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Optimization of the organic Rankine cycle power systems for waste heat recovery on heavy-duty vehicles considering the performance, cost, mass and volume of the system

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Abstract:

The use of organic Rankine cycle power systems for waste heat recovery on internal combustion engines of heavy-duty vehicles can help to mitigate the greenhouse gasses and reduce the fuel consumption of the vehicle. However, designing an organic Rankine cycle system for this application is a complex process involving trade-offs among factors such as performance, space/weight restrictions, and cost. This paper presents a multi-objective optimization study of an organic Rankine cycle unit for waste heat recovery from heavy-duty vehicles from techno-economic and sizing perspectives. The optimization was carried out for seven different working fluids using the genetic algorithm to minimize the cost, volume and mass, and maximize the net power output of the organic Rankine cycle unit. The organic Rankine cycle performances for a driving cycle of a truck were also evaluated. The results indicate that the mass, volume, cost and net power output of the organic Rankine cycle system increase with the evaporation temperature. Moreover, the results suggest that when the condensation temperature was decreased from 60 °C to 40 °C, the net power, weight, cost and volume of the organic Rankine cycle unit increases by 22 %, 12 %, 46 %, and 12 % respectively. The maximum net power output, both at the design and off-design conditions, is obtained with pentane as working fluid. For this design, the net power output of the organic Rankine cycle unit is 10.94 kW at design condition and 8.3 kW at off-design (in average) condition, and the mass, volume, and cost of the organic Rankine cycle system are 129 kg, 1.077 m³, and 8527 €, respectively.

Keywords: Waste heat recovery, Organic Rankine cycle, Internal combustion engine, Multi-objective optimization, Volume, Weight, Cost optimization

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1. Introduction

The thermal efficiency of modern internal combustion engines (ICEs) has already reached its maximum value and only marginal increments are possible [1]. Still, ICEs of heavy-duty vehicles convert only approximately 40 % of the combustion heat into mechanical energy, while the remaining heat is rejected to the environment as waste heat [2]. The use of an organic Rankine cycle (ORC) unit for waste heat recovery on an ICE of heavy-duty vehicles can help to reduce the fuel consumption of the vehicle and mitigate the greenhouse gases emissions. However, space limitations, weight, cost, control and off-design performance of the ORC unit, and implications on the ICE performance of the additional back pressure caused by the ORC unit are among the most important aspects that need to be addressed in order to make ORC systems economically attractive in truck applications.

The application of the ORC technology to recover exhaust gas heat from heavy-duty vehicles was experimentally investigated in a number of studies [3-4]. If successfully implemented on a heavy-duty vehicle, an ORC unit is expected to reduce the fuel consumption by 3 % to 10 % [5]. Recently, comprehensive studies presented the integration of an ORC-based power unit with internal combustion engines [6-9]. The major barriers for its integration are the following:

1. Weight and space restriction: the added weight of the ORC system results in an increase of the propulsion power and fuel consumption. In addition, the ORC system should be compact in order to fit into a vehicle without major modifications. Therefore, truck applications demand for a small weight and volume of the ORC system.
2. Cost of the ORC unit: the fuel saving needs to be sufficiently large to justify the cost of the ORC system in terms of payback period and other economic indicators. The specific cost of the unit (\$/kW) is considered a viable economic parameter for such analysis since the lower the specific cost the better the economic prospects.
3. Off-design conditions: due to the highly fluctuating heat source, a large fraction of the time the ORC system operates far from design conditions, which deteriorates its economic potential. Thus, the ORC system should be designed accounting for the off-design operating conditions as well.
4. Back pressure: the pressure drop of the exhaust gas in the evaporator of the ORC system results in an increase in fuel consumption of the engine. A maximum pressure drop of 40 kPa in the exhaust gas is acceptable for an ORC system for a truck application. Additional pressure drop should be avoided.
5. Control of the ORC unit: due to the very transient heat source, it is challenging to operate the ORC system at optimum power and under safe conditions (e.g. ensuring a certain degree of superheat before the expander) at all times. It is of crucial importance to consider the dynamics and control of the ORC system at the preliminary design phase.

In conclusion, a good design of the ORC system for truck application should have low weight, low volume, low cost, low pressure drop in the evaporator (especially on the gas side), high net power output, high off-design performance, and high controllability. The most appropriate way addressing all these aspects is to carry out multi-objective optimization.

In multi-objective optimizations for low-grade waste heat recovery problems solution strategies based on the genetic algorithm (GA) have been adapted. Hajabdollahi et al. [10] carried out a thermo-economic optimization of an ORC system recovering waste heat from a 20 kW diesel engine. The Pareto front results suggested that R123 provides the maximum efficiency and lowest total annual cost under given bounds. Furthermore, the results suggested 0.01 %, 4.39 %, and 4.49 % improvement of R123 for the total annual cost in comparison with R245fa, R22, and R134a, respectively. Yang et al. [11] performed thermo-economic optimization of an ORC system for waste heat recovery from a marine diesel engine. Among the considered working fluids, R1234yf performed best followed by R1234ze, R152a, and R600a, and R245fa with a specific investment cost of 0.266 W/\$, 0.252 W/\$, 0.249 W/\$, 0.247 W/\$, and 0.244 W/\$, respectively. Yang et al. [12] performed a thermo-economic study using a multi-objective optimization of an ORC system for waste heat recovery of a 247 kW diesel engine using six different working fluids. At the rated engine power, R601a and R245fa had a net power of 13.19 kW and 11.55 kW, respectively. The total investment costs of the ORC system for these two working fluids were 536990 \$ and 532710 \$, respectively. Seyedkavoosi et al. [13] performed an exergy optimization of a parallel evaporator ORC system using the working fluids R123, water, and R134a. The result suggested that R123 produce 468 kW of net output power with an exergy efficiency of 21 %. Yang et al. [14] investigated the performance of a dual loop ORC system for a compressed natural gas engine waste heat recovery and conducted a parametric optimization and heat transfer analysis. For the high-pressure cycle, the optimum evaporation temperature and superheat were 25 bar to 29 bar, and 0.43 K to 12.35 K respectively.

Bufl et al. [15] performed a robust optimization of an ORC system for heavy-duty engine waste heat recovery applications using R245fa, R245ca, R134a, R11, R113, and Novec 629. The working fluid R11 provided the maximum thermal efficiency of 20.8 %, corresponding to an evaporation pressure of 35bar, superheat of 9.8 K, and pinch point temperature difference of 8.5 K, respectively. Yang et al. [16] presented the optimization of an ORC system based on artificial neural networks for diesel engine waste heat recovery using R123 as the working fluid. A comparison of artificial neural networks and genetic algorithm optimization showed a prediction error of less than 0.2 kW in the expander power output. The maximum power of the expander was 7.13 kW, corresponding to the expander inlet and outlet pressure of 8.2 bar and 1.6 bar respectively. Zhao et al. [17] performed a global optimization of an ORC system recovering the exhaust gas heat from an 85 kW diesel engine. The simulation was carried out in

GT-Suite/Simulink co-simulation environment using the particle swarm optimizer approach. The optimization results suggested improvements of 3.24 % and 3.13 % on the power output and brake specific fuel consumption, respectively.

Negash et al. [18] carried out an optimization to maximize the net power output of the ORC system used for waste heat recovery of a construction equipment engine. Four cases were considered based on different cycle configurations and waste heat recovery streams using R134a as the working fluid. The results suggested that at half load condition, the net power increased by 11 % using also the oil system compared to the ORC system recovering heat from engine exhaust only. Zhu et al. [19] performed thermo-economic optimization to find the optimum evaporation pressure, condensation pressure and working fluids for an ORC system for exhaust waste heat recovery from a 662 kW marine diesel engine. The working fluid R141b showed the highest performance corresponding to an evaporation temperature of 450 K, condensation temperature of 320 K and provided a net power of 97 kW, a thermal efficiency of 19.87 %, and an exergy efficiency of 45.84 %. Yang et al. [20] developed a thermo-economic model of a dual loop ORC system to analyze both the thermodynamic and economic performance of several working fluid groups for waste heat recovery from a compressed natural gas engine. The results suggest that at engine maximum rated power point, the ORC system can achieve a maximum net power output per unit heat transfer area of the evaporator of 0.74 kW/m², and the ratio of maximum effective heat transfer area to the actual area of the evaporator is 69.19 %.

Galindo et al. [21] presented a mathematical model of a bottoming ORC coupled to a 2L turbocharged gasoline engine to optimize the cycle from a thermo-economic and sizing point of view using a multi-objective optimization algorithm. The results suggested that the optimum value of specific investment cost is 2515 €/kW, the area of heat exchangers is 0.48 m², and the volume coefficient is 2.62 MJ/m³. Tian et al. [22] carried a techno-economic evaluation of an ORC system used for ICE exhaust gas heat recovery and 20 working fluids were included in the evaluation. The screening criteria were based on the thermal efficiency of the ORC, the expansion ratio, the net power output per unit mass flow rate of the hot exhaust, the ratio of total heat transfer area to net power output, and the cost of electricity. The results showed that R141b, R123 and R245fa give the highest thermal efficiency and net power output values. Wang et al. [23] performed a parametric optimization of a regenerative ORC system for a diesel engine based on particle swarm optimization. The results indicated that the maximum net power output of the ORC system using n-butane is 36.57 kW, corresponding to an 11 % increase of the diesel engine power output. Hongda et al. [24] performed a multi-objective optimization of a fin-and-tube evaporator for a diesel engine ORC combined system using particle swarm optimization algorithm. Compared with the results of the original design, the multi-objective optimization showed an increase of the volume of tube bundle by 6.5 %, a decrease in the total annual cost by 71.5 % and a reduction in the exhaust pressure

drop by 27.6 %. A list of the objective functions and corresponding decision variables from multi-objective optimization studies is presented in [Table 1](#).

The literature review indicates that the most commonly employed objective functions in multi-objective optimization studies for ORC for truck applications are the total investment cost, net power, heat exchangers area, thermal efficiency and exergy efficiency. However, an optimization including the mass and volume of the ORC system is not yet presented. Due to the weight and volume restrictions on trucks, it is much more important to minimize the mass and volume of the ORC unit used for trucks than for many other applications. This paper presents a multi-objective optimization study of an organic Rankine cycle unit for waste heat recovery from heavy-duty vehicles from techno-economic and sizing perspectives. The optimization was carried out for seven different working fluids using the genetic algorithm to minimize the cost, volume and weight, and maximize the net power output of the ORC unit. In addition, the ORC performances for a driving cycle of a real truck were evaluated. The preliminary design of the ORC system was based on a 0D model, which was extended to 1D-component models. The mass, volume and cost of the ORC system were calculated using empirical correlations. Only steady-state operation was considered.

[Section 2](#) provides the details of the waste heat recovery system. [Section 3](#) presents the design, techno-economic model, weight and volume estimation and optimization approach. The optimization result and the off-design performance are discussed in [Section 4](#). Finally, [Section 5](#) provides the concluding remarks.

2. System Description

The waste heat is recovered from the exhaust gas of a 450 hp diesel engine of a long-haul truck. It is a 13 L Euro 6 engine with Selective Catalytic Reduction (SCR). Exhaust Gas Recirculation (EGR) is not used. [Figure 1](#) shows the schematic of the waste heat recovery system.

The heat from the ORC condenser can be rejected using the existing engine radiator or an additional radiator dedicated to the ORC unit. Therefore, the optimization was performed for two condensation temperatures, 40 °C, assuming that an additional radiator is used, and 60 °C, assuming that the existing engine radiator is used. The conditions for which the ORC unit is designed play a significant role to achieve an optimal design considering the operational cycle of the truck/engine. In the present study, the heat source conditions corresponding to 40 % engine load were selected as the design point of the cycle. The design point was selected based on experiences shared by a truck manufacturer. [Table 2](#) shows the operating conditions of the engine in the design point (the data were provided by the truck manufacturer). Based on the aforementioned assumptions two cases were defined:

- Case 1 for condensation temperature of 40 °C
- Case 2 for condensation temperature of 60 °C

The initial screening of working fluids was based on the global safety criteria, environmental impact, and thermodynamic characteristics. The global safety factor is summarized in one single factor based on the NFPA 704 system [25]:

$$GSF = \alpha * F + \beta * I + \gamma * H \quad (1)$$

The parameters H , F and I are health, flammability, and instability values as per NFPA 704, and α , β and γ are weighting factors for each safety factor. The major importance was given to the flammability factor ($\alpha = 0.5$), while medium importance was given to the instability factor ($\beta = 0.3$) and lowest importance was given to the health hazard ($\gamma = 0.2$). Working fluids with a global safety value higher than 2.5 and a low global warming potential and ozone depletion potential (< 1) were selected for the optimization. [Table 3](#) shows the properties of the seven working fluids selected for optimization. [Figure 2](#) shows the T-s diagram of the selected working fluids.

3. Modeling

3.1. Sizing of the ORC system

The evaporator and condenser are brazed plate heat exchangers and were selected due to their compact size. The evaporator and condenser design models are based on one-dimensional discretization in the flow direction, assuming equidistant steps of enthalpy rate change for both the fluids. [Figure 3](#) shows the one-dimensional discretization model of the evaporator and condenser along the flow direction.

Heat transfer and pressure drop in each control element were calculated by solving the heat transfer and fluid flow conditions for each control volume. The temperatures for heat transfer calculations were estimated at the control volume center points, interpolating the values between the nodes. The inlet and outlet conditions (temperature and mass flow rate) of the primary and secondary fluid were assigned as input from the cycle design. For a selected heat exchanger geometry, the model was initiated with an assumed number of plates. The design algorithm is iterative and calculates the required number of plates that satisfies the required heat load and pressure drop restriction. In order to minimize the effect of the ORC on the ICE performance, a 30 kPa pressure drop limitation was implemented in the exhaust side of the evaporator. The total pressure drop in a plate heat exchanger, ΔP , was computed as the sum of the frictional pressure drop, ΔP_{fr} , pressure drop due to acceleration, ΔP_{acc} , pressure drop due to gravity (static), ΔP_{gr} , and pressure drop in the inlet/outlet ports, ΔP_{port} .

$$\Delta P = \Delta P_{fr} + \Delta P_{acc} + \Delta P_{gr} + \Delta P_{port} \quad (2)$$

$$\Delta P_{fr} = \frac{2f\Delta LG^2}{\rho_m D_h} \quad (3)$$

$$\Delta P_{acc} = G^2 \left[\left(\frac{x}{\rho_g} + \frac{x-1}{\rho_f} \right)_{out} - \left(\frac{x}{\rho_g} + \frac{x-1}{\rho_f} \right)_{in} \right] \quad (4)$$

$$\Delta P_{gr} = g\rho_m \Delta L \quad (5)$$

$$\Delta P_{port} = 0.75 \left[\left(\frac{G^2}{2\rho} \right)_{in} - \left(\frac{G^2}{2\rho} \right)_{out} \right] \quad (6)$$

The fanning friction factor, f , during flow boiling [26] and flow condensation [27] is estimated as:

$$f_{eva} = 2.57 Re_{eq}^{-0.246} Bo_{eq}^{-1.568} We^{-0.3716} Bd^{0.0815} \quad (7)$$

$$f_{con} = 94.75 Re_{eq}^{-0.0467} Re^{-0.4} Bo^{0.5} \left(\frac{P_m}{P_{crit}} \right)^{0.8} \quad (8)$$

Empirical correlations were used to calculate the heat transfer coefficient during the flow boiling, α_{eva} , evaporation and flow condensation, α_{eva} , in plate heat exchangers [26,27]:

$$Nu_{eva} = \left(\frac{D_h \alpha_{eva}}{k} \right) 5.89 Re_{eq}^{2.509} Bo_{eq}^{1.321} We^{-0.229} Bd^{-0.76} \quad (9)$$

$$Nu_{con} = \left(\frac{D_h \alpha_{con}}{k} \right) 4.118 Re_{eq}^{0.4} Pr_l^{0.33} \quad (10)$$

For the heat sink (cooling water) and single-phase refrigerant, the Chisholm and Wanniarachchi correlation as presented in Ref. [28], and Hsieh's correlation [29] were used:

$$Nu_{ss} = \left(\frac{D_h \alpha_{ss}}{k} \right) \left[0.724 \left(\frac{6\beta}{\pi} \right)^{0.646} Re^{0.583} Pr^{0.33} \right] \quad (11)$$

$$Nu_{wf} = \left(\frac{D_h \alpha_{wf}}{k} \right) \left[0.2092 Re^{0.78} Pr^{0.33} \left(\frac{\mu_m}{\mu_{wall}} \right)^{0.14} \right] \quad (12)$$

Suitable heat transfer correlations [30] were used to estimate the heat transfer coefficient of the exhaust gas according to the Reynolds number. The heat exchanger design algorithm was validated against the results of the commercial software Unisim [31]. The total heat transfer area and pressure drop of the heat exchanger computed by the design model was compared with the results obtained using Unisim [31]. The results of the validation indicate that for the same plate geometry and working conditions, the difference in the heat exchanger area and pressure drop predicted by the models is less than ± 4 %.

The pump is characterized by a constant isentropic efficiency of 65 %. The preliminary design of the expander was performed using a mean-line model of a radial-inflow turbine [32]. For given specifications from the thermodynamic cycle (i.e., mass flow rate, inlet temperature, pressure and stage pressure ratio) and a set of decision variables related to the turbine geometry and fluid-dynamic conditions, the model solves the governing equations for mass, energy and momentum in the main turbine stations, and estimates the preliminary design and isentropic total-to-static efficiency of the

machine. The turbine model has been validated with a number of experimental test cases from the open literature and showed accuracy in the turbine isentropic efficiency within ± 3 %-points [32].

3.2. Cost model

The total cost of the system, TCS , was divided into direct cost, DC , and indirect cost, IC . The cost of the system components, C_{sys} , cost of the installation, C_{inst} , cost of working fluid, C_{wf} , cost of piping, C_{pipe} , and cost of the instrumentation and control, C_{ctr} , are included in direct cost. The engineering cost, C_{eng} , and contingency costs, C_{cont} , are included in indirect cost. The cost of working fluid pump, C_{pp} , cost of the evaporator, $C_{hx,eva}$, cost of the expander, C_{exp} , and cost of the condenser, $C_{hx,con}$, are included in the system cost. The figures for the multiplying factors included in Eqs. 16-18 and Eqs. 20-21 are based on discussions with and recommendations by manufacturers of ORC systems. The total cost of the system, direct cost and indirect cost are given by

$$TCS = DC + IC \quad (13)$$

$$DC = C_{sys} + C_{wf} + C_{pipe} + C_{ctr} + C_{inst} \quad (14)$$

$$C_{sys} = C_{pp} + C_{hx,eva} + C_{hx,con} + C_{exp} \quad (15)$$

$$C_{pipe} = 0.15 C_{sys} \quad (16)$$

$$C_{inst,ctr} = 0.2 C_{sys} \quad (17)$$

$$C_{inst} = 0.3 C_{sys} \quad (18)$$

$$IC = C_{eng} + C_{cont} \quad (19)$$

$$C_{eng} = 0.08 DC \quad (20)$$

$$C_{cont} = 0.15 C_{eng} \quad (21)$$

The economic model of the ORC system is based on the business model approach presented by Guillaume [33]. The cost estimation of the individual components is carried out assuming the type of material, the fabrication process and serial production of 20000 machines per year. Stainless steel material was selected for parts directly in contact with the working fluid while carbon steel was chosen for the others to reduce the price. Molding was the assumed machining process for the components in order to reduce the fabrication costs. The costs of the expansion machine and heat exchangers were estimated using the costing add-in function in the Solidworks software [34]. [Table 4](#) presents the cost of the ORC components. Comparisons with figures supplied by equipment suppliers indicate that the cost estimation method used here is reasonable.

3.3. Mass and volume models

The total mass of the ORC system, TM , consists of the mass of ORC components, M_{sys} , mass of piping, M_{pipe} , mass of the instrumentation/control system, M_{ctr} . An additional mass of 10 % is added to the total mass to account for the mass of the storage tank, and other auxiliary components [35]. The total mass of the ORC system is given by

$$TM = 1.1 * (M_{sys} + M_{pipe} + M_{ctr}) \quad (22)$$

$$M_{sys} = M_{pp} + M_{hx,eva} + M_{hx,con} + M_{exp} + M_{wf} \quad (23)$$

$$M_{pipe} = 0.14 M_{sys} \quad [35] \quad (24)$$

$$M_{inst,ctr} = 0.02 M_{sys} \quad [35] \quad (25)$$

The total volume of the ORC unit, TV , was calculated by adding the volume of all the individual components of the ORC system and the volume of the auxiliary components, V_{aux} :

$$TV = V_{sys} + V_{aux} \quad (26)$$

$$V_{sys} = V_{pp} + V_{hx,eva} + V_{hx,con} + V_{exp} \quad (27)$$

$$V_{aux} = 0.3 V_{sys} \quad (28)$$

The volume of the piping network, tank, and instrumentation was assumed to be 30 % of the volume of the ORC components. The models of the mass and volume of the components were based on input and suggestions provided by manufacturers/suppliers. The data obtained from the manufacturers/suppliers were used to develop polynomial fits for mass and volume of the expander, pump and heat exchangers; see [Table 5](#).

The mass and volume correlations of the expander are based on the expander power (kW), while for the pump the correlations are based on the differential pressure (in bar) across the pump. The expander correlations are valid for a turbine size from 5 kw to 25 kW. The correlations for the pumps is valid for positive displacement pumps featuring a differential pressure from 5 bar to 20 bar. The heat exchanger correlations are valid for chevron type brazed plate heat exchangers. For further details, the reader is referred to Refs. [36–38]. [Figure 4](#) depicts the industry data along with the results of the correlations for the weight and mass of the expander and the pump.

3.4. Optimization

The optimization of the ORC system was carried out by using the genetic algorithm. [Figure 5](#) shows the detailed layout of the optimization approach. A multi-objective optimization was carried out using the genetic algorithm involving both thermodynamic and economic performances. Four objectives

functions were considered including net power of the ORC system, Wn , total weight of ORC unit, TM , total volume of ORC unit, TV , and total cost of the ORC unit, TCS .

$$F_1(x) = \text{Maximize } [Wn] = \text{Maximize } [\dot{W}_{exp} - \dot{W}_{pp}] \quad (29)$$

$$F_2(x) = \text{Minimize } [TM] = \text{Minimize } [1.1 * (M_{sys} + M_{pipe} + M_{inst,ctr})] \quad (30)$$

$$F_3(x) = \text{Minimize } [TV] = \text{Minimize } [1.3 * (V_{pp} + V_{hx,eva} + V_{hx,con} + V_{exp})] \quad (31)$$

$$F_4(x) = \text{Minimize } [TCS] = \text{Minimize } [DC + IC] \quad (32)$$

Decision variables and their lower and upper bounds:

$$50\text{ }^{\circ}\text{C} \leq T_{ev} \leq 0.9 * T_{cr}$$

$$0\text{ }^{\circ}\text{C} \leq SH \leq 50\text{ }^{\circ}\text{C}$$

$$5\text{ }^{\circ}\text{C} \leq PPTD_{ev} \leq 20\text{ }^{\circ}\text{C}$$

During the optimization, the geometry of the components and other cycle parameters were kept constant and the objective functions were evaluated by varying the decision variables.

3.5. ORC off-design model

In order to evaluate the performance of the ORC along the operational profile of the diesel engine, an off-design model was implemented. The inputs for the model were the exhaust gas temperature and mass flow rate values from real driving conditions of a diesel engine of a long-haul truck. During the off-design simulations, the control strategy was based on the sliding evaporator pressure, keeping the condenser pressure constant, and controlling the working fluid mass flow rate.

The performance variation of the heat exchangers, where one fluid was a gas or vapor, were modeled based on variations in the overall UA value of the heat exchangers. In these heat exchangers, the gas or vapor side is assumed to dominate the heat transfer processes resulting in a UA value variation governed by the change in the gas or vapor mass flow rate. The UA value and pressure losses in the heat exchangers at off-design were assumed to vary according to the following function [39,40]:

$$(UA)_{od} = (UA)_{ds} \left[\frac{\dot{m}_{od}}{\dot{m}_{ds}} \right]^{0.6} ; \quad (\Delta P)_{ofd} = (\Delta P)_{ds} \left[\frac{\dot{m}_{od}}{\dot{m}_{ds}} \right]^2 \quad (33)$$

The efficiency of the pump in off-design conditions, $(\eta_{pp})_{od}$, was approximated with a third degree polynomial including the ratio of the inlet volumetric flow at the off-design condition to that at design condition [41]:

$$(\eta_{pp})_{od} = (\eta_{pp})_{ds} \left\{ -0.029265 \left[\frac{\dot{V}_{od}}{\dot{V}_{ds}} \right]^3 - 0.14086 \left[\frac{\dot{V}_{od}}{\dot{V}_{ds}} \right]^2 + 0.3090 \left[\frac{\dot{V}_{od}}{\dot{V}_{ds}} \right] + 0.8638 \right\} \quad (34)$$

$\dot{V}$ is the volume flow rate at the pump inlet and $(\eta_{pp})_{ds}$ is the pump design efficiency.

The off-design performance of the radial-inflow turbine was estimated using the mean-line model described in Ref. [32]. The values of total-to-static efficiency and mass flow rate of the turbine were computed based on the off-design values of cycle pressure ratio, and were subsequently used to estimate the ORC system performance.

Heat losses and pressure drops through the piping of the ORC unit were neglected. The simulation assumed steady state conditions and the dynamics of the ORC unit were not considered.

4. Results and discussion

4.1 Optimization results

The aim of a multi-objective optimization study is to find the Pareto frontier optimal solution. Each point of the frontier represents one potential solution in the multi-objective optimization problem. The selection of the final optimum depends on the importance of each objective. Single values of the decision variables cannot satisfy all the objective functions simultaneously. [Table 6](#) shows the value of the objective functions and the corresponding decision variables at the point where the net power has reached the maximum value. The working fluid pentane provides a maximum net power of 10.94 kW, which corresponds to a total weight of 129 kg, total volume of 1.07 m³, and total cost of 8527 €. The minimum weight and volume was obtained with the working fluid RE347mcc. The minimum weight of 116 kg and volume of 0.79 m³ corresponds to a total cost of 4316 €. These values are below the targeted values in industry. For a 300 kW truck engine, the target is to obtain a weight and cost below 150 kg and 5000 €, respectively.

The results indicate that the optimized net power is higher in case 1 than in case 2. This can be explained by the temperature difference between the heat source and heat sink. The higher temperature difference between heat and sink, the higher will be the net power. However, the higher temperature difference also results in the increase of the volume, weight and cost of the system. Each working fluid results in different values of the optimized objective functions. The highest net power is achieved for pentane and the lowest is achieved for RE347mcc.

[Figure 6](#) presents the result of optimization study for pentane for a condensation temperature of 40 °C. From the Pareto front trend it is clear that an increase in net power result in increases in the values of cost, weight and volume. The maximum net power area is highlighted in a rectangle as optimized power; the corresponding values of the weight, volume and cost are also highlighted in rectangles. The maximum value of the net power corresponding to minimum cost of the ORC system was selected as the optimized design, see the arrow in [Figure 6b](#). For each working fluid, the values of the objective

functions corresponding to the optimized net power and decision variables are shown in [Table 6](#). The red line in the [Figure 6](#) represents the optimized set of population from the selected population in the genetic algorithm. Each point of the population corresponds to different values of decision variables and, thus, has a different value of the objective functions.

In order to compare the variation of objective functions for case 1 and 2, the change in the net power output, cost, volume and weight for the ORC system with evaporation temperature for pentane is shown in [Figure 7](#). It is observed that the evaporation temperature has a higher influence on the objective functions than the other decision variables of the optimization. The values of each objective function increases with the evaporation temperature due to the increase in size of the individual components of the ORC system. A significant increase in the values of each objective function is observed at an evaporation temperature of 120 °C and above. When the condensation temperature decreases from 60 °C to 40 °C, the net power, weight, cost and volume of ORC unit increases by 22 %, 12 %, 46 %, and 12 %, respectively.

4.2 Off-design performance assessment

[Figure 8](#) depicts the data in terms of mass flow rate and temperature of the engine exhaust gases after the SCR (see [Figure 1](#)) for typical long-haul operation of a truck that was provided by a truck manufacturer (including 2700 data points in total). The data were split into 'bins', discretized over an 8x7 grid, representing groups of data of similar quantities in order to reduce the computational effort of the off-design model. Each bin was assigned mass flow rate and temperature values equal to the values at the center of each bin. Furthermore, each bin was assigned a histogram weighting based on the number of data points contained within the bin.

For the off-design performance, the design conditions of the system corresponding to the optimized values of the objective functions were used, see [Table 6](#). The average net power output in off-design conditions was calculated by weighting the value for each bin based on the number of points within each bin (representing how often a given condition occurs). [Table 7](#) shows the average net power output at design and off-design conditions.

For cases 1 and 2, the ORC unit operates 40 % of the time at heat source conditions lower than its design value. The difference between the net power in design and off-design conditions was highest for MDM. It was mainly due to lower turbine isentropic efficiency at part-load conditions. The lowest variation of the net power was observed for R1233zd. The off-design results obtained using the bin approach were also compared with the approach considering all 2700 data points. For the net power of the ORC system, the relative deviation between the two methods is less than 1 %. However, the compilation time for the

approach including all data is 17 hours, while the same simulation is performed in 43 minutes using the bin approach.

For case 1 with pentane as the working fluid, which is the design providing the highest net power output at design conditions, the variation of the net power output of the ORC unit with exhaust gas heat condition is shown in [Figure 9](#). The temperature of the exhaust gas after the SCR does not vary significantly during the driving cycle, while there is a significant variation of the exhaust mass flow rate with the engine load. During accelerating and decelerating periods, the power produced by the ORC unit is lower than 1 kW due to very low mass flow rate of the exhaust gas.

4.3 Practical implementation

The dynamics and control aspects of the ORC unit is not considered in the present study. The size of the heat exchangers plays a critical role in the dynamics performance and controllability of the ORC unit [42]. It is possible that the optimum design obtained in the present study is difficult to control in a safe manner. The controllability and safe operation of the ORC unit can be ensured by integrating the dynamic and control aspects in the preliminary design and optimization phase.

It needs to be added that some of the fluids considered in this work are flammable (e.g. pentane). In order to ensure the safe operation with a flammable working fluid on a truck, safety precautions need to be taken (e.g. the system need to be constructed by ATEX certified components and a leakage detection system may be installed). Such precautions may increase the total cost of the ORC unit.

Another important topic is the integration of the power produced by the ORC unit into the overall powertrain or driveline. The mechanical power from the ORC unit may be re-introduced into the crankshaft using a gearbox or other type of transmissions. Alternatively, the electric power from the ORC unit can be used to charge batteries, for auxiliary systems (air conditioning, cooling refrigerators, on board electrical devices) or re-introduced into the powertrain in diesel-hybrid vehicles configuration [43]. In the present study, the cost of the system integration is not considered and thus depending on the choice of system integration, the total cost of the ORC unit will increase.

5. Conclusions

A multi-objective optimization model of an organic Rankine cycle system was developed to analyze both the thermodynamic and the economic performances of several working fluids for the purpose of waste heat recovery from the exhaust gas of a diesel engine of a heavy-duty vehicle. The genetic algorithm was employed to obtain the Pareto optimal solutions in terms of maximum net power output and minimum cost, weight and volume of the organic Rankine cycle system. The off-design performance of a real driving cycle was evaluated for the organic Rankine cycle system.

In general, the results indicate that the weight, volume, cost and net power output of the organic Rankine cycle system increase with the evaporation temperature. Moreover, as expected, the net power output of the organic Rankine cycle system is higher when designed with a condensation temperature of 40 °C than designed with a condensation temperature of 60 °C. The maximum net power output, both in design and off-design conditions, is obtained for pentane as working fluid. For this design, the net power output of the organic Rankine cycle unit is 10.93 kW/8.3 kW in design/off-design (average) conditions, and the weight, volume, and cost of the organic Rankine cycle system are 129 kg, 1.077 m³, and 8527 €, respectively. When the condensation temperature decreases from 60 °C to 40 °C, the net power, weight, cost and volume of organic Rankine cycle unit with pentane as working fluid increases by 22 %, 12 %, 46 %, and 12 %, respectively. The bin approach for the off-design performance assessment reduced the computational time by 43% compared to considering all data points.

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Nomenclature

Parameters

A	Area, m ²	Pr	Prandtl number
Bd	Bond number	$\dot{Q}$	Heat transfer rate, W
Bo	Boiling number	Re	Reynolds number
C	Cost, €	S	Spacing, m
D	Diameter, m	SH	Degree of Superheat, °C
D_h	Hydraulic diameter, m	T	Temperature, °C
DC	Direct cost, €	TCS	The cost of system, €
g	Gravitational acceleration, m/s ²	TV	Total volume of system, m ³
G	Mass flux, kg/(m ² s)]	TW	Total mass of system, kg
IC	Indirect cost, €	t	Time, sec
k	Thermal conductivity, W/(m K)	U	Overall heat transfer coefficient, W/(m ² K)
L	Length, m	V	Volume, m ³
M	Mass, kg	$\dot{W}$	Power, W
$\dot{m}$	Mass flow rate, kg/s	We	Weber number
N	Angular speed, RPM	x	Vapour quality
Nu	Nusselt number	WD	Width of plate, m
P	Pressure, bar		

Greek letters

η	Efficiency, %	α	Heat transfer coefficient, W/(m ² K)
δ	Thickness, m	β	Chevron angle, °
ρ	Density, m ³ /kg	μ	Viscosity, Pa s

Abbreviations

$ATEX$	Appareils destinés à être utilisés en ATmosphères EXplosives	SF	Size factor
$BSFC$	Break specific fuel consumption	SCR	Selective catalytic reduction
EGR	Exhaust gas recirculation	$PPTD$	Pinch point temperature difference
EDR	Exergy destruction rate	ODP	Ozone depletion potential
GWP	Global warming potential		

Subscripts and superscripts

con	Condenser	in	Inlet
$cont$	Contingency	inj	Injection
$conn$	Connection	$intr$	Intermediate
ds	Design point	od	Off-design point
eq	Equivalent	ot	Outlet
eng	Engineering	pp	Pump
exp	Expander	ss	Heat sink
eva	Evaporator	sys	System
hs	Heat source	val	Valve
hx	Heat exchanger	wf	Working fluid
is	Isentropic	yr	Year

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Figures

Figure 1: Schematic of the organic Rankine cycle unit for waste heat recovery from the ICE.

Figure 2: T-s diagram of the selected working fluids.

Figure 3: Discretization scheme of heat exchanger model for a) evaporator and b) condenser.

Figure 4: Mass and volume plots of the expander and pump. The curves represent the correlations listed in Table 5, while the dots indicate data supplied by manufacturers/suppliers.

Figure 5: Flow chart of the optimization approach.

Figure 6: Pareto front of the a) net power vs mass, b) net power vs total cost, and c) net power vs volume of ORC system for Case 1 with pentane as the working fluid.

Figure 7: Variation of the net power output, weight, volume, and total cost of the ORC system with the evaporation temperature for pentane (Case 1: $T_{con} = 40$ °C, Case 2: $T_{con} = 60$ °C).

Figure 8: Mass flow rate and temperature of the exhaust gases after the SCR for typical long-haul operation.

Figure 9: Part-load map of net power output for Case 1 using pentane as the working fluid.

Tables

Table 1: Review of thermo-economic optimizations of ORC systems for WHR from ICEs.

Ref.	Objective functions		Decision variables
	Thermal	Economic	
[10]	Max η_{th}	Min TIC_{yr}	$W_{enge}, W_{pl,enge}, P_{eva}, P_{con}, \dot{m}_{wf}$
[11]	Max W_{net}	Min $TIC_{yr}, Min SIC_{yr}$	$P_{exp,out}, T_{exp,in}, \epsilon_{hx}, P_{exp,in}$
[12]	Max W_{net}	Min TIC	$P_{eva}, SH, T_{con}, T_{hs,out}$
[13]	Max $W_{net}, Max \eta_{th}$	n/a	$P_{con,in}, \eta_{pu,is}, \eta_{exp,is}, T_{hs,out}$
[14]	Max W_{net}	Min A_{hx}	$T_{eva}, T_{con}, T_{hs,out}$
[15]	Max η_{th}	n/a	$P_{eva}, SH, PPTD_{eva}$
[16]	Max W_{exp}	n/a	$P_{exp,in}, P_{exp,out}, T_{exp,in}, \dot{m}_{wf}, T_{con,out}, P_{pu,out}$
[17]	Max W_{net}	Min $BSFC$	$\tau_{engine}, N_{pu}, N_{exp}, t_{hs,val}, t_{inj}$
[18]	Max W_{net}	n/a	$T_{exp,in}, T_{con}, T_{eva}$
[19]	Max η_{exrg}	Min $\frac{A_{hx}}{W_{net}}$	T_{con}, T_{eva}
[20]	Max $\frac{W_{net}}{A_{hx}}, Min EDR$	n/a	$P_{eva}, T_{con}, T_{exp,in}$
[21]	Min $(SF)_{exp}$	Min $A_{hx}, Min TIC$	$T_{hs,out}, P_{exp,out}, SH, \dot{m}_{wf}, P_{pu,ou}$
[22]	Max W_{net}	Min $\frac{A_{hx}}{W_{net}}$	P_{eva}, T_{con}
[23]	Max $W_{net}, Min EDR$	n/a	P_{eva}, SH, P_{inter}
[24]	Min $V_{tube}, Min \Delta P_{hs}$	Min TIC_{yr}	$H_{fin}, S_{fin}, \delta_{fin}, \delta_{in,ts}$

Table 2: Engine data at 40 % load.

Engine		Exhaust gas conditions (after SCR)	
Speed (RPM)	Power (kW)	Mass flow rate (kg/h)	Temperature (°C)
1200	148	897	317

Table 3: Properties of the working fluids selected for the optimization.

Working fluid	Molar mass (kg/kmol)	Normal boiling point (°C)	Critical temperature (°C)	Critical pressure (bar)	NFPA 704 Classification			GWP 100 yrs	ODP
					H	F	I		
Ethanol	46	79	242	41	0	3	0	NA	0
MDM	237	153	291	14	0	2	0	NA	0
MM	162	100	246	19	1	4	0	NA	0
Pentane	72	36	197	34	1	4	0	7	0
R1233zd(E)	131	18	166	36	2	0	0	1	0.0003
R245fa	134	15	154	37	2	1	0	1030	0
RE347mcc	200	34	165	25	3	0	0	530	0

Table 4: Cost correlations for the ORC components [44].

Component	Dependent variable	Cost (€)
Expander	Rotor diameter (m)	$C_{exp} = 164338d_4^{2.08}$
Pump	Consumed power (W)	$C_{pp} = 150 \left(\frac{\dot{W}_p}{300} \right)^{0.25}$
Heat exchanger	Metal mass (kg)	$C_{hx} = 250 + 5M_{hx}$

Table 5: Correlations for the estimation of mass and volume of the ORC system [36–38].

Component	Correlation
Expander mass (kg)	$M_{exp} = 0.0455 (W_{exp})^2 - 0.0968 W_{exp} + 11.43$
Expander volume (m ³)	$V_{exp} = -0.00001 (W_{exp})^2 + 0.0018 W_{exp} + 0.0027$
Heat exchanger mass (kg)	$M_{hx} = M_{conn} + N_{pl} M_{pl}$
Heat exchanger volume (m ³)	$V_{hx} = L_{pl} W D_{pl} [N_{pl} (t_{pl} + S_{pl})]$
Pump mass (kg)	$M_{pp} = 0.0049 (\Delta P_{pp})^2 + 0.637 \Delta P_{pp} + 5.2$
Pump volume (m ³)	$V_{pp} = 0.000008 (\Delta P_{pp})^2 + 0.0012 \Delta P_{pp} + 0.0021$

Table 6: The optimized designs providing the highest net power output of the ORC system.

Case	Working fluid	Objective functions				Decision variables		
		Wn (kW)	TM (kg)	TV (m ³)	TCS (€)	Teva (°C)	SH (°C)	PPTDeva (°C)
1	Pentane	10.94	129	1.07	8527	167	12	13
	Ethanol	9.97	134	1.18	4054	191	13	12
	R1233zd(E)	9.68	122	0.93	6275	137	6	10
	MM	9.51	128	0.97	5118	196	3	11
	MDM	9.08	130	1.12	4455	205	4	9
	R245fa	8.94	118	0.84	5156	130	7	14
	RE347mcc	8.49	116	0.79	4361	139	15	16
2	Pentane	8.53	113	0.939	4552	173	8	10
	Ethanol	7.51	112	0.25	2470	197	11	10
	MM	7.45	109	0.36	3032	206	6	10
	R1233zd(E)	7.19	108	0.83	3296	142	8	9
	MDM	7.08	109	0.25	2714	217	7	8
	R245fa	6.48	105	0.83	2778	135	11	11
	RE347mcc	6.34	102	0.62	2581	144	12	14

Table 7: Average net power output of the ORC unit at the design and off-design conditions.

Case 1			Case 2		
Working fluid	Net power (kW)		Working fluid	Net power (kW)	
	Off-design	design		Off-design	design
Pentane	8.31	10.94	Pentane	7.04	8.53
R1233zd	7.55	9.68	R1233zd	6.04	7.19
Ethanol	7.08	9.97	Ethanol	5.92	7.51
R245fa	6.62	8.94	MM	5.71	7.45
RE347mcc	6.54	8.49	R245fa	5.25	6.48
MM	6.47	9.51	RE347mcc	5.28	6.34
MDM	5.81	9.08	MDM	5.22	7.08