Thermodynamic optimisation of a high-electrical efficiency integrated internal combustion engine – Organic Rankine cycle combined heat and power system

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HIGHLIGHTS

- An ICE-ORC CHP system design tool is developed for optimising total power or fuel use.
- Simultaneous optimisation of ICE-ORC CHP systems can increase total power output by 30%.
- Optimal ICE exhaust gas temperature increases to promote ORC power generation by 7%.
- Optimised ORCs generate up to 15% of the total power and reduce fuel consumption.
- Power optimisation increases fuel consumption; fuel-efficiency optimisation reduces it by 17%.

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ABSTRACT

Organic Rankine cycle (ORC) engines are suitable for heat recovery from internal combustion engines (ICE) for the purpose of secondary power generation in combined heat and power (CHP) systems. However, trade-offs must be considered between ICE and ORC engine performance in such integrated solutions. The ICE design and operational characteristics influence its own performance, along with the exhaust-gas conditions available as heat source to the ORC engine, impacting ORC design and performance, while the heat-recovery heat exchanger (ORC evaporator) will affect the ICE operation. In this paper, an integrated ICE-ORC CHP whole-system optimisation framework is presented. This differs from other efforts in that we develop and apply a fully-integrated ICE-ORC CHP optimisation framework, considering the design and operation of both the ICE and ORC engines simultaneously within the combined system, to optimise the overall system performance. A dynamic ICE model is developed and validated, along with a steady-state model of subcritical recuperative ORC engines. Both naturally aspirated and turbocharged ICEs are considered, of two different sizes/capacities. Nine substances (covering low-GWP refrigerants and hydrocarbons) are investigated as potential ORC working fluids. The integrated ICE-ORC CHP system is optimised for either maximum total power output, or minimum fuel consumption. Results highlight that by optimising the complete integrated ICE-ORC CHP system simultaneously, the total power output increases by up to 30% in comparison to a nominal system design. In the integrated CHP system, the ICE power output is slightly lower than that obtained for optimal standalone ICE application, as the exhaust-gas temperature increases to promote the bottoming ORC engine performance, whose power increases by 7%. The ORC power output achieved accounts for up to 15% of the total power generated by the integrated system, increasing the system efficiency by up to 11%. When only power optimisation is performed, the specific fuel consumption increases, highlighting that high-power output comes at the cost of higher fuel consumption. In contrast, when specific fuel consumption is used as the objective function (minimised), fuel consumption drops by up to 17%, thereby significantly reducing the operating fuel costs. This study proves that by taking a holistic approach to whole-system ICE-ORC CHP design and operation optimisation, more power can be generated efficiently, with a lower fuel consumption. The findings are relevant to ICE and ORC manufacturers, integrators and installers, since it informs component design, system integration and operation decisions.

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1. Introduction

In light of recent trends towards increasing the efficiency of primary energy use, reducing energy consumption and reducing emissions worldwide, the distributed cogeneration of heat and power has been identified as a viable alternative to the separate provision of these vectors. The benefits of combined heat and power (CHP) systems include higher overall energy efficiencies, lower primary energy (e.g. fuel) consumption rates, emissions and overall environmental impact, and also lower costs relative to traditional heating systems and centralised power generation, when covering the same end-use energy demands [1–3]. Crucial for the maximisation of a CHP system’s overall efficiency is the effective utilisation of the heat rejected by the prime mover. In internal combustion engines (ICE) designed for use in CHP applications, which are at the focus of this present study, more than 55% of the fuel energy is rejected as heat to the cooling jacket water.
circuit of the engine and the exhaust gas stream. By selecting a suitable energy conversion technology this heat can be used to cover the heating or cooling demands of buildings, or it can be used for additional power generation, thus reducing the overall system fuel consumption and emissions. This can be achieved with the employment of more technologically and commercially mature technologies, such as the organic Rankine cycle (ORC) and the Kalina cycle [4,5], or earlier stage technologies currently under development such as thermoacoustic [6,7] or thermofluidic [8,9] heat engines. In particular, the Non-Inertive-Feed-back Thermofluidic Engine (NIFTE) [10–12] and the Up-THERM heat converter [13–15] have been shown to be competitive with established technologies, such as ORCs [16,17], due to their small number of moving parts, and low capital and running costs. Nevertheless, ORC technology is more established, commercially available and has been selected for the present study.

ORC technology offers a particularly promising solution in high-efficiency stationary power generation applications, or CHP systems based on ICES, where the heat demand does not match the engine’s heat-to-power ratio. ORC systems available on the market are suitable for heat conversion at temperatures up to 400–500 °C, and at power output scales ranging from the order of kW to tens of MW [18,19]. Important advantages of ORC systems arise from their (relatively) simple architecture and low complexity, the use of simple and well-established components and their broad applicability and affordability even at small scales compared to alternatives [20]. To-date ORC technology has been investigated in a wide range of low/medium-grade and waste-heat recovery applications, including biomass/biogas, geothermal-heat and solar-driven power generation, waste-heat recovery in industrial plants or gas turbines (as bottoming cycles), amongst many others. Examples of ORC system performance evaluations in geothermal applications can be found in Refs. [21,22], solar-driven ORC systems have been studied in Refs. [23–25], while Lecompte et al. [26] optimised a bottoming ORC system for a CHP power plant in Belgium. Waste heat recovery from ICE engines has been investigated, e.g. in Refs. [20,27,28]. Yang et al. [29] evaluated the performance of an ORC engine coupled to an ICE, aiming to optimise the ORC evaporation and condensation pressures, as well as the superheating degree (SHD). This study revealed that the optimum ORC engine design increases the combined ICE/ORC CHP system power output by 6%. Although the use of ORC engines in building applications has been gaining a growing interest recently, due to the current trends for distributed generation and efficient building sector, the use of ORC technology in these applications is less documented. Examples of evaluating the performance of small-scale solar-ORCs for domestic buildings can be found in Refs. [30,31], while Wu et al. [32] and Chatzopoulou and Markides [33] investigated the application of ORCs in commercial buildings. Examples of the use of ORC engines to recover waste heat from ICES in buildings can be found in Refs. [34–36]. Nevertheless, in these studies the ICE performance is modelled by using either constant efficiency rates, or fits to manufacturers’ data.

There is also extensive literature on the identification of the best performing working fluids for ORC engines in different applications. The selection of suitable working fluids has significant impact on the efficiency, design and sizing of individual components, and on the economics of the plant. Examples of comparing mixtures of hydrocarbons and refrigerants against pure fluids for heat recovery can be found in Refs. [37,38]. Freeman et al. [39] evaluated the performance of a number of refrigerants and hydrocarbons for solar-driven ORC applications. Refs. [40,41] studied the performance of hydrocarbons, for a range of heat source temperatures. Yang et al. [38] evaluated the performance of zeotropic mixtures in ICE/ORC CHP applications, and reported a total system power output increase of 11%. By reviewing the available literature, it can readily be concluded that it is not possible to identify a single working fluid as the universal optimal fluid. Therefore, the selection of working fluids should be integrated into the design process of ORC systems, which should be optimised on a case-by-case basis accounting for the specific characteristics and constraints of each application.

Alternative cycle architectures have been also studied by several researchers. Maraver et al. [42] investigated the application of alternative ORC architectures for a range of heat sources, including exhaust gas streams from ICES. The optimum ORC design reported for this ICE waste heat recovery study is a recuperative system that can achieve a thermal efficiency of up to 22%. Li [43] investigated alternative ORC architectures and working fluids for a series of heat source temperature profiles and found that for high temperature waste heat recovery, the efficiency of recuperative and regenerative systems can reach 25% while for high temperature solar and biomass applications the efficiency reaches 30%. Yagli et al. [27] compared the performance of subcritical and supercritical ORCs in ICE/ORC CHP applications with maximum efficiencies of approximately 16% for both designs, while Refs. [44,45] considered interesting alternative dual-loop configurations for heat recovery from ICES.

The aforementioned studies focused mainly on optimising the power output or the investment cost of ORC systems while assuming fixed heat-source conditions (i.e. ICE designs, operational characteristics) in most cases. Nevertheless, in real integrated ICE-ORC CHP system applications, the design and operation of the ICE strongly influence its own performance, but also the conditions of the exhaust-gas stream that forms the heat source to the ORC engine, and hence, in turn, have a major impact on the bottoming ORC engine design, operation and performance, while the introduction of a heat-recovery heat exchanger (i.e. ORC evaporator) will also affect the conditions inside the ICE. Therefore, the overall performance of an integrated ICE/ORC CHP system can only be reliably optimised by considering the exhaust-gas temperature, pressure and flow-rate conditions, and how these are influenced by the interacting ICE and ORC engines, as part of the whole-system’s design and operation. Very few efforts exist that attempt to optimise the performance of such integrated ICE/ORC CHP systems by accounting explicitly for the design parameters and operational conditions of both the ICE and ORC engines, and optimising the complete system simultaneously. For example, Zhao et al. [46] investigated integrated ICE/ORC CHP system control strategies under varying ICE load conditions, by assuming fixed ICE valve size, operation and fuel injection timing, that were not adjusted based on integrated system information. Yue et al. [47] accounted for the impact of the ORC engine on the ICE operation, however, this study was not concerned with optimising the ORC engine (thermodynamic cycle, operating conditions or working fluid), and instead, a single working fluid was selected and the evaporator/condenser pressures were fixed. Furthermore, the ICE mean effective pressure was imposed as an input to the analysis, and not optimised for a scenario in which the ICE is coupled to an ORC engine (i.e. in the presence of the ORC evaporator). Xu et al. [48] investigated the response of automotive ICE-ORC systems to vehicle speed changes. Here, the ICE and ORC engine models were integrated and treated as part of one complete system, however, due to the nature of this application, the ICE operating conditions are dictated by the vehicle operation (drive cycle) and the interest is only on power generation. This is a very different application to the stationary cogeneration/CHP applications considered in our work, where the ICE operating conditions can be adjusted/optimised by the manufacturer or the plant operator to either maximise power generation or minimise fuel consumption.

In this context, the aim of this paper is to present a fully-integrated ICE/ORC CHP system tool that can be used to promote high-performance, advanced stationary cogeneration systems. The novelty and where this goes beyond previous efforts in the literature arises from: (i) the fully-integrated nature of the CHP system tool that can simultaneously optimise the design and operating conditions of both the ICE and the ORC engines, instead of assuming that these operate independently; (ii) the close consideration of the overall power and thermal performance trade-offs of the combined system; (iii) the insights provided for the design of ICES for stationary cogeneration.
applications, which can be used by ICE manufacturers when considering waste heat recovery projects with a bottoming ORC engine; and (iv) the insights provided for the design of ORC engines specifically for these applications where they are to be coupled to IC engines. To the authors’ knowledge, this is the first attempt in the literature that aims to optimise combined ICE-ORC cogeneration system performance, where the ICE-CHP and ORC engine design and operational parameters are optimised simultaneously. To achieve this, the paper is structured as follows: firstly, the thermodynamic models of the ICE and ORC engines are presented in Section 2, along with a validation of the former against data from manufacturer specification sheets; this is followed in Section 3 by the formulation of the optimisation problem and information on working-fluid selection for the ORC engine. The results of the integrated system optimisation for maximum power output and minimum specific fuel consumption (SFC) are presented in Section 4, and compared against the non-optimised (separate) ICE-CHP and ORC engine configuration performance. Optimal ORC engine designs based on different working fluids are also presented here, and the best performing fluids in terms of power output are discussed. Finally, conclusions based on the main findings from this work, as well as recommendations for future work are given in Section 5.

2. System description and thermodynamic modelling

2.1. Integrated ICE-ORC CHP system description

The complete integrated ICE-ORC CHP system proposed in this study is illustrated in Fig. 1. The main components of the ICE, which is a reciprocating engine designed for and aimed specifically at CHP applications, and referred to as “CHP-ICE” henceforth, are: (i) the prime mover, which in this case is a four-stroke spark-ignition ICE; (ii) the electric generator; (iii) the jacket water (cooling water) circuit of the ICE; and (iv) the exhaust-gas heat recovery module. The jacket water circuit thermal load is used for the provision of low temperature hot water (LTHW) and for space heating to a building. The thermal energy recovered from the exhaust gases in the heat recovery module of the CHP-ICE acts as the heat input to a bottoming ORC engine. The ORC engine is based on a subcritical and recuperative cycle, and consists of the following main components: (i) the evaporator heat exchanger (HEX), where the heat is added to the cycle from the exhaust gases; (ii) the expander, where power is generated; (iii) the recuperator HEX, where the (hot) desuperheating working fluid vapour exchanges heat with the (cold) working fluid liquid leaving the pump; (iv) the condenser HEX, where heat is rejected to a cooling circuit; and (v) the pump, which maintains the working fluid circulation in the cycle.

2.2. CHP-ICE thermodynamic model

The cogeneration system is driven by a four-stroke reciprocating spark-ignition ICE (Otto cycle), manufactured for CHP applications and featuring the necessary heat recovery hardware. The set of ordinary differential equations (ODEs) used to simulate the operation and performance of the CHP-ICE is given in Eqs. (1)–(21). The following assumptions are considered for the ICE thermodynamic modelling:

- Each engine cylinder is modelled as an open system/control volume, with the (intake) fresh air-fuel mixture charge and (exhaust) flue-gas flows crossing the system boundaries;
- Heat addition to the engine cylinder and cycle by combustion is modelled by using the Wiebe function for the heat release rate [49];
- Heat losses through the cylinder walls to the jacket-water circuit (\(Q_w\)) are calculated using the phase-averaged instantaneous heat transfer correlation of Woschni [49,50], and heat losses due to radiation are also incorporated into the model using the correlation proposed by Annand, as presented in Ref. [51];
- The mass flow rate through the intake and exhaust valves is calculated by using standard relations for compressible gas flow through nozzles [49];
- The compressor and the turbocharger are modelled using fixed isentropic efficiency.

Conservation of energy for an open system, including the heat transfer losses due to conduction and convection, as well as due to radiation, can be written as:

\[
\begin{align*}
\frac{dP}{d\delta} &= \frac{\gamma - 1}{V} \left( Q_{\text{comb}} \frac{d\omega}{d\delta} - \frac{dQ_w}{d\delta} \right) - \frac{\gamma - 1}{V} \frac{P \frac{dV}{d\delta}}{180 \omega} (m_{\text{fuel}} h_{\text{int}} - m_{\text{ex}} h_{\text{ex}}),
\end{align*}
\]

where the rate of change of cylinder pressure \(P\) with angle \(\delta\) is a function of: (i) the heat addition during combustion \((Q_{\text{comb}} \frac{d\omega}{d\delta})\); (ii) the heat losses through the wall \((dQ_w/d\delta)\); (iii) the in-cylinder positive displacement work done on the gases due to the piston movement
\( \frac{\text{PDV}}{\text{d}S} \); and (iv) the enthalpy flow rate into and out of the system \((\text{PDV}/\text{d}S)\) associated with the mass flow rate of gas entering or leaving the system boundaries \((\rho_m, \text{V}_w)\). The cylinder volume change \(\text{d}V/\text{d}t\) can be expressed as a function of the displacement volume \(\text{V}_{\text{dis}}\), the compression ratio \(\rho_c\) and the instantaneous stroke \(\text{y}(\vartheta)\):

\[
y(\vartheta) = \alpha + l - \left(\left(1 - \cos(\pi \vartheta)^{2/3} + \cos(\pi \vartheta)\right)\right)\]

\[
V(\vartheta) = \frac{\text{V}_{\text{dis}}}{\rho_c - 1} + \frac{3.14}{4} \alpha \pi \left(\cos(\pi \vartheta)^{2/3} + \cos(\pi \vartheta)\right)\]

\[
\text{d}V = \frac{\text{V}_{\text{dis}}}{2} \cos(\pi \vartheta)\left[1 + \cos(\pi \vartheta)(\alpha^2 - \sin(\pi \vartheta)^2)^{0.5}\right].
\]

Referring to the terms on the right-hand side (RHS) of Eq. (1), firstly, the heat addition/input by combustion to the cycle \(Q_{\text{comb}}\) is calculated by multiplying the mass of burnt fuel by its lower heating value \(LHV\):

\[
Q_{\text{comb}} = m_{\text{fuel}} \cdot LHV,
\]

and the heat release rate at each angle of the crankshaft’s rotation during combustion is then calculated based on the Wiebe correlation, which requires the ignition angle \(\theta_{\text{ign}}\) and duration of the combustion \(\theta_{\text{comb}}\) as inputs to the function:

\[
\frac{\text{d}V}{\text{d}q} = \frac{\kappa \alpha}{\theta_{\text{comb}}} \left[1 - \exp\left(-2\left(\frac{\theta_{\text{comb}}}{\theta_{\text{comb}}} - \frac{\theta_{\text{ign}}}{\theta_{\text{comb}}}\right)\right)\right]^{1 - 1 - \frac{1}{\kappa}},
\]

and where we have used typical values of \(\alpha = 5\) and \(n = 3\) \([49]\).

The wall heat-loss term \(dQ_w/\text{d}t\) represents the summation of the instantaneous heat transfer losses through the wall through convection/conduction, and also due to radiation:

\[
dQ_w = h(\vartheta)A(\vartheta)(\theta_{\text{comb}} - \theta_{\text{comb}}) + \frac{\beta \text{Boltzmann}}{A(\vartheta)(\theta_{\text{comb}} - \theta_{\text{comb}})}.
\]

In Eq. (7), \(h(\vartheta)\) is the instantaneous heat transfer coefficient that quantifies the heat transfer between the hot gases inside the cylinder and the walls; \(A(\vartheta)\) is the instantaneous area available for heat transfer; \(\theta_{\text{comb}}\) is the instantaneous bulk gas temperature; \(\theta_{\text{comb}}\) is the wall temperature; \(N\) is the rotational speed of the engine, in rounds per second (rps); \(\beta\) is a radiation parameter; and \(\text{Boltzmann}\) is the Stephan-Boltzmann constant. The term \(h(\vartheta)\) is mainly a function of the crankshaft angle, the average pressure and temperature inside the cylinder, and the mean piston speed.

Two correlations are commonly found in the ICE literature for the calculation of the instantaneous heat transfer coefficient in the flows in these reciprocating spaces, namely the correlations proposed by Annand \([52]\) and Woschni \([49,50]\). In this study, the Woschni correlation as provided in Ref. \([51]\) has been used, according to which the heat transfer coefficient can be estimated from the expression:

\[
h(\vartheta) = 3.26\beta(\vartheta)^{0.4} \left(U(\vartheta)^{-0.8} \text{Pr}^{-0.2} \text{T}(\vartheta)^{-0.55}\right),
\]

\[
T(\vartheta) = \frac{P^2(\vartheta) V(\vartheta)}{m(\vartheta) R},
\]

where the term \(U(\vartheta)\) is the instantaneous characteristic gas velocity, which is calculated by accounting for the mean piston speed and the instantaneous pressure variation inside the cylinder \(P(\vartheta)\):

\[
U(\vartheta) = 2.282N + 0.003247 \text{V}_{\text{dis}}^{1.2} \left(P(\vartheta)-P_0\right) \frac{V_0}{P_0},
\]

Finally, in turbocharged engines, the air compression power \(W_{\text{comp}}\) and turbocharger power \(W_{\text{turb}}\) are calculated by using constant isentropic efficiencies from the expressions below:

\[
W_{\text{comp}} = \frac{m_{\text{fuel}}(h_c - h_o)}{\eta_{\text{comp}}},
\]

\[
W_{\text{turb}} = \frac{m_{\text{fuel}}(h_c - h_\text{out})}{\eta_{\text{turb}}},
\]

The complete system of ODEs representing the ICE thermodynamic model presented in Eqs. (1)–(21) was solved in MATLAB using the non-stiff ODE solvers ode113 and ode45 \([54]\).
2.3. CHP-ICE model validation

The dynamic ICE model presented in Section 2.2 was validated against data from manufacturer’s specification sheets for natural gas CHP-ICEs, provided by EnerG [55]. Specifically, the model was validated against three CHP-ICEs labelled ‘CHP-160’, ‘CHP-230’ and ‘CHP-2500’, which includes both natural aspiration and turbocharged engines, at full and part load conditions. A summary of the geometric and operating data of the engines used in the validation process is presented in Table 1. Due to lack of information on some design and operational aspects of the engines, some assumptions have been made, based on data available in the literature. Specifically, the unavailable information includes: (i) the duration and timing of ignition; (ii) the valve sizes, opening profiles and timing; (iii) the oil circuit design and friction losses (fmep); and (iv) the turbocharger geometry and performance data. The assumptions for the validation exercise are as follows:

- The valve size was estimated as a function of the cylinder diameter, in line with Ref. [49], and the valve timing and opening size were based on information provided in Refs. [49,51];
- The ignition timing varied between 40° and 5° before top dead centre (TDC);
- The friction losses were selected to be 10% of the shaft work, as an average value found in the literature [56,57] for ICEs;
- Finally, the compressor and turbine efficiencies were selected at 75% and 70%, respectively, in line with Refs. [51,56].

Temperature and pressure profiles of the exhaust gases inside a cylinder of the turbocharged CHP-2500 ICE are shown in Fig. 2. Similar results are presented in Fig. 3 for the natural aspiration CHP-230 ICE. The turbocharged CHP-ICE reaches peak temperatures of 2,200 K, in comparison to the 3,000 K attained in the natural-aspiration engine. This is due to the higher (lean) air-to-fuel ratio (AFR) used in the CHP-2500 ICE, which has an AFR of 29.5, as opposed to the stoichiometric AFR used in the CHP-230 engine, which is equal to AFR = 16.5. As expected, the peak pressure reached in the CHP-2500 ICE is 22,500 kPa, in contrast to the maximum 8,000 kPa reached in the CHP-230. The difference in pressure is also attributed to the use of the turbocharger in the CHP-2500 ICE, which results in an initial pressure inside the cylinder of 240 kPa, even prior to the commencement of the compression stroke.

A summary of the CHP-ICE model predictions against manufacturer’s data is presented in Table 2, for all three ICEs. The results are in good agreement with the measurement data, which are quoted with a ±10% uncertainty range. Some discrepancies observed are attributed to a lack of information on some design and operational aspects that had to be assumed for the validation exercise. The highest deviation between the manufacturer’s reported power output and corresponding model-predicted results is 4.3% for CHP-160 at 100% load. This value falls well within the ±10% uncertainty range stated in the engine specification data sheets. The predicted thermal energy content of the exhaust gas stream by the model also lies within the ±10% uncertainty range from the data, with the exception of CHP-230 at 50% load, where the predicted value is 16% lower. Moreover, the exhaust-gas temperature predicted for all engines falls in the range of ±4% compared to the manufacturer’s data. The only exemption is recorded for CHP-2500 at 75% load, where the model temperature is 9.7% higher than the reported mean temperature. In addition, the deviations of the predicted heat losses to the jacket water circuit from those reported in the data sheets amounts to between 0.9 and 9%. Overall, the validation exercise reveals that the model predictions are in good agreement with the manufacturer’s reported data, so the model can be used with confidence to predict the dynamic performance of CHP systems.

Heat flows into the cycle in the evaporator (Process 2a-3). The heat input (rate) can be calculated either by using the working fluid enthalpy temperature increases; this trend is observed in the data and is captured by the model, and can be understood in terms of the lower expansion work generated by the engine and the expander, resulting in higher enthalpy content of the exhaust-gas stream. The mass flow rate in all engines decreases as the load drops to lower values. These two observations are highlighted here as they will impact the ORC engine performance under part-load conditions later in this paper. For the turbocharged engine, there is an opportunity to increase the heat source temperature ΔT, to compensate for the lower gas (heat source) mass flow rate through the ORC engine evaporator. However, for the natural aspiration engines this is not possible.

2.4. ORC thermodynamic model

A subcritical recuperative ORC is illustrated on a T – s diagram in Fig. 4, with the main components of the ORC engine also noted on the figure. Process 1-2 corresponds to liquid pumping; 2a-2 is the heat recovery process from the (desuperheated) hot vapour; 2a-3 is the heat addition from the heat carrier fluid (exhaust gas stream); 3-4 represents the expansion of the working-fluid vapour; 4a is the heat rejection to the cold liquid working fluid in the recuperator HEX; and 4a-1 indicates the cycle heat rejection to the condenser cooling-water circuit. The temperature drops experienced by the flows of the heat carrier fluid (Process 5-6) and of the cooling-water stream (Process 7-8) are also illustrated on the same diagram. A spatially lumped, (quasi) steady-state thermodynamic model of the subcritical recuperative ORC engine shown in Fig. 4 was developed in MATLAB [54], by applying mass and energy balance equations to each component in Fig. 4. The ORC model is described by Eqs. (22)-(27). The following assumptions are considered for the ORC modelling:

- Steady-state operation of the ORC components;
- All HEXs were assumed to be of a counter-flow, concentric tube-in-tube design, and adiabatic;
- The pressure drop on the working fluid side in the HEXs, piping, etc. is negligible;
- The recuperator effectiveness was taken to be equal to ηrec = 0.80, the pump isentropic efficiency was set to ηpump, = 0.65 [30], the expander isentropic efficiency to ηexp, = 0.70 [58], and the electric generator efficiency to ηel = 0.93, similar to the ICE generator performance [55].

| Parameter | CHP-160 | CHP-230 | CHP-2500 |
|-----------|---------|---------|----------|
| W (kW)    | 150     | 230     | 2,535    |
| N (rpm)   | 1,500   | 1,500   | 1,500    |
| Number of cylinders | 8     | 12     | 20       |
| r (-)     | 12      | 12      | 14       |
| Vd (m³)   | 0.0018  | 0.0018  | 0.0048   |
| s (m)     | 0.142   | 0.142   | 0.21     |
| b (m)     | 0.128   | 0.128   | 0.17     |
| t (m)     | 0.22    | 0.22    | 0.326    |
| Intake valve diameter (m) | 0.435  | 0.435   | 0.435    |
| Exhaust valve diameter (m) | 0.355  | 0.355   | 0.355    |
| AFR (-)   | 16.5    | 16.5    | 29.5     |
| T (K)     | 298     | 298     | 298      |
| Initial mass (kg/cyl) | 0.0016 | 0.0016  | 0.0144   |
| LHV (kJ/kg) | 48,074 | 48,074  | 49,152   |
| Turbocharged (Yes/No) | No     | No     | Yes      |
| ηel (%)   | 95.4    | 95.4    | 97.5     |

Table 1: CHP-ICE specification data taken from Ref. [55].
increase between States 2a and 3, or by using the heat source stream fluid temperature decrease between States 5 and 6, by assuming a constant $c_p$ during this process:

$$\dot{Q}_{\text{evap}} = m_{\text{at}}(h_{2b} - h_{2a}) = m_{\text{hs}}c_{p,\text{ev}}(T_5 - T_6).$$

(22)

The normalised cycle superheating degree ($\text{SHD}$) quantifies the actual cycle superheating temperature rise as a fraction of the maximum superheating temperature rise allowable by the heat source without violating the pinch point ($\text{PP}$) temperature difference in the evaporator HEX. The $\text{SHD}$ is defined as:

$$\text{SHD} = \frac{T_3 - T_{3v}}{T_5 - T_{\text{PP},\text{ev}}},$$

(23)

which requires the heat source inlet temperature to the ORC engine ($T_5$), the working fluid saturation temperature during evaporation, and thus also at the beginning of the superheating process ($T_{3v}$), and the working fluid temperature at the exit of the evaporator HEX following superheating ($T_3$). In the limiting case of no superheating, $\text{SHD} = 0$, and the maximum value it can attain is $\text{SHD} = 1$.

After the evaporator, the saturated or superheated vapour undergoes an expansion process to generate power. For the purposes of this study, the isentropic efficiency of the expander (0.70 [58]) and the electrical generator efficiency (0.93 [55]) are both assumed to be fixed:

$$\dot{W}_{\text{exp}} = m_{\text{at}}(h_3 - h_4)\eta_{\text{exp}} = m_{\text{at}}(h_3 - h_4)\eta_{\text{exp}}\eta_{\text{elg}}.$$  

(24)

The low-pressure superheated vapour at the expander exit flows into the recuperator, which acts as a preheater where the hot vapour heats the cold liquid working fluid after the pump:

$$\dot{Q}_{\text{rec}} = m_{\text{at}}(h_{2b} - h_3).$$

(25)

This arrangement reduces the load of the condenser, resulting in a smaller-sized component, while also reducing the overall heat input at the evaporator, which can act to improve the cycle thermal efficiency. After the recuperator, the working fluid enters the condenser where it rejects heat to the cooling water circuit:

$$\dot{Q}_{\text{cond}} = m_{\text{at}}(h_{4a} - h_1) = m_{\text{cw}}c_{p,w}(T_6 - T_7).$$

(26)

The fluid circulation and two pressure levels in the ORC engine are maintained by a pump, which consumes some of the generated power. For the purposes of this present study, a fixed value is used for the isentropic efficiency of the pump (0.65 [30]):

$$\dot{W}_{\text{pump}} = m_{\text{at}}(h_2 - h_1)\eta_{\text{pump}}.$$  

(27)

### 2.5. Exergy analysis

Exergy is defined as the maximum theoretical work that a system will generate if it undergoes a reversible process, from its initial state to the environmental (dead) state [59,60]. The exergy balance for a
Table 2
Summary of CHP-ICE model validation results for different engines and load conditions.

| Engine      | Parameter                                    | 100% Load       | 75% Load        | 50% Load        |
|-------------|----------------------------------------------|-----------------|-----------------|-----------------|
|             | Data  | Model | Deviation   | Data  | Model | Deviation   | Data  | Model | Deviation   |
| CHP-160     |       |       |             |       |       |             |       |       |             |
| W_{in} (kW) | 150   | 144   | -4.3%       | 113   | 112   | -0.6%       | 75    | 80    | 7.2%        |
| Q_{out} (kW)| 432   | 432   | 0.0%        | 345   | 345   | 0.0%        | 257   | 257   | 0.0%        |
| Q_{in}, incl. Q_{fr} & intercooler (kW) | 155 | 155 | 0.3% | 131 | 131 | 0.1% | 99 | 106 | 7.8% |
| Q_{in} at 393 K (kW) | 79 | 80 | 1.1% | 60 | 58 | -3.3% | 43 | 39 | -8.4% |
| Q_{in}, incl. friction & oil circuit (kW) | 53.6 | 61.5 | 14.6% | 34.5 | 37.2 | 7.6% | 36.8 | 28.5 | -22% |
| m_{ex} (kg/s) | 0.16 | 0.15 | -3.2% | 0.13 | 0.12 | -3.8% | 0.09 | 0.09 | -4.6% |
| T_{ex} (K) | 867   | 886   | 2.2%       | 844   | 844   | 0.0%       | 826   | 795   | -3.7%       |
| n_{ex} (-)  | 0.55  | 0.54  | -0.3%      | 0.55  | 0.55  | -1.0%      | 0.55  | 0.56  | 1.5%        |
| n_{ex} (-)  | 0.35  | 0.33  | -5.7%      | 0.33  | 0.33  | -0.6%      | 0.29  | 0.31  | 7.2%        |
| CHP-230     |       |       |             |       |       |             |       |       |             |
| W_{in} (kW) | 229   | 221   | -3.7%       | 171   | 171   | -0.2%       | 114   | 121   | 6.4%        |
| Q_{out} (kW)| 649   | 649   | 0.0%        | 519   | 519   | 0.0%        | 386   | 386   | 0.0%        |
| Q_{in}, incl. Q_{fr} & intercooler (kW) | 236 | 232 | -1.7% | 200 | 198 | -0.94% | 151 | 158 | 4.6% |
| Q_{in} at 393 K (kW) | 120 | 129 | 7.3% | 91 | 95.5 | 4.95% | 66 | 63 | -4.4% |
| Q_{in}, incl. friction & oil circuit (kW) | 88.6 | 81.9 | -7.6% | 76.7 | 74.5 | -2.95% | 69.7 | 58 | -16.7% |
| m_{ex} (kg/s) | 0.24 | 0.23 | -2.8% | 0.19 | 0.18 | -4.1% | 0.14 | 0.13 | -5.1% |
| T_{ex} (K) | 873   | 900   | 3.1%       | 850   | 890   | 4.7%       | 838   | 839   | 0.1%        |
| n_{ex} (-)  | 0.55  | 0.52  | -5.6%      | 0.56  | 0.57  | 0.9%       | 0.56  | 0.58  | 2.8%        |
| n_{ex} (-)  | 0.35  | 0.34  | -3.7%      | 0.33  | 0.33  | -0.2%      | 0.30  | 0.31  | 6.4%        |
| CHP-2500    |       |       |             |       |       |             |       |       |             |
| W_{in} (kW) | 2,530 | 2,498 | -1.3%       | 1,899 | 1,834 | -3.4%       | 1,258 | 1,210 | -3.8% |
| Q_{out} (kW)| 5,748 | 5,748 | 0.0%        | 4,391 | 4,391 | 0.0%        | 3,037 | 3,037 | 0.0% |
| Q_{in}, incl. Q_{fr} & intercooler (kW) | 1,375 | 1,380 | 0.4% | 998 | 1,088 | 9.1% | 668 | 652 | -2.4% |
| Q_{in} at 393 K (kW) | 1,147 | 1,051 | -8.4% | 966 | 991 | 2.6% | 726 | 805 | 11% |
| Q_{in}, incl. friction & oil circuit (kW) | 844 | 968 | 14.7% | 429 | 388 | -9.7% | 324 | 316 | -2.7% |
| m_{ex} (kg/s) | 3.60 | 3.51 | -2.6% | 2.70 | 2.64 | -2.2% | 1.83 | 1.78 | -2.8% |
| T_{ex} (K) | 682   | 663   | -2.8%      | 716   | 751   | 4.9%       | 750   | 823   | 9.7%        |
| n_{ex} (-)  | 0.44  | 0.40  | -9.5%      | 0.42  | 0.45  | 6.3%       | 0.43  | 0.45  | 4.8%        |
| n_{ex} (-)  | 0.44  | 0.44  | -1.4%      | 0.43  | 0.42  | -3.4%      | 0.41  | 0.40  | -3.8% |

* The term Q_{out} differs from the term Q_{in}, in that it includes all other thermal losses due to friction, exhaust-gas mass trapped inside the engine cylinder(s), tail-pipe thermal losses, etc., which are not included in the thermodynamic model.

Fig. 4. Subcritical recuperative ORC on a temperature (T) – specific entropy (s) diagram.

calculation volume of an open system, at steady state conditions, is calculated by the following equation:

\[ 0 = \int_{1}^{2} \left(1 - \frac{T_i}{T_f}\right) dQ - W + \sum_{i=1}^{n} \dot{m}_{i,1} \cdot c_{P,i,1} \ln \frac{\dot{m}_{i,1,2}}{\dot{m}_{i,1,1}} - \dot{X}_{inlet} - \dot{X}_{inlet}. \]  \hspace{1cm} (28)

The first integral in Eq. (28), represents the heat exchange between the system and its surroundings; the second term \( W \) corresponds to the work generated (or added) from (to) the system; and the third term is the summation of the exergy that enters the system, followed by the exergy that leaves the system. The last term \( \dot{X}_{inlet} \) is the exergy destruction occurring in this component. The signs indicated in Eq. (28), are based on the convention that: (i) the heat exchange is positive, when heat is added to the system; and (ii) the work is positive, when it is generated by the system [61].

For the exergy flow calculations the following assumptions are considered [62-64]:

- Magnetic and nuclear forms of exergy are neglected;
- The change in kinetic and dynamic energy is negligible, therefore the kinetic and potential exergies are also considered negligible;
- The exhaust gases inside the ORC evaporator do not react and only transfer heat to the working fluids;
- The water present in the exhaust gases is in vapour phase, so the LHV of methane has been used for the calculations;
- The ideal gas law principles are applied to air and exhaust gases;
- The combustion reaction in the Otto ICE is complete, using the AFR in the manufacturers’ data sheet;
- The reference state for the system exergy calculation is the ambient environmental conditions.

With the above assumptions, the specific physical exergy of a substance is expressed as follows:

\[ e_{x,sub}^{ph} = (h-h_{ref}) - T_{ref}(s-s_{ref}). \]  \hspace{1cm} (29)

The specific chemical exergy of reference substances is defined as [59,65]:

\[ e_{x,sub}^{ch} = R T_{ref} \ln \left( \frac{P_{ref}}{P_{sub}} \right). \]  \hspace{1cm} (30)

The total specific exergy is therefore calculated using:
\[ ex_i = ex_i^{\text{th}} + ex_i^{\text{ch}}. \]  

(31)

The chemical exergy for a mixture is expressed as [62]:

\[ ex_{\text{mix}}^\text{ch} = \sum_{i=1}^{n} x_i ex_i^{\text{ch}} + R T_0 \sum_{i=1}^{n} x_i \ln(n_i). \]  

(32)

For the exergy destruction process occurring within the ICE, Eq. (28) can be rearranged to:

\[ \dot{X}_{\text{dest,ICE}} = \dot{X}_a + \dot{X}_{\text{ex,IC}} - \dot{X}_{\text{work}} - \dot{X}_{\text{jw}}, \]  

(33)

where \( \dot{X}_{\text{air}} \) is the exergy flow of the intake air, \( \dot{X}_a \) the exergy of the fuel, \( \dot{X}_{\text{ex,IC}} \) the exergy flow of the exhaust gas stream, \( \dot{X}_{\text{work}} \) the ICE shaft work, and \( \dot{X}_{\text{jw}} \) the exergy flow to the jacket water system of the ICE. The exergy flow of the intake air is zero for a natural aspiration engine since the air is already at environmental conditions. The exergy flow of the intake air in the turbocharged engine is calculated using the air temperature and pressure after the intercooler, prior entering the cylinder:

\[ \dot{X}_{\text{air}} = \dot{m}_{\text{air}} \left[ (h_{\text{air}} - h_{\text{air,0}}) - T_0 (s_{\text{air}} - s_{\text{air,0}}) + c_{p,\text{air}} R T_0 \ln \left( \frac{P_{\text{air}}}{P_0} \right) \right]. \]  

(34)

The exergy flow of natural gas is assumed to be equal to that of methane, and is estimated using the 
LHV of methane as follows [62]:

\[ \dot{X}_a = \dot{m}_a ex_l^{\text{th}}, \]  

(35)

\[ ex_l^{\text{th}} = LHV \left( 1.033 + 0.0169 \frac{C}{H} - 0.0698 \right). \]  

(36)

Here, \( C \) stands for the number of carbon atoms in the fuel an H for the number of hydrogen atoms. In the case of methane \( C = 1 \) and \( H = 4 \).

The ICE engine shaft work is used for the estimation of \( \dot{X}_{\text{work}} \) as obtained from the ICE-ORC CHP optimisation analysis. The exergy flow rate of the exhaust gases is estimated assuming that all gases are ideal, and using the AFR of the engine to define the mass fraction of each gas in the combustion products. The physical exergy of the exhaust gases is calculated using:

\[ \dot{X}_{\text{ex,IC}} = \dot{m}_{\text{ex}} \left[ (h_{\text{ex}} - h_{\text{ex,0}}) - T_0 (s_{\text{ex}} - s_{\text{ex,0}}) \right]. \]  

(37)

The chemical exergy of the exhaust gases mixture is estimated as follows:

\[ \dot{X}_{\text{ex,IC}}^\text{ch} = \sum_{i=1}^{n} x_i ex_i^{\text{ch}} + T_0 \sum_{i=1}^{n} R x_i \ln(x_i). \]  

(38)

\[ \dot{X}_{\text{ex,IC}} = \dot{X}_{\text{ex,IC}}^{\text{ph}} + \dot{X}_{\text{ex,IC}}^{\text{ch}}. \]  

(39)

The jacket water circuit exergy flow is calculated using the physical exergy of the water stream because there is no change in the composition of the fluid:

\[ \dot{X}_{\text{jw}} = \dot{m}_{\text{jw}} c_{p,\text{jw}} \left( T_{\text{jw,in}} - T_{\text{jw,out}} - T_0 \ln \left( T_{\text{jw,in}} / T_{\text{jw,out}} \right) \right). \]  

(40)

It should be noted that Eq. (33) is applicable in natural aspiration ICES. For the turbocharged engines the exergy flow of the compressor, the intercooler heat exchanger and the turbine-expander are required. The exergy destruction rate at the compressor (\( \dot{X}_{\text{dest,comp}} \)) is calculated as follows:

\[ \dot{X}_{\text{dest,comp}} = \dot{m}_{\text{comp}} \left[ (h_{\text{comp,in}} - h_{\text{comp,out}}) - T_0 (s_{\text{comp,in}} - s_{\text{comp,out}}) \right] + \dot{W}_{\text{comp}}. \]  

(41)

The exergy destruction rate at the intercooler (\( \dot{X}_{\text{dest,intr}} \)) is calculated using:

\[ \dot{X}_{\text{dest,intr}} = \dot{m}_{\text{intr}} \left[ (h_{\text{intr,in}} - h_{\text{intr,out}}) - T_0 (s_{\text{intr,in}} - s_{\text{intr,out}}) \right] + \dot{m}_{\text{jw}} c_{p,\text{jw}} \left( T_{\text{jw,in}} - T_{\text{jw,out}} - T_0 \ln \left( T_{\text{jw,in}} / T_{\text{jw,out}} \right) \right). \]  

(42)

The exergy destruction rate of the expander (\( \dot{X}_{\text{dest,turb}} \)) is calculated:

\[ \dot{X}_{\text{dest,turb}} = \dot{m}_{\text{turb}} \left[ (h_{\text{turb,in}} - h_{\text{turb,out}}) - T_0 (s_{\text{turb,in}} - s_{\text{turb,out}}) \right] - \dot{W}_{\text{turb}}. \]  

(43)

The exergy efficiency of the ICE-CHP is therefore estimated by:

\[ \eta_{\text{ICE,ex}} = \frac{W_{\text{ICE}}}{X_f}. \]  

(44)

For the total ICE-ORC CHP system the exergy efficiency is estimated using:

\[ \eta_{\text{ORC,ex}} = \frac{W_{\text{ORC}}}{X_{\text{ex}}}. \]  

(45)

The exergy destruction rate in each ORC engine component is evaluated as detailed in Eqs. (46)–(50):

\[ \dot{X}_{\text{dest,comp}} = \dot{m}_{\text{comp}} \left[ (h_{\text{comp,in}} - h_{\text{comp,out}}) - T_0 (s_{\text{comp,in}} - s_{\text{comp,out}}) \right] + \dot{W}_{\text{comp}}. \]  

(46)

\[ \dot{X}_{\text{dest,exp}} = \dot{m}_{\text{exp}} \left[ (h_{\text{exp,in}} - h_{\text{exp,out}}) - T_0 (s_{\text{exp,in}} - s_{\text{exp,out}}) \right] + \dot{W}_{\text{exp}}. \]  

(47)

\[ \dot{X}_{\text{dest,turb}} = \dot{m}_{\text{turb}} \left[ (h_{\text{turb,in}} - h_{\text{turb,out}}) - T_0 (s_{\text{turb,in}} - s_{\text{turb,out}}) \right]. \]  

(48)

\[ \dot{X}_{\text{dest,pump}} = \dot{m}_{\text{pump}} \left[ (h_{\text{pump,in}} - h_{\text{pump,out}}) - T_0 (s_{\text{pump,in}} - s_{\text{pump,out}}) \right]. \]  

(49)

Finally, the exergy efficiency of the ORC engine is calculated as follows:

\[ \eta_{\text{ORC,ex}} = \frac{W_{\text{ORC}}}{X_{\text{ex}}}. \]  

(50)

3. Integrated ICE-ORC CHP system optimisation

A typical optimisation problem involves the minimisation or maximisation of an objective function \( F(Z) \), subject to a set of constraints. Here, vector \( Z \) contains all the decision variables, i.e. the set of independent variables the optimiser is allowed to alter in order to minimise/maximise the value of \( F(Z) \). In this work, two objective functions have been defined and optimised separately (single-objective optimisation). The first objective function calculates the combined net power outputs of the CHP-ICE (\( W_{\text{ICE}} \)) and ORC (\( W_{\text{ORC}} \)) engines, which we seek to maximise, given the heat source conditions and subject to a set of operational constraints. The second objective function calculates the SFC of the combined ICE-ORC CHP system, which we seek to minimise, again subject to operational constraints. Vector \( Z \) contains 14 decision variables related to the operation of the ICE and the ORC: (i) combustion ignition angle (\( \theta_{\text{ign}} \)); (ii) combustion duration (\( \theta_{\text{dur}} \)); (iii) intake valve (IV) timing, i.e. crankshaft angle at which this valve will open and close (\( \theta_{\text{IV,on}}, \theta_{\text{IV,off}} \)); (iv) exhaust valve (EV) timing, i.e. crankshaft angle at which this valve will open and close (\( \theta_{\text{EV,on}}, \theta_{\text{EV,off}} \)); (v) IV lift height (\( L_{\text{IV}} \)); (vi) EV lift height (\( L_{\text{EV}} \)); (vii) initial pressure conditions inside the cylinder (\( P_0 \)), and to the operation of the ORC engine: (viii) evaporating pressure (\( P_{\text{evap}} \)); (ix) condensing pressure (\( P_{\text{cond}} \)); (x) working fluid mass flow rate (\( \dot{m}_{\text{w}} \)); (xi) superheating degree (\( \Delta S_{\text{HD}} \)); and (xii) expander volume ratio (\( \nu_{\text{exp}} \)). The two objective functions are presented in Eqs. (52) and (53):

\[ \text{maximise: } \{ W_{\text{ICE}} = W_{\text{ICE}} + W_{\text{ORC}} \}. \]  

(52)
which we seek to optimise subject to:

- The SHD can only take values between 0 and $1.0 \leq \text{SHD} \leq 1$;
- The PP in any given HEX should not violate a minimum value $P_{P_{\text{min}}} \leq P$;
- The evaporating pressure should be lower than the working fluid’s critical pressure, and the condensing pressure should be lower than the evaporating pressure: $P_{\text{cond}} < P_{\text{evap}} < P_{\text{sat}}$;
- The expansion process should not violate the isentropic relation: $P_{\text{evap}}/P_{\text{sat}} \leq \eta_{\text{exp}}$;
- The heat source (exhaust gas) stream exit temperature from the evaporator should not be below a limiting temperature to avoid the risk of reaching the gas dew-point temperature: $T_{\text{dew,in}} < T_{\text{dew,out}}$;
- The heat source (exhaust gas) stream pressure drop in the ORC evaporator should not exceed 50 kPa, to avoid significant negative backpressure on the ICE operation. For every iteration during optimisation, the new exhaust-gas temperature, pressure and mass flow rate is calculated and the revised ICE power output is estimated based on the obtained backpressure. This ORC evaporator pressure drop limit is at the high end of reported values in the literature (up to 40 kPa in Refs. [66,67]);
- The maximum temperature experienced by each working fluid should not exceed the maximum allowable temperature that would ensure its chemical stability, as suggested in Ref. [68];
- The value of $\dot{m}_{\text{in}}$ should lie between $–40^\circ$ and $–5^\circ$ before TDC and that of $\dot{m}_{\text{fl}}$ from $20^\circ$ to $40^\circ$ [49];
- The mixture mass and pressure inside the ICE cylinder at the end of a complete cycle (720°) should be equal to the initial mass and pressure conditions for the steady-state assumption to be valid;
- The valve lift height can vary up to a maximum of ± 25% of the nominal design value, with the average lift height calculated based on Ref. [69];
- The AFR of each ICE-CHP engine is fixed to the nominal design value (refer to Table 1) to avoid issues with NOx formation due to the alteration of the combustion mixture composition.

The optimisation problem was solved in MATLAB [54] using the multistart structure, which repeatedly runs a local solver from a number of different starting points, predefined by the user. In this work, the interior point algorithm fmincon was used as the solver to minimise the constrained nonlinear multivariable objective functions. For the maximum power output optimisation exercise, the negative value of the power output was minimised instead. The reader can refer to Ref. [70] for more details on the mathematical formulations of the optimisation algorithm.

### 3.1. CHP-ORC optimisation cases

As discussed in earlier sections, in this work we attempt to optimise simultaneously the ICE-ORC CHP system, as a combined system rather than taking the exhaust-gas conditions exiting the ICE as fixed, and optimising only the ORC engine design. To achieve this, five cases have been investigated. In Case 1 (C1) the ICE operates at nominal conditions and only the ORC engine is optimised for maximum power output, and in Case 2 (C2) the ICE is optimised for maximum power output, and the ORC engine is sized based on these results. Case 3 (C3) considers the simultaneous optimisation of the full ICE-ORC CHP system for maximum power output. In Case 4 (C4) the ICE is optimised for minimum SFC and the ORC engine is sized based on these results. Finally, Case 5 (C5) considers the simultaneous optimisation of the full ICE-ORC CHP system for minimum SFC. These cases were examined for two ICEs:

- CHP-230 (natural aspiration ICE) and CHP-2500 (turbocharged ICE), as those were presented in Section 2. Finally, the ORC engine specification used in the current optimisation study is summarised in Table 3.

#### Table 3

| Parameter                  | Value          | Parameter                  | Value          |
|----------------------------|----------------|----------------------------|----------------|
| $P_{\text{fl,in}}$ (kPa)   | 10             | $P_{\text{fl,max}}$ (kPa)  | 0.95 $P_{\text{fl,in}}$ |
| $P_{\text{fl,evap}}$ (kPa) | 5              | $\tau_{\text{exp}}$ (–)   | 8–18           |
| $T_{\text{fl,in}}$ (K)     | 288            | $\tau_{\text{rec}}$ (–)   | 0.80           |
| $T_{\text{fl,evap}}$ (K)   | 298            | $\tau_{\text{lim}}$ (K)   | 263            |
| $m_{\text{u}}$ (kg/s)     | From ICE simulation | $\tau_{\text{cond},(–)}$ | 0.70           |
| $t_{\text{u,in}}$ (K)     | From ICE simulation | $\tau_{\text{pump},(–)}$ | 0.65           |
| $P_{\text{cond,min}}$ (kPa) | 10             | $\tau_{\text{cond},(–)}$ | 0.93           |

#### 3.2. ORC working fluid selection

Working fluid selection can strongly influence ORC engine operating conditions, performance, component size and cost. Based on increasing concerns over global warming, certain fluids such as chlorofluorocarbons (CFCs) and hydrofluorocarbons (HFCs) have been already phased out, or are set to be phased out over the next decade. Therefore, technical solutions which are not constrained by such regulations are required to maximise the market penetration of ORC technology. Adding to this, some limitations when using ORCs for harvesting thermal energy from the exhaust gases in ICE applications include the selection of fluids and operating temperatures that do not accelerate the fluids’ chemical decomposition. Fluids with high critical temperatures/pressures are considered most suitable for this application in this regard. Li [43] studied fluids such as Toluene, n-Heptane, Cyclohexane, R113 and R123 in very high-temperature applications with a heat-source temperatures of 240–290 °C, obtaining ORC thermal efficiencies up to 31%. Similarly Vescovo and Spagnoli [71] considered fluids for use at very high temperatures up to 400 °C, and report ORC thermal efficiencies up to 30–34%. Uusitalo et al. [72] considered siloxanes for heat source temperatures of up to 400 °C in a small-scale ORC experimental set-up, and achieved a thermal efficiency of approximately 16%.

Based on the above, working fluids were selected for this study on the basis of good thermodynamic characteristics, but also of low ozone depletion potential (ODP) and global warming potential (GWP) values. The selected fluids include five refrigerants: R245fa, R152a, R1233zd, R1234ze and R1234yf. R245fa is commonly used in commercial ORC engines provided by, for example, Bosch, Cryostar, Electratherm, GE and Turboden [73]. R1233zd is a very promising replacement for R123, and similarly R1234ze and R1234yf are replacement refrigerants for R134a, which is currently used in commercial ORC engines, e.g. manufactured by Cryostar [42]. Apart from these refrigerants, four hydrocarbons were also examined in the present work, specifically: (i) three alkanes (Butane, Pentane, Hexane), as these have shown promising results in previous ORC studies [33,41,74]; and (ii) one aromatic (Toluene), which is suitable for high temperature applications and is also used in ORC engines manufactured by Tri-o-gen. It should be noted that hydrocarbons are listed in Category A3 in the ASHRAE Classification [75], which includes fluids with low toxicity, but high flammability.

### 4. Results and discussion

#### 4.1. Natural aspiration CHP-230 ICE and ORC engines

The power output of the naturally aspirated CHP-230 ICE for each case and fluid investigated is presented in Fig. 5a. The highest power
output from the ICE alone is 305 kW for Case C2, when the system is optimised for maximum power output from the CHP-ICE only, followed by Case C3 with 303 kW, when the complete ICE-ORC CHP system is optimised for (total) maximum power. The optimum ICE power output in Case C2 is 37% higher than the nominal engine design value of 221 kW in Case 1 (see Tables 2 and 4). This significant increase in the engine power output can be attributed to a number of factors. Firstly, the optimised CHP-ICE has a delayed ignition angle (−5° before TDC) compared to the nominal design (−20° before TDC), which acts to increase the pressure inside the cylinder before the expansion stroke and, in turn, to increase the displacement power. Also, it is found that the optimised IV and EV lift heights are approximately 8% higher than those provided in the nominal engine design. Furthermore, for a fixed AFR, higher valve lifts allow for increased fuel flow into the cylinders, which also increases the power output, although it is noted that this will also act to increase the fuel consumption of the engine (this is discussed in subsequent sections). Adding to this, the optimal EV opens later than in the nominal design, allowing more work to be generated on the engine shaft and reducing the blow-down losses of the engine.

When the cogeneration system is optimised for minimum SFC (Cases C4 and C5), the power output from the ICE reduces to 299 kW in Case C4 (ICE-only optimised for minimum SFC), and 294 kW in Case C5 (full ICE-ORC CHP system optimisation for minimum SFC), which are 35% and 33% higher than the nominal ICE design value of 221 kW in Case 1, and as given in Tables 2 and 4. To understand these changes, we can compare the operating conditions of the ICE in Cases C2 and C4. Such a comparison reveals a change in the duration and timing of the

![Figure 5](image-url)

**Fig. 5.** (a) CHP-230 ICE power output, and (b) optimum bottoming ORC engine power output.

### Table 4

Optimum CHP-230 ICE operation results.

| CHP-ICE Parameter | Cases          |
|-------------------|---------------|
|                   | C1 (nominal)  | C2   | C3   | C4   | C5   |
| W_{\text{ICE}} (kW) | 221          | 305  | 303  | 299  | 294  |
| Q_{\text{comb}} (kW) | 649          | 840  | 833  | 821  | 803  |
| Q_{\text{w, incl. intercooler}} (kW) | 236          | 252  | 244  | 238  |
| m_{\text{ex}} (kg/s) | 0.236        | 0.292 | 0.289 | 0.287 | 0.275 |
| m_{\text{f}} (kg/s) | 0.013        | 0.017 | 0.017 | 0.017 | 0.016 |
| T_{\text{ex}} (K) | 873          | 1,002 | 1,011 | 997  | 1,011 |
| \eta_{\text{th}} (for exhaust gases cooled to 120 °C) | 0.55         | 0.52  | 0.51  | 0.52  | 0.51  |
| \eta_{\text{el}} | 0.35         | 0.36  | 0.37  | 0.37  | 0.37  |
valve operation between these two cases. For maximum power output optimisation (Case C2), the IV opens slightly earlier, and closes slightly later, allowing a higher mass flow rate into the cylinder. As was mentioned earlier, since the AFR is fixed at the manufacturer’s recommended value, the higher combustible-mixture mass flow rate implies also a higher fuel mass flow rate. This is also confirmed by looking at the fuel mass flow rate into the cylinders, which is equal to 0.0171 kg/s for the power-optimised engine (Case C2) and 0.0165 kg/s for minimum SFC operation (Case C4); see Tables 2 and 4. Additionally, for the engine optimised for maximum power output, the EV closes later, which is in line with the data reported in the literature on high performing engines that have their EV open for longer to maximise power [49].

The optimum ORC engine power output for all cases and working fluids is shown in Fig. 5b, and ranges from a minimum of 15 kW to a maximum of 41 kW, almost three times higher. The maximum ORC power output for all working fluids is found for Case C3, when the combined ICE-ORC CHP system is optimised simultaneously for total power generation. The best performing fluid is Pentane with 41 kW, followed by Butane with 39 kW, R1233zd with 37 kW, and Toluene and Hexane with 34 kW. These power outputs are 26%, 23%, 23%, 34% and 23% higher than the respective optimum ORC design for the nominal ICE (Case C1), for the same fluids. The worst fluid is R1234yf followed closely by R1234ze.

In line with the results in Table 4, it is found that when the

![Fig. 6. Power output breakdown between the CHP-230 ICE and the bottoming ORC engine for: (a) Case C1, (b) Case C2, (c) Case C3, (d) Case C4, and (e) Case C5.](image-url)
combined ICE-ORC CHP system is optimised for maximum power output (Case C3) the system optimiser attempts to rise to a slightly lower optimum power output from the CHP-ICE (303 kW in Case C3), in order to keep the exhaust-gas temperature high, but at ORC higher power output.

The breakdown of the power generated by the CHP-ICE and the ORC engine, for all investigated cases and working fluids, is presented in Fig. 6. As expected, the total power output is dominated by the CHP-ICE operation, with the ORC engine contributing between 6 and 15% of the total power output, depending on the case and fluid. It should be noted that although Cases C2 and C3 generate a similar total power output, the percentage of power coming from the ORC engine increases in Case C3, when the complete ICE-ORC CHP system is optimised. A case in the point is that in which the ORC engine with Butane in Case C2 is responsible for 10% of the total power generated, whereas in Case C3 it is responsible for 12% (20% higher). This increase in the ORC power output is also associated with a decrease in the fuel consumption of the whole system in Case C3, relative to Case C2, as indicated in Table 4.

Importantly, there are also important operational, lifetime and maintenance implications that arise from these differences, since ORC engines are generally associated with lower maintenance than ICEs.

The best performing fully optimised complete ICE-ORC CHP system generates a total of 345 kW of power, when Pentane is used as the working fluid (Fig. 6c). This is 36% higher than the nominal CHP-ICE coupled to a bottoming ORC design for the same fluid, which generates 253 kW (Fig. 6a). Overall, the fully optimised ICE-ORC CHP system generates between 31% and 37% higher power output compared to the baseline scenario (Case C1), where the CHP-ICE operates at the nominal design conditions, and the ORC engine is sized based on the exhaust-gas conditions at this operating point.

The results also reveal that optimising the ICE-ORC CHP system for maximum power output leads to solutions that consume more fuel, with higher SFCs. The SFCs for all cases investigated are summarised in Fig. 7. These range from 2.4 kW/kW in the best case (Case C3 optimised ICE-ORC CHP with Pentane) to 2.9 kW/kW in the worst case (Case C1 nominal ICE only). The absolute SFCfuel (kg/kWhnet) is also presented in Fig. 7. This varies from 0.22 kg/kWhnet (Case C1 nominal ICE only) to 0.175 kg/kWhnet for Case C3 (with Pentane). Therefore, a reduction of almost 20% is possible, with respect to the nominal case (C1) of a CHP-ICE engine only. It is also observed that when the complete ORC CHP system is optimised in Case C5 for minimum SFC, the resulting SFC is similar to the respective one achieved in Case C3 (ICE-ORC CHP optimisation for maximum power). Nevertheless, the total fuel consumption in the former case is approximately 5% lower than in the latter, even though this corresponds to a total power output reduction of only 3%. This trend highlights that by simply optimising the ICE for maximum power output, independently from the ORC engine design, will increase the operating cost of the system, due to higher annual fuel consumption.

4.1.2. ORC design considerations

The pressure ratio of the optimum ORC engine and the ORC working fluid flow rates for all cases are shown in Fig. 8. These results indicate that, with some exceptions, the optimum ORC evaporating and condensing pressure levels stay almost the same in all investigated scenarios (Cases C1-C5), with most pressure ratios (PRs) having values up to ~25, whilst the mass flow rate of the ORC working fluids varies strongly between cases, resulting in the previously observed power outputs. It is noted that some PRs are quite high (up to ~40–50), especially those in Cases 3 and 5 associated with fully optimised integrated ICE-ORC CHP systems, with Toluene. This has important implications on the optimal expander technology selection and design for this application.

In more detail, fluids such as Toluene, Pentane, R1233zd and Hexane have higher optimal PRs, varying between 25 and 50, while the mass flow rate for the same fluids is low, varying in the range 0.4–0.6 kg/s. The opposite trend is observed for fluids such as R245fa, R152a, R1234yf, and R1234ze. This observation can be understood as follows; for all fluids, the optimiser chooses to maximise the power output by first increasing the pressure difference between the condenser and the evaporator, without violating the constraints that the cycle should be subcritical. If these constraints are active, then the mass flow rate of the working fluid increases to generate more power. Since the cooling water circuit temperature is fixed (288–298 K), the saturation condensing temperature is restricted by the water temperature, and the HEX PP (PPcond = 10 K). Working fluids such as R152a, R1234ze, and R1234yf have saturation temperatures of 308 K, at pressures of 6–9 bar, whilst Pentane and Hexane have a similar saturation temperature at 1.4 and 0.5 bar, respectively. This difference in the condensation pressure, for similar saturation temperatures, results in significantly different pressure ratios amongst the investigated fluids.

The thermal loads of the various ORC engine HEX components are illustrated in Fig. 9. In agreement with the results presented and discussed previously, the dry fluids (Butane, Pentane, Hexane, Toluene, R1233zd, R1245fa) have little preheater load, because most of the preheating is performed in the recuperator that is enabled by the high superheating degree of these fluids after the expander. Due to the positive slope of their saturated vapour line, these fluids are highly superheated after the expansion process, offering great opportunity for recuperating some of that heat, for preheating the cold liquid fluid exiting the pump. This is also illustrated in Fig. 10, where for the same dry fluids approximately 70% of the heat rejection is done in the recuperator and desuperheater. On the contrary, for fluids such as R1234ze and R1234yf (isentropic fluids) there is an important

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Fig. 7. SFC and SFCfuel for the different cases investigated, based on CHP-230 ICE with and without a bottoming ORC engine.
preheater load (Fig. 9), whilst about 50% of the heat rejection for the same fluids occurs in the two-phase condensing zone (Fig. 10).

As expected, the highest heat input (Fig. 9) is recorded for the ORC engines with the highest power output. Pentane generates the highest power output, and it has the highest heat input of 320 kW, followed by Butane with 310 kW (the figures quoted correspond to the Case C3 results). It should be noted that R1233zd has a slightly lower heat input than Hexane, although the former (R1233zd) generates more power than the latter (Hexane). This can be understood by considering the thermal efficiency of these cycles. The thermal efficiency of the ORC engines are presented in Fig. 11. The highest thermal efficiency is observed for Pentane 25%, followed by Butane with 22%, R1233zd with 20.5%, Hexane with 19.5% and Toluene with 17.5%. R1233zd operates with higher thermal efficiency than Hexane, therefore it requires less heat input for a similar generated power output. By comparing the thermal efficiencies of the ORC engines across the different scenarios (cases, fluids), it emerges that the efficiency of the cycle remains at similar levels, with the efficiency of Case C5 being slightly higher than the other cases.

4.2. Turbocharged CHP-2500 ICE and ORC engines

4.2.1. CHP system level performance

The power output of the turbocharged CHP-2500 ICE for each case and fluid investigated is shown in Fig. 12a. As expected, and similarly to the results for the naturally aspirated CHP-230 ICE, the maximum power output by the ICE alone is recorded in Case C2, when the system is optimised for maximum power output of the ICE only. The maximum power output amounts to 2,540 kW, which is 2% higher than the nominal design output. The exhaust-gas exit temperature for the engine in Case C2 is 661 K, which is close to that of the nominal design (663 K), while the fuel consumption is higher than the nominal conditions by 45 kW (see Table 5). The ICE efficiency in Case C2 is also slightly higher at 44%.

A closer examination of the ICE operating conditions in Case C2 reveals that the EV stays open for longer than in Case C1, which agrees with the data reported in the literature for high performing engines. Also, the IV and EV lift height is approximately 8% larger than in the nominal design, reducing the blow-down losses of the engine. On the contrary, when the system is optimised for minimum SFC (Cases C4 and C5), the ICE power output drops by ∼10% relative to Case C1. However, this is accompanied by a significant reduction in fuel consumption, by up to 15% compared to Case C1. These results also reveal that all optimum designs for maximum power output have lower exhaust-gas temperatures than the designs for minimum SFC (Table 5), which results in lower power outputs from the CHP-ICE, but also allows higher power generation from the ORC engine.

The optimised ORC engines for all investigated cases and fluids are shown in Fig. 12b. When the ORC engine is designed (sized) for the nominal CHP-ICE operating conditions (Case C1), the maximum power output recorded is 188 kW for Pentane, and 186 kW for Toluene, followed by Butane with 184 kW, R1233zd with 181 kW, Hexane with 174 kW, and R245fa with 152 kW. In Case C2, the maximum power output of the ORC engine is either similar or slightly lower than the
nominal design, due to the reduced exhaust-gas temperature. The maximum ORC power output is recorded for every fluid in Case C3, when the combined ICE-ORC CHP system is optimised for maximum power output. This trend is in line with the results obtained for the naturally aspirated CHP-230 ICE. The best performing ORC in Case C3 generates 195 kW with Pentane; Toluene generates 192 kW, Butane 191 kW, and R1233zd 187 kW. The power output of the ORC engine in Case C3 increases by approximately 4% in comparison to the baseline case, whilst the ICE power output decreases slightly in this scenario.

The breakdown of total generated power between the CHP-2500 ICE and ORC engines for the various cases and fluids is presented in Fig. 13. As expected, the total power output is dominated by the CHP-ICE, with the ORC engine contributing between 4 and 8% of the overall power output, depending on the working fluid. These values are lower than those obtained for the natural aspiration engine CHP-230, mainly due to the lower exhaust-gas temperatures of the CHP-2500 ICE. It should be noted that although Cases C2 and C3 have similar total power outputs, the fraction of power coming from the ORC engine increases in Case C3, when the integrated ICE-ORC CHP system is optimised. It is interesting to note that with Toluene in Case C2 the ORC engine is responsible for 6.5% of the total generated power, whereas in Case C3 it is responsible for 7.5%. Importantly, this increase of the ORC power output is also accompanied by a decrease in total fuel consumption compared to Case C2, as shown in Table 5. As noted with the CHP-230 ICE, these differences can have important operational, lifetime and maintenance implications, since ORC engines are generally associated with lower maintenance than IC engines.

The maximum ICE-ORC CHP total power output of 2,732 kW is observed in Case C3, when either Pentane or Toluene are used as ORC working fluids (Fig. 13c). This power output value is 2.5% higher than the nominal design power output in Case C1 (Fig. 13a). The results also reveal that optimising the system for maximum power output leads to a higher fuel consumption, and an increase in the SFC. The SFC for all investigated cases and fluids is summarised in Fig. 14. For the ICE-ORC CHP system with the best performing ORC fluids (Pentane and Toluene) in Cases C4 and C5, the SFC drops to 2.1 kW/kW. This amounts to a reduction of approximately 7% compared to the nominal/baseline Case C1. The absolute SFC abs is also presented in Fig. 14. The ICE-ORC CHP with pentane consumes 0.155 kg/kWh (Case C5), which is 4.5% lower than the integrated ICE-ORC CHP in Case C1 for the same fluid, and 8% lower than the stand alone ICE, when operating at nominal conditions.

The results indicate that the optimisation performed for minimum SFC (or SFC abs) returns similar SFC (or SFC abs) values as the complete system optimisation for maximum power (Case C3), however, the power output breakdown differs by promoting the ORC power output. These findings highlight that maximising the system power output results in higher fuel consumption, thereby also affecting the running costs.

4.2.2. ORC engine design considerations

The ORC working fluid mass flow rates and operating pressure ratios are shown in Fig. 15. Fluids with high PRs operate with lower mass flow rates, and vice versa. For all cases, the optimum PR per fluid remains the same, indicating that the constraints for the minimum and maximum pressure levels in the cycle are active. Therefore, the ORC power output can only improve further by increasing the superheating degree and/or the mass flow rate of the working fluid. It should be also highlighted that the optimum pressure ratios in the ORC engines coupled to the CHP-2500 ICE are lower than those recorded for CHP-230 ICE, mainly due to the exhaust-gas temperature exiting CHP-2500 ICE is 1,000 K, whereas that exiting the CHP-2500 ICE.
it is approximately 680 K. This finding indicates that transcritical ORC engines may be suitable for high temperature applications. However, the additional cost of these systems might risk the overall viability of the project, and need to be carefully considered.

The heat input to and the heat rejected from the ORC are presented in Figs. 16 and 17, also showing the breakdown across the different HEXs of the ORC engine. Fluids with high SHD have high recuperator loads, since the working fluid after the expander is still highly superheated. This offers a great opportunity for recovering heat to preheat the cold liquid working fluid leaving the pump. Butane, R152a and R1233zd, for example, have approximately 53% of the heat input delivered in the superheater, and 30% in the recuperator (Fig. 16). The same fluids experience more than 50% of the heat rejection in the recuperator and the desuperheater (Fig. 17). On the contrary, fluids such as Pentane and Toluene have low SHD, and thus low heat rejection to the recuperator and desuperheater. This can be explained by the evaporation saturation temperatures of those fluids, which are 467 K and 445 K respectively, which are higher than the saturation temperatures of R152a (384 K) or Butane (422 K), allowing the latter fluids to have higher SHDs without violating the evaporator PP. These findings have an important impact on the design and sizing of the HEXs of these engines, since fluids with a similar power output (such as Toluene and Pentane) will have very different component design requirements.

Finally, the thermal efficiency of the ORC engines is presented in Fig. 18. The highest thermal efficiency is observed for Butane 21%, followed by R1233zd with 20.5%, R1233zd with 20.5%, Hexane with

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Table 5
Optimum CHP-2500 ICE operation results.

| CHP-ICE Parameter | Cases          |
|-------------------|---------------|
|                   | C1 (nominal)  | C2  | C3  | C4  | C5  |
| $\dot{W}_{IC E}$ (kW) | 2,498         | 2,540| 2,537| 2,220| 2,230|
| $\dot{Q}_{comb}$ (kW) | 5,748         | 5,787| 5,778| 5,038| 5,087|
| $\dot{Q}_{w}$ incl. $\dot{Q}_{w}$ & intercooler (kW) | 1,380         | 1,381| 1,147| 1,234| 1,230|
| $\dot{m}_{ex}$ (kg/s) | 3.51          | 3.50 | 3.52 | 3.06 | 3.09 |
| $\dot{m}_{f}$ (kg/s) | 0.117         | 0.118| 0.118| 0.103| 0.104|
| $T_{ex}$ (K) | 663           | 661 | 668 | 676 | 680 |
| $n_{th}$ (for exhaust gases cooled to 120 °C) | 0.40          | 0.40 | 0.37 | 0.42 | 0.42 |

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18%, Pentane with 17%, and Toluene with 16.5%. It is noted that the best performing fluids in terms of thermal efficiency do not coincide with the best performing ORC engines in terms of power output. Pentane operates with lower thermal efficiency than Butane, although the former generates more power. Also, by comparing the best performing ORC engines for CHP-230 ICE and CHP-2500 ICE, it is observed that for CHP-230 ICE the optimum working fluids are Pentane, Butane, R1233zd and Toluene, whilst for CHP-2500 ICE Toluene has similar performance to Pentane, followed by R1233zd, and Butane.

### 4.3. Exergy analysis

The integrated ICE-ORC CHP system exergy destruction rate is presented in Fig. 19, along with the ORC engine exergy efficiency and the integrated ICE-ORC CHP system exergy efficiency, for CHP-230. In line with the results, the higher power output of the system comes at the cost of higher exergy destruction rate (in absolute terms). However, while the complete system is optimised the exergy efficiency also increases, indicating that we utilise better the available exergy of the fuel. These results are in agreement with those reported in Section 4.1.1 where the electric efficiency of the ICE increases from Cases C1 to C5, while the ORC thermal efficiency also increases when the integrated
ICE-ORC CHP system is optimised (Section 4.1.2). The maximum integrated system exergy efficiency is recorded for Pentane (47%), followed by Butane with 46%, and R1233zd with 45%, in Case C5.

The exergy efficiency of the complete system increases while moving from Case C1 to C5, for CHP-2500 as well (Fig. 20). It is noted that the complete system exergy efficiency reaches 54%, which is higher than the one achieved for CHP-230. It is highlighted that the absolute exergy destruction rate for this engine also drops while the system is optimised for minimum SFC. This signifies that to maximise the power output of the engine does not mean that the utilisation of the available useful work is also maximised.

Exergy destruction breakdowns amongst the various system components are shown in Figs. 21 and 22 for pentane and R1233zd. As expected, the majority of the exergy loss occurs, as an inevitable consequence of the combustion process, in the ICE. Depending on the case, this corresponds to 83–91% of the total destruction.

Fig. 14. SFC and SFC$_{abs}$ for the different cases investigated based on CHP-2500 ICE, with and without a bottoming ORC engine.

Fig. 15. Optimum ORC engine working fluid flow rate and pressure ratio for the CHP-2500 ICE.

Fig. 16. ORC engine heat input load breakdown for all working fluids and for all investigated cases (Cases C1-C5 from left to right).
Exergy destruction is higher than pentane, corresponding up to 51%. The latter observation is due to the larger temperature differences between the fluids entering and exiting the evaporator of the R1233zd ORC engine, when coupled to the CHP-230 engine. The results presented in Figs. 21 and 22 are representative of the findings for the other working fluids investigated. Therefore, the evaporator heat exchanger geometry and design should be very carefully selected to maximise its effectiveness, while reducing the backpressure effect on the exhaust gas side. The expander is the second biggest source of irreversibility, therefore further research is required on expanders’ design suitable for high pressure/high temperature applications. These findings are aligned with other studies in the literature, such as Ref. [43].

5. Summary and conclusions

An integrated ICE-ORC CHP whole-system optimisation framework has been developed for high-performance, advanced stationary cogeneration systems. This work differs from earlier efforts in that it accounts explicitly for the design parameters and operational conditions of both the ICE and ORC engines to optimise overall system performance, by capturing trade-offs between optimum ICE and ORC design and operation. This holistic approach allows us to identify, amongst other, the
optimum ICE valve size, lift, and timing, along with the optimum ORC working fluid and flow rate, evaporation/condensation pressures and superheating degree, so that together the two engines have the maximum total power output, or minimum total specific fuel consumption. The framework includes a validated dynamic ICE model, and a steady-state model of subcritical recuperative ORC engines. Both naturally aspirated and turbocharged ICEs are considered, of two different sizes/capacities.

Based on the application of this framework, an optimised integrated ICE-ORC CHP system (Case C3) is proposed that achieves up to 30% higher power output than a nominal (Case C1) ICE design, with the ORC engine contributing between 4 and 15% of the total power output, depending on the case and fluid. We note that the percentage improvement of the whole-system power output is higher for the CHP-230 ICE engine than for the CHP-2500 ICE, indicating that heat recovery from ICEs may be more promising for medium-sized engines. The ORC engine contribution to the total power output is higher in the case of the smaller ICE, improving the system fuel efficiency. The integrated ICE-ORC CHP efficiency (Case C3) increases by up to 21% in comparison to a stand-alone ICE at nominal operation, and by up to 11% in comparison to Case C1, when an ORC engine is optimised based on the nominal ICE design conditions. The ICE efficiency also increases by up to 7.5%

**Fig. 20.** ICE-ORC CHP exergy destruction, and exergy efficiency for CHP-2500 ICE and all investigated cases (Cases C1-C5 left to right).

**Fig. 21.** Exergy destruction breakdown: (a) integrated ICE-ORC CHP system with pentane, (b) ORC engine with pentane, (c) integrated ICE-ORC CHP system with R1233zd, (d) ORC engine with R1233zd, for CHP-230 ICE. Results presented here refer to Case C3, but are representative of all investigated cases (Cases C1-C5).
(Case C5) in comparison to the nominal ICE. The importance of the integrated ICE-ORC CHP system is further highlighted when comparing these figures to other studies in the literature. For example, efficiency improvements of no more than 6% were reported for ICEs within ICE-ORC CHP systems in Ref. [76], with other studies even reporting a decrease in ICE efficiency by 2–3% [47], with total (ICE + ORC) system efficiency improvements not higher than 8% [46,47,77].

When only the ICE power output is maximised (Case C2), the ICE optimum operating conditions result in lower exhaust-gas temperatures exiting the ICE, which maximises the power delivered to the ICE shaft. In contrast, when the complete ICE-ORC CHP system is optimised for maximum power output (Case C3) the exhaust-gas leaving the ICE is at a higher temperature and pressure, so as to promote ORC generation, but at the cost of slightly reduced ICE power output. This highlights the importance of designing the ORC evaporator to impose minimal pressure drop on the exhaust gas stream and to avoid giving rise to antagonistic effects that deteriorate the ICE performance. It is also observed that when the optimisation objective is to maximise the power output, the fuel consumption of the system increases compared to the nominal ICE design (Case C1). Therefore, the higher power output comes at a cost of a higher fuel consumption. On the contrary, when the complete system is optimised specifically for minimum fuel consumption (Case C5) this can drop by up to 17%, also allowing for an important reduction to the annual fuel-purchase costs of the system.

The type of ICE aspiration affects the exhaust-gases temperature entering the ORC evaporator, and thus the ORC engine design and efficiency. The ORC efficiency is higher when coupled to CHP-230 ICE (natural aspiration) where the exhaust-gas temperature reaches 1,000 K. The best performing ORC with Pentane has a thermal efficiency of 25%, which drops to 20% for CHP-2500 ICE (turbocharged engine) where the exhaust-gas temperature after the turbine does not exceed 700 K. The higher temperature difference observed in the ORC evaporator designed for CHP-230 ICE also results in lower heat transfer area requirements from the heat exchanger. In turn, the pressure drop experienced in this smaller heat exchanger by both the exhaust-gas and the working-fluid streams will be lower. The heat exchanger will also be less expensive, reducing the ORC engine capital cost. The ORC pressure ratios observed between the two engines also differ. In the CHP-230 ICE case (high temperature exhaust gases), the pressure ratio ranges between 7 and 55, whilst in the case of the CHP-2500 ICE it does not exceed 44. The difference in pressure ratios highlight the need for further developments towards suitable expander technologies for this application, while also suggesting that multi-stage machines may prove to be a more suitable solution for the very high pressure ratio cases. In terms of working fluid selection, this study provides strong evidence that new hydrofluoroolefins such as R1233zd, are very promising for such high-temperature ORC applications, while being non-toxic, having a low flammability in comparison to hydrocarbons, and a low global warming potential. So, it is concluded that these fluids should be considered by the ORC vendors in the design of next generation ORC engines.

In closing, when designing ORC engines as a bottoming cycle for stationary cogeneration/CHP applications, the impact of the optimised ICE design, operation and performance on the overall design is very important, also affecting the ORC engine design and power output. This study has proven that these systems should be addressed in an integrated manner. Overall the methodology developed here can be used by: (i) ICE manufacturers to provide guidelines on suitable ICE designs for waste heat recovery projects with ORC technology; and (ii) ORC manufacturers to inform working-fluid selection, component (e.g.
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