Advancements in Organic Rankine Cycle System Optimisation for Combined Heat and Power Applications: Components Sizing and Thermoeconomic Considerations

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Abstract:

There is great interest in distributed combined heat and power (CHP) generation in the built environment due to the higher overall efficiencies attained in comparison to separate provision of these vectors. Organic Rankine cycle (ORC) systems are capable of generating additional electricity from the thermal outputs of CHP engines, improving the electrical conversion efficiency and power-to-heat ratio of such systems. Thermodynamic analysis and technical feasibility are at the core of the development of these systems, while a critical factor for the wider adoption of ORC systems concerns their economic proposition. Obtaining credible estimates of system costs requires correct sizing of individual components. This work focuses on the thermodynamic optimisation, sizing and costing of ORC units in CHP applications, over a range of heat-source temperatures. The working fluids examined include R245fa, R1233zd, Pentane and Hexane, due to their good performance and favourable environmental characteristics. The optimal cycles obtained can increase the power-to-heat ratio of the complete CHP-ORC system by up to 65%. Alternative equipment sizing methods are then applied for each fluid and the resultant component sizes are compared. The cost estimates obtained from the alternative methods are also compared to real ORC application. Based on this, a hybrid costing method is proposed and applied to an ORC system design, in order to obtain the specific investment cost (SIC). The results indicate that as the heat source temperature increases, the power output increases, resulting in larger and more expensive components. Nevertheless, the SIC drops from 17 GBP/W for low-power outputs to 1.1 GBP/W for high-temperature/high-power outputs.

Keywords:
Organic Rankine cycle; Cogeneration; Optimisation; Component sizing; Thermoeconomics; Energy efficiency.

1. Introduction

Distributed combined heat and power (CHP) generation has been gaining growing interest, due to the higher overall efficiency of the system, in comparison to the separate provision of heat and power. This results in lower energy consumption and costs for the user, whilst the system has overall lower environmental impact, than the traditional heating systems and the centralised power generation. Typical small-medium scale CHP systems are comprised of an internal combustion engine (ICE) coupled with a generator for electricity production. The exhaust gas stream, after the power generation, is used for heating (or other processes), or in case there is no heating load, it is released to the atmosphere as waste heat. The release of unusable heat to the atmosphere reduces the efficiency of the system. The amount of waste heat can vary significantly, depending on the heating demand. A case in the point is that in the industry, studies have shown that waste heat corresponds up to 50% of the overall heat generated [1].

Organic Rankine cycle (ORC) units can utilise the waste heat of CHP systems for generating additional power, increasing the power-to-heat ratio of the system, especially in periods of the year when the heating demand is low. The power-to-heat ratio of CHP systems is defined as the ratio of the power generated over the thermal output available. The lower the heating demand, the higher the power-heat ratio should be to minimise the heat being wasted, which reduces the overall system efficiency. ORC engines operate using the conventional Rankine thermodynamic cycle, but the working fluid is an organic compound (refrigerant, hydrocarbon, etc.). By selecting the appropriate working fluid given the heat source conditions (in this case the ICE waste heat), the ORC can be optimised for maximum power output. Several studies can be
found in the literature analysing the ORC thermodynamic performance for different working fluids and heat sources. Maraver et al. [2] have optimised thermodynamically ORC units for six working fluids, for both high temperature waste heat sources and low temperature geothermal systems. Aljundi [3] and Oyewunmi et al.[4] have done a parametric analysis for a range of working fluids including hydrocarbons, refrigerants, and mixtures for geothermal applications. Li [5] have analysed the performance of alternative ORC architectures for solar, geothermal and waste heat recovery applications, for 15 alternative working fluids. Less documented is however the coupling of ORC units to CHP systems, with limited examples found in Lecompte et. al [1] and Shokati et. al [6]. Based on a review of the extended literature, it can be concluded that not a single working fluid can be identified as the optimal one. On the contrary, the most suitable working fluids vary with the application, and the heat source conditions. This finding leads to the need for a systematic integration of working fluids selection process for ORC engines, for the alternative applications.

Apart from the system thermodynamic performance, a critical factor for the wider deployment of the ORC units in the market is the capital expenditure (CAPEX), and the operating and maintenance (O&M) costs. To obtain credible estimates of the ORC investment cost, the sizing of the main ORC components is of paramount importance. These estimates form the basis of the cost calculations to be undertaken when evaluating the financial viability of ORC units’ deployment. In contrast to the prolific literature related to the thermodynamic analysis of ORCs, there is limited research on ORC components systematic sizing and costing. Walraven et al. [7] have done a comparison of shell and tube, and plate heat exchangers design for ORC units, focusing on the equipment sizing only. Thermo-economic analysis of ORC units recovering geothermal energy can be found in Oyewunmi and Markides [4], for solar driven ORC in Guaraccino et. al [8]; and for CHP bottoming cycles in Lecompte et al. [1]. Feng et al. [9] and Quoilin et al.[10] have done thermo-economic analysis of ORC driven by high grade waste heat, achieving specific investment cost (SIC) of 2,800 EUR/kW.

These studies have shown the importance of accounting for the components specific design characteristics and costs, when assessing the feasibility of ORC systems. There is a great range of sizing methods for the ORC main components (evaporator, condenser, etc.) based on alternative Nusselt number correlations, such as [11-13]. These correlations vary depending on the type of heat exchangers (HEXs) selected, the flow regime and the type of evaporation/condensation taking place. In a similar manner, there is a variety of costing methods available, mostly originating from the chemical industry, which were traditionally used to provide estimates for chemical plants costs [14]. Typical costing methods include the module costing technique [15,16].

Based on the above, the aim of this paper is to provide a systematic ORC engine component sizing and costing methodology. The ORC is first optimised thermodynamically for alternative working fluids, and then a number of different correlations for components sizing are applied. Next, the best suited correlation is used to obtain the components cost and the system SIC. The novelty of this work lies in the fact that: i) it considers alternative sizing methods to come up with credible equipment design guidelines; and ii) it compares the system cost calculations to real applications, to establish the validity of the cost estimates. The ORC systems examined in this work are coupled with CHP units driven by ICE, but the methodology proposed is applicable to any ORC system design. The paper structure is as follows; firstly, the ORC model set up is presented, along with the optimisation problem. Then, the thermodynamic optimisation results are discussed. Next, alternative methods for the ORC components sizing are examined, to obtain the equipment size. The economic analysis of the ORC unit follows. Finally, conclusions and recommendations for the design of ORC units are discussed.

2. Modelling methodology

2.1. Organic Rankine cycle (ORC) unit

A schematic of an ORC is presented in Fig. 1 along with a typical subcritical non-regenerative cycle on a T-s diagram. The system is composed of four major components: i) the evaporator, where heat inputs the cycle; ii) the expander, where power is generated; iii) the condenser, where heat is released; and iv)
the circulating pump. Process 1-2 represents liquid pumping, 2-3 the heat addition, 3-4 the expansion of the working-fluid vapour, and 4-1 the heat rejection. The heat carrier fluid (5-6) and condenser cooling-water (7-8) temperature changes are also illustrated on the same diagram.

Fig. 1. (a) ORC schematic diagram, and (b) subcritical non-regenerative ORC on a T-s diagram.

The HEX selected for the evaporator and the condenser are of tube-in-tube construction, which is a relatively simple and low cost design, in comparison to shell-and-tube HEXs for example [11]. The evaporator has been modelled by splitting it into three distinct zones: i) the preheating section, a single phase liquid zone where the working fluid is heated up to the evaporation saturation temperature (States 2-3l); ii) the evaporating section, a two-phase zone where the phase change occurs (States 3l-3v); and iii) the superheating section, where the saturated vapour is heated further in the dry vapour zone (States 3v-3). Similarly, the condenser has been modelled by dividing it into two distinct zones: i) the desuperheating section, a single phase zone where the dry vapour is cooled down to the condensing saturation temperature (States 4-4v); and ii) the condensing zone, a two-phase zone where the working fluid changes phase (States 4v-1). The HEX system is sized to maintain turbulent flow in both working fluid streams to maximise the heat transfer rate. The HEXs have been split into n-segments and in each i-segment the temperature and the quality of the working fluid are calculated. Based on those, the heat transfer coefficient, \( h_i \), is estimated for each segment, using a number of alternative correlations. Using the Logarithmic Mean Temperature Difference (LMTD) method [17] the area, \( A_i \), of each segment is then calculated. By summing the \( A_i \) calculated, the total area requirements of the HEX are obtained, for each working fluid examined and each correlation. More details on the HEX sizing analysis are presented in subsequent section.

Finally, the expander is modelled by using an isentropic efficiency value of 0.7, as representative of high-performance reciprocating expanders (for details on the loss mechanisms refer to Sapin et. al [18]), and the pump used is a centrifugal pump with a typical isentropic efficiency of 0.65 [19]. A reciprocating expander has been selected due to its wide range of operating volume and pressure ratios.

### 2.2. Optimisation model

In a typical optimisation problem, there is an objective function \( F(\vec{X}) \) which is to be minimised or maximised, subject to a set of constraints. Vector \( \vec{X} \) contains all the decision variables, i.e. the set of independent variables the optimiser will alter, in order to minimise/maximise the value of \( F(\vec{X}) \). For this study the objective function is the calculation of the power output of the ORC (\( \dot{W}_{\text{net}} \)), which we seek to maximise, given the heat source conditions, and subject to a number of operational constraints. Vector \( \vec{X} \) contains six decision variables: i) the evaporating pressure \( P_{\text{evap}} \) (Pa); ii) the condensing pressure \( P_{\text{con}} \) (Pa); iii) the working fluid mass flow rate \( \dot{m}_{\text{ref}} \) (kg/s); iv) the superheating degree \( \text{SHD} \) (-), normalised over the maximum possible, given the heat source inlet conditions to the cycle; v) the evaporator pinch point \( P_{\text{evap}} \) (K); and vi) the volume ratio of the expander \( r_{\text{exp}} \)(-). The objective function and the constraints are listed below:

\[
\begin{align*}
\text{Objective:} & \quad W_{\text{net}} = F(\vec{X}) \\
\text{Constraints:} & \quad \begin{align*}
& P_{\text{evap}} \geq P_{\text{con}} \\
& \dot{m}_{\text{ref}} \geq 0 \\
& \text{SHD} \leq 1 \\
& P_{\text{evap}} \leq 1000 \text{K} \\
& r_{\text{exp}} \leq 1
\end{align*}
\end{align*}
\]
maximise \{\dot{W}_{\text{net}}\} \quad \text{subject to:} \quad \frac{T_3-T_\text{in}}{T_\text{in}-p_{\text{exp}}-T_3}.
\] Equation (2) describes the constraint related to the pressure levels of the cycle; the pressure at the condenser must be lower than the evaporation pressure, which should be lower than the working fluid critical pressure, for a subcritical ORC. Equation (3) is the normalised superheating degree restricted between 0 and 1; it is expressed as the ratio of the superheating degree the cycle actually operates at, over the maximum allowable, accounting for the pinch point in the evaporator. The SHD is defined as:

\[ SHD = \frac{T_3-T_\text{in}}{T_\text{in}-p_{\text{exp}}-T_3} \]

Equation (4) ensures that the minimum pinch point of the HEXs is not violated, while Eq. (5) ensures that the expansion process is isentropic. The next constraint (Eq. (6)) ensures that the temperature after the expansion is higher than the saturation condensing temperature, and finally, Eq. (7) has been added to limit the exhaust gases leaving temperature from the evaporator, to minimise the risk of reaching the dew-point temperature. For exhaust gases from natural gas combustion, the dew point can vary from 373 K (60 °C) to as low as 308 K (35 °C), for very high oxygen concentration [20,21]. Other researchers report typical exhaust gases dew point temperature of approximately 323-343 K (50-70 °C) [22-24]. Typical CHP units with natural gas ICE cool the exhaust gases down to approximately 373-393 K (100-120 °C) [2,25] to ensure that no condensation occurs. In this study for the design of the ORC unit, the heat carrier fluid temperature exiting the evaporator (T_{\text{lim}}) has been restricted to 353 K (80 °C), which is a temperature well above the reported dew point levels and close to the current state of the art figures reported in the CHP system design.

### 2.3 Working fluids selection

In this paper, the exhaust gases stream from the ENER-G 2500 natural gas CHP engine [25] has been used as the heat source to the ORC (m_{\text{ex}} = 3.52 kg/s). The power output of the ORC has been maximised for different temperatures of the exhaust gases (T_{\text{hs,in}}), accounting for situations when alternative heat recovery for heating purposes occur during the operation of the CHP system.

A number of working fluids has been investigated in this optimisation exercise, in order to obtain the most suitable candidate for the CHP-ORC application. Due to the high computational cost of the optimisation, a pre-selection of working fluids has been made. Apart from the good thermodynamic performance characteristics of the working fluids (high vapour density, low specific heat capacity, and high latent heat), their impact to the environment has been also gaining growing interest. This is due to the introduction of strict regulations imposing the gradual phase out of the use of refrigerants with high ozone depletion potential (ODP) and global warming potential (GWP). In the EU the so-called F-gas regulation has imposed a number of limitations to the use of refrigerants in the air-conditioning and refrigeration industry, which may be applied in the near future to the power generation units.

Based on the above, the following refrigerants were selected for this study as having good thermodynamic characteristics and low ODP and GWP values: R245fa, R152a, R1233zd, R1234ze, and R1234yf. R245fa is commonly used in commercial ORC units made by Bosch, Turboden, GE Clean start, Cryostar and Electratherm [26]. R1233zd is a very promising replacement of R123, similarly R1234ze and R1234yf are replacement refrigerants for R134a which is currently used in ORC units manufactured by Cryostar [2]. Apart from refrigerants, some hydrocarbons have also shown very promising results in previous performance studies [27]. In this work Butane, Pentane and Hexane have been examined. It should be highlighted that hydrocarbons are listed as A3 category in the ASHRAE...
Classification [21] which includes fluids with low toxicity, but high flammability. Finally, Toluene has been also examined, which is mostly suitable for high temperature waste heat, and it is used in ORC units manufactured by Triogen.

3. Thermodynamic cycle analysis

In Fig. 3a, the power output for every working fluid investigated is illustrated for different heat carrier fluid temperature. From the refrigerants investigated, R1233zd has the highest power output reaching 130 kW, for heat source temperature of 623 K (350 °C). This is followed by R245fa with power output of up to 116 kW, and R152a of up to 104 kW. R1234ZE and R1234YF generate up to 85 kW and 70 kW, respectively. The variation in power output is mainly attributed to the different pressure ratios used in the optimum cycle configurations for the different fluids. The higher the pressure ratio (PR) over the expander the higher the power output of the cycle. R1233zd and R1245fa have the highest power ratios among all the refrigerants investigated, 21 and 19 respectively, whereas the other fluids operate with PR of up to 6-7 (Fig. 3b). The difference in the PR achieved is explained by looking at the condensing pressure of the cycle for each fluid (low pressure side of the cycle). For all fluids, the optimised cycle chooses a condensing temperature as low as possible, close to the cooling fluid temperature profile, while also the condenser pressure must by higher or equal to 101 kPa (1 bar) to do not violate the constraints. Based on these, for fluids such as R1233zd and R245fa the condensing pressure is 1-2 bar, whilst for the other refrigerants this is equal to 6-8 bar. Therefore, the PR cannot be as high without violating the constraint that the evaporating pressure must be less than the fluid critical pressure to maintain subcritical cycle operation. These trends explain the difference in the power output of the alternative refrigerants.

![Fig. 3. ORC unit results for different working fluids and heat-source temperatures: (a) maximum power output, and (b) operating pressure ratio.](image)

Looking at the other working fluids in Fig. 3a, Pentane and R1233zd have the highest power output (up to 130 kW for a 623 K (350 °C) heat-source temperature), followed by Butane and Hexane (111-112 kW), and Toluene (96 kW). However, from Fig. 3(b) it can also be seen that Pentane and Hexane generally operate at higher cycle pressure ratios (PRs). This is due to the higher gamma values of these fluids (up to 1.3 for high temperatures) in comparison to 1.1-1.15 of the refrigerants. For the isentropic expansion assumed in this study the high gamma values result in high pressure ratios across the expander. Butane, which also generates similar power output as Pentane and Hexane, operates however in much lower PR. The difference can be explained by looking at: i) the evaporating temperature, once the maximum allowable evaporating pressure is reached; and ii) at the gamma values of Butane. For the peak evaporating pressure of the cycle operating with Butane, the saturation temperature is 420 K (147 °C) (for $P_{\text{evap}} = 36$ bar), so while the heat source temperature increases the system increases the superheating degree, rather than increasing the pressure at the evaporator further, in order to increase the enthalpy difference across
the expander. For fluids such as Hexane on the contrary, the respective evaporating temperature at high pressure is 505 K (232 °C), allowing for much lower degree of superheating. Adding to this, the gamma value of Butane is lower than those of Pentane and Hexane, resulting in lower PR.

So depending on the fluid selection, different types of expanders will be more suitable. For fluids such as R245fa and Butane, expanders operating at low expansion ratios will be suitable, whilst expanders with high expansion ratios will be required for fluids such as Pentane and Hexane.

In Fig. 4, the first and second law efficiency of the ORC for every working fluid is presented. In line with the results for all fluids while the heat source temperature increases up to 500 K (232 °C) approximately the thermal efficiency increases up to 9%-14% depending on the fluid, and then stays constant, while the heat carrier fluid temperature increases further. The only exemption is Toluene for which the thermal efficiency constantly increases, while the heat carrier fluid temperature increases, although is much lower than the other fluids until the heat source temperature reaches 600 K (327 °C). These results indicate that some fluids are more suitable for low temperature heat sources (e.g. R1233zd), than others which maximise their performance at high temperatures (e.g. Toluene).

Fig. 4. Optimum ORC thermal and exergy efficiencies for different working fluids and heat-source temperatures. The exergy efficiency shows a different trend to the thermal efficiency, with the former increasing as the heat-source temperature increases up to approximately 470 K (200 °C) and then drops (Fig. 4). The highest exergy efficiency is 43% for R1233zd and Pentane. The drop in the exergy efficiency implies that the additional heat input is not being utilised as efficiently by the cycle, although the power output increases.

Finally, the improvement of the overall CHP-ORC system thermal efficiency and power-to-heat ratio is shown in Fig. 5, using the optimum ORC results for $T_{\text{hs,ain}}$ of 623 K (350 °C). The total system efficiency improves by up to 5%, whereas the power-to-heat ratio shows significant increase of up to 65%, for working fluids such as R1233zd and Pentane. These findings illustrate that the CHP system efficiency can be decoupled from the heating demand of the building by using an ORC unit, maximising the system performance.

Fig. 5. CHP-ORC efficiency and power-to-heat ratio variation for different working fluids.
4. Equipment sizing analysis
In what follows, component sizing is performed for the optimum ORC systems identified in Section 3.

4.1. Heat transfer calculations
A tube-in-tube heat exchanger (HEX) construction was selected that was split into \( n \)-segments in which the temperature and quality of the fluids were calculated. In order to size the HEX, the surface area requirements of each one of these zones (\( A_i \)) is required, which in turn requires the heat transfer coefficient (HTC) \( h_i \) in each zone “\( i \)”.. Extensive literature exists on developing different Nusselt number correlations in the context of modelling heat transfer in HEXs. In the single phase zone, for flow in tubes (liquid only or vapour only), two correlations are common in the literature, namely by Dittus Boelter [17] and Gnielinski [28]. In this study, the Dittus Boelter correlation has been used. A key difficulty lies in the calculation of the HTC in the two-phase zone during phase change. Again, numerous correlations have been developed for different working fluids, accounting for different types of boiling and condensation in tubes. To provide reliable estimates of the surface area requirements in the two-phase zone, five different correlations have been used for both the evaporating and condensing sections, and the average value of the different areas (\( A_i \)) calculated, is then used for the costing exercise that follows.

Two main types of methods can be found in the literature for boiling in tubes: i) asymptotic methods that account for both thermal mechanisms of nucleate boiling and convection boiling; and ii) nucleate pool-boiling methods only [29]. The Nusselt number correlations selected in this paper cover both methods for comparison purposes: i) Steiner [29,30], Dobson [12], and Zuber [11] are the correlations used for the asymptotic method; and ii) Cooper [11] and Gorenflo [30] are those used for the nucleate pool boiling. The reader can refer to the respective references for details on the formulations of these correlations.

For condensation, the correlations are divided into those accounting for gravity driven condensation, or shear driven condensation or a combination of the two. In this study, correlations covering both types of condensation have been used for comparison purposes. The correlations of Dobson [12], Shah [13] and Chaddock and Chato [11] have been used for combined shear and gravity driven phenomena, whereas Nusselt [11] has been used for gravity only, and Mikheev [11] for shear condensation only.

4.2. ORC heat exchanger sizing
4.2.1. Evaporator
The evaporating section area requirements for R245fa and different heat source temperatures are presented in Fig. 6a, for different correlations. For low temperatures the deviation of the two-phase zone area is significant, between the various correlations used. In these conditions, the surface area requirements vary from 3.5 \( \text{m}^2 \) (if nucleate boiling correlations are used) up to approximately 6 \( \text{m}^2 \) (if convective boiling also occurs). However, for high heat source temperature the deviation becomes negligible. In line with the results, correlations accounting for 100% nucleate boiling result in higher HTC's, and lower evaporator surface area requirements, than those accounting for convective and nucleate boiling phenomena. For high temperatures, all correlations seem to result in similar HTC and surface area. This is due to the fact that for high heat source temperature nucleate boiling conditions prevail, therefore all correlations result in very similar HTC. Additionally, when vertical tubes are used, for the same flow regime, the HTC is higher than the respective one in horizontal configuration (refer to Steiner results in Fig. 6a), resulting in lower surface area requirements. Based on the results, for cycles using low temperature heat source, the design of the evaporator HEX should aim to maximise the HTC in convection, in order to reduce the system size and costs. Similar trends, for the two-phase evaporating section area variation, are observed for all working fluids examined.

Looking at the overall evaporator area requirements for R1233zd and Pentane (Fig. 6b), it can be seen that the preheating section has the highest surface area requirements. As discussed, for the maximum ORC power output the optimiser chooses the highest evaporating pressure feasible. However, while the evaporating pressure increases, the preheating section load dominates, since close to the critical pressure there is very low latent heat required for the phase change. Therefore, the preheater surface area
requirements increase significantly up to heat source temperatures of 493-523 K (220-250 °C). For the same temperature range, depending on the working fluid, the two-phase zone has high area requirements (for $T_{hs} < 480$ K (207 °C)) or the superheating section has high surface area requirements ($T_{hs} > 480$ K).

![Fig. 6. Heat exchanger sizing results for: (a) the evaporating two-phase zone, and (b) the entire evaporator.](image)

For each fluid and zone, it is observed that the area requirements peak at some operating points and then drop, while the heat source temperature increases. This is explained by two key trends: i) the LMTD in the preheating/superheating sections increases significantly for high heat carrier fluid temperatures, resulting in lower surface area requirements; and ii) the HTC increases while the mass flow rate of the working fluid increases. In the two-phase zone, apart from these two phenomena, the transition to nucleate boiling conditions results in high HTC and low area requirements.

### 4.2.3. Condenser

The condensing section surface area requirements for R245fa are presented in Fig. 7a, for different correlations. Correlations that account for both gravity and shear forces driven condensation result in lower surface area requirements than the gravity driven only condensation. It should be noted that based on the flow conditions in this study, the flow regime is such that falls within the shear condensation regime. Similar results, for the condensing section are found for all working fluids investigated. In contrast with the evaporator results, the two-phase condensing zone area is much higher than the single phase (desuperheating) area for all working fluids examined. This is explained by the fact that the optimiser selects the lowest condensing saturation pressure feasible, without violating the constraints, therefore there is high latent heat load to be rejected in the condensing zone. Adding this the optimum cycles for all working fluids selected have minimum superheating degree.

![Fig. 7. Heat exchanger sizing results for: (a) the condensing two-phase zone, and (b) the entire condenser.](image)
The average (total) condenser area requirements are presented in Fig. 7b for the best performing working fluids, derived by using all five correlation results to obtain an average value. The area requirements increase, while the heat source temperature increases for all working fluids. This is in line with the increase of the condenser heat load being rejected in the condenser water circuit. It should be highlighted that Hexane has the lowest condenser area requirements among the fluids investigated. This is attributed to the fact that the saturation condensing temperature of Hexane at 101 kPa, is 343 K (70 °C) and it is the highest among the fluids investigated. The higher saturation temperature results in a high LMTD across the HEX, reducing the heat transfer area requirements.

5. Economic analysis

Based on the equipment sizing results, the mean surface area of the two-phase zone has been used for the equipment costing exercise, for both the evaporator and the condenser. For the components costing, alternative methods found in the literature have been reviewed, and those with the best fit to real applications data have been used for the rest of the study.

5.1. Module costing techniques

In line with a cost methods comparison conducted by Lemmens [14], among the costing methods available, the module costing technique is the one providing the most accurate results, when compared to real large scale installations costs. In this study, the module costing method deployed uses correlations provided by Turton et al. [15], and Seider et al. [16]. The general format of the module costing techniques is as follows:

Turton et al. [15]: \[ C_p^0 = F \times 10^{(K_1 + K_2 \log_{10} Z + K_3 \log_{10}^2 Z)} \] (9)

Seider et al. [16]: \[ C_p^0 = F \exp(K_1 + K_2 \ln Z + K_3 \ln^2 Z + K_4 \ln^3 Z + K_5 \ln^4 Z) \] (10)

where \( C_p^0 \) is the component purchase cost (or free of board, FOB, cost); \( F \) is a material factor accounting for the component manufacturing; \( Z \) corresponds to the specific component attribute based on which the costing is conducted; \( K_i \) are the cost coefficients described in Turton et al. [15] and Seider et al. [16], and depend on the type of component. The cost coefficients used to estimate the purchase cost of each piece of equipment are summarised in Table 1. For the pump cost calculation, the specific attribute required is:

\[ S = \frac{v_{wf}}{0.0631} \left( \frac{P}{0.0299} \right)^{0.5} \] (11)

where \( v_{wf} \) is the volume flow rate of the working fluid in L/s, and \( P \) is the pump head in bar.

Table 1. Cost coefficients used for the economic analysis

<table>
<thead>
<tr>
<th>Component</th>
<th>Specific attribute (Z)</th>
<th>( F )</th>
<th>( K_1 )</th>
<th>( K_2 )</th>
<th>( K_3 )</th>
<th>( K_4 )</th>
<th>( K_5 )</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator/Condenser</td>
<td>Area (m²)</td>
<td>1</td>
<td>9.5638</td>
<td>0.532</td>
<td>-0.0002</td>
<td>-</td>
<td>-</td>
<td>[16]</td>
</tr>
<tr>
<td>Preheater/Desuperheater</td>
<td>Area (m²)</td>
<td>1</td>
<td>10.106</td>
<td>-0.4429</td>
<td>0.0901</td>
<td>-</td>
<td>-</td>
<td>[16]</td>
</tr>
<tr>
<td>Expander</td>
<td>Power (kW)</td>
<td>1</td>
<td>2.2897</td>
<td>1.3604</td>
<td>-0.1027</td>
<td>-</td>
<td>-</td>
<td>[15]</td>
</tr>
<tr>
<td>Pump</td>
<td>( S ) ( \text{gpm} \sqrt{f_{head}} )</td>
<td>2.7</td>
<td>9.2951</td>
<td>-0.6019</td>
<td>0.0519</td>
<td>-</td>
<td>-</td>
<td>[16]</td>
</tr>
<tr>
<td>Pump-motor</td>
<td>Power (HP)</td>
<td>1.4</td>
<td>5.83</td>
<td>0.134</td>
<td>0.0533</td>
<td>0.0286</td>
<td>0.00355</td>
<td>[16]</td>
</tr>
</tbody>
</table>

The costs are converted into present values using the Chemical Engineering Plant Cost Index (CEPCI). For Turton et al. [16], CEPCI\(_{2006} \) = 500 has been used, and for Seider et al. [15] CEPCI\(_{2001} \) = 397. All values have been converted to 2016 values using CEPCI\(_{2016} \) = 556.8. Also, for Turton et al. [16] the USD values in 2006 have been converted to GBP, using 1 GBP = 1.84 USD, and for Seider et al. [15], USD values in 2001 have been converted to GBP, using 1 GBP = 1.439 USD [31].

5.2. ORC specific investment cost (SIC)

The variation of the SIC for the best performing fluids (R245fa, R1233zd, Butane, Pentane, Hexane) are presented in Fig. 8a. As expected, while the power output of the system increases this has a positive impact on the SIC which drops exponentially. The lowest SIC is achieved by R1233zd and Pentane; this is equal
to 1,100 GBP/kW for 130 kW of net power generated. On the contrary, the SIC for low power outputs reaches 18,000 GBP/kW for 8 kW of power output for R1233zd, and 6,240 GBP/kW for 23 kW of power output for Pentane. The results are in good agreement with the data provided by [14,32], which summarise real ORC units cost. Based on this data, for ORC units of 130 kW capacity the SIC falls within the range of 1,000-4,000 GBP/kW. For small units of 1-10 kW the SIC reported in Viking [32] has a greater range between as low as 2,600 GBP/kW and up to 17,850 GB/kW (all values converted from EUR to GBP, using 1 EUR = 0.85 GBP, Dec. 2016 rate). This proves that the hybrid method deployed in this study performs well for low power outputs falling within the average values of the real applications costs, whilst sits at the lower end of the SIC range for high power outputs.

Fig. 8. For the best-performing working-fluids investigated: (a) optimum ORC-unit SIC (GBP/kW), and (b) equipment purchase cost (GBP).

The absolute costs of each component for R1233zd and Pentane are shown in Fig. 8b. The expander costs dominate the total purchase costs at high temperatures (high pressure ratios), amounting to approximately 50% of the ORC unit purchase cost (for $T_{hs} = 630$ K). The condenser is the second most expensive component (up to 30% of the total), followed by the evaporator and the pump. These findings are in agreement with the costs data reported for a heat recovery ORC facility in Belgium of 375 kWe [14]. When the oil circuit and installation costs are excluded, the expander and the pump are responsible for approximately 60% of the investment cost, followed by the condenser (22%), and the evaporator (13%) [14]. For lower power outputs (low temperature and pressure ratios) the expander costs are comparable to the cost of the condenser and the evaporator (Fig. 8b), corresponding to 27%, 42%, and 31% respectively (for $T_{hs} = 450$ K). Based on these findings, it is concluded that for low-medium temperatures, and low pressures, the cost of the HEXs is significant, therefore it is of high importance to design them to maximise the heat transfer. In this study, tube-in-tube HEXs have been used for both the evaporator and the condenser, so the results might vary in case other types of HEXs are used. For higher power outputs, on the contrary, the selection and design of the expander and then of the condenser is key factor for reducing the ORC costs. Finally, it should be noted, that the cost correlations, when used for very small capacities, should be treated with caution, because these loads are close to the low end of the range of applicability of those correlations.

6. Conclusions and recommendations

Combined heat and power (CHP) systems have great potential to reduce the cost of energy generation, while having low environmental impact. Key for maximising the CHP performance, is the utilisation of the excess thermal energy. In this study, a thermo-economic analysis has been performed aiming i) to evaluate the financial viability of coupling ORC units to CHPs to recover the exhaust heat; and ii) to establish a systematic ORC system design methodology.
Firstly, thermodynamic optimisation of the ORC unit has been performed for maximising the power output, given the exhaust gases conditions. A range of working fluids has been examined and the best performing fluids, namely R1233zd and Pentane, were found to generate up to 130 kW of additional power. This results in an increase of the power-to-heat ratio of the system of up to 65%, decoupling the CHP performance from variations on the heating demand side. Adding to this, depending on the fluid selection, the optimum ORC cycle operates with different pressure levels, and superheating degree, influencing the components selection. A case in the point is that, for fluids, such as R1233zd, R245fa and Butane, expanders operating at low expansion ratios will be suitable, whilst expanders efficient at high expansion ratios will be required for fluids such as Pentane and Hexane.

Next, the individual components sizing has been performed, with particular focus on the evaporator and condenser HEX design. Key for the sizing of HEXs is the surface area requirements calculation, which in turn depends on the estimation of the HTC. There is a plethora of correlations for describing the heat transfer phenomena in HEXs, with the most complex related to the 2-phase zone. In this work, multiple Nusselt number correlations have been compared. Results for the evaporator have shown that for high temperature levels- where nucleate boiling prevails- all correlations seem to result in similar surface area requirements, whilst for lower temperatures the existence of combined convection and nucleate boiling can reduce the HTCs. Looking at the condenser, the majority of the correlations result in similar surface area requirements, with the exemption of correlations for gravity driven condensation.

Following the system sizing, an economic evaluation of the ORC system has been performed. This work illustrates that alternative costing methods can be found in the literature, resulting in different cost estimations. Therefore, it is of great importance to compare the costing calculation results, to real applications costs. However, this has been proved to be a very onerous task, due to the lack of publicly available cost data for ORC installations. In this paper, a hybrid costing method has been used, combining well established cost correlations, and the results were compared to a limited number of real large scale installation data. The cost estimates are in good agreement with the real ORC units costs, however a broader range of ORC units and components data should be used in the future to establish the validity range of the method. While the ORC units’ capacity increases with high heat sources temperature, the SIC cost drops exponentially, reaching up to 1,100 GBP/kW for the best performing fluids investigated in this study (R1233zd and Pentane). Depending on the unit capacity, the HEXs will form between 40%-75% of the total investment cost. Therefore, the HEX sizing has significant impact on the ORC units cost and financial viability.

To conclude, this work has established the need for a holistic approach when designing ORC units, accounting for both the thermodynamic and economic aspects. Depending on the application, the most suitable working fluids, and components design may vary significantly. The cost correlations used to-date, originate from the chemical industry, and should be used for comparing alternative designs, where the relative results are more important than the exact cost figures obtained. Finally, the findings reveal there is need for validating the cost calculations against real ORC applications, to establish a validity range for the cost estimates. Having validated the cost estimates of the ORC system, optimisation can then be performed on the basis of minimising the SIC of the project.

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