Multi-objective optimization of organic Rankine cycle systems considering their dynamic performance

Pili, Roberto; Jørgensen, Søren Bojer; Haglind, Fredrik

Published in:
Proceedings of the 34th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems 2021

Publication date:
2021

Document Version
Peer reviewed version

Link back to DTU Orbit

Citation (APA):
Multi-objective optimization of organic Rankine cycle systems considering their dynamic performance

Roberto Pilia, Søren Bojer Jørgensen and Fredrik Haglind

Abstract:
The Organic Rankine cycle system is a well-established technology for converting medium/low temperature waste heat into mechanical or electrical power. Inefficiencies in the internal combustion engines for road transportation lead to large amounts of waste heat that are not exploited. Because of the engine load changes during a driving cycle, the mass flow rate and temperature of the heat source fluctuate rapidly over a broad range. This poses high requirements to the control of the organic Rankine cycle unit, in order to prevent the formation of liquid droplets at the turbine inlet and acid gas corrosion in the evaporator if the exhaust gas temperatures are too low, which reduce the system lifetime. In addition, the fluctuations in the heat source degrade the efficiency of the organic Rankine cycle unit, because of part-load operation. Furthermore, the penalty on the transportable vehicle payload caused by the increase in system mass should be considered. This paper presents a novel design method for organic Rankine cycle systems subject to highly fluctuating heat sources, ensuring safe and efficient operation. An integral optimization code developed in MATLAB®/Simulink® combining the design of the thermodynamic cycle, the system evaporator and the control system with a dynamic simulation model is presented. The multi-objective optimization maximizes the organic Rankine cycle net power output over a driving cycle of a heavy-duty truck, while minimizing the mass of the evaporator. The results indicate that, in order to ensure safe operation, the degree of superheating of the working fluid as well as the exhaust gas temperature leaving the evaporator at design conditions should be higher than what classical steady-state thermodynamic analyses suggest. This work provides a unique benchmark for the optimization of organic Rankine cycle systems subject to high fluctuating heat sources that will be of benefit both for academia and industry.

Keywords:
Organic Rankine Cycle; Sustainable Transport; Waste Heat Recovery; Dynamic Modelling; Multi-objective Optimization.

1. Introduction
The transport sector is responsible for almost one quarter of the greenhouse gas emissions in Europe, of which 70% are caused by transport on road [1]. Especially in the last two decades, particular focus has been set on the development of more environmental-friendly vehicle powertrains. For instance, electrical drives have been proposed, in particular for smaller vehicles like passenger cars [2]. The electrification of heavy-duty trucks, which cover approximately one third of the emissions of road transport, appears to be quite challenging, mainly because of the high energy density that the batteries should have and the lack of infrastructure for trucks supplied by overhead lines [3]. For these reasons, internal combustion engines are still a relevant option for the propulsion of heavy-duty vehicles, possibly fuelled by an increasing share of power-to-X fuels from renewable energy sources [4]. A considerable increase in efficiency of internal combustion engines can be achieved by recovering the energy that is wasted in the form of heat [5]. Organic Rankine cycle (ORC) power systems are a cost-effective technology to recover medium/low temperature heat and convert it into mechanical or electrical power. ORC systems have extensively been installed to recover the waste heat of stationary diesel engines, gas turbines and industrial facilities [6]. In the context of heavy-duty vehicles, the ORC technology has faced several technical challenges, which have hindered its commercialization so far [7–9]: i) the mass flow rate and temperature of the waste heat fluctuate rapidly over a vast range, leading to a complex design and inefficiencies during part-load operation; ii) safe operation must be ensured despite the large fluctuations in waste heat; iii) the installation of the ORC unit requires additional space and mass that need to be transported penalizing the vehicle loading capabilities and iv) the backpressure of the internal combustion engine needs to be minimized in order to keep a high engine efficiency. These requirements, combined with the goal of maximum ORC power output, result in a trade-off among multiple objectives. In this way, the design of the ORC unit for heavy-duty trucks consists of a complex inter-disciplinary optimization problem, which involves thermodynamic design, component design, part-load operation, system dynamics and control.
Several authors optimized ORC units affected by a variable heat source combining the design and the off-design operation in the same global optimization routine. The fluctuations of waste heat were handled by using a quasi-steady state approach, where dynamic effects were neglected [10–12]. For example, Lecompte et al. [10] carried out a combined design and off-design optimization of an ORC unit recovering waste heat from a diesel engine by developing a grid of stationary off-design points and optimizing for minimum specific investment costs per unit average power output. The advantages of an integrated design and off-design optimization were also highlighted for waste heat recovery from a billet reheat furnace by Pili et al. [11]. Considerably different results in terms of degree of superheating and evaporation pressure at design point resulted from the integration of design and off-design in the optimization. Guillaume and Lemort [13] performed a techno-economic optimization of an ORC unit recovering heat from a heavy-duty vehicle. The optimization was carried out for several stationary operating points of the engine, which were weighted according to their frequency of occurrence. The average net power output was the objective function to be maximized. Different working fluids, expansion machines and cycle architectures were investigated. Ethanol was found to be the most appropriate fluid in combination with a screw expander and engine exhaust gas recirculation.

Especially for road transport, it can be beneficial to consider the mass and volume of the ORC system already in the design phase, since they strongly affect the economic feasibility of the ORC integration [8]. Some authors included the trade-off among the ORC net power output and the mass and volume of the ORC unit in the design optimization [14,15]. Maclán et al. [14] optimized the ORC unit for maximum net power output and then discarded the options that did not satisfy the volume constraint of 0.2 m$^3$. Water was compared to R245fa as working fluids. Their results suggested that water can achieve a higher power output at the expenses of a larger volume. Pili et al. [15] optimized an ORC unit for truck applications and looked at the impact of working fluid selection on the power-to-weight and power-to-volume ratios. Although acetone and ethanol could provide the largest power output, isobutene showed larger power-to-weight and power-to-volume ratios. Other authors suggested a more holistic approach based on a multi-objective optimization of the ORC unit, where a Pareto front of optimal solutions is attained [16–20]. In particular, Imran et al. [19] performed a multi-objective optimization of an ORC unit recovering waste heat from the exhaust gas of a heavy-duty vehicle driven by a 13L diesel engine. The optimization targets were the unit net power output (to be maximized) and the unit overall cost, mass and volume (to be minimized). Several working fluids were investigated, among which pentane showed the best performance, reaching 8.3 kW on average on a truck driving cycle at a condensation temperature of 40 ºC. The system dynamics were neglected, and a quasi-steady state approach was used for the off-design operation. The results of Ref. [19] also indicated that there is a trade-off between the unit power output and the cost, mass and volume. Holik et al. [20] optimized the design of an ORC unit for heavy-duty diesel trucks by maximizing the net power output and minimizing the total heat exchanger surface area, although for fixed mass flow rate and temperature of the heat source. Their results suggested that ethanol is the best fluid.

Given the large and rapid fluctuations in mass flow rate and temperature of the waste heat from heavy-duty trucks during a driving cycle, the dynamic response of the ORC system significantly affects the ORC performance, and should carefully be considered in order to ensure safe operation and maximum net power output. However, only a few previous works considered the dynamic performance in the ORC design phase [21,22]. Pierobon et al. [21] focused on the waste heat recovery from an offshore oil platform and analysed a posteriori the feasibility of different designs in terms of dynamic performance, excluding the designs that did not satisfy the required specifications. An integrated optimization including the working fluid selection and the dynamic performance was carried out by Tillmanns et al. [22], although the dynamic model of the system (first-order system) and the time behaviour of the heat source (sine wave) were strongly simplified and far from realistic conditions.

The literature review suggest that only limited aspects of the integration of ORC units in heavy-duty vehicles have been included in the design optimization procedure so far. No holistic approach that includes the complex trade-off among multiple objectives, such as maximum power output and minimum component mass, and other essential aspects of this application, such as dynamic performance and control design, has been developed yet. This work presents a novel design approach that includes and integrates the thermodynamic design, the component design, the dynamics and the controller design in the same global optimization loop. In this way, an optimal and feasible design is achieved, ensuring optimal performance and safe operation. The multi-objective optimization problem is solved by using the genetic algorithm and numerical models developed in MATLAB®/Simulink®. This work is of great interest not only for the truck application but also for other applications affected by transient operation, such as road, maritime and aviation transport, offshore platforms, as well as remote areas on land with no access to interconnected power grids.

2. Waste heat recovery with organic Rankine cycle power system

The present work investigates the waste heat recovery of the exhaust gas of a 450-hp 13L turbocharged diesel engine installed in a heavy-duty truck. The exhaust gas data were provided by a truck manufacturer, and refer to a 45-minute trip. The time behaviour of the mass flow rate and temperature of the exhaust gas is shown in
Fig. 1a, whereas the profile duration curve is given in Fig. 1b. It can be seen in Fig. 1a that the mass flow rate of the exhaust gas fluctuates quite rapidly over a broad range between 0.05 kg/s and 0.517 kg/s, whereas the temperature shows a slower trend fluctuating between 270 ºC and 334 ºC. The profile duration curve in Fig. 1b shows that the distribution of the mass flow rate and temperature of the exhaust gas over time is uniform and no particular value occurs more often than others do. The available waste heat rate $\dot{Q}_{av}$ can be determined by the following equation:

$$\dot{Q}_{av} = m_{eg} c_{p,eg} (T_{eg, in} - T_{eg, ref})$$

(1)

where $m_{eg}$ is the mass flow rate, $c_{p,eg}$ the specific heat at constant pressure, $T_{eg, in}$ the inlet temperature and $T_{eg, ref}$ the reference temperature of the exhaust gas (subscript 'eg'). The specific heat at constant pressure $c_{p,eg}$ is a function of the chemical composition of the exhaust gas, which depends in turn on the operating conditions of the truck internal combustion engine and the exhaust gas after-treatment system. Since this information is not available, $c_{p,eg}$ is approximated by using the properties of air. The reference temperature of the exhaust gas $T_{eg, ref}$ is set to the minimum temperature allowable for the exhaust gas to avoid corrosion of the exhaust pipes. A value of $T_{eg, ref} = 100$ ºC is chosen based on literature [23,24]. Given these assumptions, the available waste heat rate $\dot{Q}_{av}$ fluctuates between 10 kW and 114 kW, with an average value of 51.6 kW.

The exhaust gas is taken as the heat source of the ORC unit. The ORC plant consists of a simple ORC system without recuperator, since the recuperator increases costs, mass, volume and complexity of the ORC system. The layout of the ORC unit is shown in Fig. 2. The pump forwards the working fluid from point 0 to point 1, where it is preheated, vaporized and superheated to point 2 by receiving heat from the exhaust gas. The vapour at point 2 is then expanded in a turbine to point 3, where it is condensed back to point 0 while rejecting heat to a cooling medium, which is typically air or water. The ORC system can be controlled by manipulating two variables: i) the mass flow rate of the pump $m_{wrf,p}$ (or correspondingly the pump rotational speed) and ii) the opening of the exhaust gas bypass valve VO.

3. Multi-objective optimization routine

In this section, the novel integrated multi-objective approach for the design of the ORC unit is explained. The optimization routine uses the genetic algorithm available in the Global Optimization Toolbox of MATLAB® [25] to solve the multi-objective optimization problem. The genetic algorithm repeatedly changes the decision variables in order to find a Pareto front that minimizes or maximizes the objective functions. A set of ten decision variables is used for the optimization and summarized in Table 1. The first five variables refer to the thermodynamic design of the ORC system, while the last five variables refer to the design of the ORC evaporator, which is used to assess the mass of this component and, most importantly, to assess the dynamic off-design performance of the ORC unit over the driving cycle. It is worth highlighting that, since the waste heat fluctuates over time, the exhaust gas mass flow rate and temperature at design point need to be included in the set of decision variables, as shown in Table 1. Two objective functions were selected to demonstrate the feasibility of the proposed approach: i) the net power output of the ORC unit, which has to be maximized and ii) the mass of the evaporator, which needs to be minimized. The mass of the ORC evaporator can also be considered as an indicator of its volume as well as of its cost, since the mass is connected to the amount of material needed for the manufacturing of the component.
Table 1. Decision variables for the multi-objective optimization.

<table>
<thead>
<tr>
<th>Decision variable</th>
<th>Symbol</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exhaust gas mass flow rate</td>
<td>$m_{eg}$</td>
<td>kg/s</td>
</tr>
<tr>
<td>Exhaust gas temperature</td>
<td>$T_{eg,in}$</td>
<td>°C</td>
</tr>
<tr>
<td>Evaporation pressure</td>
<td>$p_{wf,EVA}$</td>
<td>bar</td>
</tr>
<tr>
<td>Degree of superheating</td>
<td>$SH_{wf,EVA,out}$</td>
<td>K</td>
</tr>
<tr>
<td>Temperature difference between exhaust and working fluid at saturated liquid</td>
<td>$\Delta T_{EVA,satL}$</td>
<td>K</td>
</tr>
<tr>
<td>Tube outer diameter</td>
<td>$d$</td>
<td>m</td>
</tr>
<tr>
<td>Tube length</td>
<td>$l$</td>
<td>m</td>
</tr>
<tr>
<td>Fin height</td>
<td>$f_{height}$</td>
<td>m</td>
</tr>
<tr>
<td>Fin pitch</td>
<td>$f_{pitch}$</td>
<td>m</td>
</tr>
<tr>
<td>Fin spacing</td>
<td>$f_{spacing}$</td>
<td>m</td>
</tr>
</tbody>
</table>

Fig. 3 illustrates the main steps of the optimization routine. Given the waste heat profile and a suitable working fluid, the first five decision variables are used to design thermodynamically the ORC, i.e. to define the temperatures, pressures and mass flow rates of the cycle. Once the thermodynamic design is known, the evaporator can be designed, so that the number of tubes, heat transfer area and mass of this component can be determined. The ORC thermodynamic and evaporator design data are used as parameters for the dynamic model of the ORC system. The next steps involve the tuning of the controllers. For this purpose, the dynamic model of the ORC unit is linearized at design point. After controller tuning, the dynamic behaviour of the ORC system subjected to the transient heat source profile is simulated. From the simulation, the average net power output of the ORC unit is determined and together with the evaporator mass, the two objective functions are known. The routine discards all designs that cannot ensure i) a positive degree of superheating, ii) a backpressure of the internal combustion engine below the upper limit or iii) an outlet temperature of the exhaust gas above the lower limit. Details on each single optimization step and on the constraints used for the global optimization are provided in the next subsections.

3.1 Thermodynamic organic Rankine cycle design

The thermodynamic design routine is implemented in MATLAB®. This routine is essential for the subsequent component design, controller tuning and dynamic simulation steps. The main assumptions are as follows: i) all components are considered in steady-state and thermodynamic equilibrium; ii) heat and pressure losses in all components are neglected; iii) the exhaust gas is considered as pure air at 1 bar and iv) the working fluid properties are retrieved from the REFPROP database [26]. The isentropic efficiencies of the pump, as well as the mechanical and electrical efficiency of the pump motor and inverter are summarized in Table 2. The isentropic efficiency of the expander, assumed here to be an axial-flow turbine, is estimated from the correlations from Macchi and Astolfi [27]. These correlations allow including the impact of the working fluid and of the thermodynamic design on the isentropic efficiency of the turbine. In this way, unfeasible or inefficient expander designs can be discarded already in this phase. The isentropic efficiency is a function of the isentropic volumetric ratio, turbine size parameter and the number of turbine stages. The selection of the number of turbine stages is based on the work from Martelli et al. [28], which set a maximum change in specific enthalpy for single stage to 130 kJ/kg and maximum actual volumetric ratio to 15. If one of these constraints is exceeded, the number of stages is increased by one, until the conditions are satisfied. The mechanical and electric losses of the turbine and the generator are summarized in Table 2.
In the thermodynamic design routine, the ORC evaporator is modelled as three subsequent zones depending on the working fluid phase. The routine monitors the temperatures of the exhaust gas and working fluid at the inlet and outlet of each zone to avoid temperature crossings. The condenser is assumed to operate at a fixed condensation temperature of 50 ºC. For some working fluids, however, this condensation temperature corresponds to a condensation pressure below the atmospheric pressure. This case has the drawback of requiring a vacuum system to avoid air infiltration, resulting in larger mass, volume and cost of the ORC system. For this reason, two cases are considered, one with and one without the constraint of minimum condensation pressure equal to 1 bar. If this constraint is active, the minimum condensation temperature is equal to the saturation temperature corresponding to 1 bar.

Table 2. Efficiencies of several ORC components.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isentropic efficiency of pump</td>
<td>(\eta_P)</td>
<td>70</td>
<td>%</td>
</tr>
<tr>
<td>Combined pump, motor and inverter mechanical and electrical efficiency</td>
<td>(\eta_{PMI})</td>
<td>70</td>
<td>%</td>
</tr>
<tr>
<td>Isentropic efficiency of turbine</td>
<td>(\eta_T)</td>
<td>according to Ref. [27]</td>
<td>%</td>
</tr>
<tr>
<td>Combined turbine and generator mechanical and electrical efficiency</td>
<td>(\eta_{TG})</td>
<td>82</td>
<td>%</td>
</tr>
</tbody>
</table>

3.2 Evaporator design

The evaporator design code is also developed in MATLAB®. Here, the heat transfer area and the mass of the heat exchanger are determined, together with the heat transfer coefficients of the exhaust gas and the working fluid. The dynamic model presented in the next subsection will use this information. The evaporator design routine was originally developed and presented in Pili et al. [15]. The exhaust gas flows in cross-counterflow on the shell-side of a fin-and-tube heat exchanger, whereas the working fluid flows inside the tubes (see Fig. 4). The evaporator is a once-through heat exchanger where the preheating, vaporization and superheating zones are considered as three consecutive lumped zones across the heat exchanger. The evaporator design routine determines the required amount of tubes corresponding to the design heat transfer rate. For the optimization, five decision variables are used, i.e. the fin height, tube outer diameter, fin pitch, tube length and fin spacing. Further details on the modeling and heat transfer correlations used in the evaporator design code are given in Pili et al. [15]. Differently from that work, the tubes are considered here in staggered arrangement in order to enhance the heat transfer. In addition, after a sensitivity analysis, the fin thickness was fixed to 0.5 mm instead of being considered a decision variable of the optimization.

The evaporator causes a rise in the engine backpressure, which penalizes the engine performance in terms of fuel consumption and power. Therefore, this value needs to be minimized. Based on advices from a truck manufacturer, the maximum allowed pressure drop on the exhaust gas side was set to 30 mbar. This value is expected to decrease the engine power output by less than 1 % [29]. The pressure drop of the working fluid was set not to exceed 5 % of the design evaporation pressure in order not to penalize the net power output of the ORC system.
Fig. 4. Geometry of fin-and-tube evaporator, adapted from Pili et al. [15]: a) tube cross-sectional view, b) tube axial view and c) heat exchanger arrangement.

3.3 Dynamic model and simulation

In order to ensure safe and efficient operation of the ORC unit subjected to the highly-transient waste heat profile, a dynamic model is included in the optimization. The model is developed in Simulink® [30]. The dynamic model mainly focuses on the ORC evaporator, which is the link between the heat source and the ORC unit. A moving boundary approach is used, because it provides a reasonable trade-off between model accuracy and computational speed. In the moving boundary approach, the evaporator is divided into three regions depending on the working fluid phase. During operation, the interfaces among the regions can shift, depending on the heat transfer rate. The model and its verification are described in Ref. [31]. The deviations in terms of degree of superheating from the moving boundary evaporator model available in the TIL library [32] in Dymola [33] based on a 20 % step change in working fluid mass flow rate were below 1 %. To allow for high computational speed, the heat transfer coefficients $\alpha$ of both the exhaust gas and the working fluid at off-design ('OD') are corrected from the design values ('D') according to the mass flow rate $\dot{m}$:

$$\alpha_{OD} = \alpha_D \left( \frac{\dot{m}_{OD}}{\dot{m}_D} \right)^\tau$$

where $\tau$ is an exponent derived from fitting and reported in Table 3. Particular attention is paid to the pressure drop of the exhaust gas $\Delta p_{eg}$, because of the negative effect of the engine backpressure on its performance. At off-design, the design exhaust pressure drop is corrected as a function of the square of the exhaust mass flow rate through the evaporator [34]:

$$\Delta p_{eg,OD} = \Delta p_{eg,D} \left( \frac{\dot{m}_{eg,OD}}{\dot{m}_{eg,D}} \right)^2$$

Table 3. Exponents for correction of heat transfer coefficients at off-design.

<table>
<thead>
<tr>
<th>Heat exchanger side</th>
<th>Region</th>
<th>Symbol</th>
<th>Exponent $\tau$</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube side</td>
<td>Liquid</td>
<td>$\alpha_{wf,L}$</td>
<td>0.92</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Two-phase</td>
<td>$\alpha_{wf,LV}$</td>
<td>0.67</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Vapour</td>
<td>$\alpha_{wf,V}$</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
<td>Shell side</td>
<td>All</td>
<td>$\alpha_{eg}$</td>
<td>0.54</td>
<td>-</td>
</tr>
</tbody>
</table>

The ORC pump and turbine are modelled at steady-state, since their dynamic responses are much faster than that of the evaporator. While the isentropic efficiency of the pump is kept constant at the design value because of its limited impact on the net power output of the ORC unit, the isentropic efficiency of the turbine is corrected as a function of the mass flow rate according to a correlation presented in Vetter [35]. In addition, while the pump speed is manipulated by the proportional-integral controller, the speed of the expander is assumed to stay constant, because of the lack of suitable correlations to estimate the part-load efficiency of the turbine at variable speed. It is worth mentioning that the constant-speed operation, although less complex, leads to a lower part-load efficiency of the expander with respect to the case where the rotational speed is optimized as a function of the operating point. Given the low speed of sound of organic fluids, it is reasonable to assume sonic conditions to be reached at the turbine nozzle. Therefore, the turbine inlet thermodynamic quantities are related to the mass flow rate at off-design by using the Stodola equation corrected for real gases [36]:
\[ m_{\text{wf},T} = k_T \frac{p_{\text{wf},T,\text{in}}}{\sqrt{\gamma_{\text{wf},T,\text{in}}} Z_{\text{wf},T,\text{in}} T_{\text{wf},T,\text{in}}} \]  

(4)

where \( m_{\text{wf}} \) is the mass flow rate of the working fluid, \( p_{\text{wf},T,\text{in}} \) the pressure, \( \gamma_{\text{wf},T,\text{in}} \) the ratio of the specific heats, \( Z_{\text{wf},T,\text{in}} \) the compressibility factor and \( T_{\text{wf},T,\text{in}} \) the temperature at turbine inlet.

Similarly to the thermodynamic design, the condensation temperature and pressure is fixed at 50 °C or at the condensation temperature corresponding to 1 bar, if the constraint on minimum pressure is applied.

3.4 Controller tuning

Two controllers are used to operate the ORC system in a safe and efficient manner: i) one proportional-integral controller manipulates the mass flow rate of the pump to achieve the desired set point in terms of degree of superheating at turbine inlet and ii) one proportional controller manipulates the exhaust gas bypass when the evaporation pressure exceeds 90 % of the critical pressure.

The first controller has the task of ensuring that the turbine always operates with positive degree of superheating, thus avoiding damage and erosion of the turbine blades due to liquid droplets. The set point in degree of superheating could be optimized online or offline, depending on the waste heat mass flow rate and temperature. However, given the rapid fluctuations of the waste heat, a rapid change in set point at low degree of superheating can cause undershoots, and potentially lead to liquid droplets reaching the turbine. For this reason, the set point in degree of superheating is kept constant at the design value, and a constraint in the dynamic simulation excludes designs that cannot satisfy a positive degree of superheating over the driving cycle. By acting on the exhaust bypass valve, the second controller ensures that the system does not work near the critical point, where the working fluid thermodynamic properties can change considerably and the system can become difficult to control. In addition, if working at higher pressures, the evaporator should be designed to withstand larger mechanical stresses and prevent leakage of the working fluid, resulting in higher costs and mass. The pump and exhaust bypass valve controllers are tuned by linearizing the dynamic model around the design point. After that, the proportional and integral constants of the controller are found by using the standard ‘pidtune’ command from the Control System Toolbox from MATLAB® [25].

4. Results

This section demonstrates the effects of including the dynamic performance in the ORC design procedure, by considering the following two cases: i) a conventional design procedure, where the ORC net power output is maximized and the evaporator mass is minimized considering a steady-state heat source at the time-weighted average mass flow rate and temperature of the profile in Fig. 1 (i.e. 0.223 kg/s and 307 °C) and ii) the novel approach proposed here, where the average ORC net power output over the profile is maximized and the evaporator mass is minimized considering the dynamic performance of the ORC unit.

4.1 Steady-state optimization at time-weighted average conditions of the waste heat source

The multi-objective optimization is carried out for eight working fluids, which previous publications suggested as suitable candidates for waste heat recovery from heavy-duty trucks [13–15,37]. The results of the optimization are shown in Fig. 5. Since the dynamic performance of the system is not investigated here, all parameters correspond to the design point. The Pareto front in Fig. 5a shows for all working fluids that the larger the evaporator mass, the larger the ORC net power output. This means that a trade-off exists between the net power output and the evaporator mass. Based on suggestions from a truck manufacturer, the whole ORC system should not exceed 250 kg for a feasible installation. If a share of the evaporator mass between 30 % and 40 % with respect to the mass of the entire ORC unit is assumed [38], the maximum allowed evaporator mass is between 75 kg and 100 kg. The working fluids that could reach the largest net power output are toluene and ethanol with condensation pressure below ambient (‘LP’ case, which stands for ‘low pressure’), ranging from 3 kW at 23 kg to more than 6 kW at 110 kg. If the constraint of condensation pressure above ambient is imposed, pentane can reach the largest net power output between approximately 3 kW and 4 kW, followed by R1233zd(E), ethanol and R245fa.

Fig. 5b shows that the evaporation pressure is not significantly affected by the Pareto solution. Only for isobutene, there is an increase in evaporation pressure, which ranges from 19 bar to 23 bar. On the contrary, as indicated in Fig. 5c, the degree of superheating shows a large variation along the Pareto front. For all working fluids except ethanol LP, the degree of superheating drops significantly. The degree of superheating is the lowest for toluene, followed by pentane and ethanol, varying between 8 K and 30 K. The trend of the exhaust gas temperature at evaporator outlet is depicted by Fig. 5d. It can be seen that the outlet exhaust gas temperature drops approximately linearly with increasing ORC net power output. Since the mass flow rate and inlet temperature of the exhaust gas are the same for all designs (they are equal to the time-weighted average of the profile in Fig. 1), there is a larger heat transfer rate in the evaporator for larger ORC net power output. The maximum heat transfer rate is reached for most working fluids at the minimum heat source temperature at evaporator outlet of 100 °C.
Fig. 5. Steady-state multi-objective optimization at time-weighted average conditions of the waste heat: (a) Pareto front, (b) evaporation pressure, (c) degree of superheating and (d) exhaust gas temperature at evaporator outlet as a function of the ORC net power output.

It is important to highlight that the steady-state optimization leads to some of the best Pareto-front solutions having a very low degree of superheating (below 10 K) at design condition, which is in agreement with the results of Ref. [19], where the multi-objective optimization was also based on a steady-state heat source close to the time-weighted average. The solutions with low degree of superheating could turn out to be unfeasible, because the fluctuations of the waste heat profile could easily lead to a violation of the constraint of positive degree of superheating during the dynamic operation. Analogously, the exhaust gas outlet temperature is for some optimal designs very close to the lower bound of 100 °C, which will most likely be violated during the driving cycle.

4.2 Optimization including the dynamic performance

In order to highlight the potential of including the dynamic performance of the ORC unit in the multi-objective optimization, only the best working fluids from the steady-state multi-objective optimization are considered: toluene LP, ethanol LP and pentane. The multiple objectives are the ORC net power output and the evaporator mass as in the previous case, but the net power output refers now to the time-weighted average value at off-design from the dynamic simulation.

The results of the optimization are shown in Fig. 6. The results indicate that the Pareto fronts have similar trends as those presented in Fig. 5a, but the absolute values are shifted downward by approximately 1 kW in net power output. Toluene LP reaches the highest net power output, followed by ethanol LP and pentane. Differently from the steady-state optimization, the design mass flow rate and inlet temperature of the exhaust gas are not fixed but they are decision variables of the optimization. Nonetheless, it is interesting to see in Figs. 6b and 6c that both the design mass flow rate and inlet temperature of the exhaust gas are very close to...
Fig. 6. Multi-objective optimization including the dynamic performance of the ORC unit: (a) Pareto front, (b) exhaust gas mass flow rate, (c) exhaust gas inlet temperature, (d) evaporation pressure, (e) degree of superheating and (f) exhaust gas temperature at evaporator outlet at design point as a function of the average ORC net power output over the waste heat profile.

the time-weighted average value used for the steady-state optimization, suggesting that is reasonable to use the time-weighted average values for the design of the ORC unit. The maximum deviations from the time-weighted average values are found for ethanol LP at an average net power output over the profile below 3 kW, reaching approximately 0.12 kg/s and 35 K. The design inlet temperature of the exhaust gas has a slight
increasing trend with rising average net power output for all working fluids, with a deviation between -40 K and 20 K from the time-weighted average. The trend of the design evaporation pressure is shown in Fig. 6d. The results suggest that this quantity remains approximately constant with increasing ORC net power output, analogously to the results shown in Fig. 5c. On the contrary, the design degree of superheating is much larger than for the steady-state optimization, with values above 35 K for all working fluids. The degree of superheating drops for pentane and toluene with increasing net power output, whereas ethanol LP shows an increase of degree of superheating at design with increasing ORC net power output, analogously to the steady-state optimization case. The design exhaust gas temperature at evaporator outlet is shown in Fig. 6f. The results indicate that analogously to Fig. 5d the exhaust gas outlet temperature drops with increasing net power output, although ethanol LP has some oscillations below 3 kW. The lower exhaust gas temperature at evaporator outlet corresponds to a larger heat transfer rate at design, and therefore larger net power output of the ORC unit. For all working fluids the margin of the exhaust gas temperature at the evaporator outlet to the lower bound of 100 °C is larger than those of the steady-state case, being above 5 K for Pareto-front optimal solutions.

To summarize, the results presented in Figs. 5 and 6 indicate that including the dynamic performance results in higher degrees of superheating and exhaust gas temperatures at evaporator outlet at design condition to avoid the violation of the constraints in terms of superheating degree and exhaust gas temperature at evaporator outlet, ensuring safe operation and avoiding damaging components of the ORC unit.

5. Conclusions

This work presented a novel integrated multi-objective optimization approach for the design of organic Rankine cycle power systems recovering the waste heat from a heavy-duty truck. Two objectives were selected for the optimization routine, i.e. the maximization of the unit net power output and the minimization of the evaporator mass. Unlike conventional steady-state design optimization, the multi-objective optimization proposed in this work includes the dynamic performance of the organic Rankine cycle unit in the optimization routine, thus accounting for the performance degradation of the system at off-design and ensuring safe operation also when the system dynamics play an important role. A comparison with a conventional steady-state multi-objective optimization based on the time-weighted average values of the heat source mass flow rate and temperature is provided. In the steady-state conventional design, the results suggest that there is a trade-off between the maximum net power output and the minimum evaporator mass, corresponding to a Pareto front of optimal solutions. In addition, the design evaporation pressure does not vary significantly along the Pareto-front solutions, whereas the degree of superheating and the exhaust gas temperature at evaporator outlet vary over a broad range, reaching their minimum at high values of net power output and evaporator mass for most of the working fluids. When the dynamic performance of the organic Rankine cycle unit is included the degree of superheating and the exhaust gas temperature at evaporator outlet increase considerably for most working fluids (more than 35 K and 5 K, respectively), because the waste heat fluctuations during the driving profile require larger margins to satisfy the constraints of positive degree of superheating and exhaust gas temperature above the acidic dew point. The results suggest that both for the conventional steady-state optimization and when the dynamic performance is included, toluene achieves the highest net power output for the same evaporator mass, followed by ethanol. However, if the condensation pressure is constrained to be above ambient to avoid the installation of a vacuum system, pentane results in the highest net power output. The multi-objective optimization routine presented in this work highlights the importance of including the dynamic performance already in the design optimization phase in order to avoid unfeasible or suboptimal solutions. This work provides a methodology through which the optimal working fluid and the optimal set of decision variables can be found, even for a challenging problem as the waste heat recovery from the highly-transient exhaust gas profile of heavy-duty trucks.

Acknowledgments

This research was developed as part of the project “ACT-ORC: Advanced control of organic Rankine cycle for increased energy efficiency of heavy-duty trucks”, funded by the European Union's Horizon 2020 research and innovation program under the Marie Sklodowska-Curie grant agreement no. 754462 (EuroTechPostdoc).

Nomenclature

Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>LP</td>
<td>Low Pressure (case)</td>
</tr>
<tr>
<td>ORC</td>
<td>Organic Rankine Cycle</td>
</tr>
</tbody>
</table>

Symbols

- $c_p$: Specific heat at constant pressure, J/kgK
- $l$: Tube length, m
\[ d \quad \text{Tube outer diameter, m} \]
\[ \dot{m} \quad \text{Mass flow rate, kg/s} \]
\[ \Delta p \quad \text{Pressure drop, Pa} \]
\[ \dot{Q} \quad \text{Heat transfer rate, W} \]
\[ \Delta T \quad \text{Temperature difference, K} \]
\[ S_h \quad \text{Degree of superheating, K} \]
\[ \tau \quad \text{Power law exponent} \]

Greek symbols
\[ \alpha \quad \text{Heat transfer coefficient} \]
\[ \gamma \quad \text{Ratio of specific heats} \]
\[ \eta \quad \text{Efficiency} \]
\[ \tau \quad \text{Power law exponent} \]

Superscripts and subscripts
\[ a \quad \text{Available} \]
\[ D \quad \text{Design} \]
\[ eg \quad \text{Exhaust gas} \]
\[ EVA \quad \text{Evaporator} \]
\[ in \quad \text{Inlet} \]
\[ L \quad \text{Liquid phase} \]
\[ LV \quad \text{Two-phase} \]
\[ OD \quad \text{Off-design} \]
\[ out \quad \text{Outlet} \]
\[ P \quad \text{Pump} \]
\[ PMI \quad \text{Pump, motor and inverter} \]
\[ satL \quad \text{Saturated liquid} \]
\[ T \quad \text{Turbine} \]
\[ TG \quad \text{Turbine and generator} \]
\[ V \quad \text{Vapour phase} \]
\[ wf \quad \text{Working fluid} \]

References


Mathworks®. MATLAB® 2019.


