work should not be used directly to simulate the ORC system and its heat exchangers (especially for the BPHEXs). It is therefore proposed to modify these correlations in order to better fit the experimental measurements. To this end, three correction methods are investigated for all the models presented in Table 2:

- Correction #1: only the most influential correlation in the heat exchanger (i.e. the one referring to the highest thermal resistivity) is scaled by means of a single factor c, i.e.

$$Nu_{new} = c.Nu \qquad (3)$$

- Correction #2: idem as correction #1, except that all the correlations in the heat exchanger are corrected with independent factors c_j, i.e.

$$Nu_{new,j} = c_j.Nu_j \qquad (4)$$

  Therefore, two (resp. three) correction factors are used for correcting each BPHEX (resp. condenser) model.

- Correction #3: idem as correction #2, except that, additionally, the empirical exponents on the Reynolds number are also scaled by factors c_k, i.e.

$$Nu_{new,jk} = c_j.Nu_{init}(Re^{c_k.m}) \qquad (5)$$

  In total, four (resp. six) correction factors are used for modifying each BPHEX (resp. condenser) model.

For each model proposed in Table 2, the correction factors c_{jk} are tuned by minimizing the residuals between the model predictions and the experimental data. More specifically, the following objective function F is optimized by means of an interior-point algorithm [24], i.e.

$$\min_{c_{jk}} F = \beta.NRMSE \; + (1-\beta) \sum_{j=1}^{M} \frac{(1-c_{jk})}{M} \qquad (6)$$

where the first term is referring to the Normalized Root Mean Square Error (as defined in (7)), the second term accounts for the correction applied to the correlations (aimed to be minimized as well) and β is a weighting factor (set to 0.95 in this work).

$$NRMSE = \frac{1}{\dot{Q}_{max} - \dot{Q}_{min}} \sqrt{\frac{\sum_{i=1}^{N}\left(\dot{Q}_{exp,i} - \dot{Q}_{sim,i}\right)^2}{N}} \qquad (7)$$

In order to compare the effect of the different correction methods, averaged values of the NRMSE related to each modelling approach are given in Figure 3. As expected, the larger the modifications applied to the correlations, the better the fitting of the experimental data. The effect of each correction method on the individual models is detailed in the next section (Figure 4).

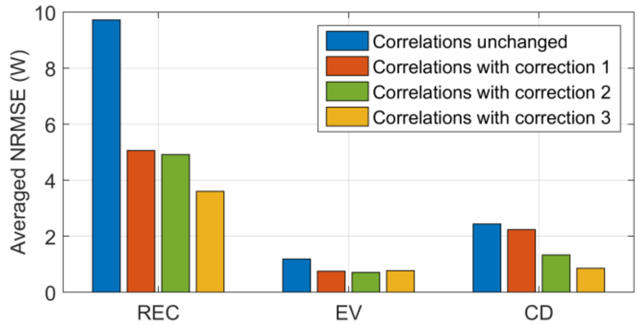


Figure 3. Average values of the NRMSE for the heat exchangers in function of the correction method applied to the correlations

4. Void fraction correlations analysis

As mentioned in the introduction, the density of a flowing two-phase mixture is not function of its thermodynamic state only, but it also depends on the spatial fraction occupied by the liquid and vapour phases. To account for this effect, the flow can be characterized by the void fraction α, defined as:

$$\alpha = \frac{A_v}{A_{tot}} = \frac{1}{1+\frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)S} \qquad (8)$$

where A_v and A_{tot} are respectively the vapour flow and the total flow cross-sectional areas, x is the fluid quality, ρ_v and ρ_l are the saturated vapour and saturated liquid densities, and S is the slip (velocity) ratio between the vapour phase and the liquid phase. Based on the void fraction α, the mass M_{tp} enclosed in a two-phase zone of length L and of cross section area A can be calculated by

$$M_{tp} = M_v + M_l = A\,\left(\rho_l \int_0^L (1-\alpha)\,dl + \rho_v \int_0^L \alpha\,dl\right) \qquad (9)$$

As for the heat transfer coefficients, several correlations may be found in the scientific literature to characterize the void fraction in function of the flow operating conditions. In this work, four of the most common void fraction models are considered, namely:

- The homogenous model, which assumes a slip-ratio S equal to 1;
- The model proposed by Zivi [25], which computes the slip-ratio such as $S = (\rho_v/\rho_l)^{-1/3}$;
- The model proposed by Lockhart-Martinelli [26], which calculates α as an empirical function of the parameter X_{tt} (see the reference paper for further details);
- The empirical mass-flux-dependent model proposed by Hughmark [27].

By using these four void fraction models, the total mass of working fluid enclosed in the ORC system is computed over the whole dataset (40 points in total) and compared to the experimental charge. The total charge of R245fa is calculated as the sum of the masses enclosed in the various components of the ORC system, i.e.

$$M_{ORC} = \overbrace{M_{ev} + M_{cd} + M_{rec}}^{heat\ exchangers} + \overbrace{M_{pipings} + M_{pp} + M_{exp} + M_{liq,receiver}}^{other\ components} \qquad (10)$$

To numerically quantify the reliability of the mass prediction, both the mass mean value (i.e. the average mass computed over the 40 experimental points) and the mass standard deviation (to illustrate the mass scattering) are computed for the different modelling approaches. Because there are two models of the condenser, nine models for the BPHEXs and 4 correction methods, all independent from each other, 72 different models of the ORC system are built (i.e. all the combinations possible are evaluated). For each of these 72 ORC models, the four void fraction methods are used to compute the total mass of working fluid.

Ultimately, 288 charge-sensitive simulations are performed and the results are given in Figure 4. In this figure, to each abscissa position corresponds a unique combination of BPHEX model, condenser model and correction method (refer to the legends for better clarity). The three top subplots illustrate the residuals (NRMSE as defined in (7)) related to the heat transfer prediction for each heat exchanger (in comparison to the experimental data). The fourth subplot depicts the mean mass value computed for the total system over the 40 reference points (aimed to be close to the experimental charge of 26 ± 0.5kg). Finally, the last subplot depicts the standard deviation while evaluating the mass over the 40 reference experimental points (aimed to be as low as possible).

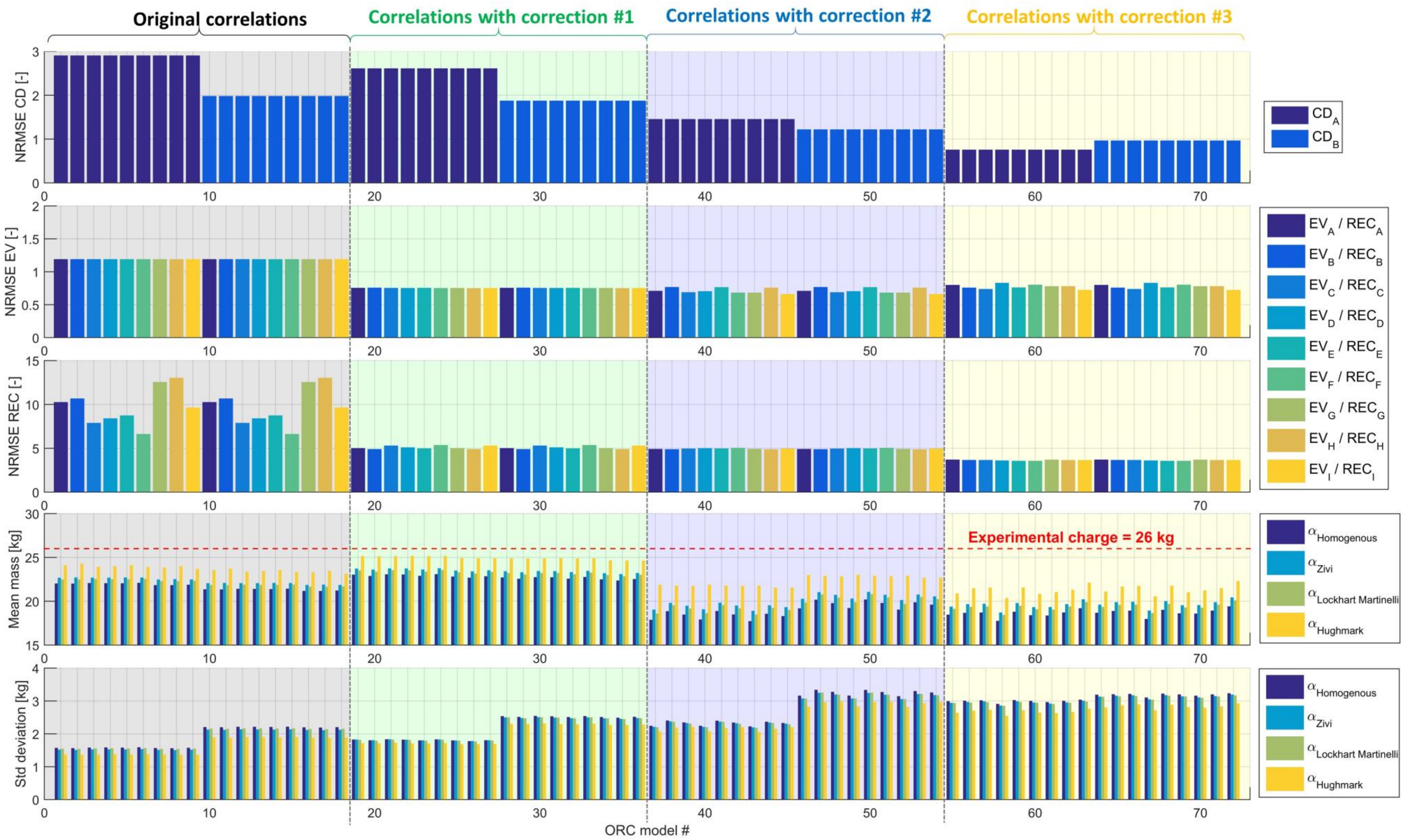


Figure 4. Global results of the comparative study for the ORC system in terms of heat transfer and charge inventory predictions. Each abscise (from 1 to 72) is referring to a single ORC model, i.e. a unique combination of BPHEX model ($EV_A/REC_A \rightarrow EV_I/REC_I$), condenser model ($CD_A$ or CD_B) and correction method (original correlation or with correction #1 to #3). The 3 top subplots depict the NRMSE committed on the heat transfer prediction of each heat exchangers. The 4^{th}subplot depicts the mean total mass predicted over the 40 experimental points. The 5^{th} plot illustrates the standard deviation in the mass prediction over the dataset.

Based on this figure, the following elements can be highlighted:

- Regarding the condenser, the model CD_B shows a slightly better thermal performance (i.e. it better fits the experimental data in terms of heat transferred in the condenser), but it induces larger scattering in the global mass calculation (i.e. larger standard deviations of M_{ORC}) whatever the correction method employed.
- In all situations, the homogenous and the Hughmark's void fraction models lead to the lowest and the highest mass estimations, respectively. On average, the difference between their total mass predictions is 2.54kg. On the other hand, the void fraction proposed by Zivi and Lockhart-Martinelli computes similar intermediate charge inventories (generally closer to the predictions of the homogenous model). Whatever the correction method and the heat transfer correlations employed, Hughmark's void fraction model (which accounts for the influence of the fluid mass flux) appears to be the best modelling approach to assess the global charge of refrigerant in the system. In every case, it leads to the lowest error in the prediction of the charge and to the lowest standard deviation.
- The mass standard deviation is not much influenced by the void fraction model, but rather by the correction methods applied to the heat transfer correlations. As shown in the previous section, the larger the modifications applied to the correlations, the better the fitting of the heat power transferred in the heat exchangers. However, from Figure 4 it can be seen that the larger the modifications applied to the correlations, the wider the scattering in the mass predictions. By altering the heat transfer correlations, the thermal performance of the heat exchanger models is improved but the charge inventory is deteriorated. Furthermore, the correction method also influences the mean mass prediction over the entire dataset. According to this analysis, the most suited correction method to be applied to the heat transfer correlations is the method #1. By only scaling one heat transfer correlation, this correction method offers the best trade-off between:
    - improving the heat exchangers models thermal predictions;
    - estimating the global charge of working fluid in the system both in terms of good mean mass value and low mass prediction scattering.

From these results, the best modelling approach to perform a charge-sensitive simulation of the system appears to be the ORC model #19 (i.e. the models $EV_A + REC_A + CD_A$, all modified by means of the correction method #1) using Hughmark's void fraction model. The detailed charge inventory predicted by this model for the 40 points of the reference dataset is depicted in Figure 5. As can be seen, the mass mean value (25.12kg) fits well the experimental charge enclosed in the ORC system (26±0.5kg). The scattering of the global mass prediction is fairly limited, with a standard deviation of 1.71kg. The deviation observed from the experimental charge is maximum 13.5% ($M_{ORC,max}$ = 29.5 kg) and drops to 6.2% on average. According to the simulations, the largest part of the fluid is found in the condenser, the liquid receiver and then the evaporator. Without any weight measurements of the individual system components, this charge-sensitive model cannot be further validated.

5. Conclusion

The goal of this work is to investigate and compare different empirical correlations for simulating the heat exchangers in a charge-sensitive ORC simulation tool. To this end, a 2kWe recuperative ORC unit is used as case study and experimental measurements are gathered as reference dataset. The ORC system features two brazed plate heat exchangers and one fin coil condenser. For these three heat exchangers, various empirical correlations derived from the scientific literature are tested in order to evaluate both the fluids void fractions and their convective heat transfer coefficients. The reliability of these models is assessed by confronting the simulation predictions to the experimental data.

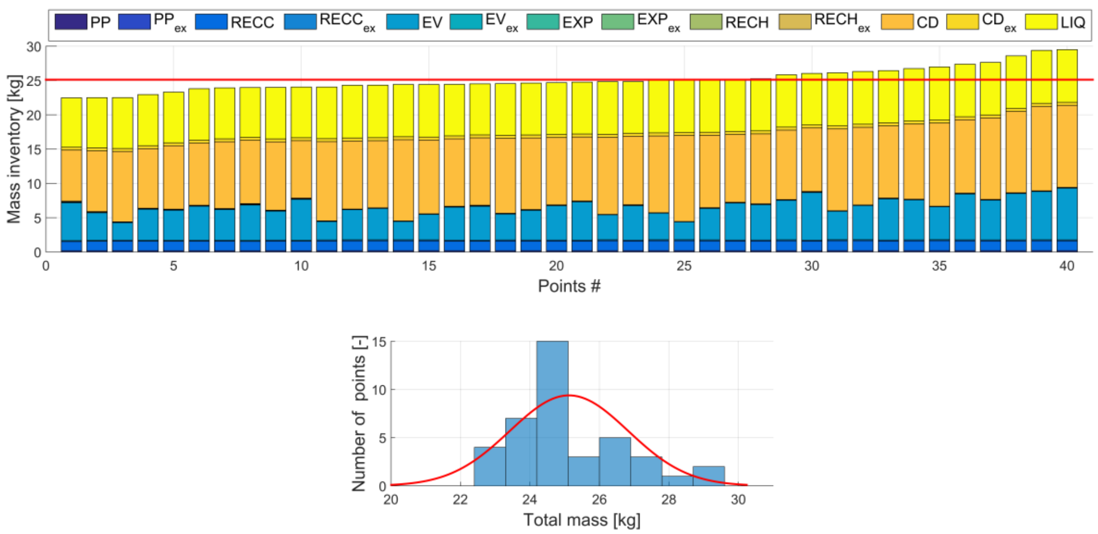


Figure 5. Charge inventory predicted by the ORC model #19 with Hughmark's void fraction

More specifically, the current study compares:

- 9 models of brazed plate heat exchangers built by combining different heat transfer correlations of single-phase, condensing and boiling flows;
- 2 models of air-cooled fin coil condenser;
- 3 correction methods to enhance the models thermal prediction;
- 4 void fraction models (homogenous, Zivi, Lockhart-Martinelli and Hughmark).

Ultimately, 288 different models of the ORC are compared for predicting both the heat exchangers thermal performance and the total mass enclosed in the ORC unit. The main outcomes of the study are the following ones:

- The original correlations characterizing the heat transfer coefficients in the brazed plate heat exchangers should not be considered trustful and they require to be corrected for improving their thermal performance predictions. The models of the condenser, on the other hand, demonstrate better agreement with the experimental data.
- By altering the heat transfer correlations, the thermal performance of the heat exchanger models is improved but the charge inventory of the ORC system is deteriorated. The best correction method applied to the heat transfer correlations appears to be the less intrusive (correction method #1). By only scaling one correlation in the heat exchanger models (see equation (3)), it is possible to much better predict the heat transfer in the heat exchangers while ensuring a proper mass estimation in the ORC unit.
- The best void fraction model identified is the one proposed by Hughmark which accounts for the influence of the fluid mass flux. It is interesting to note that similar analyses applied to HVAC systems [e.g. [5], [28]] also identified this model to be the best for characterizing the fluids void fraction.
- Accounting for the fact that the experimental charge in the ORC system was 26kg, the best modelling approach permits to estimate a mean mass of 25.12kg over 40 different operating conditions with a standard deviation of 1.7kg. The deviation from the experimental charge is maximum 13.5 % and it drops to 6.3% in average.
- In the current analysis, the best modelling method is chosen to be the one offering a good compromise between reliable thermal performance predictions of the heat exchanger models and proper mass estimation in the ORC system. Future work would require to better assess

the influence of these effects on the performance modelling of the global ORC system. Indeed, it might be interesting to give more credit to one criteria over the other (i.e. the proper predictions in the heat exchangers over the mass modelling in the system) if its effect was demonstrated to be more important on the simulation of the entire system.

- In this work, the charge-sensitive ORC model is validated in terms of global charge prediction only. In order to better validate the model and its charge inventory predictions, experimental measurements of the mass repartition *between* the different components would be required.

Nomenclature

Acronyms

BPHEX	Brazed plate heat exchangers
CD	Condenser
EV	Evaporator
EXP	Expander
HEX	Heat Exchangers
HVAC	Heating Ventilation Air Conditioning
NRMSE	Normalized Root Mean Square Error
ORC	Organic Rankine Cycle
PP	Pump
REC	Recuperator

Variables

A	Surface area, m²
c	Correction factor, -
M	Mass, kg
Nu	Nusselt number, -
$\dot{Q}$	Heat power, W
Re	Reynolds number, -
S	Slip ratio, -
V	Volume, m³
X_{tt}	Lockhart-Martinelli factor, -
x	Quality -

Greek symbols

α	Void fraction, -
β	Weighting factor, -
ρ	Density, kg/m³
ω	Volume fraction, -

Subscripts

i,j,k	indices
exp	experimental
sim	simulated
min	minimum
max	maximum
v	vapour
l	liquid
tot	total
tp	two-phase
pp	pump

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