A novel design method of organic Rankine cycle system harvesting waste heat of heavy-duty trucks based on off-design performance

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Abstract
The organic Rankine cycle (ORC) system can effectively recover waste heat from engines of heavy-duty trucks, and is a promising method to improve the efficiency of on-board engines. However, engine operating conditions fluctuate greatly while driving, the waste heat recovery system must often work under off-design conditions, which significantly affects system performance. Further, different component structures can also affect the off-design performance of the system. Thus, a novel design method of preheating organic Rankine cycle (P-ORC) system harvesting waste heat of heavy-duty trucks based on off-design performance is proposed in this study. The design method includes selection of the optimal types of components and design point to optimize the comprehensive performance of the waste heat recovery system in all road conditions. In this study, different heat exchanger combinations are applied to the P-ORC system to obtain six different design systems. According to the principle of uniform coverage, the scatter diagram of exhaust temperature and mass flow rate of the engine under real road conditions is discretized into 19 alternative design points. Each system is designed with 19 discretized design points, and a total of 114 design systems are obtained. The optimal heat exchanger combination and design point are selected based on the off-design performance. It is concluded that a P-ORC system using a combination of plate preheater, finned tube air cooler, and shell-tube evaporator is the optimal system. The optimal design point number is 10, and the corresponding engine speed at $DP_{10}$ is 1471 rpm, the engine torque is 474 Nm, the occurrence probability is 14.48%, the exhaust temperature is 350°C, the exhaust mass flow rate is 0.11 kg/s, and the maximum combined net power output is 4.26 kW. The results reveal that the optimal design point of the system can be selected at the design point with medium engine load and high occurrence probability. It guides the system design toward a more practical direction, so as to obtain an optimal system that could operate efficiently and recover more waste heat under the full working conditions of the engines. This novel design method can be extended for other cycle configurations.
1 | INTRODUCTION

The continuous development of human society has resulted in increasing energy demands and significant environmental problems. The road transport sector, with the internal combustion engine (ICE) as its main source of power, is a key driver of economic activity and energy demand, accounting for 42.2% of global oil consumption.\(^1\) Heavy-duty trucks (HDVs) are the second largest source of global oil consumption in road transport. The high dependence of HDVs on oil poses environmental concerns; HDVs account for 7% of carbon dioxide emissions for the entire industry.\(^1\) Thus, improving the fuel consumption of ICEs in HDVs is a significant factor in alleviating the pressure of energy demand, and the international community has reached a consensus on the need for improvements in the fuel consumption performance of ICEs.

According to the research of Fu et al\(^2\) and Shu et al,\(^3\) only 30%-45% of ICE fuel energy is converted into effective work; most of the remaining energy is dissipated in the form of residual heat. Therefore, waste heat recovery is an effective means to improve the fuel efficiency of ICEs. A technical report published by Argonne National Laboratory indicated that waste heat recovery technology exhibits the greatest potential for ICE performance improvement among various energy-saving technologies.\(^4\) It is also pointed out by the\(^5,6\) that the organic Rankine cycle (ORC) system can effectively recover and utilize the waste heat of exhaust and jacket water from the engine, and is a promising scheme to improve the efficiency of on-board engines. In recent years, ORC system has been widely studied for its advantages such as simple structure, high thermal efficiency, easy integration and environmental friendliness. There have been reviews focusing on ORCs for ICE exhaust heat recovery. Sprouse III et al\(^7\) presented the history of ORC technology for ICEs since the 1970s. This paper focused on the expander and working fluid selection. Chintala et al\(^8\) focused on compression ignition engines and reviewed major research advances in ORCs with respect to working fluids, expanders, heat exchangers, back pressure, and performance analysis. Shi et al\(^9\) reviewed the improved waste heat recovery systems of ICEs, which mainly include the studies focusing on modified ORCs to achieve a better performance from the aspects of cycle and fluid during the past decade. Lion et al\(^10\) provided an overview of ORC to utilize various waste heat sources from ICEs in HDVs, paying particular attention to the multiple engine operation profiles for various typical vehicle applications. Xu et al\(^11\) provided a preliminary introduction to ORC in heavy-duty diesel engines, focusing on the system architecture evaluation, heat exchanger and expander selection, working fluid selection, power optimization, control strategy evaluation, simulation and experimental work overview, and limiting factors.

Research focusing on the ORCs as a promising waste heat recovery technology of heavy-duty diesel engines has been increasing continuously in the last decade. One of the major issues concerns on the optimization of the system design methodology.\(^12-16\) Macian et al\(^17\) proposed an optimized design method for the ORC system for vehicle waste heat recovery considering the selection of different working fluids and heat sources; the thermodynamic optimization of the cycle system was also studied. Guillaume et al\(^18\) proposed a comprehensive design method for a waste heat recovery system for diesel engines with cycle structure selection, expander selection, and cycle working fluid selection. Denny et al\(^19\) proposed a new method to optimize the ORC system design; in the ORC design stage, the selection of main components and the design process were conducted simultaneously. Simone et al\(^20\) discussed a comprehensive design methodology for optimization of the ORC system considering a wide range of design variables with component limitations and costs. The design process comprises three steps: heat source selection, working fluid selection, and thermodynamic cycle optimization.

All these heuristic studies regarding optimization of the system design methodology contribute to the development of the ORC systems in waste heat recovery of heavy-duty diesel engines. However, a common characteristic of the aforementioned research is that the ICE waste heat recovery system was designed with a specific heat source condition, normally at the rated engine condition. In other words, off-design performance of the design system was not considered. While the waste heat recovery system is a passive system of the internal combustion engine, its performance depends greatly on the characteristics of the heat source. However, the working conditions of the ICE and the corresponding waste heat source change frequently and suddenly. When the heat source fluctuates, the system is forced to operate under off-design conditions, which may lead to the deterioration of system performance and decreased component efficiency. Experimental studies\(^21\) have shown that the exhaust temperature of a heavy diesel engine under large load conditions can be more than 300 K higher than under small load conditions, and the exhaust mass flow rate is more than 3.6 times higher than under small load conditions. In practice,
the waste heat recovery system of an ICE operates far from its design condition, owing to changes in engine load, which results in performance degradation or even system failure. The research results of Usman indicate that off-design operation of the waste heat recovery system is the main reason for its performance deterioration. Previous studies indicated that for ICE waste heat recovery systems, the thermal efficiency, maximum net power output, and exergy efficiency all decrease linearly with decreasing engine load. Xie et al. simulated an ORC system under a driving cycle. The results showed that the thermal efficiency of an ORC system varies significantly with different operating conditions, and due to fluctuations of the heat source, the thermal efficiency of the system is only 3.63%, less than half of the designed 7.77%. Ringler et al. reported that a 51% net power decrease could be observed when the engine load decreases from 100% to 50%. Thus, selection of the optimal design point for the waste heat recovery system under the fluctuating heat source of the engine is particularly necessary, and requires a system design method integrating off-design performance in the design stage.

Further, the design method should also include the selection of main components based on off-design performance. There is evidence suggesting that heat exchanger architecture also affects the off-design performance. Chatzopoulou et al. analyzed the off-design performance of waste heat recovery systems adopting two different heat exchanger types: plate type and shell-tube type. They concluded that systems with different types of heat exchangers have different off-design performance. However, current design method research rarely involves the selection of components based on off-design performance, especially the selection analysis of heat exchangers. Mavridou et al. proposed a method for the selection and analysis of diesel engine exhaust heat exchangers for truck applications, including a comparison of traditional and strengthened heat exchangers. Zhang et al. selected three different types of heat exchangers including a plate heat exchanger, a shell-and-tube heat exchanger, and a finned tube heat exchanger, as the evaporator and condenser of an ORC system; a thermo-economic evaluation and a comparison of different ORC configurations were presented. The comparative analysis of different types of heat exchangers in these studies is based on the design performance; these studies do not consider differences in the off-design performance of the system caused by the selection of heat exchangers. Few existing studies have analyzed the selection of heat exchangers based on the off-design performance of the ORC system under all road conditions.

The main objective of this paper is to integrate system off-design performance during the system design phase and reduce the adverse effects of off-design working conditions on the waste heat recovery system. In this work, a novel comprehensive design method of an ORC waste heat recovery system for ICEs in HDVs based on off-design performance is proposed to optimize the selection of different forms of heat exchanger combinations and design points, so as to obtain an optimal system that could operate efficiently and recover more waste heat even when it faces the possible variations of the heat sources.

The proposed method discretizes the scatter diagram of exhaust temperature and mass flow rate of the ICE under real road conditions into 19 alternative design points according to the principle of uniform coverage. Different types of heat exchangers are combined to obtain six different types of waste heat recovery systems. For these six systems, 19 discretized alternative design points are used for system design, and a total of 114 design systems are obtained. The combined net power output of the waste heat recovery system is used as the evaluation index. Considering the pressure drop and weight of the heat exchangers, the occurrence probability of each alternative design point, the applicability of off-design points under the design system, the energy utilization efficiency, and other factors, the optimal heat exchanger combination and design point are selected based on the off-design performance. The design method proposed in this study considers the off-design performance in the design stage of the waste heat recovery system, including the selection of optimal components and design point, which can guide the system design in a more practical direction.

TABLE 1 Specifications of the target engine

| Item                | Parameter | Unit |
|---------------------|-----------|------|
| Cylinder number     | 4         | —    |
| Cylinder bore       | 110       | mm   |
| Stroke              | 135       | mm   |
| Rated speed         | 2200      | rpm  |
| Max. speed          | 2300      | rpm  |
| Rated power         | 169       | kW   |
| Displacement        | 5.13      | L    |

2 | SYSTEM DESCRIPTION

This section introduces the relevant specifications of the target engine used in the heavy-duty truck, and the description of the cycle structure and cycle process of the preheating organic Rankine cycle (P-ORC) system.

2.1 | Engine introduction

The internal combustion engine (ICE) in this study is a four-cylinder four-stroke engine used in a heavy-duty truck. The main parameters of the engine are listed in Table 1.
2.2 | Organic Rankine cycle system

Figure 1 and Figure 2 show the structural schematic and T-s diagram of the P-ORC system, respectively. In Figure 1, the purple circuit represents the jacket water source, the red circuit represents the engine exhaust heat source, the green circuit represents the circulating working fluid, and the blue circuit represents the cooling source. In Figure 2, the solid blue line represents the P-ORC system, the dashed purple line represents the jacket water source, the dashed red line represents the engine exhaust heat source, and the dashed blue line represents the cooling source. R245fa is selected as the working fluid because it is nonflammable, nontoxic, and has relatively low environmental impact (low ODP and low GWP).

The preheater and evaporator transfer the engine jacket water heat and exhaust waste heat to the working fluid (state: 2-30-3-4-5). The working fluid absorbs the heat, becomes a superheated vapor, and enters the expander; mechanical energy is produced in the expander during superheated vapor expansion (state: 5-6). The exhaust at the outlet of the expander is condensed by the cooling air in the condenser (state: 6-7-1) and is pumped back to the preheater (state: 1-2) for the next cycle process.

3 | DESCRIPTION OF DESIGN METHOD

This section introduces the specific design method and operation process of the design system and the off-design system. The main component modeling of the P-ORC system is described, including the design model and off-design model of each heat exchanger, pump, and expander. MATLAB 2018a software was used for modeling.

The novel design method selects the optimal heat exchanger combination and system design point based on off-design performance. The P-ORC design process based on off-design is shown in Figure 3. Six heat exchanger combinations are used in six waste heat recovery systems, defined as design systems $DS_i$ ($i = 1, 2, 3, 4, 5, 6$). Each design system is used to conduct off-design performance analysis. For example, in $DS_1$ (defined as a design system with a PFS heat exchanger combination, where PFS represents the combination of plate preheater, finned tube air cooler, and shell-tube evaporator), the first of 19 alternative design points $DP_1$ is used for system design, based mainly on the area of each heat exchanger. The remaining 18 alternative design points are taken as off-design points, and the off-design performance of each under the design system is calculated and analyzed. The other alternative design points $DP_n$ ($DP_2, DP_3, ..., DP_{19}$) are each used as design points, with the remaining 18 alternative design points used as off-design points for comprehensive performance analysis; the comprehensive performance of each off-design system under $DS_i$ is obtained. The performance calculations for the remaining design systems $DS_i$ ($i = 2, 3, 4, 5, 6$) are obtained using the same process. This study uses $W_{net,i}$ as the performance evaluation index of each design system and uses $W_{net}$ as the final performance evaluation index.

The calculation of $W_{net}$ comprehensively considers the influence of heat exchanger weight and pressure drop. The specific model is as follows:

$$W_{net} = \max(W_{net,i})$$

$$W_{net,i} = \max(W_{net,n})$$

$$W_{net,n} = \sum_{m=1}^{19} (W_{net,m} \cdot P_m)$$
where $P_m$ is the occurrence probability of each alternative design point:

$$W_{net,m} = W_{rec,m} - W_{\Delta P,m} - W_{\Delta WT,m}$$  \hspace{1cm} (4)$$

where $W_{\Delta P,m}$ and $W_{\Delta WT,m}$ represent the power loss caused by pressure drop and weight, respectively:

$$W_{\Delta P,m} = \Delta P_m \cdot A \cdot v$$  \hspace{1cm} (5)$$
where $A$ and $v$ are the cross-sectional area and the mass flow rate of the working fluid at the heat exchanger inlet, respectively;

$$W_{WT,m} = \frac{\Delta W_T}{100} \cdot \sigma \cdot \rho \cdot q \cdot \frac{\eta_s}{3600}$$  \hspace{1cm} (6)

where $\rho$ is the density of fuel ($\rho = 0.725$ kg/L), $q$ is the calorific value of fuel ($q = 0.46 \times 10^7$ J/kg), $\eta_s$ is the power conversion efficiency of fuel oil ($\eta_s = 40\%$), $\sigma$ is the fuel consumption caused per 100 kg weight increase, which is 0.7 L in this study. $^3$$^5$

### 3.1 Modeling methodology of design system

The specific operation process of the design system is shown in Figure 4. The heat transfer parameters of each heat exchanger are obtained according to the heat source parameters of the design point, and the heat exchangers are sized to achieve the minimum heat transfer areas within the allowable maximum pressure drop.

For the preheater in the design system, models of the plate preheater, shell-tube preheater, and double-pipe preheater are established in this study; for the evaporator in the design system, models of the shell-tube evaporator and double-pipe evaporator are established. The main correlations of different evaporator types are shown in Table 2; the main correlations of different preheater types are shown in Table 3. Further information on the heat exchanger calculations can be found in the Appendix.

Because the purpose of this study is to apply the waste heat recovery system to heavy-duty trucks, a finned tube air cooler is selected as the condenser; the condenser model refers to the finned tube air cooler model in $^4$$^4$.
In this study, a scroll expander is selected and a steady-state model is established. The total output expansion work $W_{\text{exp}}$ is expressed as the product of isentropic expansion output work and efficiency $\eta_{\text{exp}}$, as shown in Equation (7).

$$W_{\text{exp}} = m (h_5 - h_6) \eta_{\text{exp}} \quad (7)$$

The relationship between a given expander's rotational speed and the relevant working fluid's mass flow rate is given by Equation (8).\(^4\)

$$m = \frac{\text{FF} \cdot \rho_{\text{exp,in}} \cdot V_s \cdot N_{\text{exp}}}{60} \quad (8)$$

where $\text{FF}$ is the filling factor, which has a physical meaning equivalent to the volumetric efficiency in a compressor, and $V_s$ is the volume of the expander's suction chamber (cm$^3$).

According to the relevant parameters of the expander provided by the manufacturer, the $\text{FF}$ and $V_s$ values are 3.25 and 73, respectively. The design efficiency of the expander $\eta_{\text{exp}}$ is 0.7.

A rotary vane pump is selected and a conservative isentropic efficiency of 0.7 is assumed for all systems at the design stage. Although the pump efficiency is a critical parameter that influences cycle performance, we consider a reasonable efficiency that can be realized with current technology.

### 3.2 Modeling methodology of off-design system

In the calculation of the off-design model, the mass flow rate and temperature of the heat source vary with the off-design point. Figure 5 presents the flowchart for the off-design simulation modeling. The heat transfer areas of the heat exchangers in the off-design model are given by the design value. When the design system is running at off-design points, the parameters of each state point are calculated according to the flowchart in Figure 5. The heat transfer areas should be equal to the designed areas within ±1%, which is considered feasible. If the error is greater than ±1%, the heat transfer area required by the off-design point is too different from the designed heat transfer area; that is, the available energy of the off-design point is not enough for the design system to operate normally.

In the design model and the off-design model, the heat transfer correlations for each heat exchanger are the same. The difference is that the heat transfer area of the heat exchanger in the off-design model is fixed, and can be calculated by the design model. Regarding the evaporator, the exhaust entering the heat exchanger is adjustable, which means the exhaust can be bypassed. The total heat exchange of the system can be reduced by decreasing the exhaust gas mass flow rate through the evaporator, allowing normal system operation when the designed heat transfer area is small. For a large designed heat transfer area, some off-design points are infeasible because the exhaust energy is not sufficient to meet the heat exchange requirements. The convergence condition of the off-design model requires that the error between the heat exchange area required by the design system and the designed area at off-design points is maintained within ±1%.

The isentropic efficiency of the expander at off-design points is related mainly to the expander's rotational speed $N_{\text{exp}}$, the inlet pressure of the working fluid $P_{\text{exp,in}}$, and the expansion ratio $r_p$. In the actual working process, to ensure the frequency of power generation is stable, the expander's rotational speed should be maintained at a constant value for as long as possible. The inlet pressure of the working fluid is correlated with the expansion ratio when the cold source temperature is determined. The isentropic

### TABLE 2 Heat transfer and pressure drop correlations of evaporator

|                      | Shell-tube evaporator | Double-pipe evaporator |
|----------------------|-----------------------|------------------------|
| **Fluid side**       |                       |                        |
| Heat transfer coefficient (single-phase) | 36 | 40 |
| Heat transfer coefficient (two-phase) | 37 | 41,42 |
| Pressure drop (single-phase) | 38 | 41 |
| Pressure drop (two-phase) | 25.38 | 41 |
| **Gas side**         |                       |                        |
| Heat transfer coefficient | 25.39 | 40 |
| Pressure drop         |                       | 41                     |

### TABLE 3 Heat transfer and pressure drop correlations of preheater

|                      | Plate preheater | Shell-tube preheater | Double-pipe preheater |
|----------------------|----------------|----------------------|-----------------------|
| **Fluid side**       |                |                      |                       |
| Heat transfer coefficient | 43  | 36 | 40 |
| Pressure drop         | 38 | 41 |                |
| **Gas side**         |                |                      |                       |
| Heat transfer coefficient | 43  | 25.39 | 40 |
| Pressure drop         |                | 41 |                |
The efficiency of the pump at off-design points is determined by a third-degree polynomial of the ratio of the inlet volumetric flow with respect to the design point. The maximum isentropic efficiency of the pump is set to 0.7, and is assumed to occur at the design points. The coefficients of the polynomial are fitted according to the performance curve of a commercial pump; further information is available in Appendix D.

4 | CASE STUDY

4.1 | Heat exchanger combination definition

For different types of heat exchangers, the heat transfer effect, weight, and pressure drop have certain differences. The difference in heat transfer effect is directly reflected in the difference in net power output. To facilitate comparison and selection, the influence of heat exchanger weight and pressure drop on the cycle system is also transformed into the influence on net power output. As for the influence of vehicle
weight on fuel consumption, Reynolds et al.\textsuperscript{47} reported that for every 100 kg increase in vehicle weight, per 100 km fuel consumption increases by 0.7 L. This study converts the additional fuel consumption due to weight into effective power loss, as shown in Equation (6). As for the influence of pressure drop on power loss, the power loss model proposed in reference\textsuperscript{43} is used for calculation and analysis in this study, as represented in Equation (5).

To select the best heat exchanger combination, the plate preheater, shell-tube preheater, and double-pipe preheater are each combined with the shell-tube evaporator and double-pipe evaporator in this study. A finned tube air cooler is used as the condenser. In this way, six different heat exchanger combinations are obtained, denoted as PFS, SFS, DFS, PFD, SFD, and DFD. Six different design systems are obtained by applying the heat exchanger combinations to the P-ORC system, denoted as $\text{DS}_1$: P-ORC-PFS, $\text{DS}_2$: P-ORC-SFS, $\text{DS}_3$: P-ORC-DFS, $\text{DS}_4$: P-ORC-PFD, $\text{DS}_5$: P-ORC-SFD, and $\text{DS}_6$: P-ORC-DFD, as shown in Figure 6.

### 4.2 Design point definition

The operating conditions of the target engine in this study are determined according to actual road conditions for heavy-duty trucks. Using standard highway conditions for heavy-duty trucks, the scatter diagram of exhaust temperature and mass flow rate is shown in Figure 7; the scatter diagram of engine speed and torque is shown in Figure 8. Figure 7 and Figure 8 are the simulation results of the existing dynamic simulation model of heavy-duty trucks under standard highway conditions from our research team. The simulation speed of the model matches well with the actual target speed.

To find the optimal design point of the waste heat recovery system under the working conditions of the engine, the scatter diagram of exhaust temperature and mass flow rate of the engine under whole working conditions (black dots) is discretized into 19 alternative design points (red dots), as shown in Figure 7 by the principle of uniform coverage; the corresponding scatter diagram of engine speed and torque is shown in Figure 8. For the convenience of subsequent description, the 19 alternative design points are defined as $\text{DP}_1$.
The occurrence probability of alternative design points is obtained according to the proportion of working conditions around each alternative design point (exhaust temperature range of ±10°C, exhaust flow range ±0.01 kg/s) to the working conditions on the whole surface. The specific parameters of each alternative design point are shown in Table 4 and Figure 9.

4.3 | Results and discussion

4.3.1 | Selection of the optimal heat exchanger combination

$DS_1, DS_2, DS_3, DS_4, DS_5,$ and $DS_6$ represent the six different design systems described in Section 4.1: their performance under off-design points are analyzed under the whole road conditions of the target engine. Each system is designed with 19 discretized design points; a total of 114 design systems are obtained. The off-design performance is calculated and analyzed for each design system, and the comprehensive performance of each system is obtained. The combined net power output $W_{net}$ is used as the evaluation index for comparing different design systems. As shown in Figure 10, the trend of the combined net power output of the six design systems at different design points is similar. On the whole, the combined net power output of the P-ORC-PFS system is the best in all design conditions, and the combined net power output of the P-ORC-DFD system is the worst in all design conditions. Therefore, the optimal heat exchanger combination selected in this study is the PFS combination.

The difference between heat exchanger combinations is caused mainly by differences in weight and pressure drop of different heat exchangers. As shown in Figure 11, the additional power loss of the system is affected mainly by weight and pressure drop loss. On the whole, the power loss caused by the weight of each heat exchanger is larger than the power loss caused by pressure drop. In comparing different types of evaporators, it is observed that the weight of the double-pipe evaporator is significantly greater than that of the shell-tube evaporator, while the power loss caused by the pressure drop is smaller. It is concluded that the combined effect of the shell-tube evaporator is better than that of the double-pipe evaporator. Comparing different preheaters, the combined weight and pressure drop of the plate preheater have less influence on the system power loss than the other two preheaters; the plate preheater is the optimal selection.

4.3.2 | Selection of the optimal design point

In this study, the error between the heat transfer area required by the off-design system and the heat transfer area of the

| Alternative design point number | Exhaust temperature/°C | Exhaust mass flow rate/kg/s | Engine speed/(rpm) | Engine torque/(Nm) | Occurrence probability/(%) |
|---------------------------------|--------------------------|-----------------------------|---------------------|---------------------|-----------------------------|
| DP1                             | 290                      | 0.05                        | 1257                | 205                 | 0.35                        |
| DP2                             | 290                      | 0.07                        | 1477                | 168                 | 1.06                        |
| DP 3                            | 310                      | 0.07                        | 1401                | 238                 | 3.66                        |
| DP 4                            | 330                      | 0.07                        | 1258                | 315                 | 0.91                        |
| DP 5                            | 310                      | 0.09                        | 1510                | 288                 | 4.72                        |
| DP 6                            | 330                      | 0.09                        | 1441                | 351                 | 7.21                        |
| DP 7                            | 350                      | 0.09                        | 1229                | 450                 | 1.73                        |
| DP 8                            | 370                      | 0.09                        | 1116                | 539                 | 0.53                        |
| DP 9                            | 330                      | 0.11                        | 1525                | 408                 | 9.23                        |
| DP 10                           | 350                      | 0.11                        | 1471                | 474                 | 14.48                       |
| DP 11                           | 370                      | 0.11                        | 1378                | 551                 | 2.17                        |
| DP 12                           | 390                      | 0.11                        | 1111                | 731                 | 0.41                        |
| DP 13                           | 350                      | 0.13                        | 1623                | 494                 | 9.14                        |
| DP 14                           | 370                      | 0.13                        | 1561                | 556                 | 10.81                       |
| DP 15                           | 390                      | 0.13                        | 1384                | 692                 | 2.55                        |
| DP 16                           | 370                      | 0.15                        | 1671                | 572                 | 7.09                        |
| DP 17                           | 390                      | 0.15                        | 1653                | 618                 | 11.93                       |
| DP 18                           | 410                      | 0.15                        | 1615                | 681                 | 8.97                        |
| DP 19                           | 410                      | 0.17                        | 1693                | 690                 | 3.05                        |
The design system is set to be within ±1% to satisfy the system constraints. The off-design points beyond this range cannot operate normally under this design system. Figure 12 shows the applicability of off-design points under typical design systems. As the available energy of design points increases, the applicability of off-design points in the design system decreases; the applicability of off-design points is better in design systems with less available energy.

A design point with a smaller number has a smaller exhaust temperature and mass flow rate, and thus less available energy; a design point with a larger number contains more available energy. Infeasible off-design points are caused mainly by design heat transfer areas of the heat exchanger that are too large, resulting in continuous heat transfer between the working fluid and the heat source during the recovery of waste heat. As a result, the outlet temperature of the heat source in the evaporator is too low. When the outlet temperature is lower than the acid dew point, a certain degree of corrosion results in the evaporator, which is not conducive to safe operation of the waste heat recovery system. In addition, some of the off-design points contain too little available energy, insufficient to overheat the working fluid, resulting in the working fluid at the inlet of the expander not overheating.
which has a negative impact on the life of the expander. Thus, in selecting the best design point, the applicability of off-design points under the design system is an important factor.

However, design points with less available energy do not correspond to better off-design performance. The energy utilization efficiency of off-design points in different design systems should be considered. As shown in Figure 13, a lower design point number (e.g., DP1, DP3) corresponds to lower energy utilization efficiency in the corresponding system for off-design points; a larger design point number (e.g., DP17, DP19) corresponds to higher energy utilization efficiency in the corresponding system for off-design points. When the system is designed at design points with less available energy, the performance of off-design points with more available energy is limited due to the lack of heat transfer capacity of the design system, and only part of the available energy can be recovered.

Through the analysis, a contradiction is found between the applicability of off-design points and the efficiency of energy utilization under the design system. Although a design system with less available energy has better applicability of off-design points, the energy utilization efficiency of off-design points is lower in a design system with less available energy. The double restriction of these two factors should be considered when selecting the optimal design point of the system.

This study uses $W_{\text{net}}$ as the final performance evaluation index, and Equation (3) indicates that the occurrence probability of each alternative design point $P_m$ affects the selection of the optimal design point. The occurrence probability distribution of each alternative design point is shown in Figure 14. Considering the factors affecting the net power output of the system, the combined net power output of the P-ORC-PFS system is shown in Figure 15. It is observed in Figure 15 that the design system corresponding to alternative design point 10 has the maximum combined net power output of 4.26 kW in whole road working conditions, thus $DP_{10}$ is selected as the optimal system design point. The occurrence probability at $DP_{10}$ is 14.48%, the exhaust temperature is 350°C, and the exhaust mass flow rate is 0.11 kg/s. The corresponding engine speed is 1471 rpm and the torque is 474 Nm, according to Table 4, indicating a medium load condition.

5 | CONCLUSIONS

This study proposes a comprehensive design method of organic Rankine cycle system harvesting waste heat of heavy-duty trucks based on off-design performance. This novel...
Design method not only incorporates the optimization of component combinations but also integrates the off-design performance evaluation at the design phase, which allows simultaneous selection of the optimal types of heat exchangers and optimal design point. The proposed method improves the reliability and robustness of the waste heat recovery system design, guiding the system design toward a more practical direction. The main conclusions are as follows:

1. Different heat exchanger types will greatly influence the comprehensive benefit of ORC system. The P-ORC-PFS system, combination of plate preheater, finned tube air cooler, and shell-tube evaporator, at different design points demonstrates the best comprehensive performance.

2. Considering the off-design performance integrated at the design phase, a condition with medium engine load is recommended for the system design to achieve more benefit. Taking $W_{\text{net}}$ as the evaluation index, $DP_{10}$ is selected as the best system design point. The corresponding engine speed is 1471 rpm, the engine torque is 474 Nm, the exhaust temperature is 350°C, the exhaust mass flow rate is 0.11 kg/s, and the maximum benefit $W_{\text{net}}$ is 4.26 kW.

3. The probability of occurrence of the engine conditions significantly influences the design point selection. The best design point moves toward the engine conditions of greater weight. Specifically, the design point with a 14.48% probability of occurrence in current work is selected as the optimal design point.

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NOMENCLATURE

| Symbol | Definition |
|--------|------------|
| $A$    | heat transfer area ($m^2$) |
| $D$    | diameter (m) |
| $m$    | mass flow rate (kg/s) |
| $P$    | pressure (Pa) |
| $\Delta P$ | pressure drop (Pa) |
| $Q$    | heat capacity (W) |
| $T$    | temperature (K) |
| $W$    | power output (W) |
| $x$    | dryness |
| $\rho$ | density ($kg/m^3$) |
| $\eta$ | efficiency |

**Subscripts**

- cond: condenser
- design: design condition
- evap: evaporator
- exp: expander
- i: inside
- in: inlet
- lim: limit
- net: net
- o: outside
- off: off-design condition
- out: outlet
- pp: pinch point
- pre: preheater
- th: thermal
- wf: working fluid
- wt: weight

**Abbreviations**

- Cair: cooling air; DP: design point; DS: design system; Exh: exhaust gas; GWP: global warming potential; HDV: heavy-duty truck; HEX: heat exchanger; ICE: internal combustion engine; JW: jacket water; ODP: ozone depression potential; OP: off-design point; ORC: organic Rankine cycle; P-ORC: preheating organic Rankine cycle; PFS: combination of plate preheater, finned tube air cooler, and shell-tube evaporator; SFS: combination of shell-tube preheater, finned tube air cooler, and shell-tube evaporator; DFS: combination of double-pipe preheater, finned tube air cooler, and shell-tube evaporator; PFD: combination of plate preheater, finned tube air cooler, and double-pipe evaporator; SFD: combination of shell-tube preheater, finned tube air cooler, and double-pipe evaporator; DFD: combination of double-pipe preheater, finned tube air cooler, and double-pipe evaporator.

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APPENDIX A.

Shell-tube evaporator modeling

In the shell-tube evaporator, the working fluid (R245fa) flows on the tube side, and the high-temperature exhaust flows on the shell side.

The total heat transfer coefficient is calculated by:

\[
U = \frac{1}{\frac{1}{a_{cf}} + \frac{d_{wall}}{\lambda} + \frac{1}{a_{h}}} \] \(\text{A.1}\)

For the cases where no phase change occurs, the convective heat transfer coefficient in the single-phase zone on the tube side is calculated by the Petukhov and Popov correlation\(^{36}\):

\[
f = (0.782 \cdot \lg \text{Re}^{-1.51})^{-2} \] \(\text{A.2}\)

\[
Nu = \frac{\left(\frac{\dot{m}}{\dot{m}_h}\right) \cdot \text{Re} \cdot Pr}{12.7 \cdot \left(\frac{\dot{m}}{\dot{m}_h}\right) \cdot \left(Pr^{\frac{2}{3}} - 1\right) + 1.07} \] \(\text{A.3}\)

\[
h = \frac{Nu \cdot k}{d_i} \] \(\text{A.4}\)

When the working fluid boils in the evaporator, the convection heat transfer coefficient in the two-phase zone on the tube side is calculated by the Liu and Winterton correlation\(^{37}\):

\[
h = \sqrt{(Fh_l)^2 + (Sh_{pool})^2} \] \(\text{A.5}\)

\[
F = \left[1 + xPr \left(\frac{Pr}{Pr} - 1\right)\right]^{0.35} \] \(\text{A.6}\)

\[
S = \frac{1}{\left(1 + 0.055F^{0.1}Re^{0.16}\right)^{0.25}} \] \(\text{A.7}\)

\[
h_l = 0.023 \left(\frac{\dot{m}}{d_i}\right) \text{Re}^{0.8} \text{Pr}^{0.4} \] \(\text{A.8}\)

\[
h_{pool} = 55 \text{Pr}^{0.12} q^{0.5} (-lgPr)^{-0.55} \] \(\text{A.9}\)

where \(F, h_p, S\) and \(h_{pool}\) represent the forced convective heat transfer enhancement factor, the liquid heat transfer coefficient, the suppression factor and pool boiling heat transfer coefficient.

The pressure drop in single-phase zone on the tube side is calculated by\(^{38}\):

\[
\Delta P_{tube} = \Delta P_a + \Delta P_N \] \(\text{A.10}\)

\[
\Delta P_a = \frac{2f_p \cdot G^2 \cdot L}{\rho \cdot D_{eq}} \] \(\text{A.11}\)

\[
\Delta P_N = 1.5 \cdot \frac{\rho v^2}{2} \] \(\text{A.12}\)

\[
\text{Re} < 10^5, \quad f_p = \frac{0.316}{\text{Re}^{0.25}} \] \(\text{A.13}\)

\[
\text{Re} \geq 10^5, \quad f_p = \frac{1}{\left[0.79 \cdot \lg(\text{Re}) - 0.64\right]^2} \] \(\text{A.14}\)

The pressure drop in the two-phase zone on the pipe side is calculated by:

\[
\Delta P_{t,lp} = \phi \cdot \Delta P_{t,lp} \] \(\text{A.15}\)

\[
\Delta P_{u,lp} = \phi \cdot \Delta P_{u,lp} \] \(\text{A.16}\)
\[
\phi = 1 + \frac{12}{X} + \frac{1}{X^2} [\langle label \rangle (A.14) < /label > (A.14)
\]

\[
X = 18.65 \left( \frac{\rho_s}{\rho_f} \right)^{0.5} \left( \frac{1-x}{x} \right) \left( \frac{\text{Re}_{\text{ib}}^{0.1}}{\text{Re}_{\text{in}}^{0.5}} \right) [\langle label \rangle (A.15) < /label > (A.15)
\]

In the shell-tube evaporator, the fluid on the shell side is the engine exhaust without phase change. When calculating the shell side heat transfer coefficient, the Delaware-Bell method is usually adopted\textsuperscript{38,39} which adopts correction factor to modify the transfer factor of ideal tube bundle.

\[
h = j_H \cdot j_\gamma \cdot j_b \cdot G \cdot C_p \cdot \left( \frac{\mu_f}{\mu_{\text{wall}}} \right) [\langle label \rangle (A.16) < /label > (A.16)
\]

\[
\begin{align*}
\Delta P_{\text{ib}} &= 4 \cdot f_{\text{ib}} \cdot \frac{m^2 \cdot N_c}{2 A_r \cdot \rho} [\langle label \rangle (A.17) < /label > (A.17) \\
\Delta P_{\text{sk}} &= \frac{m^2}{2 \cdot A_r} \left( 2 + 0.6 \cdot N_c \right) [\langle label \rangle (A.18) < /label > (A.18) \\
\Delta P_{\text{shel}} &= \left( N_c - 1 \right) \cdot \Delta P_{\text{ib}} \cdot R_b + N_b \cdot \Delta P_{\text{sk}} + R_f \left( 1 + \frac{N_c}{N_c} \right) [\langle label \rangle (A.19) < /label > (A.19)
\end{align*}
\]

**APPENDIX B.**

**Double-pipe evaporator modeling**

In the calculation of double-pipe evaporator, it is divided into preheating zone, two-phase zone and overheat zone. Dittus-Boelter\textsuperscript{25} correlation was adopted for single-phase flow of working fluid:

\[
\text{Nu}_{\text{ib}} = 0.023 \cdot \frac{\text{Re}_{\text{ib}}^{0.8} \cdot \text{Pr}_{\text{ib}}^{0.4}}{\text{Pr}_{\text{ib}}} [\langle label \rangle (B.1) < /label > (B.1)
\]

Similarly for engine exhaust heat source (single-phase flow), Dittus-Boelter\textsuperscript{37} correlation is also applied:

\[
\text{Nu}_{\text{sh}} = 0.023 \cdot \frac{\text{Re}_{\text{sh}}^{0.8} \cdot \text{Pr}_{\text{sh}}^{0.4}}{\text{Pr}_{\text{sh}}} [\langle label \rangle (B.2) < /label > (B.2)
\]

In the two-phase zone of the double-pipe evaporator, the correlation proposed by Zuber and Chen\textsuperscript{41,42} is then applied:

\[
f = 0.046 \cdot \text{Re}^{-0.2} [\langle label \rangle (B.9) < /label > (B.9)
\]

\[
\begin{align*}
\text{FIGURE C1} & \text{ Plate heat exchanger structure} \\
\text{in this paper, in which both nucleate boiling and forced convection boiling are considered:}
\end{align*}
\]

\[
F = 2.35 \cdot \left( \frac{1}{X_H} + 0.213 \right)^{0.736} [\langle label \rangle (B.4) < /label > (B.4)
\]

\[
\alpha_{\text{ibf}} = F \cdot \alpha_{\text{ibf}} + S_j \cdot \alpha_{\text{ibf}} [\langle label \rangle (B.3) < /label > (B.3)
\]

\[
X_H = \left( \frac{1-x}{x} \right)^{0.875} \left( \frac{\rho_{\text{ibf}}}{\rho_{\text{ibf}}} \right)^{0.5} \left( \frac{\mu_{\text{ibf}}}{\mu_{\text{ibf}}} \right)^{0.125} [\langle label \rangle (B.5) < /label > (B.5)
\]

\[
\begin{align*}
\Delta P_{\text{ibf}} &= \frac{\lambda_{\text{ibf}}}{D_{\text{in}}} \cdot \text{Re}^{0.8} \cdot \text{Pr}_{\text{ibf}}^{0.4} [\langle label \rangle (B.6) < /label > (B.6) \\
S_j &= \frac{1}{1 + 2.53 \times 10^{-0.6} \cdot \text{Re}^{0.17}_{\text{ibf}}} [\langle label \rangle (B.7) < /label > (B.7)
\end{align*}
\]

\[
\begin{align*}
\alpha_{\text{ibf}} &= 0.023 \cdot \frac{\lambda_{\text{ibf}}}{D_{\text{in}}} \cdot \text{Re}^{0.8}_{\text{ibf}} \cdot \text{Pr}_{\text{ibf}}^{0.4} \cdot \text{Pr}_{\text{ibf}}^{0.4} [\langle label \rangle (B.8) < /label > (B.8)
\end{align*}
\]

The pressure drop of the single-phase zone is calculated as follows, and the pressure drop of the two-phase zone is referenced\textsuperscript{41}:

\[
f = 0.046 \cdot \text{Re}^{-0.2} [\langle label \rangle (B.9) < /label > (B.9)}}
\[
\Delta P_{dp} = 4f \frac{L \rho y^2}{D} \left\langle \frac{y}{D} \right\rangle < \text{label} > (B.10) < \text{/label} >
\]

**APPENDIX C.**

Plate preheater modeling
The plate heat exchanger structure is shown in Figure C.1, Chisholm and Wanniarachchi correlation\(^43\) are used to calculate the Nussel coefficient of the plate preheater.

\[
Nu = 0.724 \left( \frac{6\theta}{\pi} \right)^{0.646} \text{Re}^{0.583} \text{Pr}^{0.13} < \text{label} > (C.1) < \text{/label} >
\]

\[
f = \frac{32}{\text{Re}} < \text{label} > (C.2) < \text{/label} >
\]

\[
\text{Re} = \frac{GD}{\mu} < \text{label} > (C.3) < \text{/label} >
\]

\[
D = \frac{4 \cdot c \cdot d \cdot p \cdot w}{2 (c \cdot d + p \cdot w)} < \text{label} > (C.4) < \text{/label} >
\]

\[
\Delta P = \frac{2G^2 p \cdot l}{\rho D} < \text{label} > (C.5) < \text{/label} >
\]

**APPENDIX D.**

Off-design modeling

**D.1 | Expander**

**TABLE D1** Expander off-design model coefficient table

| Coefficients | \(a_1\) | \(b_1\) | \(c_1\) |
|--------------|--------|--------|--------|
| \(a_1\)     | 6.2215 | -0.3132 | -0.3418 |
| \(a_2\)     | -7.6713 | 0.3291 | 0.4722 |
| \(a_3\)     | 0.8852 | -0.7253 | -0.0216 |

The isentropic efficiency expression of the expander under off-design conditions is as follows.\(^46\)

\[
\eta_{\text{exp,off}} = a_1 \cdot \alpha + a_2 \cdot \beta + a_3 \cdot \gamma + b_1 \cdot \alpha^2 + b_2 \cdot \beta^2 + b_3 \cdot \beta^2 + c_1 \cdot \alpha \cdot \beta + c_2 \cdot \alpha \cdot \gamma + c_3 \cdot \beta \cdot \gamma < \text{label} > (D.1) < \text{/label} >
\]

\[
\alpha = \ln \left( N_{\text{exp}} \right), \beta = \ln \left( P_{\text{exp, in}} \right), \gamma = \ln \left( r_\rho \right)
\]

The corresponding values of each coefficient is shown in the table below.

**D.2 | Pump**

The coefficients of the polynomial are fitted according to the performance curve of a commercial pump.\(^48\)

\[
\eta_{\text{pp,off}} = \left( d_1 \cdot \left( \frac{m_{\text{off}}}{m_{\text{design}}} \right)^3 + d_2 \cdot \left( \frac{m_{\text{off}}}{m_{\text{design}}} \right)^2 + d_3 \cdot \left( \frac{m_{\text{off}}}{m_{\text{design}}} \right) + d_4 \right) \cdot \eta_{\text{pp,design}} < \text{label} > (D.2) < \text{/label} >
\]

where \(d_1, d_2, d_3, d_4\) are all fitting coefficients, and the corresponding values are -0.439, 0.466, 0.453 and 0.519 respectively.