Performance of working-fluid mixtures in an ORC-CHP system for different heat demand segments

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Abstract:

Organic Rankine cycle (ORC) power systems are being increasingly deployed for waste heat recovery and conversion to power in several industrial settings. In the present paper, we investigate the use of working-fluid mixtures in ORC systems operating in combined heat and power mode (ORC-CHP) with shaft power provided by the expander/turbine and heating provided by the cooling-water exiting the condenser. The waste-heat source is a flue gas stream from a refinery boiler with a mass flow rate of 560 kg/s and an inlet temperature of 330 °C. When using working fluids comprising normal alkanes, refrigerants and their subsequent mixtures, the ORC-CHP system is demonstrated as being capable of delivering over 20 MW of net shaft power and up to 15 MW of heating, leading to a fuel energy savings ratio (FESR) in excess of 20%. Single-component working fluids such as pentane appear optimal at low hot-water supply temperatures, and fluid mixtures become optimal at higher temperatures, with the combination of octane and pentane giving an ORC-CHP system design with the highest efficiency. The influence of heat demand intensity on the global system conversion efficiency and optimal working fluid selection is also explored.

Keywords:

Organic Rankine Cycle, Heating, Combined Heat and Power, Heat Recovery, Working-Fluid Mixtures.

1. Introduction

The rising global energy-demand is a major driver for increasing the energy efficiency of existing energy processes. The utilization of wasted heat from such processes (especially at temperatures up to 300-400 °C) is a promising way to increase overall fuel-use efficiency. At the same time, the use of alternative sources of low-/medium-grade heat, such as geothermal or solar heat, can play a key role in decreasing our dependence on fossil fuels. Low-/medium grade heat can be converted into useful power such as electricity, or recovered to provide heating for buildings, or a combination of the two. A noteworthy feature of lower-grade (temperature) heat is that it is available from numerous sources in the industrial, tertiary, residential and transportation sectors [1]; however, its temporal profile is strongly influenced by the specific process that generates this heat source for CHP conversion.

The Carnot efficiency gives the maximum possible theoretical efficiency attainable by a closed thermodynamic power-cycle for a given temperature difference between the heat source and heat sink to the cycle. The Carnot efficiencies of low-grade heat conversion technologies are, therefore, inherently low compared to conventional (e.g. fossil fuel or nuclear) power plants, which are very often based on Rankine cycles. However, the overall efficiency of industrial processes can be enhanced by recovering and reusing low-grade heat. Furthermore, once the heat recovery infrastructure is in place and there are no further significant operational cost incurred in continuously generating the energy input (waste-heat in this case) to relevant systems, the baseload waste-heat recovery operation to maximize the power output is often the most profitable strategy [2]. However, the possibility to serve a low temperature heat demand (heat sink to the cycle) with the discharged cogenerated heat could further increase the global conversion efficiency of the conversion system (i.e. the sum of thermal and electric efficiencies). However, the increase of thermal efficiency (i.e. of the temperature of heat sink) unavoidably reduces the power output, which is justified when the low temperature heat demand profile well matches the high temperature waste heat source one.

A number of technologies exist for the conversion of lower-grade heat to useful power. The Kalina cycle, for example, uses a mixture of ammonia and water, whereas the organic Rankine cycle (ORC), employs different organic working fluids and their mixtures, such as hydrocarbons, refrigerants, or siloxanes [3-6]. A significant effort has been placed in recent years on the development and improvement of ORC power systems in different applications [7-12]. In particular, the utilization of a heat source to provide combined heat and power (CHP) is of interest, as the overall usefulness and 'total efficiency' of this system can be very high [13-17]. In such CHP systems, part of the thermal energy available in the heat source is used to provide heating for buildings and/or industrial processes, while another part is converted into useful work and power. A number of studies have focused on the optimal selection of heat to electricity ratio in such ORC-CHP systems in order to maximize global energy performance, on the basis of the heat demand profile, the quality of thermal energy required by the load and the influence of discharged heat sink on the CHP output power [18-19], while other studies have focused on the part load efficiency of different CHP configurations and best operational strategies selection on the basis of the electricity and heating demand profiles [20,21].

In this paper, we investigate the utilization of an industrial waste-heat stream for power generation by an ORC system and the simultaneous provision of variable-load low-temperature heating for blocks of buildings (residential/tertiary) or to cover more constant heat-demand profiles (industrial processes) by using the ORC condenser cooling-stream in a CHP application. The exergy of the resulting ORC cooling / CHP heating stream (exiting the condenser) is calculated as a quantifiable measure of the 'quality' of this stream. Based on the power output from the ORC system and the exergy input from the heat source, an overall exergy efficiency of the ORC-CHP system can be calculated. This efficiency measure is optimized for different outlet temperatures of the CHP heating stream. A higher outlet temperature increases the exergy of the generated heating stream but decreases the power output of the expander. Conversely, a low outlet temperature allows for a high power output, but a low heating stream exergy and hence a low potential to heat buildings or match other industrial thermal energy demand(s). These trade-offs are investigated with a mathematical model of the CHP system, with *n*-alkane and refrigerant working fluids and their binary mixtures. Moreover, the waste-heat supply and the low-temperature heating demand profiles are often not well matched. In particular, industrial waste-heat supply is strictly related to the specific industrial process and can relatively constant over the time, while heating demand profiles in buildings are strongly affected by typical daily and seasonal variations. This means that, without a proper thermal storage system, cogenerated heat from the ORC-CHP can be discharged over large periods of the year. This is particularly true when waste heat availability makes profitable a baseload CHP operation to maximize the power output, instead of thermal load following operations. The optimal working fluid for a given low-temperature heat demand resulting from the optimization procedure may not be the optimal one if the heating demand is affected by high temporal variations and does not match the CHP output profile. For this reason, the influence of the heat demand profile on optimal working fluid selection and global CHP conversion efficiency is explored.

2. Methodology

2.1. Thermodynamic ORC model

A schematic diagram of the ORC engine is displayed in Figure 1. The corresponding temperature – specific entropy (*T*–*s*) diagram is shown in Figure 2 using the same states around the cycle.

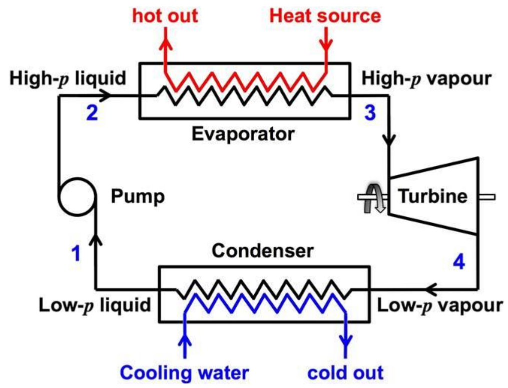

Figure 1: Schematic of the non-regenerative ORC engine.

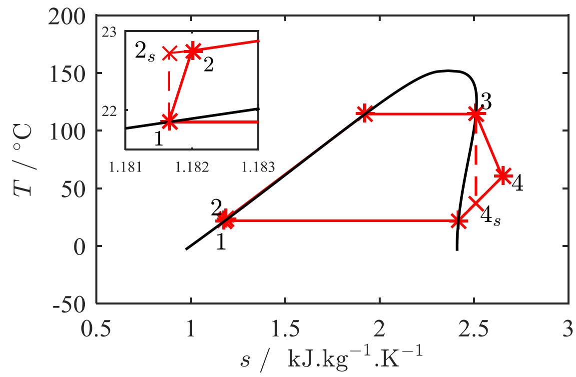

Figure 2: T–s diagram for a dry single-component working fluid.

The required power of the pump is modelled by using the following equation:

$$\dot{W}_{\mathrm{pump}} = \dot{m}_{\mathrm{wf}}(h_2 - h_1) = \dot{m}_{\mathrm{wf}}(h_{2,s} - h_1)/\eta_{\mathrm{s,pump}}, \quad (1)$$

where $\dot{m}_{\mathrm{wf}}$ is the mass flow rate of the working fluid, h is the enthalpy and $\eta_{\mathrm{s,pump}}$ is the isentropic efficiency of the pump, which is set to 75%.

In the evaporator, the minimum pinch temperature-difference is set to 10 °C. It is assumed that there are no heat losses in the heat exchanger. The temperature of the working fluid at State 3 can vary between the dew point temperature at the evaporation pressure (no superheating) and 320 °C, which is the maximum temperature when the pinch point is at the heat source inlet, corresponding to the maximum degree of superheating d_{SH}:

$$d_{\mathrm{SH}} = \frac{T_3 - T_{\mathrm{dew}}(P_{\mathrm{evap}})}{320 - T_{\mathrm{dew}}(P_{\mathrm{evap}})}, \quad (2)$$

Assuming the heat-addition process to be isobaric, the rate of heat input from the heat source is:

$$\dot{Q}_{\mathrm{in}} = \dot{m}_{\mathrm{wf}}(h_3 - h_2) = \dot{m}_{\mathrm{hs}} c_{\mathrm{p,hs}}(T_{\mathrm{hs,in}} - T_{\mathrm{hs,out}}), \quad (3)$$

where $c_{\mathrm{p,hs}}$ is the heat capacity of the heat source stream fluid.

The exergy input into the cycle can be calculated by:

$$\dot{E}_{\mathrm{in}} = \dot{m}_{\mathrm{hs}}[c_{\mathrm{p,hs}}(T_{\mathrm{hs,in}} - T_{\mathrm{hs,out}}) - T_0(s_{\mathrm{hs,in}} - s_{\mathrm{hs,out}})], \quad (4)$$

where s is the specific entropy and T_0 the reference 'dead-state' temperature, which is set to ambient temperature in this work (20 °C).

The power that can be extracted from the cycle in the expander is given by:

$$\dot{W}_{\text{exp}} = \dot{m}_{\text{wf}}(h_3 - h_4) = \eta_{\text{s,exp}}\dot{m}_{\text{wf}}(h_3 - h_{4\text{s}})\,, \tag{5}$$

with the isentropic efficiency of the expander $\eta_{\text{s,exp}}$ set to 75%.

Heat from the cycle is rejected in the condenser to the heat sink. The heat sink stream is utilized to provide heating for low temperature heat demand. The heat rejected from the cycle is:

$$\dot{Q}_{\text{out}} = \dot{m}_{\text{wf}}(h_4 - h_1) = \dot{m}_{\text{cs}}c_{p,\text{cs}}\left(T_{\text{cs,out}} - T_{\text{cs,in}}\right). \tag{6}$$

Similarly to the evaporator, the pinch temperature-difference in the condenser is set to 10 °C. The inlet temperature of the heat sink $T_{\text{cs,in}}$ is set to 20 °C, while the outlet temperature $T_{\text{cs,out}}$ can vary between 30 °C and 90 °C. This temperature is equal to the supply temperature of the heat stream that is used to serve the heat demand. The return temperature of this stream is set to 30 °C, so that for $T_{\text{cs,out}} = 30$ °C no heating is provided and for $T_{\text{cs,out}}$ = 90 °C the maximum heating is provided.

To calculate the quality of the heat rejected from the ORC engine at the sink, i.e. the exergy available for heating, the exergy flow-rate of this stream is calculated from:

$$\dot{E}_{\text{SH}} = \dot{m}_{\text{cs}}\left[c_{\text{p,cs}}\left(T_{\text{cs,out}} - T_{\text{cs,in}}\right) - T_0\left(s_{\text{cs,out}} - s_{\text{cs,in}}\right)\right]. \tag{7}$$

Along with the net power output and the heating exergy, the third performance indicator considered in this paper is the exergy efficiency:

$$\eta_{\text{ex}} = \frac{\dot{W}_{\text{exp}} + \dot{E}_{\text{SH}} - \dot{W}_{\text{pump}}}{\dot{E}_{\text{in}}}\,. \tag{8}$$

In this paper the heat source is considered to be a flue gas from an industrial process with a mass flow rate of 560 kg/s and the temperature at the inlet of 330 °C.

The above-mentioned performance indicators go beyond the traditional ‘energy utilization factor’ EUF (or total/overall efficiency) and ‘fuel-energy savings ratio’ FESR of CHP systems, nevertheless, salient EUF and FESR values are also mentioned.

2.2. Optimization algorithm

An optimization algorithm is employed to find the maximum exergy-efficiency of the aforementioned CHP system, which necessitates an objective function and constraints to be defined. The first constraint (Equation 10) ensures that the pinch conditions in the evaporator and condenser are satisfied. The temperature at the turbine T_4 outlet has to be higher than or equal to the dew point temperature at the condensation pressure (Equation 11) to prevent liquid droplet formation in the expander. This means that working fluid at the turbine outlet is always in the vapour state. For the cycle to be subcritical, the evaporation pressure has to be lower than or equal to the critical pressure (Equation 12). In addition, by definition, the degree of superheating must be between 0 and 1 (Equation 13). Finally, the condensation pressure must be equal to or larger than 1 bar (ambient) to avoid sub-atmospheric pressures in the cycle and expensive solutions to avoid air ingress.

$$\max\{\,\eta_{\text{ex}}\} \tag{9}$$

$$\text{s.t.}\,\Delta T_{\text{pinch}} \geq 10\ ^{\circ}\text{C} \tag{10}$$

$$T_4 \geq T_{\text{dew}}(P_{\text{cond}}) \tag{11}$$

$$P_{\text{evap}} \leq P_{\text{crit}} \tag{12}$$

$$0 \leq d_{\text{sh}} \leq 1 \tag{13}$$

$$P_{\text{cond}} \geq 1\ \text{bar} \tag{14}$$

2.3. ORC working-fluid selection

In this paper, we consider both pure working-fluids and working-fluid mixtures. The pure working-fluids are the refrigerants R245fa and R227ea, and the *n*-alkanes from butane to octane. We also consider binary mixtures of promising pure fluids. While single-component working-fluids evaporate isothermally at isobaric conditions, fluid mixtures exhibit a non-isothermal evaporation ('glide'), which raises the average temperature of heat addition. This can be advantageous for the efficiency of an ORC-CHP system, since the temperature profile of the heat source can be matched by the working fluid, reducing losses in the evaporator. Similarly, the heat rejection process is also non-isothermal for mixtures, which raises the average temperature of heat rejection. This is expected to be detrimental for the cycle efficiency. However, given that in this paper the heat sink is used to provide useful heating, the temperature profile of the heat sink can be better matched by using a working-fluid mixture, which can improve the overall exergy efficiency of the cycle.

2.4. Energy demand modelling and global CHP efficiency

In order to take into account the influence of the heating demand profile on the global energy conversion efficiency of the CHP with different working fluids, the coefficient X_{CHP} is introduced, representing the ratio of equivalent operating hours of the system in the cogenerative configuration (when both heat and power are delivered) vs. the total operating hours over an annual time horizon:

$$X_{\text{CHP}} = \frac{h_{\text{CHP}}}{h_{\text{TOT}}} \tag{15}$$

In case of baseload CHP operation (h_{TOT} of 7,500 hr/year), this coefficient typically ranges between 0.15-0.25 for residential heating demand, 0.20-0.30 for tertiary demand, and 0.40-0.65 for industrial demand; it is affected, case by case, by factors such as climate conditions, building energy efficiency, industrial process operations, etc. The CHP exergy efficiency η_{CHP} over a year can be calculated from:

$$\eta_{\text{CHP}} = \frac{(\dot{W}_{\text{exp}} - \dot{W}_{\text{pump}}) + \dot{E}_{\text{SH}}\, X_{\text{CHP}}}{\dot{E}_{\text{in}}}\ . \tag{16}$$

3. Results and discussion

In this section, we present results of the ORC-CHP system with the various pure working-fluids. The important indices of comparison are the heating demand exergy, the net power-output, the overall exergy-efficiency and the CHP-exergy efficiency. The heating demand exergy is the amount of exergy available from the ORC cooling water to provide low temperature heating to the surrounding buildings or other demand segments. The net power output is the difference between the gross power output from the ORC expander/turbine and the power required in pumping the working fluid around the various components of system. The overall exergy efficiency was thus defined as in Equation 9.

3.1 Performance of pure working-fluids

The ORC-CHP model was simulated with different working fluids, by maximizing the overall system exergy-efficiency at different hot-water supply temperatures and assuming $X_{\mathrm{CHP}} = 1$. In Figure 3, the maximum overall exergy-efficiency and the corresponding net power-output and heating demand exergy of the system are presented as functions of the hot-water supply temperature, which is varied from 30 °C to 90 °C. The optimal design variables are also presented in Table 1 for the cases of hot-water supply at temperatures of 30 °C and 90 °C.

For all the working fluids considered here, the net power-output generally decreases with the hot-water supply temperature (this is also the temperature of the cooling water exiting the ORC engine). The ORC inlet cooling water is provided at 20 °C, thus higher exit temperatures imply larger temperature gradients across the condenser. This has the tendency of increasing the working fluid condensation temperature and pressure thereby reducing the power output from the ORC expander. An evidence of this is found in Table 1 where the condensation pressures of butane, pentane and R245fa are more than doubled when the supply temperature is increased from 30 °C to 90 °C.

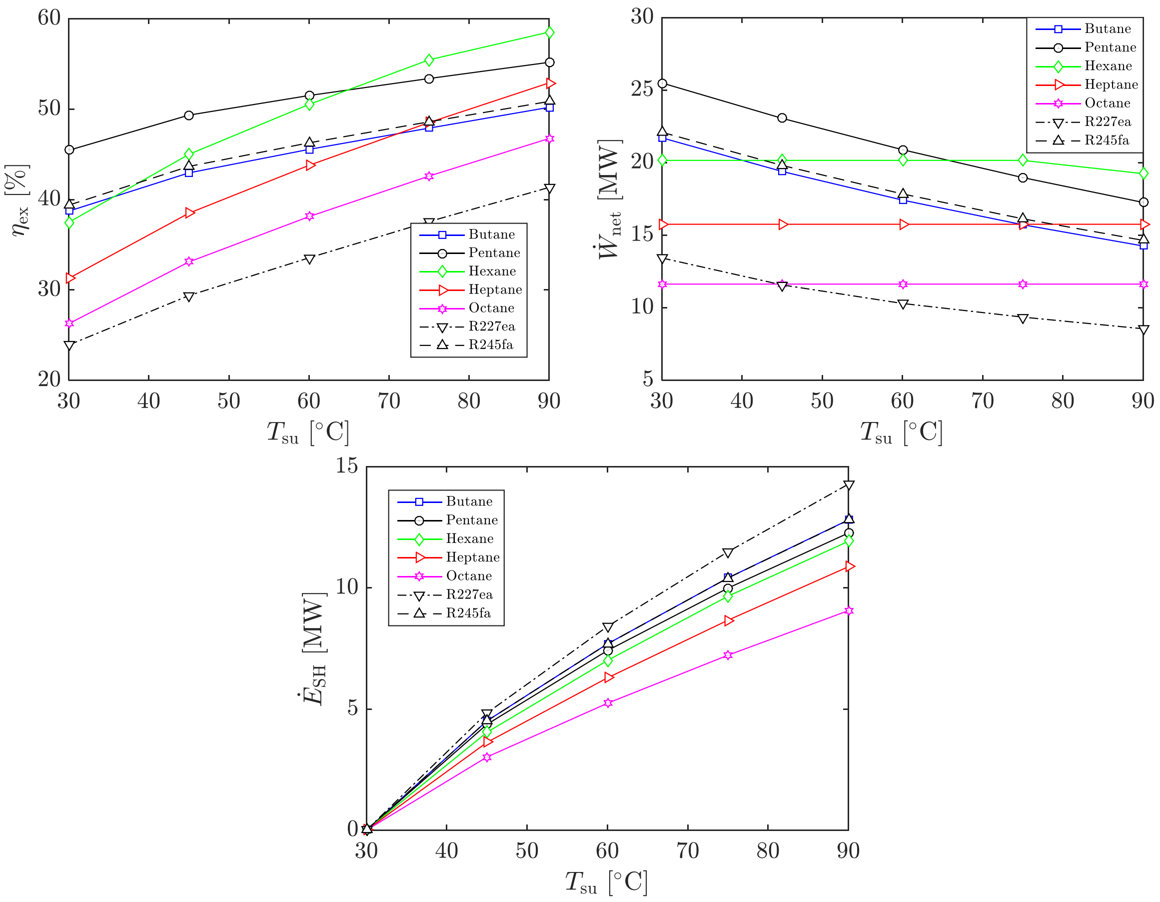

Figure 1: The overall exergy efficiency (top left), net power output (top right) and heating-demand exergy (bottom centre) of the ORC-CHP unit with selected single-component working fluids as functions of the hot water supply temperature.

Table 1: Optimal operating conditions for the ORC-CHP system

Fluid	30 °C supply temperature				90 °C supply temperature			
	P_{evap} [bar]	P_{cond} [bar]	d_{sh}	$\dot{m}_{\mathrm{wf}}$ [kg/s]	P_{evap} [bar]	P_{cond} [bar]	$\dot{m}_{\mathrm{wf}}$ [kg/s]	d_{sh}
Butane	36.1	3.41	0.317	245	36.1	6.61	178	0.745
Pentane	32.0	1.03	0.192	239	32.0	2.65	197	0.495
Hexane	28.8	1.00	0.027	238	28.8	1.07	204	0.264
Heptane	26.0	1.00	0.000	211	26.0	1.00	211	0.000
Octane	23.7	1.00	0.000	170	23.7	1.00	170	0.000
R227ea	27.8	6.19	0.288	750	27.8	8.98	404	1.000
R245fa	34.7	2.21	0.365	462	34.7	4.86	345	0.812

For working fluids such as hexane, heptane and octane, the ORC net power-output is seen (from Figure 3) to remain constant irrespective of the hot-water supply temperature. This is because of the condensation taking place at a constant pressure of 1 bar as a result of the constraint in Equation 14. This constraint ensures that the cycle operates above atmospheric pressure, thus eliminating the need for expensive vacuum expanders and condensers. At this condensation pressure of 1 bar, the working-fluid temperature is generally greater than the cooling-water exit temperature. For example, the saturation temperatures of heptane and octane at 1 bar are 97.9 °C and 125 °C, respectively. Thus, increasing the cooling-water exit temperature does not have an effect on the working-fluid condensation temperature and pressure.

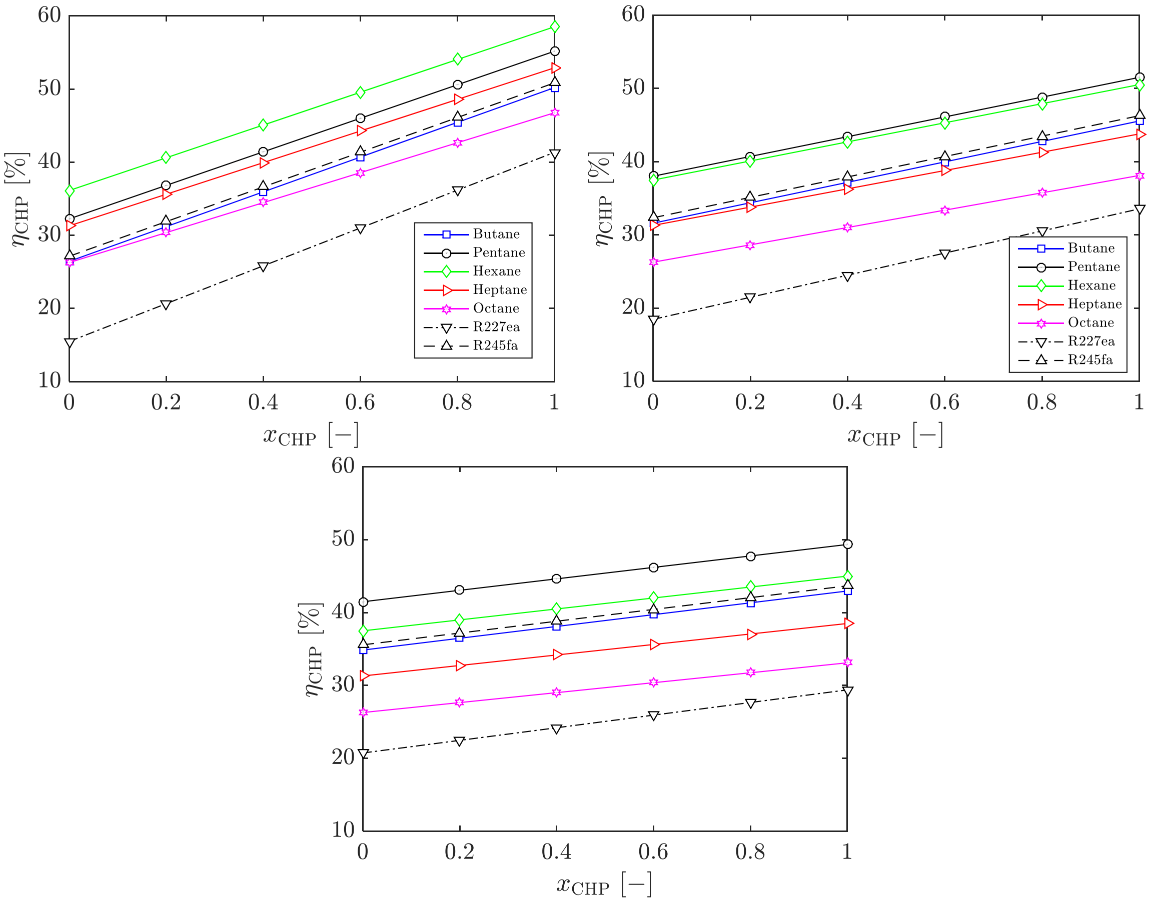

Figure 4. CHP exergy-efficiency of the ORC-CHP unit with selected single-component working fluids as functions of the heating demand intensity (coefficient X_{CHP} of Eqn. 15) and hot water supply temperature respectively of 90°C (top left), 60°C (top right) and 45°C (bottom). The navy and red bars represent the range of X_{CHP} for residential and industrial heating demand respectively

Although the power output is relatively insensitive to the cooling-water exit temperature (for hexane, heptane, octane), the heating-demand exergy increases with the exit temperature due the higher exergy made available to the heat-sink stream at higher temperatures. Similarly, for the other working fluids, the heating-demand exergy generally increases with the heat-supply temperature. Thus, the heating-demand exergy, which increases with the heat-supply temperature, appears to be in competition with the ORC power output, which decreases with the supply temperature.

This trade-off is further explored in Figure 4 that shows the CHP exergy efficiency over a year as a function of the heating demand profile for selected working fluids and low temperature heat supply. The navy and red bars represent the typical ranges of the CHP coefficient X_{CHP} in case of residential and industrial energy demand. As expected, the global CHP exergy efficiency η_{CHP} decreases at lower heat demand intensities. This effect is stronger at higher heat sink supply temperature.

Table 1 summarizes the effect of the operating conditions on the optimal exergy efficiency of the ORC-CHP system. Due to the high heat source temperature (which is higher than the critical temperatures of the working fluids) and the sub-critical nature of the ORC-CHP architecture, the optimal evaporation pressure is limited by the critical pressure such that it remains constant irrespective of the hot-water supply temperature. The optimal condensation pressure, however, varies with the heat-supply temperature as noted earlier, and only with some fluids (heptane, octane) is it limited by the atmospheric pressure. The optimal mass flow-rate is that which maintains the pinch temperature-difference in the evaporator at the minimum specified value of 10 °C (Equation 10). The optimal superheating degree increases with the heat-supply temperature except for very dry fluids, specifically heptane and octane, for which it is negligible. This is a peculiar feature of these dry fluids, where optimal performance is attained by direct expansion from the saturated vapour line, without superheating.

In terms of the system's overall exergy-efficiency, the ORC-CHP system is generally more efficient for designs with higher hot-water supply temperatures, notwithstanding the fact that the system delivers lower power-outputs at higher heat-supply temperatures. This is possible because of the higher space-heating potential available at higher heat-supply temperatures. From Figure 3, it is clear that, while the net power-output decreases by a maximum of about 8 MW (between hot-water supply temperatures of 30 °C and 90 °C), the space-heating exergy increases by at least 9 MW over the same temperature range. Thus, the increase in space-heating exergy with heat-supply temperature is steeper than the decrease in net power-output, leading to an overall increase in the ORC-CHP system's exergy efficiency with the hot-water supply temperature.

From Figure 3 we can draw some conclusions concerning the performance of different working fluids in the ORC-CHP system. The refrigerants have the highest space-heating exergy followed by the alkanes; the space-heating exergy decreases progressively as the alkanes get heavier (from butane to octane). This is evident regardless of the hot-water supply temperature. While R227ea provides the largest space-heating exergy across all supply temperatures, it leads to an ORC-CHP design with the lowest exergy-efficiency due to its comparatively low power-output. Amongst the alkanes, octane provides the lowest space-heating exergy and power output and thus leads to the least efficient CHP-ORC design.

Working fluids such as the alkanes butane, pentane, and hexane lead to designs with the highest power outputs, sometimes in excess of 20 MW. They also have high values of space heating exergy (although slightly lower than that of R227ea) and are thus seen to exhibit the highest overall system exergy efficiency. In particular, ORC-CHP systems with pentane as working fluid have the highest overall exergy efficiency and net power output, up till about 65 °C, after which systems with hexane as the working fluid are the most efficient and most powerful.

3.2. Working-fluid mixture performance

Having considered using single-component organic fluids for harnessing a low temperature heat source in an ORC-CHP system, it is interesting to considering what opportunities mixtures of such fluids offer. In ORC systems, working-fluid mixtures, due to their temperature glide during isobaric evaporation/condensation provide a better thermal match to the heat source/sink streams thereby reducing systems' exergy losses and improving overall performance. It should be noted that working-fluid mixtures could lead to deterioration in heat transfer performance especially during evaporation and condensation. Systems with such working-fluid mixtures may thus eventually require larger heat transfer equipment compared to those with single-component working fluids.

Here, however, we limit the analyses to the thermodynamic effect(s) of such working-fluid mixtures on the system. Hence, we simulate the ORC-CHP model with mixtures of working fluids from the earlier-presented alkanes and refrigerants. Amongst the pure fluids, pentane and hexane lead to cycle designs with the highest overall exergy efficiencies and net power outputs. Thus, we start the investigation of working-fluid mixtures in the system with a mixture of pentane and hexane. Other alkane mixtures and the refrigerant mixtures are considered and evaluated later in this section.

Presented in Figure 5 are the performance indices of the ORC-CHP system with a working-fluid mixture of hexane and pentane; mixtures here and hereafter are defined on a mass fraction basis. As with the single-component fluids, the ORC net power decreases with increasing hot water supply temperature while the heating-demand exergy and the overall exergy efficiency both increase with the hot water supply temperature. The use of working-fluid mixtures seems to have a negligible effect on the heating demand exergy as the working-fluid mixtures only allow a small margin of improvement over the pure fluids (hexane, pentane). In addition, the fuel energy savings ratio (FESR) for the ORC-CHP system is presented, demonstrating the economic benefits of such a system in comparison to a conventional power station and/or a conventional boiler. The ORC-CHP system is seen to perform favourably in comparison to the conventional systems with FESR values up to 30%. The working-fluid mixtures do however result in lower fuel savings than the pure working fluids in the ORC-CHP system.

The working-fluid mixtures do however present a considerable improvement to the performance of the ORC-CHP system in terms of its net power output and exergy efficiency as illustrated in Figure 5. In particular, a working-fluid mixture is usually seen to provide a higher system exergy efficiency or power output of the two constituent pure fluids, at the varying values of the hot water supply temperature. This is generally favoured by the non-isothermal condensation profile of the mixtures which offers a better thermal match with the heat sink and thus, allowing lower condensation pressures (and higher evaporation pressures) possible. This then leads to the higher power output exhibited by the systems with working-fluid mixtures.

Furthermore, the better thermal match between the working-fluid mixture and the heat source/sink (compared to pure working fluids) also minimizes the average temperature difference between the working fluid and the heat source/sink, offering a reduction in the exergy destruction in the evaporator/condenser and thereby leading to higher overall exergy efficiency. The optimal composition that maximizes both the power output and the net exergy efficiency however varies with the hot water supply temperature (and the temperature gradient of the cooling water stream). At the lowest hot water supply temperature of 30 °C, pentane (i.e. mass fraction of unity) is the optimal working fluid and at the highest temperature, hexane (i.e. mass fraction of zero) is optimal. At other supply temperatures, the optimal pentane composition varies progressively between 1 and 0, with a value of 0.5 for a hot water supply temperature of 60 °C, further illustrating the crossover in optimal single-component working fluid from pentane to hexane as seen in Figure 3.

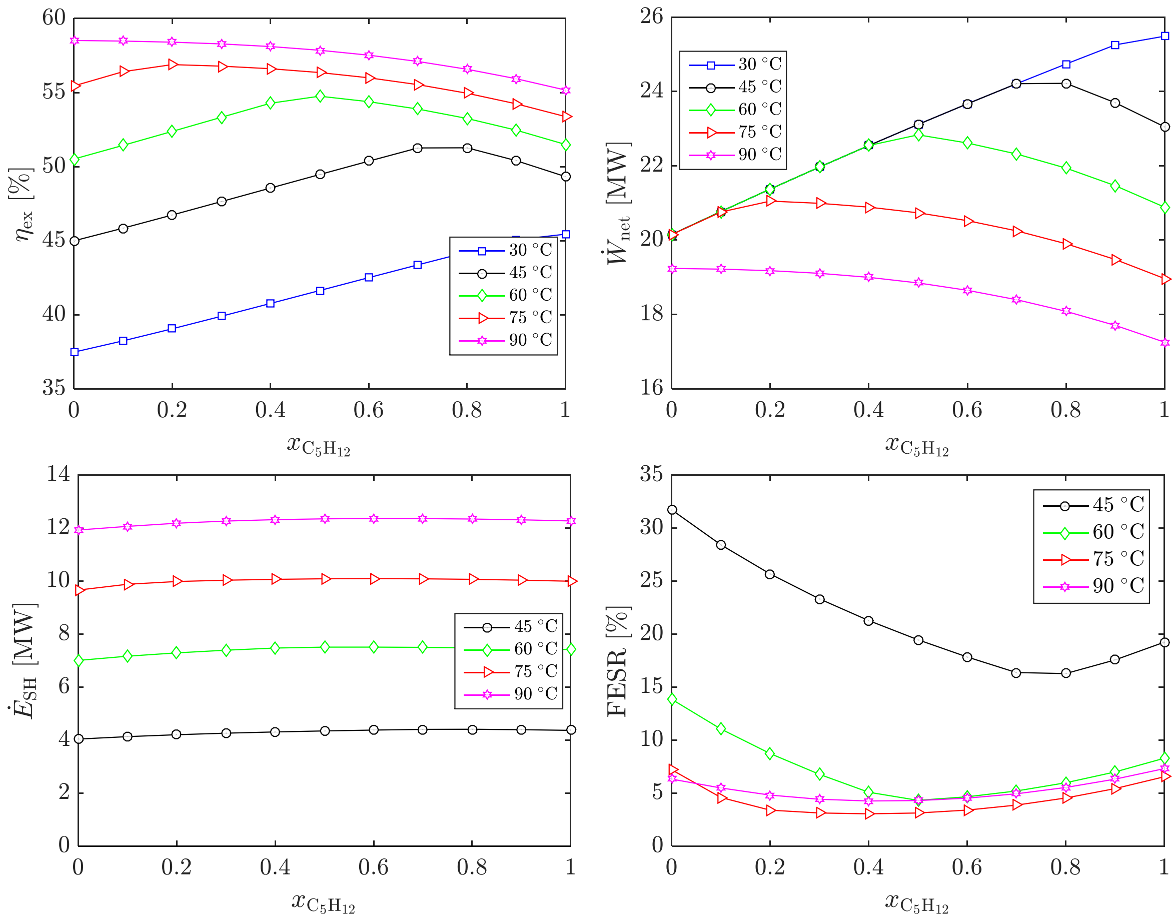

Figure 5: The overall exergy efficiency (top left), net power output (top right), space heating exergy (bottom left) and fuel energy savings ratio (bottom right) of the ORC-CHP system with hexane + pentane working-fluid mixtures as functions of the hot water supply temperature.

Other working fluid mixtures were also investigated for use in the ORC-CHP system; these results are presented in Figure 6 further illustrating the performance improvements presented by working-fluid mixtures over the pure fluids in ORC-CHP systems. However, it is not in all the cases that the working-fluid mixtures perform better than their constituent pure fluids. Cycle performance with R227ea +R245fa mixtures and butane + pentane mixtures are seen to be no better than those with pure R245fa and pure pentane respectively; the overall exergy only varies linearly between the constituent pure fluids and are thus excluded from Figure 6. In addition, for most of the mixtures at a hot water supply temperature of 30 °C, the exergy efficiency varies linearly between the constituent pure fluids with no mixing ratio posing a better alternative to the pure fluids. Only the working-fluid mixture of butane + hexane gives a better performance over the pure fluids, with a mixture having a hexane mass fraction of 0.8 maximizing the overall exergy efficiency.

As the hot-water supply temperature is increased, more mixtures deliver better a performance than their constituent pure fluids. At a hot water supply temperature of 60 °C, working-fluid mixtures of pentane perform better than the constituent pure fluids. This is facilitated by the cooling water stream temperature profile now being better matched to the working fluid condensation temperature profile. Thus, the working-fluid mixtures of pentane result in slightly superior performance when compared to the pure fluids. In particular, the working-fluid mixtures 50% hexane + 50% pentane, 20% heptane + 80% pentane and 20% octane + 80% pentane, maximize the overall exergy efficiency and perform better than their constituents (pure pentane, hexane, heptane and octane).

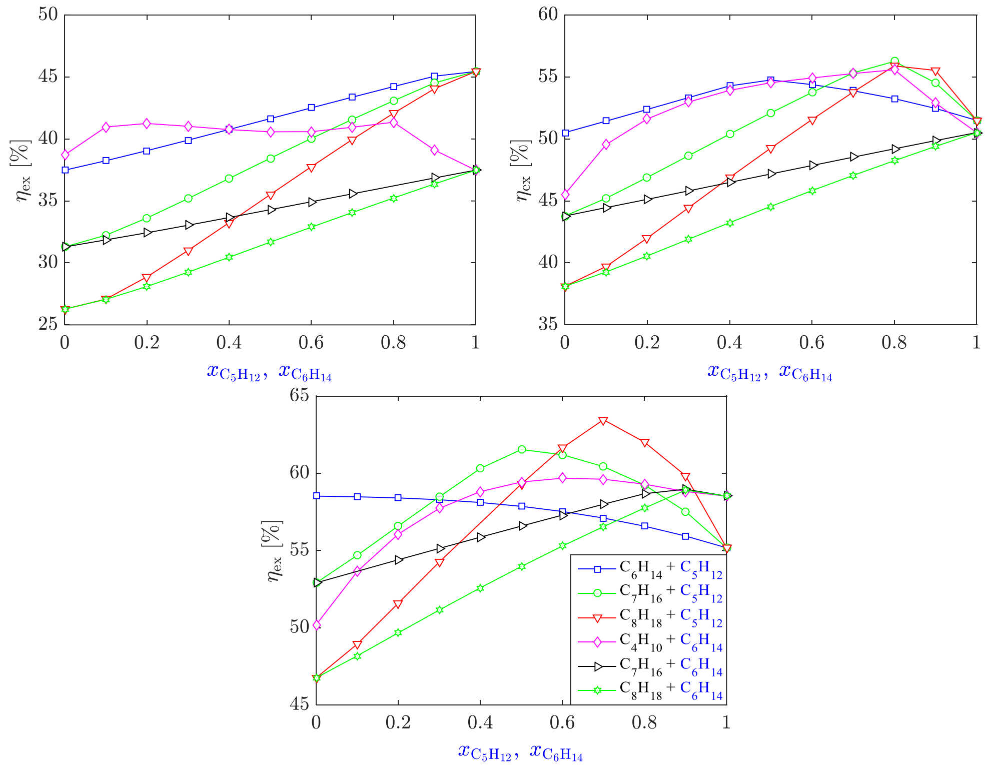

Figure 6: Overall ORC-CHP system exergy efficiency at hot water supply temperatures of 30 °C (top left), 60 °C (top right) and 90 °C (bottom). The horizontal axes are defined in terms of the composition of the second fluid listed in each of the fluid combinations, i.e. pentane and hexane respectively.

The performance improvement achieved by employing working-fluid mixtures is even higher at a hot water supply temperature of 90 °C, especially for working-fluid mixtures containing pentane and heptane or octane, as the temperature profile of the cooling stream and the condensing working fluid become more perfectly matched. The working-fluid mixtures of octane + pentane generally have the largest temperature glide and are thus better matched to the heat sink, resulting in a maximum exergy efficiency at a pentane mass fraction of 0.7. The working-fluid mixtures of hexane (with heptane and octane as the second constituent) perform less significantly than those of pentane because they have lower temperature glides and are not as good as a match with the cooling stream as do the working-fluid mixtures with pentane as a constituent. If the low temperature heat demand profile is taken into account and the coefficient X_{CHP} ranges between 0 to 1, the global CHP exergy efficiency values of Figure 7 are obtained, for the different working fluid mixtures and heat sink temperatures of 90 to 60°C.

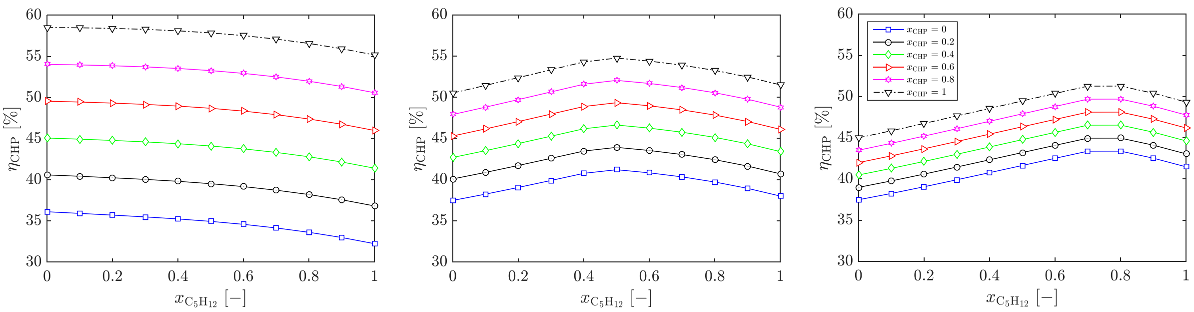

Figure 7. CHP exergy-efficiency of the ORC-CHP unit with various working fluid mixtures as functions of the heating demand intensity (coefficient X_{CHP}) and hot water supply temperature of 90 °C (left), 60 °C (middle) and 45 °C (right). The horizontal axes are defined in terms of the composition of the second fluid listed in each of the fluid combinations, i.e. pentane and hexane respectively.

4. Conclusions

The aim of the present study was to evaluate the performance of working-fluid mixtures in a combined heat and power system based on an organic Rankine cycle engine (ORC-CHP) capable of utilizing heat from industrial waste-heat sources. The heat source considered was a flue-gas stream from a refinery preheater at a flow rate of 1 kg/s and a temperature of 330 °C. Power is generated by the expansion of the working fluid through the expander/turbine. Low temperature cogenerated heating is also provided by utilizing the cooling-water exiting the ORC condenser. The single-component working fluids considered are straight-chained alkanes from butane to octane and the refrigerants R245fa and R227ea; working-fluid mixtures were subsequently derived from these pure substances.

The performance of the system was quantified in terms of the net power-output from the ORC system, the heating-demand exergy available from the cooling water exiting the ORC condenser, the fuel energy savings ratio (FESR) and the overall exergy-efficiency. The global CHP exergy efficiency was quantified as a function of a CHP coefficient representative of the rate of useful heat delivered to the heating demand, i.e. of the energy demand profile (residential vs. industrial). The results indicate that the net power-output and the heating-demand exergy are competing objectives, with the heating demand exergy increasing and the power output reducing with the hot-water supply temperature. It is found, however, that the overall exergy-efficiency increases with the hot-water supply temperature. Amongst the single-component working fluids, pentane and hexane emerged as the optimal fluids, leading to ORC-CHP system designs with the highest net power-outputs and highest overall exergy-efficiencies. In addition, the ORC-CHP system appears to be more economical than conventional power station and gas boilers, due to its high fuel energy saving ratios, which are in excess of 20%.

Although the working-fluid mixtures were shown to have a negligible effect on the heating exergy, they exerted a considerable influence on the power output and overall exergy-efficiency of the system. While the pure fluids (especially pentane) showed best performance at low hot-water supply temperatures due to the small temperature increase of the cooling stream (this providing a better match with the pure fluid condensation profile than that of the mixtures), designs featuring fluid mixtures were shown to deliver higher power outputs and exergy efficiencies at higher hot-water supply temperatures. This is especially true for mixtures with condensation temperature glides matching that of the heat sink as in the case of pentane + octane working-fluid mixtures, which are shown to be optimal at a hot-water supply temperature of 90 °C.

The heat demand segment plays a role in the global CHP efficiency over a long-term operation period (1 year). In fact, even if the ORC-CHP is designed to maximize the overall exergy efficiency at a given low temperature heating demand, the global CHP performance can be significantly decreased when a mismatch between heat sink supply and heating demand determines large amounts of cogenerated heat to be discharged, which is the case in particular for residential energy demand segments.

Further work aims to include energy demand patterns in ORC-CHP working-fluid selection, operational strategy optimization, and to address cost vs performance trade-offs by means of thermo-economic optimization methodologies, minimizing the levelized cost of energy and maximizing profitability.

Nomenclature

c_p	heat capacity at constant P, kJ/(kg K)	h	enthalpy, kJ/kg
$\dot{E}$	exergy flow-rate, MW	P	pressure, bar

$\dot{Q}$	heat, MW
s	entropy, kJ/(kg K)
T	temperature, °C
$\dot{W}$	power, MW

Greek symbols

η	efficiency, %

Subscripts and superscripts

0	reference state
1	ORC condenser outlet/pump inlet
2	ORC pump outlet/evaporator inlet
3	ORC evaporator outlet/expander inlet
4	ORC expander outlet/condenser inlet
con	condensation
crit	critical
dew	dew point
evap	evaporation
ex	exergy
exp	expander
hs	heat source
cs	cooling source (building heating supply)
in	inlet
out	outlet
pump	pump
s	isentropic
th	thermal
wf	working fluid

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