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

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

Date: 2017-11-07T00:00:00+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 online optimization and control implementation of both evaporation pressure and condensation pressure. To this end, an efficiency prediction model for the ORC waste heat recovery system was established. Based on the first law of thermodynamics model of the ORC system, capturing the macroscopic 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. Additionally, using a method combining simplified mechanism and data-driven modeling, an empirical model for the condenser' s 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 proposed. Road simulation results demonstrate the effectiveness of the efficiency model and optimization algorithm: the effective power generation time of the ORC system reaches 94%, steam superheat is controlled within 5–15 K, and the ratio of condenser heat dissipation power consumption to power generation is maintained below 20%.

Full Text

Preamble

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: The ORC (organic Rankine cycle) system is regarded as one of the most potential ways for exhaust heat recovery of heavy-duty vehicle diesel engines. But the exhaust energy fluctuates widely during driving cycles, also the ORC cooling will introduce extra energy consumption. Consequently, it is a big challenge to optimize and control the evaporation pressure and condensation pressure in the driving cycle. To this end, a control-oriented ORC efficiency prediction model was completed at first: basing on a foundation model of the first law of thermodynamics; focusing on the dominant feature of the ORC system step response, an effective heat absorption dynamic model and operating modes corrected factor were built up; and an empirical model was established to describe the energy consumption for ORC cooling. Afterwards, a sequence optimization algorithm was proposed to seek optimum values of the evaporation pressure and condensation pressure dynamically. The validation results based on a simulation plat were encouraging: the effective working time percentage achieves 94%, the superheat degree is controlled as 5K-15K, and the cooling power consumption is kept less than 20% of the ORC power generation.

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

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

1.2 Problem Description

During real-world driving conditions, vehicle engines primarily operate under dynamic transient 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 demonstrate pronounced dynamic behavior during driving cycles, including fluctuations in steam parameters and switching between operating modes, which deteriorates the actual fuel-saving effect. Research by Tona et al. at the French Petroleum Institute (IFP) [8] identified that optimal control of ORC systems under dynamic conditions represents a major bottleneck for their practical application.

Furthermore, according to thermodynamic principles, a heat engine cannot extract heat from a single source and completely convert it into useful work without other effects. While the ORC system recovers exhaust energy from the engine to generate electricity, it simultaneously releases low-temperature heat to the engine cooling system through the condenser, increasing the cooling system' s heat rejection load and power consumption. Studies by Horst et al. [9] demonstrated that ORC system cooling can increase water pump power consumption, thereby reducing its fuel-saving potential by 1.3%-1.8%. The more waste heat energy the ORC system recovers, the greater its cooling demand becomes, and consequently, the higher the cooling system power consumption. There exists a certain profit-loss coupling relationship between ORC power generation and engine cooling system power consumption [10].

In summary, when recovering exhaust energy from vehicle engines, ORC systems exhibit strong dynamic and coupling characteristics, posing severe challenges for both optimal decision-making and control implementation under dynamic operating conditions. This paper focuses on these dynamic and coupling characteristics of ORC systems, aiming to establish an efficiency prediction model oriented toward optimal control to support online decision-making and control of evaporation pressure and condensation pressure.

1.1 Research Object

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

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

$$\eta_{\text{System}} = \frac{W_{\text{RC}} - W_C}{E_{\text{Ex}}} = f(E_{\text{Ex}}, R_{\text{RC}}, X_{\text{RC}})$$

where η_{System} is system efficiency, E_{Ex} is exhaust energy, W_{RC} is the net work output of the ORC system, W_C is the power consumption of cooling accessories, R_{Ex} represents exhaust boundary conditions, and X_{RC} represents the operating state of the ORC system, including evaporation pressure, superheat degree, condensation pressure, and subcooling degree.

To maintain the ORC system operating at its optimal state and achieve maximum system efficiency throughout the entire dynamic operation process, it is necessary to optimize and control the evaporation pressure, superheat degree, condensation pressure, and subcooling degree of the ORC system at each control step based on exhaust boundary conditions including exhaust temperature and flow rate. Recent studies [6,10] have shown that the optimal values for superheat degree and subcooling degree are relatively simple and do not vary with operating conditions. However, the optimal regulation law for evaporation pressure and condensation pressure remains unclear, primarily for the following three reasons:

1.2.1 Dynamic Characteristics

Steady-state research [6] indicates that the optimal evaporation pressure is positively correlated with exhaust temperature. While exhaust temperature and evaporation pressure have a one-to-one correspondence in terms of numerical values, they are not temporally consistent during dynamic operation due to a certain time lag. This discrepancy arises because the time scales for energy conversion differ significantly between the engine and ORC systems, as shown in Table 1 .

The reciprocating operation of the engine and the explosive combustion energy release enable the conversion of fuel chemical energy to mechanical energy within one working cycle, resulting in a fast dynamic time scale for energy conversion equivalent to one engine cycle. However, the ORC system operates continuously; transient exhaust energy from the engine is transferred to the ORC working fluid through wall heat transfer, heating and evaporating the fluid to drive the expander for power generation. The system' s thermal and flow inertia results in a much larger dynamic time scale for ORC energy conversion, requiring tens of seconds or even minutes, as shown in Table 1. This dynamic characteristic makes the steady-state control law of one-to-one correspondence between evaporation pressure and exhaust temperature unimplementable during dynamic operation, necessitating online optimization of evaporation pressure.

1.2.1 Operating Mode Characteristics

Tona et al. [8] studied that from cold start, the ORC system gradually heats up, increases pressure until complete evaporation and superheat are achieved, and finally drives the expander to output work, experiencing a series of dynamic processes. Xie et al. [6] defined these as four basic operating modes: start-up mode, acceleration mode, power generation mode, and protection mode, and pointed out that fluctuations between operating modes reduce the effective power generation time of the ORC system, thereby affecting fuel-saving performance. Therefore, during driving cycles, the greater task of ORC system optimization and control is to regulate evaporation pressure to prevent the ORC system from falling into protection mode, mitigate operating mode fluctuations, and increase the effective power generation time. Thus, regulating evaporation pressure is not only about optimizing system efficiency at each control step but more importantly, about avoiding fluctuations in the ORC system' s operating modes.

[Figure 2: see original paper] ORC System Dynamic Working Process and Mode Definition

1.2.1 Coupling Characteristics

According to thermodynamic principles, a heat engine cannot extract heat from a single source and completely convert it into useful work without other effects. While the ORC system recovers engine exhaust energy for power generation, it simultaneously releases low-temperature heat to the engine cooling system through the condenser, increasing the cooling system' s heat rejection load and power consumption. Yang et al. [10] demonstrated that the more waste heat energy the ORC system recovers, the greater its cooling demand becomes, and the higher the cooling system power consumption. There exists a strong coupling relationship between ORC power generation and engine cooling system power consumption, as shown in Figure 3. The value of condensation pressure has a strong regulating effect on 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] 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 during driving cycles, the optimization and control of evaporation pressure and condensation pressure pose severe challenges. Establishing an efficiency prediction model oriented toward ORC system optimal control is key to overcoming these challenges.

2.1 Control Architecture

Considering the dynamic characteristics, operating mode features, and coupling characteristics of the ORC system, an optimal control architecture is proposed as shown in Figure 4 [Figure 4: see original paper]. For given exhaust boundary conditions E_{Ex}, the manager and efficiency observer work collaboratively to optimize the optimal evaporation pressure $P_{\mathrm{RC},3}$, which together with steam temperature $T_{\mathrm{RC},3}$, turbine speed N_{RC}, and condensation pressure $P_{\mathrm{RC},4}$ serve as control targets Y_{Set} for the controller. At the control level, mature PID control algorithms are directly employed to control ORC system turbine speed, expansion ratio, working fluid pump flow rate, and cooling system accessory power. The key to this architecture lies in establishing a macroscopic, predictive efficiency model.

2.2 Efficiency Model

According to Figure 4, the core function of the efficiency model is: at any decision moment κ of the manager, based on the exhaust parameters $E_{\mathrm{Ex}}(\kappa)$ and ORC system state parameters $X_{\mathrm{RC}}(\kappa)$ at that moment, if the manager provides decision variables $X_{\mathrm{RC,Opt}}(\kappa)$, it needs to predict the operating mode, output power, and cooling power consumption of the ORC system at moment $\kappa + 1$ and feed them back to the manager. The system efficiency η_{System} is defined by Equation (2).

[Figure 4: see original paper] Control Framework of the ORC Exhaust Heat Recovery System

During the establishment of the efficiency model, the following two assumptions are made first: 1) The decision variables provided by the manager can be executed perfectly, without considering the actuator execution process and execution delay. 2) The hardware execution processes of the expander, generator, working fluid pump, cooling system, and valves are not considered; only their output quantities are focused on, including expansion ratio π_{RC}, power generation load $T_{\mathrm{RC,load}}$, cooling accessory power L_m, and valve opening V_{CW}.

Based on the dynamic characteristics of the ORC system, the engine working cycle is the basic time unit for its energy conversion. If the engine working cycle is used as the time reference, the dynamic characteristics of the ORC system become very significant, with its energy conversion time scale being much larger than one engine cycle. Therefore, the power generation of the ORC system is a dynamic process represented as $W_{\mathrm{RC}}(\tau)$, where τ represents time. Correspondingly, the cooling power consumption required for ORC power generation is denoted as $W_C(\tau)$. Additionally, the efficiency model of the ORC system must have an important function: predicting the operating mode $m(\xi)$ of the ORC system, which is a 0-1 relationship for system efficiency η_{System}. Accurately predicting the operating mode of the ORC system is key to avoiding mode fluctuations and increasing the proportion of time spent in power generation mode.

In summary, the efficiency model of the ORC system describes the ORC system and its controller, and needs to dynamically predict the operating mode $m(\xi)$, power generation $W_{\mathrm{RC}}(\tau)$, and cooling power $W_C(\tau)$. The modeling approach is shown in Figure 5 [Figure 5: see original paper].

The real-time prediction model for ORC system heat absorption in the evaporator is established using the heat transfer dynamic effects of the evaporator. Considering the manager' s decision variables and the controller' s execution capability, the output power, heat rejection, and superheat degree of the ORC system are predicted based on the thermodynamic model of the ORC system. The cooling power consumption and operating mode are predicted according to the cooling power consumption model and operating mode model, and the efficiency of the ORC system is calculated on this basis.

[Figure 5: see original paper] 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 established efficiency model needs to be calculated in real-time in the controller, the model cannot be overly complex and requires a trade-off between description accuracy and computational real-time performance, with the inaccurate parts compensated by advanced control algorithms. Therefore, the description of the evaporator convective heat transfer process adopts an average value model based on the lumped parameter method.

On the exhaust side, the governing equations for the heat transfer process of exhaust gas flowing over staggered tube bundles include the convective heat transfer equation, energy conservation equation, and logarithmic mean temperature difference equation, as shown in Equations (2), (3), and (4):

$$Q_{\mathrm{Ex}} = h_{\mathrm{Ex}} A_{\mathrm{Ex}} \Delta T$$

$$Q_{\mathrm{Ex}} = c_m \dot{m}_{\mathrm{Ex}} (T_{\mathrm{Ex,in}} - T_{\mathrm{Ex,out}})$$

$$\Delta T = \frac{\Delta T_{\mathrm{Ex,in}} - \Delta T_{\mathrm{Ex,out}}}{\ln\left(\frac{\Delta T_{\mathrm{Ex,in}}}{\Delta T_{\mathrm{Ex,out}}}\right)}$$

where h_{Ex} is the average heat transfer coefficient on the exhaust side, which follows the Zhukauskas heat transfer correlation and can be enhanced with a calibration factor λ_{Ex} for the exhaust heat transfer coefficient; A_{Ex} represents the effective heat transfer area on the exhaust side; $\dot{m}_{\mathrm{Ex}}$ represents the exhaust mass flow rate; $\Delta T_{\mathrm{Ex,in}}$ and $\Delta T_{\mathrm{Ex,out}}$ are the temperature differences between the exhaust at the evaporator inlet/outlet and the average tube wall temperature T_{EPT}; c_m is the specific heat capacity of diesel exhaust, typically valued at 1.088 kJ/(kg · K) in engineering applications.

By solving these equations simultaneously, we obtain:

$$Q_{\mathrm{Ex}} = c_m \dot{m}_{\mathrm{Ex}} (T_{\mathrm{Ex,in}} - T_{\mathrm{EPT}}) \left(1 - e^{-\frac{h_{\mathrm{Ex}} A_{\mathrm{Ex}}}{c_m \dot{m}_{\mathrm{Ex}}}}\right)$$

Equation (5) can be interpreted as two parts: the maximum heat transfer $Q_{\mathrm{Ex,Max}}$ and the exhaust side heat transfer efficiency $\eta_{\mathrm{Ex,E}}$, which can be expressed as:

$$Q_{\mathrm{Ex,Max}} = c_m \dot{m}_{\mathrm{Ex}} (T_{\mathrm{Ex,in}} - T_{\mathrm{EPT}})$$

$$\eta_{\mathrm{Ex,E}} = 1 - e^{-\frac{h_{\mathrm{Ex}} A_{\mathrm{Ex}}}{c_m \dot{m}_{\mathrm{Ex}}}}$$

In Equation (7), the exponential term characterizes the ratio of actual exhaust heat flux to the maximum transferable heat flux density, which can be further expressed as:

$$\eta_{\mathrm{Ex,E}} = 1 - e^{-\mathrm{St} \times \zeta}$$

where St is the Stanton number, a dimensionless parameter characterizing forced convection intensity. The larger the St, the stronger the convective heat transfer between exhaust and evaporator tube wall. ζ is the ratio of exhaust heat transfer area to flow area. The larger the ζ, the more complete the heat transfer process between exhaust and evaporator tube wall, and the higher the heat transfer efficiency.

After calculating the exhaust heat transfer Q_{Ex}, the effective heat absorption $Q_{\mathrm{RC,eff}}$ of the ORC working fluid can be calculated based on the lumped heat transfer model of the evaporator tube wall, with the governing equation:

$$c_m \frac{dT_{\mathrm{RC,EP}}}{d\tau} = Q_{\mathrm{Ex}} - Q_{\mathrm{RC,eff}}$$

where $T_{\mathrm{RC,EP}}$ is the tube wall temperature corresponding to moment κ.

As mentioned earlier, under the assumption of perfect controller execution, the boundaries between liquid, two-phase, and vapor phases are relatively stable, so the average temperature $T_{\mathrm{RC,ET}}$ of the working fluid in the evaporator can be considered to be uniquely controlled by the evaporation pressure $P_{\mathrm{RC},3}$. Higher evaporation pressure results in higher average temperature, where $T_{\mathrm{RC,ET}}$ is a function of $P_{\mathrm{RC},3}$. Under such conditions, Equation (9) and Equation (10) cannot obtain analytical solutions, and the computational burden of solving the differential equation system is unacceptable for engine controllers, requiring an alternative approach.

Assuming the evaporation pressure at moment κ is $P_{\mathrm{RC},3}(\kappa)$, and the manager provides an evaporation pressure decision value $P_{\mathrm{RC},3}(\kappa+1)$ at moment $\kappa+1$, the average temperature of the working fluid will change accordingly, making $T_{\mathrm{RC,ET}}(\kappa+1) \neq T_{\mathrm{RC,ET}}(\kappa)$. Due to the heat storage effect of the evaporator tube wall, its temperature $T_{\mathrm{RC,EP}}$ will not change immediately. According to Equation (10), the effective heat absorption $Q_{\mathrm{RC,eff}}$ will produce a pulse response due to the change in working fluid average temperature, with the pulse energy magnitude equal to the energy released or absorbed by the evaporator tube wall due to temperature change. If $P_{\mathrm{RC},3}(\kappa+1) < P_{\mathrm{RC},3}(\kappa)$, the evaporator tube wall releases energy; otherwise, it absorbs energy, characterized as:

$$Q_{\mathrm{RC,eff}}(\kappa+1) = Q_{\mathrm{Ex}}(\kappa+1) + \delta Q_{\mathrm{EP}}(\kappa)$$

where $\delta Q_{\mathrm{EP}}(\kappa)$ can be expressed as a linear relationship with the change in working fluid saturation temperature $\Delta T_{\mathrm{RC,EP}}$ caused by evaporation pressure change, with coefficient c_m:

$$\delta Q_{\mathrm{EP}}(\kappa) = c_m \Delta T_{\mathrm{RC,EP}} = c_m (T_{\mathrm{RC,EP}}(\kappa+1) - T_{\mathrm{RC,EP}}(\kappa))$$

Considering the complexity of predicting the working fluid side heat transfer coefficient, its inertia time $T_{\mathrm{RC,CHT}}$ can be obtained through excitation tests. Accounting for the effects of flow rate and temperature on Reynolds number and property parameters, the inertia time obtained from excitation tests is corrected with flow rate and temperature correction factors $\lambda_{\mathrm{RC,m}}$ and $\lambda_{\mathrm{RC,T}}$. On this basis, according to Equations (5) and (12), the effective heat absorption of the ORC system evaporator can be predicted in real-time:

$$Q_{\mathrm{RC,eff}}(\kappa+1)=\frac{Q_{\mathrm{Ex}}(\kappa+1)+\delta Q_{\mathrm{EP}}(\kappa)}{1+\frac{T_{\mathrm{RC,CHT}}}{\Delta\tau}}$$

where $T_{\mathrm{RC,CHT}}$ represents the inertia time constant of evaporator dynamic heat transfer, characterizing the ratio of evaporator heat storage capacity to working fluid heat transfer capacity.

2.2.2 Operating Mode Factor

The operating mode $m(\xi)$ is a 0-1 variable. Under given effective heat absorption $Q_{\mathrm{RC,eff}}(\kappa)$, it is influenced by both the decision variable $P_{\mathrm{RC,3}}$ and the controller' s regulation of mass flow rate $\dot{m}_{\mathrm{RC}}$. However, the manager needs to know the possibility that when decision variable $P_{\mathrm{RC,3}}$ is executed and mass flow rate $\dot{m}_{\mathrm{RC}}$ has not yet been adjusted, the ORC system' s operating mode $m(\xi)$ might fall into protection mode.

For given effective heat absorption $Q_{\mathrm{RC,eff}}(\kappa)$ and decision variable $P_{\mathrm{RC,3}}$, we calculate the instantaneous operating mode $m(\xi)$ of the ORC system when the decision variable and controller-regulated mass flow rate have not yet affected the steam state. That is, at moment κ, the ORC system working fluid temperature at the evaporator inlet $T_{\mathrm{RC,2}}(\kappa)$ and mass flow rate $\dot{m}_{\mathrm{RC}}(\kappa-1)$ are the same as at moment $\kappa-1$. Then, to achieve the safe temperature $T_{\mathrm{RC,3,min}}$ corresponding to evaporation pressure $P_{\mathrm{RC,3}}(\kappa)$ at the evaporator outlet, the minimum required energy $Q_{\mathrm{RC,eff,min}}(\kappa)$ is:

$$Q_{\mathrm{RC,eff,min}}(\kappa)=\dot{m}_{\mathrm{RC}}(\kappa-1)(h_{\mathrm{RC,3,min}}-h_{\mathrm{RC,2}}(\kappa))$$

[Figure 6: see original paper] Thermodynamic Cycle Diagram of the ORC System

Then the operating mode $m(\xi)$ is:

$$m(\xi)=\begin{cases}1, & \text{if } Q_{\mathrm{RC,eff}}(\kappa)>Q_{\mathrm{RC,eff,min}}(\kappa)\\ 0, & \text{otherwise}\end{cases}$$

The meaning is that if the effective heat absorption $Q_{\mathrm{RC,eff}}(\kappa)$ is greater than the minimum required heat $Q_{\mathrm{RC,eff,min}}(\kappa)$, the ORC system operates in power generation mode with $m(\xi)$ valued at 1; otherwise, the ORC system enters protection mode with $m(\xi)$ valued at 0.

2.2.3 Cooling Power Consumption

The ORC system power generation $W_{\mathrm{RC}}(\tau)$ is the predicted value after the decision variable is fully executed by the controller within one decision cycle. The ORC working fluid pump inlet is state point 1, with saturated pressure $P_{\mathrm{RC},1}$ and specific enthalpy $h_{\mathrm{RC},1}$. Based on the working fluid pump efficiency $\eta_{\mathrm{RC,PS}}$, state point 2 can be calculated, represented by variables $(P_{\mathrm{RC},2}, T_{\mathrm{RC},2})$ with specific enthalpy $h_{\mathrm{RC},2}$. State point 3 is controlled by decision variables $(P_{\mathrm{RC},3}, T_{\mathrm{RC},3})$ with specific enthalpy $h_{\mathrm{RC},3}$. Based on the expander isentropic efficiency $\eta_{\mathrm{RC,TS}}$, the specific enthalpy at point 4 can be calculated. Using REFPROP software, the cycle efficiency of the ORC system under these conditions can be calculated as:

$$\eta_{\mathrm{RC,Cycle}} = \frac{(h_{\mathrm{RC},3} - h_{\mathrm{RC},4}) - (h_{\mathrm{RC},2} - h_{\mathrm{RC},1})}{h_{\mathrm{RC},3} - h_{\mathrm{RC},2}}$$

The ORC system' s power generation and heat rejection in the condenser are respectively:

$$W_{\mathrm{RC}}(\tau) = Q_{\mathrm{RC,eff}}(\tau) \cdot \eta_{\mathrm{RC,Cycle}}$$

$$Q_{\mathrm{Con}}(\tau) = Q_{\mathrm{RC,eff}}(\tau) \cdot (1 - \eta_{\mathrm{RC,Cycle}})$$

The ORC cooling system power consumption mainly includes water pump power, fan power, and power loss due to ram air resistance.

For the water pump, its power consumption can be expressed as:

$$W_{\mathrm{CP}} = \frac{V_{\mathrm{CP}} \Delta P_{\mathrm{CP}}}{\lambda_1}$$

where λ_1 is the reciprocal of pump efficiency, V_{CP} represents the volumetric flow rate through the pump, and ΔP_{CP} represents the actual pump head. For the cooling water flow loop, the pump head is actually used to overcome pipeline resistance, including frictional losses and local losses. Therefore, the pump head ΔP_{CP} can be approximated as:

$$\Delta P_{\mathrm{CP}} = \lambda_2 V_{\mathrm{CP}}^2$$

Ignoring the first-order term error in the calibration coefficient, Equations (18) and (19) can be combined to obtain:

$$W_{\mathrm{CP}} = \lambda_3 V_{\mathrm{CP}}^3$$

Generally, cooling water flow rate is directly related to heat rejection. Equation (20) can be transformed into:

$$W_{\mathrm{CP}} = \lambda_3 \left(\frac{Q_{\mathrm{Con}}}{\lambda_1 \lambda_2}\right)^{3/2}$$

where Q_{Con} is the heat rejection in the condenser. Based on Equation (21), λ_1, λ_2, and λ_3 can be combined to obtain:

$$W_{\mathrm{CP}} = \lambda_3 Q_{\mathrm{Con}}^{3/2}$$

Similarly, characteristic models for fan power consumption and ram air resistance power can be derived, as shown in Equations (23) and (24):

$$W_{\mathrm{CF}} = \beta_3 Q_{\mathrm{Rad,F}}^{3/2}$$

$$W_{\mathrm{CD}} = \gamma_3 Q_{\mathrm{Rad,D}}^{3/2}$$

where $Q_{\mathrm{Rad,F}}$ and $Q_{\mathrm{Rad,D}}$ represent the heat removed from the radiator by fan cooling and ram air cooling, respectively. Ram air cooling is passive and affected by driving conditions, while fan-assisted cooling can be actively regulated, but their efficiencies differ significantly. For fan-assisted cooling, the power consumption is:

$$W_{\mathrm{CF}} = \frac{V_{\mathrm{CF}} \Delta P_{\mathrm{CF}}}{\beta_1}$$

For ram air cooling, the power loss due to resistance is approximately:

$$W_{\mathrm{CD}} = \frac{V_{\mathrm{CD}} \Delta P_{\mathrm{CD}}}{\gamma_1}$$

Ram air cooling is direct with no energy conversion 环节, with efficiency approximately 100%; whereas the fan is a low-efficiency component with isentropic efficiency typically below 30%. A ram air cooling ratio factor ω is introduced to characterize the integration degree between the ORC system and engine cooling system. When the proportion of ram air cooling increases, the energy efficiency ratio of the ORC heat rejection process increases. Combining Equations (23) and (24) and based on energy conservation, the heat rejection in the condenser equals that in the radiator:

$$Q_{\mathrm{RC,Con}} = Q_{\mathrm{Rad}} = Q_{\mathrm{Rad,F}} + Q_{\mathrm{Rad,D}}$$

The total power consumption during the heat rejection process can be obtained from Equations (22) and (27):

$$W_C = (1-\omega)\lambda_3 Q_{\mathrm{RC,Con}}^{3/2} + \omega\gamma_3 Q_{\mathrm{RC,Con}}^{3/2}$$

Define the sub-term "$\frac{W_C}{Q_{\mathrm{RC,Con}}}$" in Equation (29) as the ORC cooling system coefficient of performance CorrCOP, which is controlled by condensation pressure. According to Yang et al. [10], when fan cooling is fully adopted:

$$\mathrm{CorrCOP} = 3.2\times 10^{-4} P_{\mathrm{RC,4}}^2 + 4.2\times 10^{-2} P_{\mathrm{RC,4}} + 0.429$$

2.2.4 Efficiency Model Integration

Based on the efficiency model establishment process, discretizing Equation (12) using backward difference yields:

$$Q_{\mathrm{RC,eff}}(\kappa+1) = \frac{Q_{\mathrm{Ex}}(\kappa+1) + \delta Q_{\mathrm{EP}}(\kappa)}{1 + \frac{T_{\mathrm{RC,CHT}}}{\Delta\tau}}$$

where the exhaust heat release at moment κ is:

$$Q_{\mathrm{Ex}}(\kappa) = c_m \dot{m}_{\mathrm{Ex}}(\kappa)(T_{\mathrm{Ex,in}}(\kappa) - T_{\mathrm{EPT}}(\kappa))\left(1 - e^{-\frac{h_{\mathrm{Ex}}A_{\mathrm{Ex}}}{c_m \dot{m}_{\mathrm{Ex}}(\kappa)}}\right)$$

The average evaporator tube wall temperature uses the value from moment $\kappa-1$, which can be calculated from Equation (33):

$$T_{\mathrm{EPT}}(\kappa) = T_{\mathrm{EPT}}(\kappa-1) + \frac{Q_{\mathrm{Ex}}(\kappa-1) - Q_{\mathrm{RC,eff}}(\kappa-1)}{c_m}\Delta\tau$$

The pulse energy $\delta Q_{\mathrm{EP}}(\kappa)$ absorbed or released by the evaporator tube wall thermal capacity due to decision variable changes at moment κ is:

$$\delta Q_{\mathrm{EP}}(\kappa) = c_m(T_{\mathrm{RC,EP}}(\kappa+1) - T_{\mathrm{RC,EP}}(\kappa)) = c_m\lambda_{\mathrm{EP}}\Delta T_{\mathrm{RC,S}}(\kappa)$$

where $\Delta T_{\mathrm{RC,S}}(\kappa)$ is the difference between the saturation temperature corresponding to the decision variable $P_{\mathrm{RC,3}}(\kappa+1)$ at moment $\kappa+1$ and that corresponding to $P_{\mathrm{RC,3}}(\kappa)$ at moment κ.

The operating mode $m(\xi)$, power generation $W_{\mathrm{RC}}(\kappa)$, and cooling power $W_C(\kappa)$ at moment $\kappa+1$ are calculated separately according to Equations (14), (16), and (29), with their discrete forms being:

$$m(\xi) = \mathrm{sign}\left(Q_{\mathrm{RC,eff}}(\kappa) - Q_{\mathrm{RC,eff,min}}(\kappa)\right)$$

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

$$W_C(\kappa) = \mathrm{CorrCOP}(\kappa) \cdot Q_{\mathrm{RC,eff}}(\kappa) \cdot (1 - \eta_{\mathrm{RC,Cycle}}(\kappa))$$

Thus, the system efficiency of the ORC system at moment κ+1 can be calculated as:

$$\eta_{\mathrm{System}}(\kappa + 1) = \frac{W_{\mathrm{RC}}(\kappa) - W_C(\kappa)}{E_{\mathrm{Ex}}(\kappa)} \cdot m(\xi)$$

A total of four coefficients need to be calibrated: the exhaust side heat transfer coefficient scaling factor λ_{Ex}, the inertia time flow correction factor $\lambda_{\mathrm{RC,m}}$ and temperature correction factor $\lambda_{\mathrm{RC,T}}$, and the evaporator tube wall thermal capacity pulse energy temperature difference correction factor λ_{EP}.

3 Efficiency Model Validation

Based on the simulation platform shown in literature [6], the established efficiency model is validated and applied to the control architecture shown in Figure 4 for online optimization and management of the ORC system, further verifying the effectiveness of the efficiency model from an application perspective.

3.1 Model Accuracy Verification

The efficiency model is calibrated and validated at four operating points from low speed and low load to high speed and high load, with parameter settings shown in Table 2 . The calibration results are shown in Figure 7 [Figure 7: see original paper].

Although sufficient experimental data cannot be obtained to validate the established efficiency model, if the efficiency model can completely match the detailed simulation model in terms of trends, it has the potential to replace the detailed simulation model for online optimization and control of ORC waste heat recovery systems.

[Figure 7: see original paper] Calibration of the Efficiency Model

3.2 Model Application Verification

The established efficiency model is applied to the optimal control architecture of the ORC system shown in Figure 4 to dynamically optimize evaporation pressure and condensation pressure online, improving the dynamic characteristics, operating mode features, and coupling characteristics of the ORC system. The application effect of the efficiency model is verified under HWFET driving cycle conditions with a fully loaded vehicle and complete fan cooling. It should be noted that the HWFET cycle duration is 765 seconds, which is limited; therefore, the ORC system' s thermal capacity needs to be controlled, otherwise the startup time would be too long to validate the control method' s effectiveness. In this verification process, a relatively low thermal capacity of 50 kg stainless steel equivalent is adopted. Under this condition, due to the poor heat storage capacity of the evaporator tube wall and insufficient buffering capability against exhaust disturbances, its impact on the ORC system' s steam state is significant. Verifying the proposed control architecture and efficiency model under such unfavorable conditions should be more universally applicable. The verification results are shown in Figure 8 [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, the implementation process of online optimization and control for the ORC waste heat recovery system is elaborated as follows:

1) A low-pressure startup strategy is adopted to shorten the ORC system startup time. The evaporation pressure decision value is set to 0.8 MPa during startup, which is limited by the working fluid pump inlet pressure in bench tests. This decision keeps the cooling system inactive, reducing cooling water pump and fan power consumption on one hand, and preventing heat loss to promote rapid ORC system warm-up and startup on the other hand.

2) After startup and acceleration, the ORC system enters power generation mode. In this mode, the manager and efficiency model work collaboratively to complete three major functions: (1) operating mode management to avoid the ORC system entering protection mode as much as possible; (2) dynamic efficiency optimization to control the ORC system' s evaporation pressure to adapt to exhaust temperature over a period, improving ORC system efficiency; and (3) balancing the profit-loss coupling relationship by managing condensation pressure to optimize the trade-off between ORC power generation and cooling system power consumption. Specific implementation results are as follows:

    a) When the vehicle suddenly decelerates and exhaust energy is insufficient, steam temperature and superheat degree decrease. By reducing evaporation pressure, steam superheat degree is maintained to avoid entering protection mode due to dropping below safety limits in a short time. Meanwhile, the controller also reduces working fluid pump flow rate to coordinate with evaporation pressure. Throughout the driving cycle, the ORC system' s steam superheat degree is consistently controlled at around 10K.

    b) When the vehicle cruises at high speed or accelerates with sufficient exhaust energy and adequate steam superheat degree, evaporation pressure gradually increases to maintain high-efficiency operation of the ORC system.

c) To balance the profit-loss relationship between ORC power generation and cooling system power consumption, when the vehicle accelerates with large continuous heat absorption and rejection in the ORC system and high cooling system load, condensation pressure control target is increased to enlarge cooling temperature difference and coefficient of performance, reducing cooling system power consumption. When the vehicle decelerates or cruises with small continuous heat absorption and rejection in the ORC system and low cooling system power consumption, condensation pressure control target is reduced to prioritize improving ORC system conversion efficiency.

Overall, using the established efficiency model and control architecture, the ORC system' s effective power generation time reaches 94%, steam superheat degree is controlled between 5K-15K, and the ratio of heat rejection power consumption to power generation is maintained below 20%, achieving fuel-saving effects of 2.2%-2.8% in the HWFET driving cycle.

4 Conclusions

To break through the bottleneck of online optimal control for ORC waste heat recovery systems, this paper comprehensively considers the dynamic characteristics, operating mode features, and coupling characteristics of ORC systems, establishes an efficiency prediction model for ORC systems, and proposes an online optimal control architecture and strategy based on the efficiency prediction model. Simulation results verify 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 calibration-based optimal control laws unimplementable during dynamic operation and necessitating online optimization of evaporation pressure and condensation pressure.

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

3) The established efficiency prediction model has a calibration error of less than 5% on the simulation platform and completely matches the detailed simulation model in terms of trends, making it suitable for online optimization and control of ORC waste heat recovery systems.

4) HWFET driving cycle simulation results validate the effectiveness of the

established efficiency prediction model from an application perspective, achieving 94% effective power generation time, controlling steam superheat degree between 5K-15K, and maintaining the ratio of heat rejection power consumption to power generation below 20%.

References

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

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,

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:

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.