

Transient Power Optimization of an Organic Rankine Cycle Waste Heat Recovery System for Heavy-Duty Diesel Engine Applications

Bin Xu, Adamu Yebi, Simona Onori, and Zoran Filipi
 Clemson-ICAR

Xiaobing Liu, John Shutty, and Paul Ansel
 BorgWarner Inc.

Mark Hoffman
 Clemson-ICAR

ABSTRACT

This paper presents the transient power optimization of an organic Rankine cycle waste heat recovery (ORC-WHR) system operating on a heavy-duty diesel (HDD). The optimization process is carried on an experimentally validated, physics-based, high fidelity ORC-WHR model, which consists of parallel tail pipe and EGR evaporators, a high pressure working fluid pump, a turbine expander, etc. Three different ORC-WHR mixed vapor temperature (MVT) operational strategies are evaluated to optimize the ORC system net power: (i) constant MVT; (ii) constant superheat temperature; (iii) fuzzy logic superheat temperature based on waste power level. Transient engine conditions are considered in the optimization. Optimization results reveal that adaptation of the vapor temperature setpoint based on evaporation pressure strategy (ii) provides 1.1% mean net power (MNP) improvement relative to a fixed setpoint strategy (i). The highest net power is produced by setpoint strategy (iii), which exhibited a 2.1% improvement compared strategy (i), revealing importance of utilizing engine conditions during reference trajectory generation. These results serve as the benchmark for the ORC system net power optimal control.

CITATION: Xu, B., Yebi, A., Onori, S., Filipi, Z. et al., "Transient Power Optimization of an Organic Rankine Cycle Waste Heat Recovery System for Heavy-Duty Diesel Engine Applications," *SAE Int. J. Alt. Power*: 6(1):2017, doi:10.4271/2017-01-0133.

INTRODUCTION

Recent U.S. regulations on CO₂ emission calls for a 20% reduction in 2020 relative to 2010 [1]. Heavy duty vehicles account for 7% of all on-road vehicles and approximately 30% of the U.S. transportation energy usage [2]. However, over 40% of energy consumed by heavy duty diesel engines (HDD) is wasted as heat [3]. Thus, waste heat recovery (WHR) technologies have been an active area of recent research [4, 5, 6, 7, 8, 9, 10, 11]. There are three main methodologies for vehicle WHR: thermoelectrics, turbo-compounding, and implementation of an organic Rankine cycle (ORC) system.

Thermoelectric strategies generate electricity utilizing the temperature difference between the exhaust gas and the thermoelectric material coolant stream [9, 12, 13]. These devices are compact and simple in structure, but their efficiency is limited by the low ZT value of current thermoelectric materials.

Turbocompounding either combines the turbocharger with an electric generator or couples the device to the engine crankshaft [10, 11, 14]. Turbocompounding recovers kinetic energy from the exhaust gas, which amounts to only a fraction of the total waste heat energy. Much of the waste heat thermal energy still passes to ambient via the

exhaust. In addition, turbocompounding generally doesn't consider the waste heat contained in the exhaust gas recirculation (EGR) flow, which comprises nearly 1/3 of the total waste heat in the HDD engine depending on engine operating conditions [15].

An ORC system is able to directly extract thermal energy from tail pipe exhaust gas, EGR, charge air, and even the water jacket [16, 17]. Thus, ORC is the WHR technology with the highest ceiling for recovery heat energy from the multiple streams present on a HDD engine. Literature on ORC-WHR can be divided into two groups: experimental studies and modeling/control studies.

Experimental ORC Studies

Cummins conducted a series of ORC-WHR HDD experiments [18, 19, 20, 21] with funding from U.S. Department of Energy. R245fa was selected as working fluid and a turbine was chosen as the expansion device. They noted a 10% brake thermal efficiency improvement when both the tail pipe exhaust gas and EGR streams were utilized as heat sources [21]. AVL also extensively investigated ORC-WHR research on HDD engine [3, 22, 23, 24, 25, 26, 27]. Ethanol was selected as the working fluid because of its low

evaporation pressure, allowing for lighter and lower cost system components [3]. AVL achieved a 3-5% fuel economy improvement utilizing a turbine as the expansion device. Additionally, Bosch experimentally obtained 2.1kW, 5.3kW and 9.0kW turbine generated power from B25, B50, and B75 engine operating conditions with an ORC system coupled to a 12L HDD engine, respectively [28].

ORC-WHR Modeling and Control

S. Quoilin et al created a finite volume (FV) model of a small-scale R245 ORC-WHR system which utilized a scroll expander [29, 30]. Three control strategies were evaluated for power optimization. The first strategy combined a constant evaporating temperature (via pressure control) and constant superheating temperature for establishing ORC working fluid setpoints. The second strategy utilized an optimal evaporating temperature and constant superheating temperature. In the second case, the optimal evaporating temperature was expressed as a first order polynomial function of working fluid mass flow rate, condensation temperature and heat source inlet temperature through a steady state analysis with Engineering Equation Solver. A third strategy examined optimal working fluid pump speed with a fixed working fluid superheating temperature. Only minimal ORC performance differences were revealed between proposed three control strategies.

Hou et al proposed a power plant scale, 100kW ORC-WHR system model with a turbine expander and R245 as the working fluid [31]. The evaporator was modeled with moving boundary (MB) method. A linear quadratic regulator (LQR) was incorporated with a PI controller to track references of: evaporation pressure, vapor temperature, and power output. The same group proposed a linear model predictive control for the ORC-WHR system and showed a satisfactory working fluid setpoint reference tracking performance over mild transient conditions [32].

Peralez et al presented a model-based control for a steam Rankine cycle waste heat recovery system utilizing a volumetric expander [33]. The evaporator was modeled with the MB method and a reduced order model was derived from the full order MB model. Working fluid superheat temperature and evaporation pressure reference tracking were achieved by a feedback controller and a feedforward controller. The same team proposed transient power optimization for the ORC-WHR via dynamic programming [34]. A 0D heat exchanger model was utilized for computational efficiency, while the engine conditions for the optimization were a mild step-change transient.

Xu et al proposed an ORC-WHR system model for HDD application [4]. The parallel EGR and exhaust tailpipe evaporators were modeled with FV method, a turbine was selected as the expansion device, and ethanol was chosen as working fluid. Steady state turbine power optimization was conducted by analyzing the relationship between the expansion turbine and four actuators: working fluid pump speed, mass flow distribution between the parallel evaporators, turbine speed, and coolant pump speed [15].

Yebi et al proposed a nonlinear model predictive controller operating over parallel MB evaporator models for HDD engine ORC-WHR system extracting heat from both the EGR and tailpipe streams [35]. Comparisons between MPC and PID control over the mixed vapor temperature at the outlet of parallel evaporators showed the merit of MPC utilization during working fluid vapor setpoint changes at constant engine conditions.

Transient ORC-WHR Optimization

Even with the wealth of prior ORC research, transient power optimization for an ORC system operating on a HDD engine is rarely researched. Even though Quoilin et al considered different control strategies to optimize the ORC-WHR system power output, that ORC system was designed for a small scale waste heat source with a low temperature range (120-300°C), which is below the typical temperature of HDD engine exhaust gas and EGR streams (300-500°C) [29]. Meanwhile, the optimal temperature was calculated based on steady state, which does not account for heat exchanger thermal inertia. Additionally, while Peralez et al optimized ORC power with dynamic programming over transient engine conditions, the transient condition was merely a mild step change rather than a transient drive cycle [34]. This paper studies the HDD engine power optimization over highly transient engine conditions with a experimentally validated, physics-based, high fidelity ORC-WHR model [4].

Working fluid saturation temperature is the optimal vapor temperature reference with respect to maximizing turbine generated power. However, if saturation temperature is utilized as the control reference, then the ORC system controller must perfectly maintain the exact saturation temperature or risk harming the turbine expander via exposure to working fluid with vapor quality below unity. Therefore, for imperfect controllers, the working fluid temperature reference trajectory must be set greater than the saturation temperature. The greater the working fluid temperature oscillation allowed by the controller, the larger this buffer in control reference temperature must be.

In this work, three ORC working fluid vapor temperature reference creation strategies are described and evaluated over a HDD highly transient driving cycle. The strategies focus on control of the mixed working fluid vapor temperature (i.e. the pre-turbine condition where which combines the outlets of the parallel EGR and tailpipe evaporators). The pre-turbine mixed working fluid reference setpoint strategies addressed herein are: (i) constant working fluid vapor temperature (ii) constant working fluid superheat temperature, and (iii) a fuzzy logic working fluid vapor temperature.

This paper is organized as follows: The ORC-WHR system configuration overview is described and then the component modeling section follows to give a brief introduction of the FV model. After that, the transient optimization problem is formulated and optimization results are analyzed. Lastly, the paper ends with discussion and conclusions.

EXPERIMENTAL ORC-WHR SYSTEM CONFIGURATION

Modern diesel engines typically operate with high pressure exhaust gas recirculation (EGR), making the EGR stream an attractive heat source for a diesel WHR system. The inclusion of EGR as a heat source (along with the exhaust flow) creates options for the connection of the two heat exchangers: in series or in parallel. The parallel configuration is suitable for high mass flow with less pressure loss because the cross sectional area of working fluid flow path increases. A series evaporator connection produces higher working fluid vapor temperatures because the long flow path aides superheat. Considering higher mass flow has been proven fortuitous for power generation [5], the parallel configuration was chosen, and is shown in Figure 1.

The ORC system components are connected in a standard ORC loop, specifically: (i) a low pressure working fluid supply pump, (ii) a high pressure working fluid pump, (iii) two mass flow valves which distribute the working fluid between the evaporators, (iv) two parallel evaporators, (v) an exhaust bypass valve, (vi) a valve upstream of the turbine expander, (vii) a valve before the turbine to bypass liquid working fluid when the system is in warm-up or cool-down state, (viii) a turbine expander for electricity generation, (ix) a condenser downstream of the turbine, and (x) an expansion tank between condenser and low pressure pump to buffer the working fluid flow. This system utilizes ethanol as the working fluid.

A 13L heavy duty diesel engine is connected with the ORC-WHR system. The only connections between engine and ORC-WHR system are the tail pipe (TP) and EGR evaporators, shown in Figure 1. During normal operating conditions, exhaust gas flows through the TP evaporator and transfers heat to the working fluid. A TP evaporator bypass valve is installed to protect the ORC-WHR working fluid from overheat and subsequent working fluid degradation when engine operates at high load. The EGR evaporator replaces the stock EGR intercooler. Due to EGR evaporator outlet temperature constraints, there is no bypass valve installed in the EGR line. The working fluid mass flow to the parallel evaporators is controlled by valves located between the high pressure pump and their respective evaporator.

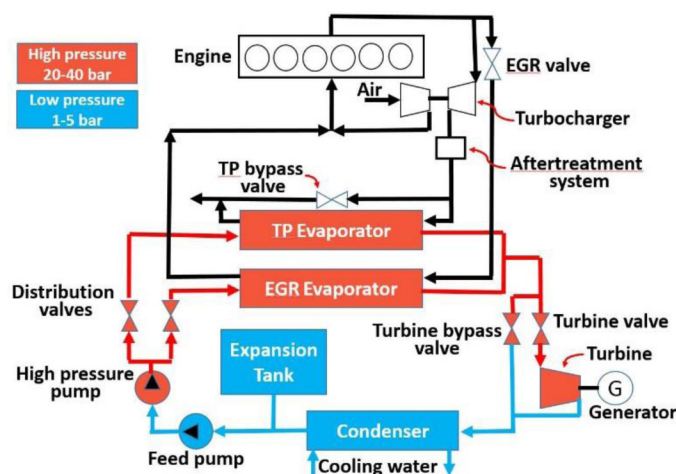


Figure 1. Schematic of ORC-WHR system

ORC COMPONENT MODEL DESCRIPTIONS

The ORC-WHR system is simulated with a physics-based high fidelity model for this transient evaluation. A complete description of the model and its governing equations can be found in [4]. However, the following is a basic summary of the model's principles.

The evaporators are modeled with the FV method, which discretizes the working fluid flow volume, exhaust flow volume and wall volume into 30 cells. Mass balance, energy balance and momentum balance equations are solved in each cell. Working fluid phase boundaries can be interpolated by the vapor quality of each cell. Meanwhile, the high pressure pump supplying working fluid to the evaporators is a positive displacement type and the mass flow rate varies linearly with pump revolution speed.

There are five valves in the system. The two mass flow distribution valves located before the parallel evaporators are modeled as control valves with exposure to incompressible liquid because these two valves control pure liquid ethanol at all times. Another control valve, located before the turbine expander, is modeled as controlling compressible vapor since it opens only when the vapor quality is sufficiently high to protect the turbine from damage. The control valve parallel to the turbine expander interacts with both incompressible liquid and compressible vapor working fluid as the system warms, and is modeled accordingly. The TP bypass valve is modeled as simple on/off binary valve and pressure drop is not considered in the open position. Again, complete modeling details of each component within the ORC system can be found in [4]. However, some updates have been made, namely:

The turbine expander model has been updated since the referenced work was published and now takes the form:

$$P_{turb} = \eta_{thermal} \dot{m}_{turb} (h_{turb,in} - h_{turb,out,is}) \quad (1)$$

where P_{turb} is turbine generated power, $\dot{m}_{turb}$ is turbine mass flow rate, $h_{turb,in}$ is turbine inlet enthalpy, and $h_{turb,out,is}$ is turbine outlet isentropic enthalpy. Thermal efficiency $\eta_{thermal}$ is interpolated from a turbine map which contains the functional dependences shown in (Eq. 2)

$$\eta_{thermal} = \text{map}(N_{turb}, p_{in}, p_{out}) \quad (2)$$

where N_{turb} is turbine speed, P_{in}, p_{out} are turbine inlet and outlet pressure. The operating expansion ratio for the turbine expander ranges from 5-30, which is relatively high compared to typical turbo-charger [36, 37]. The high expansion ratio leads to choked flow of the turbine and thus the mass flow rate has linear relationship with the inlet pressure:

$$\dot{m}_{turb} = a_{turb,1} p_{in} + a_{turb,2} \quad (3)$$

where $a_{turb,1}$ and $a_{turb,2}$ are two constant coefficients. Turbine outlet enthalpy is calculated as follows: