

Simulation of Organic Rankine Cycle Power Generation with Exhaust Heat Recovery from a 15 liter Diesel Engine

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ABSTRACT

The performance of an organic Rankine cycle (ORC) that recovers heat from the exhaust of a heavy-duty diesel engine was simulated. The work was an extension of a prior study that simulated the performance of an experimental ORC system developed and tested at Oak Ridge National laboratory (ORNL). The experimental data were used to set model parameters and validate the results of that simulation. For the current study the model was adapted to consider a 15 liter turbocharged engine versus the original 1.9 liter light-duty automotive turbodiesel studied by ORNL. Exhaust flow rate and temperature data for the heavy-duty engine were obtained from Southwest Research Institute (SwRI) for a range of steady-state engine speeds and loads without EGR. Because of the considerably higher exhaust gas flow rates of the heavy-duty engine, relative to the engine tested by ORNL, a different heat exchanger type was considered in order to keep exhaust pressure drop within practical bounds. The effectiveness-NTU based model of the shell-and-tube heat exchanger in the original simulation was replaced with direct finite-difference modeling of a counter-flow finned heat exchanger. A full-scale version of the finned heat exchanger was built from aluminum extrusions and tested for pressure drop in the exhaust of a 6.7 liter Cummins turbo diesel, for comparison with model predictions. In the simulation the engine exhaust heated the refrigerant R245fa working fluid which was expanded in an integrated turbo-generator. Optimizing heat exchanger pressure drops with heat transfer rates was found to be essential. The predicted increase in overall engine/ORC thermal efficiency was approximately 5% across the entire speed/load range of the engine. Refrigerant mass flow rates for peak ORC system power were in the range of 0.7 to 1.45 times the engine exhaust flow rate.

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INTRODUCTION

A thermodynamic model was developed to simulate the performance of an organic Rankine cycle (ORC) applied to the exhaust of a 15 liter turbodiesel engine operated over a range of steady-state speeds and loads. The exhaust gas temperature and flow data for this engine were obtained from Southwest Research Institute (SwRI) [1]. The model is based on a previous model of Valentino and Hall [2] developed to simulate the performance of an experimental ORC developed and tested at Oak Ridge National Laboratory (ORNL) and applied to a light-duty automotive-type 1.9 liter turbodiesel [3].

Of primary interest are the predicted ORC power output, thermal efficiency, heat exchanger effectiveness and pressure loss, and the sensitivity of these parameters to ORC refrigerant flow rate. The biggest different between the present and previous model is the heat

exchanger sub-model. Within the heat exchanger, heat is extracted from the hot engine exhaust and used to vaporize and superheat the ORC refrigerant. The previous model used an effectiveness-NTU method to model the heat transfer in the series of shell-and-tube heat exchangers used by ORNL in their experiments. The heat transfer sub-model in the simulation presented here considers a straight-finned extruded aluminum heat exchanger based on one constructed by the authors and installed in the exhaust of a Cummins 6.7 liter diesel. Experimental data from this heat exchanger were used to validate pressure drop and pumping loss predictions from the model. In the present model the effectiveness-NTU model was not used since the simpler geometry of the finned heat exchanger lent itself to direct numerical simulation of heat exchanger performance using well established correlations.



ORC Background Literature

The use of an organic Rankine cycle to recover waste heat from engine exhaust for the purpose of increasing engine thermal efficiency is a topic of recurring interest. Seminal studies were performed and reported in the literature by Patel and Doyle in 1976 [4]. They determined a potential fuel efficiency improvement of 15% for heavy-duty engines and verified this experimentally with a Mack 676 diesel. Since then there have been quite a few studies of ORCs applied to heavy-duty engines. These include both experimental studies [5, 6] and modeling/analytical studies [5, 7, 8, 9]. The background presented below focuses on more recent ORC studies.

Kunte, and Seume [10] performed a modeling study of an ORC system for both heavy-duty truck and passenger car applications. The primary focus of the study was to optimize characteristics of the impulse turbine used to expand the refrigerant. Engine exhaust temperatures and flow rates were assumed based on the exit conditions from emission controls. Heat exchanger operating conditions were estimated based on a study by Span et al. [11].

Yamaguchi et al. [12] modeled the improvement in fuel consumption of a diesel-Rankine combined cycle based on experimental engine exhaust temperatures and flow rates from a high-boost diesel engine. They predicted an improvement in BSFC of 2.6-3.0% for full load (BMEP = 20 bar). They did not model the heat exchangers and used assumed temperatures, pressures, and refrigerant flow rates in the system based on the thermodynamic properties of various Rankine cycle working fluids that included water, methanol, toluene, R134a and R245fa.

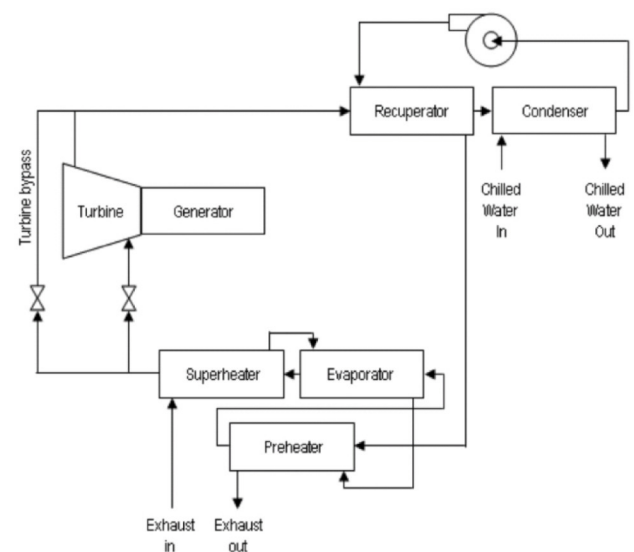
Latz et al. [13] theoretically evaluate expansion devices for Rankine type waste-heat recovery systems based on their normalized operating speed and rotor diameter. They studied the effect of expansion-pressure ratio and type of working fluid to identify conditions that would accommodate the use of turbine expanders. Their results suggest that the “efficient use of turbine expanders requires changes in the design of the Rankine cycle that have negative effects on the performance of the waste-heat recovery system as a whole”, and “is primarily because of the need to use lower expansion-pressure ratios than designs based on reciprocating expanders.” They did not consider heat exchanger performance.

Shu et al. [14] performed simulations of a bottoming organic Rankine cycle coupled to a diesel engine capturing heat from both the exhaust and water jacket. Their predicted results indicated that the maximum power and efficiency of the ORC were as high as 18.8 kW and 9.6%. They studied two engine output conditions of 138 and 243 kW. They modeled the performance of 7 counter-flow flat-plate heat exchangers based on manufacture's data. They assumed a dual-fluid heat exchanger system in which oil was heated by the engine exhaust flow and the oil was then used to heat the working fluid in a second heat exchanger. This was done to prevent decomposition of the working fluid at high temperature. A fixed flow rate for the working fluid (R245fa) was assumed. No pressure drop data were presented.

Hossain and Bari [15] experimentally investigated the ability of shell and tube heat exchangers to extract heat from the exhaust of a diesel engine with parallel and counter flow arrangements using water as the working fluid. Heat exchanger parameters such as number and diameter of the tubes, length of the heat exchanger and the diameter of the shell were optimized using numerical simulations. They proposed design modifications that with a predicted improvement in heat exchanger performance of 23.7% additional power at 30 bar pressure yielded a 20% improvement of brake specific fuel consumption.

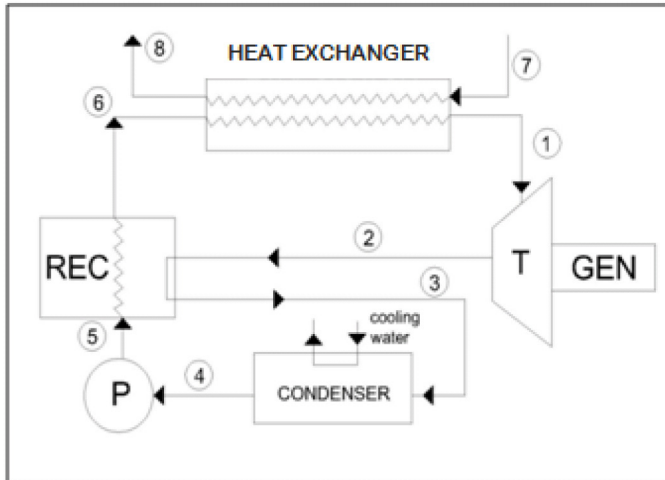
Sprouse and Depcik [16] simulated ORC performance for eight dry fluids (butane, pentane, hexane, cyclopentane, benzene, toluene, R245fa, and R123) based on the exhaust from a single-cylinder diesel engine. They found pentane to give the best performance, yielding at 10% improvement in BSFC. They assumed a two-stage heat exchanger system and concluded that a parallel flow evaporator arrangement can operate more efficiently as a result of the high temperature difference in the area of low heat transfer coefficients within the heat exchanger. They used a log-mean-temperature difference analysis for a two zone heat exchange with assumed area and heat transfer coefficients. They found predicted ORC power to be a strong function of turbine pressure ratio for pressure ratios less than approximately 10.

Feru et al. [17] performed a modeling study of ORC performance based on a Euro-VI heavy-duty diesel engine assuming the use of a piston expander and ethanol working fluid. They considered a two evaporator system, one heated by the EGR flow and the other with the exhaust flow. The heat exchanger performance was modeled based on an energy balance and three heat transfer coefficients corresponding to the liquid, superheated and mixed-phase regimes of the working fluid. The evaporator heat transfer coefficients varied by approximately a factor of 5 between the liquid phase and the superheated vapor. The model was validated against measurements of temperatures and flow rates in the evaporators and expander.



(a)

Figure 1. Schematic of ORNL setup (a) [2] and modified recuperator (b). State points (b).



(b)

Figure 1. (cont.) Schematic of ORNL setup (a) [2] and model representation with State points (b).

Hossain and Bari [18] presented a combined experimental and modeling study of ORC performance. A pair of shell-and-tube heat exchangers was used for vaporizing and superheating the working fluid, respectively. These were coupled to the exhaust of a 40 kW diesel generator. They experimentally investigated heat transfer to both water and HFC-134a working fluids. The simulations were used to optimize the geometry of the heat exchangers in terms of number of tubes, tube length and baffle design. Based on their model and heat transfer experiments they predicted an improvement in overall engine bsfc of 12% and 11% respectively for water and HC-134a for an applied ORC.

Two important performance aspects that are not discussed in many of the previous studies are the effect of refrigerant flow rate on ORC performance and the effects of the pressure drops through the heat exchangers with the resulting parasitic flow-work losses. Pressure losses can be considerable for both shell-and-tube and flat-plate heat exchangers.

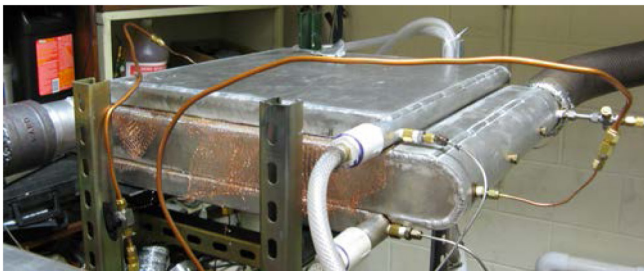


Figure 2. Photo of heat exchanger as installed in the exhaust of Cummins 6.7 liter engine.

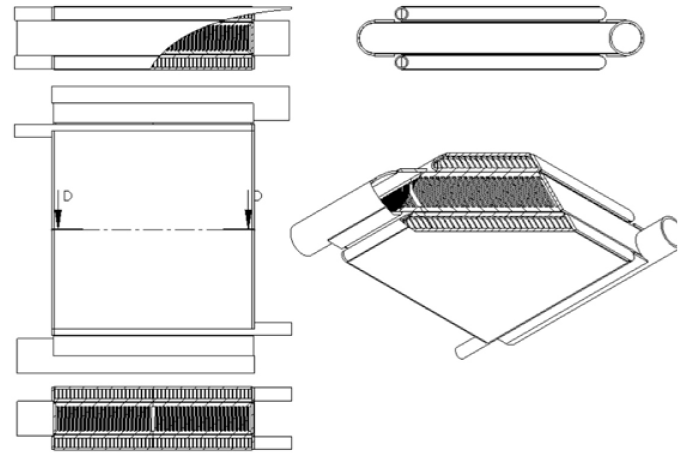


Figure 3. Engineering drawing of the experimental heat exchanger.

Methodology

As mentioned above, the ORC model presented here was originally developed based on performance data from the ORC experimental study performed by Briggs et al. [3] at ORNL. A schematic showing the layout of their experimental ORC components is shown in Fig. 1a. Figure 1b shows a schematic of our model's simplified interpretation of that system's components along with the thermodynamic state points used in the model. In the ORNL experiments the R245fa refrigerant preheater, evaporator and super heater were mounted on carts. The turbine-generator, recuperator, and refrigerant pump were on another cart. The condenser was mounted separately on another test-stand. The superheater and evaporator were built from shell-and-tube EGR coolers from a heavy-duty truck; the preheater and recuperator were industrial flat-plate heat exchangers. Chilled water at 10°C cooled the condenser; this was certainly colder than would be typical of the air-cooled condenser that would be used on-board a vehicle, but conceptually similar. The Barber-Nichols Inc. integrated turbine-generator had a peak operating speed of 80,000 rpm and was relatively compact at approximately 38 cm long and 15 cm in diameter. The turbine-generator used in the model presented here was assumed to have a larger capacity, but with the same generating efficiency. Various component efficiencies as measured in the ORNL experiments are presented in Table 1. These values were used in the model and assumed constant. More details on the ORC system can be found in references 2 and 3.

Table 1. ORC parameters derived for the baseline operating conditions of the ORNL experiments.

Baseline ORC Parameters	
ORC Turbine/generator efficiency*	0.65
Recuperator efficiency*	0.13
Refrigerant pump efficiency*	0.39

*Taken as constant throughout analysis

The major simplification in the original model was that the three experimental heat exchangers, the preheater, evaporator, and superheater were modeled as a single shell-and-tube heat exchanger. This configuration, as shown in Fig. 1b, was retained for the present study, but with the substitution of the finned heat exchanger for the shell-and-tube heat exchanger.

The present study had two primary objects. The first was to model the performance of a larger engine for which the application of an ORC would likely be more practical. The second was to use, as the basis of the model, an engine-exhaust-gas to refrigerant heat exchanger that was more compact than those used by ORNL. The heat exchanger modeled in the present study is based on one we designed, built, and tested at the University of Texas. Figure 2 is a photo of the heat exchanger installed in the exhaust of a Cummins 6.7 liter turbodiesel. Figure 3 is an engineering drawing of the experimental heat exchanger. This heat exchanger was originally designed to recover engine exhaust heat for delivery to thermoelectric modules [19].

The experimental heat exchanger was a straight-fin type and was constructed from aluminum heat-sink stock. It consisted of three separate heat exchangers, the thicker middle section for the engine exhaust and the two outer sections designed for coolant flow. The thicker middle section was constructed using two identical sections of heat-sink material. The fins of one side, while facing the fins of other, were inserted into the gaps between the fins of the opposing side resulting in a fin spacing one-half that of the heat sink material. The fin height was therefore equal to the 6.3 cm fin height of the original heat sink stock. For this hot-side heat exchanger the fin thickness was 0.30 cm and the spacing between fins was 0.20 cm.

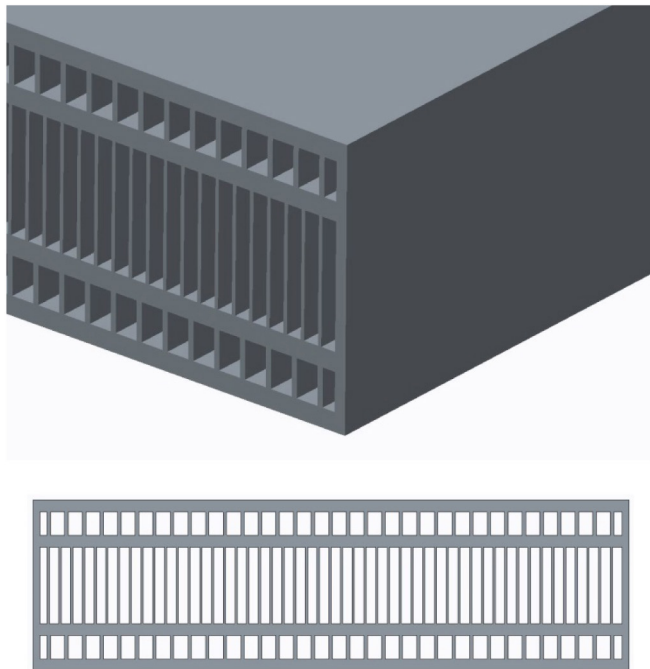


Figure 4. Configuration of conceptual heat exchanger used in the model. Note: for clarity the fin spacing shown is different from that of the experimental heat exchanger and that used in the model.

The geometry of the heat exchanger used in the model was somewhat different from that of the experimental heat exchanger. Instead, a geometry similar to that shown in Fig. 4 was used in the model in which the exhaust hot-side and the two outer refrigerant cold-sides were taken to be combined as a single extruded aluminum piece. The largest uncertainty in modeling the heat transfer for the experimental heat exchanger geometry was the thermal contact resistance between the hot-side and each of the two cold-side heat exchangers. Contact resistance is very sensitive to factors such as surface finish, surface flatness, and coupling force [20]. Contact resistance and its uncertainties were eliminated by considering the configuration shown in Fig. 4 in which the gap between the hot-side and the cold-sides was eliminated, resulting in the one piece design. Other thermal resistances can be found with much greater accuracy from well-established convective flow correlations, 1-D conduction through the aluminum walls, etc.

Another difference between the experimental and model heat exchangers was that, in the model, the fins were assumed to be attached to both sides of the heat exchanger rather than to just one side; this had the effect of making each fin one-half its actual length, effectively, extending to the middle of the hot-side adjoining the fin from the opposite side. Because of these differences between the experimental and model heat exchangers, experiments were conducted to validate only the pressure drop and pumping work predictions of the model and not the heat transfer. From a flow perspective the model heat exchanger had the same geometry as the experimental heat exchanger on the hot-side. As given below, the fin geometry on the cold-sides had to be modified in the model from that of the experimental heat exchanger.

ORC Model Description

The simulations were performed using a thermodynamic Rankine cycle model developed for the project; it utilized state properties and component efficiencies derived from the ORNL experiments. The model was coded in MatLab. Thermodynamic properties of the R245fa refrigerant were sourced from an on-line property calculator developed and maintained by the National Institute of Standards and Technology (NIST) [21]. The refrigerant preheater, evaporator, and superheater were modeled, collectively, as a counter-flow finned heat exchanger. Thermodynamic properties such as specific heat of the diesel exhaust gases were determined from its molar composition, derived from known air and fuel flow rates and assuming complete combustion of the fuel having assumed molecular structure of dodecane ($C_{12}H_{26}$).

Convection heat transfer and pumping power were modeled using standard empirical correlations for either laminar or turbulent flow, as applicable. The flow properties (dynamic viscosity, and thermal conductivity) were calculated as a function of temperature using correlations assuming properties of air. The following correlations for Nusselt number based on hydraulic diameter (Nu_D) and friction factor (f) were used [22]:

$$Nu_D = \frac{8.24}{4.36} 0.023 Re_D^{4/5} Pr^{1/3}$$


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