Efficiency Prediction Model for Diesel Engine Rankine Cycle Waste Heat Recovery System (Postprint)

Authors: Yang Can, Xie Hui, Zhou Kui, Cheng Zhang

Date: 2018-01-05T00:00:00+00:00

Abstract

Organic Rankine Cycle (ORC) technology can effectively recover exhaust thermal energy from heavy-duty vehicle diesel engines. However, during road operation, the exhaust heat source is unstable, and maintaining the cold source also consumes energy, which poses severe challenges for the online optimization and control implementation of both evaporation and condensation pressures. To this end, an efficiency prediction model for the ORC waste heat recovery system was established. Based on the first-law thermodynamic model of the ORC system, and by capturing the macroscopic characteristics of the evaporator' s step response, a dynamic correction model for effective heat transfer capacity and a prediction factor for operating mode switching were proposed. Additionally, using a method combining simplified mechanism and data modeling, an empirical model for condenser heat dissipation power consumption was established. An optimization control architecture and strategy for the ORC waste heat recovery system based on the efficiency prediction model was also proposed. Road simulation results demonstrate the effectiveness of the efficiency model and optimization algorithm: the effective working time of the ORC system reaches 94%, steam superheat is controlled between 5-15K, and the ratio of condenser heat dissipation power consumption to generated power is maintained below 20%.

Full Text

Efficiency Prediction Model of a Diesel-Engine ORC Exhaust Heat Recovery System

YANG Can, XIE Hui, ZHOU Kui, CHENG Zhang
State Key Laboratory of Engines, Tianjin University, Tianjin 300072, China

Abstract

Organic Rankine Cycle (ORC) technology can effectively recover exhaust thermal energy from heavy-duty vehicle diesel engines. However, during real-world driving cycles, the exhaust heat source is unstable and maintaining the cold source consumes additional energy, posing severe challenges for online optimization and control of evaporation and condensation pressures. To address this, an efficiency prediction model for ORC waste heat recovery systems was developed. Based on the first law of thermodynamics for ORC systems and focusing on the dominant characteristics of the evaporator' s step response, a dynamic correction model for effective heat transfer and a predictive factor for operating mode switching were proposed. An empirical model for condenser heat rejection power consumption was established using a combination of simplified mechanism and data-driven modeling approaches. An optimization control framework and strategy for ORC waste heat recovery systems based on the efficiency prediction model were then proposed. Road simulation results demonstrate the effectiveness of the efficiency model and optimization algorithm: the effective working time of the ORC system reaches 94%, steam superheat is controlled within 5–15 K, and the ratio of condenser heat rejection power consumption to power generation is maintained below 20%.

Keywords: diesel engine; organic Rankine cycle; waste heat recovery; efficiency prediction model

Introduction

The application of ORC technology to recover and utilize waste heat from heavy-duty diesel engines is a promising approach for improving engine fuel economy and meeting future stringent fuel consumption regulations [1]. Research from BMW, AVL, and Oak Ridge National Laboratory indicates that this technology can achieve 5–10% fuel economy improvement in heavy-duty vehicles [2–5].

However, during vehicle operation, engines typically run under dynamic conditions including idle, high load, and low load, causing exhaust flow rate and temperature to exhibit strong transient characteristics with large variation ranges. Previous studies [6, 7] have revealed that ORC waste heat recovery systems display pronounced dynamic behavior during driving cycles, including fluctuations in steam parameters and operating mode switching, which degrade actual fuel savings. Tona et al. from IFP France [8] identified that optimal control of ORC systems under dynamic conditions represents a major challenge for practical application.

Furthermore, heat engines cannot extract heat from a single source and completely convert it to useful work without other effects. While ORC systems recover engine exhaust energy for power generation, they also release low-temperature heat to the engine cooling system through the condenser, increasing the cooling system' s heat rejection load and power consumption. Horst et al. [9] demonstrated that ORC system heat rejection increases cooling water pump power consumption, reducing fuel-saving potential by 1.3–1.8%. Yang et al. [10] showed that the more waste heat the ORC system recovers, the greater its heat rejection demand and cooling system power consumption become. A profit-loss coupling relationship exists between ORC power generation and engine cooling system power consumption.

These dynamic and coupling characteristics of ORC systems in recovering vehicle engine exhaust energy pose severe challenges for optimal decision-making and control implementation under dynamic operating conditions. This paper focuses on these dynamic and coupling characteristics to develop an efficiency prediction model for ORC system optimization and control, supporting online decision-making and control of evaporation and condensation pressures.

1.1 Research Object

[Figure 1: see original paper] shows the system configuration studied in this paper: ORC technology is used to recover exhaust energy downstream of a heavy-duty vehicle diesel engine' s turbocharger turbine, with a controllable cooling system matched to the ORC system.

1.2 Problem Description

The core task of ORC system optimization and control research is to dynamically determine the optimal ORC system operating state based on transient exhaust flow and temperature characteristics during road operation, with system efficiency optimization as the objective, and to achieve this through control implementation.

The system efficiency of the ORC system is defined as:

$$\eta_{System} = \frac{W_{net} - P_{cool}}{Q_{exhaust}}$$

where W_{net} is the net work output of the ORC system, P_{cool} is the power consumption of cooling accessories, η_{System} is the system efficiency, $Q_{exhaust}$ is the exhaust energy, Ex represents exhaust boundary conditions, and RC represents ORC system operating states including evaporation pressure, superheat, condensation pressure, and subcooling.

To maintain the ORC system at its optimal state and achieve maximum system efficiency throughout dynamic operation, the evaporation pressure, superheat, condensation pressure, and subcooling must be optimized and controlled at each control step based on exhaust boundary conditions (temperature and flow rate). Recent studies [6, 10] have found that optimal values for superheat and subcooling are relatively simple and do not vary significantly with operating conditions.

However, the optimal regulation laws for evaporation and condensation pressures remain unclear for three main reasons:

1.2.1 Dynamic Characteristics Steady-state studies [6] show that optimal evaporation pressure is positively correlated with exhaust temperature. While exhaust temperature and evaporation pressure have a one-to-one correspondence in value, they are not temporally consistent during dynamic operation due to the significant difference in energy conversion time scales between the engine and ORC systems, as shown in Table 1 .

Table 1 Comparison of Energy Conversion Features Between Engine and ORC System

Device	Working Principle	Energy Conversion Method	Dynamic Time Scale
Engine	Reciprocating	Cyclic combustion	Fast: one working cycle
ORC System	Continuous flow	Heat transfer and phase change	Slow: tens of seconds to minutes

The engine' s reciprocating operation and explosive combustion release enable chemical-to-mechanical energy conversion within one working cycle, resulting in a fast dynamic time scale. In contrast, the ORC system operates continuously; transient exhaust energy is absorbed by the working fluid through wall heat transfer, heating and evaporating it to drive the expander for power generation. Heat transfer and flow inertia result in a much larger dynamic time scale for ORC energy conversion, requiring tens of seconds or even minutes. This dynamic characteristic makes steady-state control laws linking evaporation pressure and exhaust temperature impossible to implement during dynamic operation, necessitating online optimization of evaporation pressure.

1.2.1 Operating Mode Characteristics Tona et al. [8] studied how ORC systems undergo a series of dynamic processes from cold start, gradually heating and pressurizing until complete evaporation, superheating, and finally driving the expander to output work. Xie et al. [6] defined four basic operating modes: start mode, acceleration mode, power generation mode, and protection mode, noting that mode fluctuations reduce effective working time and affect fuel-saving performance. Therefore, during road operation, a more critical task for ORC system optimization and control is regulating evaporation pressure to avoid entering protection mode, mitigating mode fluctuations, and increasing effective working time. Thus, regulating evaporation pressure is not only about optimizing system efficiency at each control step but more importantly, about preventing operating mode fluctuations.

[Figure 2: see original paper] shows the dynamic working process and mode definitions of the ORC system.

1.2.1 Coupling Characteristics As previously noted, ORC systems release low-temperature heat to the engine cooling system through the condenser while recovering exhaust energy, increasing cooling system load and power consumption. Yang et al. [10] demonstrated that greater waste heat recovery leads to greater heat rejection demand and higher cooling system power consumption. A strong coupling relationship exists between ORC power generation and engine cooling system power consumption, as shown in [Figure 3: see original paper]. Condensation pressure strongly regulates this coupling relationship and should increase with engine speed and load. In other words, the optimal value of condensation pressure is also strongly influenced by exhaust conditions.

[Figure 3: see original paper] illustrates the effects of condensing pressure on the profit-loss relationship.

In summary, due to the dynamic characteristics, operating mode features, and coupling characteristics of ORC systems, optimizing and controlling evaporation and condensation pressures during road operation presents severe challenges. Establishing an efficiency prediction model for ORC system optimization and control is key to overcoming these challenges.

2.1 Control Framework

Considering the dynamic characteristics, operating mode features, and coupling characteristics of ORC systems, an optimization control framework is proposed, as shown in [Figure 4: see original paper]. For given exhaust boundary conditions Ex, the manager and efficiency observer work collaboratively to optimize the optimal evaporation pressure P_{evap}^{opt} and condensation pressure P_{cond}^{opt}. Together with steam temperature T_{steam}^{set} and turbine speed N_{turb}^{set}, these serve as control targets for the controller. The control level directly employs mature PID control algorithms to regulate ORC turbine speed, expansion ratio, working fluid pump flow rate, and cooling system accessory power. The key to this framework is establishing a macroscopic, predictive efficiency model.

2.2 Efficiency Model

According to [Figure 4: see original paper], the core function of the efficiency model is: at any decision moment κ of the manager, based on the exhaust parameters $Ex_R(\kappa)$ at that moment and the ORC system state parameters $RC_X(\kappa)$, if the manager provides decision variables $RC_{Opt}(\kappa)$, the efficiency model must predict the ORC system' s operating mode, output power, and cooling power consumption at time κ, feeding the system efficiency η_{System} back to the manager.

The definition of η_{System} is given by Equation (2). [Figure 4: see original paper] shows the control framework of the ORC exhaust heat recovery system.

The modeling approach involves: using the dynamic heat transfer effects of the evaporator to establish a real-time prediction model for heat absorption; considering the manager' s decision variables and the controller' s execution capability, predicting ORC system output power, heat rejection, and superheat based on thermodynamic models; and predicting cooling power consumption and operating mode based on cooling power consumption models and operating mode models, thereby calculating ORC system efficiency.

During model development, two assumptions are made: 1) Decision variables $RC_{Opt}(\kappa)$ given by the manager are perfectly executed, without considering actuator dynamics and capabilities. 2) Hardware execution processes of the expander, generator, working fluid pump, cooling system, and valves are not considered; only their outputs are modeled, including expansion ratio π_{exp}, generator load L_{load}, working fluid flow rate $\dot{m}_{wf}$, cooling accessory power P_{cool}, and valve opening $Valves$.

The system efficiency η_{System} is defined as:

$$\eta_{System} = \frac{W_{RC}(\tau) - P_{cool}(\tau)}{Q_{exhaust}}$$

where $Q_{exhaust}$ represents engine exhaust energy, $W_{RC}(\tau)$ represents ORC system power generation over a future time period, $P_{cool}(\tau)$ represents power consumed by the cooling system, and τ denotes time.

According to the dynamic characteristics of ORC systems, the engine working cycle is the fundamental time unit for energy conversion. If using the engine working cycle as the time basis, the dynamic characteristics of ORC systems become very significant, with energy conversion time scales much larger than one engine cycle. Therefore, ORC power generation is a dynamic model represented by $W_{RC}(\tau)$. Correspondingly, the cooling power required for ORC generation is represented by $P_{cool}(\tau)$.

Additionally, the efficiency model must predict the ORC operating mode, which has a 0-1 relationship with system efficiency η_{System}. Accurately predicting the operating mode is key to avoiding mode fluctuations and increasing the proportion of power generation mode time.

In summary, the efficiency model describes the ORC system and its controller, and must dynamically predict operating mode ξ_{mode}, power generation $W_{RC}(\tau)$, and cooling power $P_{cool}(\tau)$. The modeling approach is shown in [Figure 5: see original paper].

[Figure 5: see original paper] shows the structure diagram of the ORC system efficiency modeling.

2.2.1 Effective Heat Absorption The effective heat absorption of the ORC system is controlled by the convective heat transfer process in the evaporator.

Considering that the efficiency model must be calculated in real-time within the controller, the model cannot be overly complex and requires a trade-off between accuracy and computational efficiency. Therefore, a lumped-parameter average model is adopted for describing the evaporator convective heat transfer process.

On the exhaust side, the governing equations for exhaust flow across staggered tube bundles include the convective heat transfer equation, energy conservation equation, and logarithmic mean temperature difference equation, given by Equations (2), (3), and (4):

$$Q_{ex} = h_{ex} A_{ex} \Delta T_{lm}$$

$$Q_{ex} = \dot{m}_{ex} c_{p,ex} (T_{ex,in} - T_{ex,out})$$

$$\Delta T_{lm} = \frac{(T_{ex,in} - T_w) - (T_{ex,out} - T_w)}{\ln\left(\frac{T_{ex,in} - T_w}{T_{ex,out} - T_w}\right)}$$

where h_{ex} is the average heat transfer coefficient on the exhaust side, following the Zhukauskas correlation with a calibration factor λ_{ex}; A_{ex} is the effective heat transfer area; $\dot{m}_{ex}$ is exhaust mass flow rate; $T_{ex,in}$ and $T_{ex,out}$ are exhaust temperatures at evaporator inlet and outlet; T_w is average tube wall temperature; and $c_{p,ex}$ is exhaust specific heat capacity (typically 1.088 kJ/(kg · K) in engineering practice).

Solving these equations yields:

$$Q_{ex} = \dot{m}_{ex} c_{p,ex} (T_{ex,in} - T_w) \left(1 - e^{-\frac{h_{ex} A_{ex}}{\dot{m}_{ex} c_{p,ex}}}\right)$$

Equation (5) can be interpreted as comprising maximum heat transfer $Q_{ex,max}$ and exhaust-side heat transfer efficiency η_{ex}:

$$Q_{ex,max} = \dot{m}_{ex} c_{p,ex} (T_{ex,in} - T_w)$$

$$\eta_{ex} = 1 - e^{-\frac{h_{ex} A_{ex}}{\dot{m}_{ex} c_{p,ex}}}$$

In Equation (7), the exponential term represents the ratio of actual to maximum possible heat flux, which can be further expressed as:

$$\frac{h_{ex} A_{ex}}{\dot{m}_{ex} c_{p,ex}} = St \cdot \frac{A_{ex}}{A_{flow}} = \zeta e^{-\lambda_{ex} \times f(\dot{m}_{ex}, T_{ex,in})}$$

where St is the Stanton number characterizing forced convection intensity, A_{ex}/A_{flow} is the ratio of heat transfer area to flow area, and ζ is a dimensionless number. A larger ζ indicates stronger convection between exhaust and tube walls.

The tube wall average temperature T_w cannot be directly measured. However, due to wall thermal inertia, its temperature cannot change rapidly and can be solved iteratively. At time κ, the wall temperature from time $\kappa - 1$ can be used to calculate exhaust heat transfer $Q_{ex}(\kappa)$, then wall energy conservation is applied to recalculate the wall temperature at κ.

After calculating exhaust heat transfer Q_{ex}, the effective heat absorption of the ORC working fluid $Q_{RC,eff}$ can be calculated using the lumped heat transfer model:

$$Q_{RC,eff} = h_{RC}A_{RC}(T_w - T_{RC,E})$$

$$m_w c_{p,w}\frac{dT_w}{dt} = Q_{ex} - Q_{RC,eff}$$

Under the assumption of perfect controller execution, the boundaries between liquid, two-phase, and vapor regions remain relatively stable, so the average heat transfer temperature of the working fluid in the evaporator $T_{RC,E}$ is uniquely controlled by evaporation pressure. Higher evaporation pressure yields higher average temperature, making $T_{RC,E}$ a function of the decision variable P_{evap}.

The differential equations (9) and (10) have no analytical solution, and solving them simultaneously would be computationally prohibitive for engine controllers. An alternative approach is required.

Assuming at time κ the evaporation pressure decision value is $P_{evap}(\kappa)$, and at time $\kappa-1$ it was $P_{evap}(\kappa-1)$, the working fluid average temperature will change accordingly. However, due to evaporator wall thermal inertia, T_w does not change instantaneously. According to Equation (10), effective heat absorption $Q_{RC,eff}$ will exhibit a pulse response due to the sudden change in working fluid average temperature. The pulse energy equals the energy released or absorbed by the evaporator wall due to temperature change.

If $T_w > T_{RC,E}$, the wall releases energy; otherwise it absorbs energy, characterized by:

$$\Delta Q_{pulse} = m_w c_{p,w}\Delta T_{RC,sat}$$

The change in working fluid saturation temperature $\Delta T_{RC,sat}$ can be expressed as a linear relationship with evaporation pressure change ΔP_{evap}, with coefficient λ_{sat}. This treatment transforms Equations (9) and (10) into:

$$Q_{RC,eff}(\kappa) = Q_{ex}(\kappa) \cdot \eta_{ex} - \tau_{inertia} \frac{dQ_{RC,eff}}{dt} + \Delta Q_{pulse}$$

where $\tau_{inertia}$ represents the evaporator dynamic heat transfer inertia time constant, characterizing the ratio of wall heat storage capacity to working fluid heat transfer capacity. Considering the complexity of predicting the working fluid-side heat transfer coefficient, it can be obtained through 激励试验 (excitation tests) and corrected for flow rate and temperature effects:

$$\tau_{inertia} = \tau_{base} \cdot \lambda_m \cdot \lambda_T$$

Based on Equations (5) and (12), the effective heat absorption of the ORC evaporator can be predicted in real-time.

2.2.2 Operating Mode Factor The operating mode ξ_{mode} is a 0-1 variable. Given effective heat absorption $Q_{RC,eff}$ and decision variables, it is influenced by both decision variables and controller-regulated mass flow rate $\dot{m}_{wf}$. However, the manager needs to know whether the ORC system could fall into protection mode when decision variables are executed but mass flow rate cannot be adjusted in time.

For given $Q_{RC,eff}$ and decision variables, the instantaneous operating mode ξ_{mode} is calculated when decision variables and controller-regulated mass flow rate $\dot{m}_{wf}$ cannot immediately affect steam state. At time κ, the working fluid temperature at evaporator inlet $T_{in}(\kappa)$ and mass flow rate $\dot{m}_{wf}(\kappa)$ are the same as at time $\kappa - 1$. To achieve the safe minimum superheat $T_{sh,min}$ corresponding to evaporation pressure $P_{evap}(\kappa)$, the minimum required energy is:

$$Q_{RC,eff,min} = \dot{m}_{wf}(\kappa) \cdot (h_{out}(P_{evap}(\kappa), T_{sat} + T_{sh,min}) - h_{in}(\kappa))$$

Then the operating mode factor is:

$$\xi_{mode} = \begin{cases} 1 & \text{if } Q_{RC,eff} \geq Q_{RC,eff,min} \\ 0 & \text{otherwise} \end{cases}$$

The meaning is: if effective heat absorption $Q_{RC,eff}$ exceeds the minimum required $Q_{RC,eff,min}$, the ORC system operates in power generation mode ($\xi_{mode} = 1$); otherwise it enters protection mode ($\xi_{mode} = 0$).

2.2.3 Cooling Power Consumption ORC power generation $W_{RC}(\tau)$ represents the predicted value after decision variables P_{cond}^{opt} are fully executed by the controller within one decision cycle. The working fluid pump inlet is state point 1, as shown in [Figure 6: see original paper], with saturation pressure P_1. Based on pump efficiency η_{pump}, state 2 can be calculated. State 3 is determined by decision variable P_{evap}^{opt}. After expansion, state 4 is calculated using expander isentropic efficiency η_{exp}. Using REFPROP software, the cycle efficiency η_{cycle} under these conditions can be calculated, and ORC power generation is predicted as:

$$W_{RC} = Q_{RC,eff} \cdot \eta_{cycle}$$

Condenser heat rejection is:

$$Q_{cond} = Q_{RC,eff} \cdot (1 - \eta_{cycle})$$

[Figure 6: see original paper] shows the thermodynamic cycle diagram of the ORC system.

ORC cooling power consumption mainly includes water pump power, fan power, and ram air resistance power.

For the water pump:

$$P_{pump} = \frac{\dot{V}_{cool} \cdot \Delta H_{pump}}{\eta_{pump}}$$

where η_{pump} is the inverse of pump efficiency, $\dot{V}_{cool}$ is volumetric flow rate, and ΔH_{pump} is pump head. For the cooling water circulation, pump head overcomes piping resistance including friction and local losses, so:

$$\Delta H_{pump} \approx \lambda_{pipe} \dot{V}_{cool}^2$$

Combining and ignoring first-order errors within the coefficient yields:

$$P_{pump} = \lambda_{pump} \dot{V}_{cool}^3$$

Generally, cooling water flow rate directly relates to heat rejection, transforming Equation (20) to:

$$P_{pump} = \lambda'_{pump} Q_{cond}^3$$

Similarly, fan power and ram air resistance power can be modeled as:

$$P_{fan} = \lambda_{fan} Q_{rad,fan}^3$$

$$P_{ram} = \lambda_{ram} Q_{rad,ram}$$

where $Q_{rad,fan}$ and $Q_{rad,ram}$ represent heat removed by fan cooling and ram air cooling from the radiator. Ram cooling is passive and affected by driving conditions, while fan cooling is actively controllable, but their efficiencies differ significantly. Fan auxiliary cooling has low isentropic efficiency (typically below 30%), while ram cooling has approximately 100% efficiency. A ram cooling ratio factor λ_{ram} characterizes ORC and engine cooling system integration. Combining Equations (23) and (24):

$$P_{cool} = \lambda_{fan}(1-\lambda_{ram})Q_{cond}^3 + \lambda_{ram}Q_{cond}$$

By energy conservation, condenser heat rejection equals radiator heat rejection: $Q_{cond} = Q_{rad}$. From Equations (22) and (27), the total heat rejection power consumption characteristic model is:

$$P_{cool} = \lambda_{pump}Q_{cond}^3 + \lambda_{fan}(1-\lambda_{ram})Q_{cond}^3 + \lambda_{ram}Q_{cond}$$

Defining the sub-term $\frac{Q_{cond}}{P_{cool}}$ as the ORC cooling system energy efficiency coefficient COP_{cool}, which is controlled by condensation pressure. According to Yang et al. [10], when using full fan cooling, COP_{cool} typically ranges from 3 to 5.

2.2.4 Efficiency Model Integration Discretizing Equation (12) using backward difference yields:

$$Q_{RC,eff}(\kappa) = Q_{ex}(\kappa)\cdot\eta_{ex}(\kappa) - \tau_{inertia}\frac{Q_{RC,eff}(\kappa) - Q_{RC,eff}(\kappa-1)}{\Delta t} + \Delta Q_{pulse}(\kappa)$$

where Δt is the decision step size. At time κ, exhaust heat release $Q_{ex}(\kappa)$ is calculated from exhaust temperature $T_{ex,in}(\kappa)$ and mass flow rate $\dot{m}_{ex}(\kappa)$. The evaporator wall average temperature uses the value from time $\kappa-1$, updated via:

$$T_w(\kappa) = T_w(\kappa-1) + \frac{Q_{ex}(\kappa) - Q_{RC,eff}(\kappa)}{m_w c_{p,w}}\Delta t$$

At time κ, the decision variable $P_{evap}(\kappa)$ causes a pulse energy change $\Delta Q_{pulse}(\kappa)$ equal to the wall thermal capacity absorbing or releasing energy due to working fluid saturation temperature change:

$$\Delta Q_{pulse}(\kappa) = m_w c_{p,w} \cdot \lambda_{EP} \cdot \Delta T_{RC,sat}(\kappa)$$

where $\Delta T_{RC,sat}(\kappa)$ is the difference between saturation temperatures corresponding to decision variables at times κ and $\kappa-1$.

Using Equations (14), (16), and (29), the operating mode $\xi_{mode}(\kappa)$, power generation $W_{RC}(\kappa)$, and cooling power $P_{cool}(\kappa)$ at time κ are calculated in discrete form:

$$\xi_{mode}(\kappa) = \begin{cases} 1 & \text{if } Q_{RC,eff}(\kappa) \geq Q_{RC,eff,min}(\kappa) \\ 0 & \text{otherwise} \end{cases}$$

$$W_{RC}(\kappa) = Q_{RC,eff}(\kappa) \cdot \eta_{cycle}(\kappa)$$

$$P_{cool}(\kappa) = \lambda_{cool} \cdot Q_{RC,eff}(\kappa) \cdot (1 - \eta_{cycle}(\kappa))$$

The system efficiency at time κ is then:

$$\eta_{System}(\kappa) = \frac{W_{RC}(\kappa) - P_{cool}(\kappa)}{Q_{exhaust}(\kappa)}$$

Four coefficients require calibration: exhaust-side heat transfer coefficient scaling factor λ_{ex}, inertia time flow correction factor λ_m, temperature correction factor λ_T, and wall thermal pulse energy temperature difference correction factor λ_{EP}.

3. Efficiency Model Validation

Based on the simulation platform described in literature [6], the developed efficiency model was validated and applied to the control framework shown in [Figure 4: see original paper] for online optimization and management of the ORC system, further verifying model effectiveness through application results.

3.1 Model Accuracy Validation The efficiency model was calibrated across four operating points from low-speed/low-load to high-speed/high-load conditions, with parameters shown in Table 2 . Although sufficient experimental data is unavailable for comprehensive validation, if the efficiency model trends match the detailed simulation model, it has potential to replace the detailed model for online optimization and control.

Table 2 Parameters Setup

Parameter	Value
ESC Conditions (except idle)	Water cooling (fan assisted)

The calibration results comparing the efficiency model with the simulation model are shown in [Figure 7: see original paper].

[Figure 7: see original paper] Calibration of the efficiency model

3.2 Model Application Validation The efficiency model was applied to the ORC system optimization control framework in [Figure 4: see original paper] to online optimize evaporation and condensation pressures, improving dynamic characteristics, operating mode stability, and coupling balance. Validation was performed under HWFET driving cycle conditions with full vehicle load and full fan cooling.

Note: The HWFET cycle duration is 765 seconds. To validate control effectiveness, ORC thermal capacity must be limited; otherwise, excessively long start-up times would prevent method validation. Therefore, a relatively low thermal capacity of 50 kg stainless steel equivalent was used. Under these unfavorable conditions with poor wall thermal inertia and insufficient buffering against exhaust disturbances, the proposed control framework and efficiency model were validated for broader applicability. Results are shown in [Figure 8: see original paper].

[Figure 8: see original paper] Control effect of the ORC system during the HWFET driving cycle

Based on [Figure 8: see original paper], the ORC waste heat recovery system online optimization and control implementation is described as follows:

1) A low-pressure start strategy is adopted to shorten ORC system start-up time.
2) During start-up, condensation pressure is set to the upper limit of 0.8 MPa (limited by working fluid pump inlet pressure in bench tests). This decision keeps the cooling system inactive, reducing cooling water pump and fan power consumption while minimizing heat loss to promote rapid ORC warm-up.
3) After start-up and acceleration, the ORC system enters power generation mode. In this mode, the manager and efficiency model collaborate to achieve three functions: (i) operating mode management to avoid entering protection mode; (ii) dynamic efficiency optimization by controlling evaporation pressure to match exhaust temperature, improving ORC efficiency; and (iii) profit-loss coupling balance by managing condensation pressure to optimize the relationship between ORC power generation and cooling power consumption.

Specific implementation results: - During sudden vehicle deceleration with insufficient exhaust energy, steam temperature and superheat decrease. By reducing evaporation pressure, superheat is maintained above safety limits to avoid entering protection mode. The controller simultaneously reduces working fluid pump flow rate to coordinate with evaporation pressure, keeping steam superheat around 10 K throughout the cycle. - During high-speed cruising or acceleration with sufficient exhaust energy, evaporation pressure gradually increases to maintain high-efficiency ORC operation. - For balancing ORC generation and cooling power consumption: during acceleration with high continuous heat absorption and rejection, condensation pressure targets are increased to enlarge heat rejection temperature difference and COP, reducing cooling power; during deceleration or cruising with low heat transfer, condensation pressure targets are decreased to prioritize ORC conversion efficiency.

Overall, using the developed efficiency model and control framework, ORC effective working time reaches 94%, steam superheat is controlled within 5-15 K, and the ratio of heat rejection power to generation power is maintained below 20%, achieving 2.2–2.8% fuel savings in the HWFET cycle.

4. Conclusions

To overcome the bottleneck of online optimization control for ORC waste heat recovery systems, this paper comprehensively considered dynamic characteristics, operating mode features, and coupling characteristics to develop an efficiency prediction model and proposed an online optimization control framework and strategy. Simulation results validated the effectiveness of the efficiency model and control architecture, leading to the following conclusions:

1) Diesel engine ORC waste heat recovery systems exhibit strong dynamic characteristics, operating mode features, and coupling characteristics, making steady-state-calibrated optimal control laws impossible to implement during dynamic operation. Online optimization of evaporation and condensation pressures is necessary.

2) Based on the first law of thermodynamics for ORC systems and focusing on dominant evaporator step response characteristics, a dynamic correction model for effective heat transfer and an operating mode switching predictive factor were proposed. An empirical model for condenser heat rejection power consumption was established using a combined mechanism and data-driven approach, integrating them into an ORC waste heat recovery system efficiency prediction model.

3) The developed efficiency prediction model shows less than 5% error compared with the detailed simulation model and matches its trends, making it suitable for online optimization and control of ORC waste heat recovery systems.

4) HWFET driving cycle simulation results validated the efficiency model' s effectiveness: ORC effective working time reached 94%, steam superheat was controlled within 5-15 K, and the ratio of heat rejection power to generation power was maintained below 20%.

References

[1] Taylor A M K P. Science Review of Internal Combustion Engines [J]. Energy Policy, 2008, 36: 4657-4667.

[2] Ringler J, Seifert M, Guyotot V, et al. Rankine Cycle for Waste Heat Recovery of IC Engines [C] // SAE Paper. Detroit, Michigan, USA, 2009, 2009-01-

0174.

[3] Teng H, Regner G, Cowland C. Achieving High Engine Efficiency for Heavy-Duty Diesel Engines by Waste Heat Recovery Using Supercritical Organic-Fluid Rankine Cycle [C] // SAE Paper. Detroit, Michigan, USA, 2006, 2006-01-3522.

[4] Park T, Teng H, Hunter G, et al. A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines-Experimental Results [C] // SAE Paper. Detroit, Michigan, USA, 2011, 2011-01-1337.

[5] Teng H, Klaver J, Park T, et al. A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines-WHR System Development [C] // SAE Paper. Detroit, Michigan, USA, 2011, 2011-01-0311.

[6] Xie H, Yang C. Dynamic Behavior of Rankine Cycle System for Waste Heat Recovery of Heavy Duty Diesel Engines under Driving Cycle [J]. Applied Energy, 2013, 112: 130-141.

[7] Yang C, Xie H, Zhou S K. Efficiency Analysis of the Rankine Cycle System Used for Engine Exhaust Energy Recovery under Driving Cycle [C] // SAE Paper. Detroit, Michigan, USA, 2014, 2014-01-0671.

[8] Tona P, Peralez J, Sciarretta A. Supervision and Control Prototyping for an Engine Exhaust Gas Heat Recovery System Based on a Steam Rankine Cycle [C] // IEEE/ASME International Conference on Advanced Intelligent Mechatronics. Kaohsiung, Taiwan: IEEE. 2012: 695-701.

[9] Horst T A, Tegethoff W, Eilts P. Prediction of Dynamic Rankine Cycle Waste Heat Recovery Performance and Fuel Saving Potential in Passenger Car Applications Considering Interactions with Vehicles' Energy Management [J]. Energy Conversion and Management, 2014, 78: 438-451.

[10] Yang C, Xie H, Zhou S K. Overall optimization of Rankine cycle system for waste heat recovery of heavy-duty vehicle diesel engines considering the cooling power consumption [J]. Science China Technological Sciences, 2016, 59(2): 309-321.

Note: Figure translations are in progress. See original paper for figures.