

Simulation of Organic Rankine Cycle Electric Power Generation from Light-Duty Spark Ignition and Diesel Engine Exhaust Flows

Ryan Valentino and Matthew J. Hall
Univ. of Texas-Austin

Thomas Briggs
Southwest Research Institute

ABSTRACT

The performance of an organic Rankine cycle (ORC) used to recover waste heat from the exhaust of a diesel and a spark ignition engine for electric power generation was modeled. The design elements of the ORC incorporated into the thermodynamic model were based on an experimental study performed at Oak Ridge National Laboratory in which a regenerative organic Rankine cycle system was designed, assembled and integrated into the exhaust of a 1.9 liter 4-cylinder automotive turbo-diesel. This engine was operated at a single fixed-load point at which Rankine cycle state point temperatures as well as the electrical power output of an electric generator coupled to a turbine that expanded R245fa refrigerant were measured. These data were used for model calibration.

The model was used to predict the steady-state power output, thermal efficiency, and state point temperatures of the ORC as a function of refrigerant flow rate and engine speed/load for the diesel engine used in the experiments and for a spark ignition engine based on experimental exhaust flow data.

Net power of the ORC was predicted to increase linearly with engine speed and load, and the performance characteristics were very similar for the two engines despite the large differences in exhaust temperatures and exhaust mass flow rates characteristic of diesel versus S.I. engines. While the maximum predicted net power output of the ORC was about 11.5 kW for the S.I. engine at high speed and load, for light loads in the range of 2-4 bmep and engine speeds of 2000 rpm and lower, the predicted net power was in the range of 1 kW or less, calling into question its practicality for light-duty vehicles.

CITATION: Valentino, R., Hall, M. and Briggs, T., "Simulation of Organic Rankine Cycle Electric Power Generation from Light-Duty Spark Ignition and Diesel Engine Exhaust Flows," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-1644.

INTRODUCTION

With oil prices, energy demands, and the global population increasing, there has recently been much interest in energy efficiency. As transportation is responsible for 30-40 % of industrial nations' energy use, automobile efficiency is a particularly relevant matter [1]. The Corporate Average Fuel Economy (CAFE) standards require automobile manufacturers' vehicle fleets to meet specified fuel economy thresholds in the United States. The minimum acceptable fuel economy by CAFE standards for passenger vehicles is set to nearly double from 27.5 miles per gallon (mpg) in the 1990s and 2000s to 54.5 mpg by 2025 [2]. There is thus sufficient interest in vehicle efficiency to warrant investigation.

Advances in engine technology have brought engine efficiency closer and closer to its peak achievable value as limited by the laws of thermodynamics. Indeed irreversibilities in the combustion process result in the loss of 20-30% of the fuel's exergy, which limits the achievable efficiency of an internal combustion engine [3]. Recent focus on the combustion process and the air/fuel mixture have resulted in the Homogeneous Charge Compression Ignition (HCCI) and Gasoline Direct-Injection (GDI) technologies that have yielded gains in thermal efficiency [1, 4]. Other options for improving engine thermal efficiency lie in the conversion of waste heat in engine exhaust to useful work, either electrical or mechanical. Alternative methods of improving vehicle efficiency have thus been the subject of recent studies. In fact, around 70 % of the fuel's chemical



energy is lost to the environment through the exhaust or through heat transfer to the environment for a standard vehicle [5]. Various alternative energy harvesting methods for waste heat recovery include turbocharging, turbocompounding, and thermoelectric generation. Ringler et al. [4] compare many options for waste heat recovery, and their results are perhaps best summarized in Figure 1.

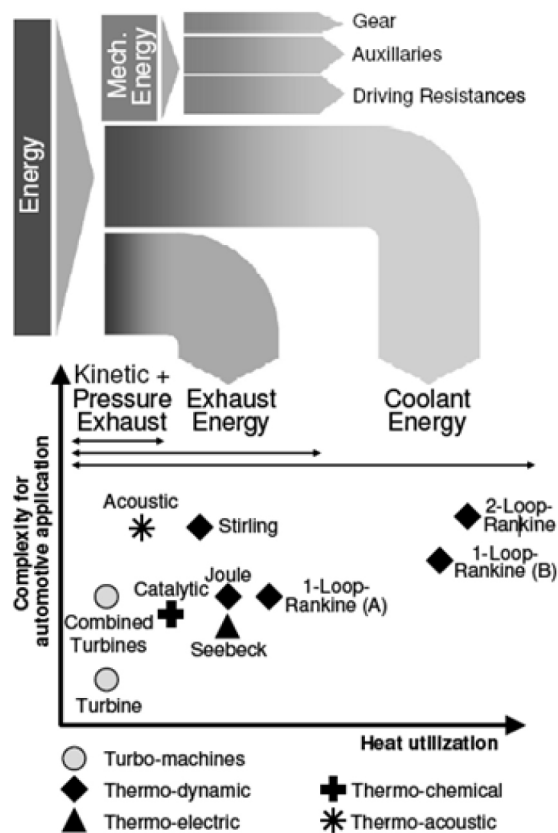


Figure 1. Energy utilization vs. complexity of different heat recovery systems [4].

As can be seen in the figure, the organic Rankine cycle (ORC) is one of the best candidates for waste heat recovery. In a Rankine cycle a working fluid is pumped to a high pressure, superheated, expanded, condensed, and the cycle repeats. Work is extracted from the fluid during the expansion process. Energy loss in the exhaust through both pressure and temperature differentials can be recovered, which is an advantage of the Rankine cycle over some of the alternative methods [4].

There have been many studies, both analytical [6, 7, 8, 9] and experimental [6, 10], demonstrating the viability of Rankine cycle based waste-heat recovery when coupled with heavy duty diesel engines. Indeed as early as the 1970s, during the energy crisis, studies showed that implementing a Rankine cycle waste-heat recovery system could improve vehicle efficiency. Patel and Doyle calculated a potential fuel efficiency improvement of 15% in heavy duty applications and verified this with an experimental setup using a Mack 676 [11].

While these studies are interesting and modifications in heavy duty vehicles can significantly contribute to the increased efficiency of the total transportation fleet, the application of Rankine cycles in light duty vehicles has been less extensively studied. This study presents a thermodynamic model used to investigate the viability of an ORC coupled with a light duty engine; both a diesel and a S.I. engine are considered. A few other studies have been done on the topic. BMW researchers, Ringer et al. [4], found that an ORC system could yield power outputs of 0.7 to 2 kW or up to a 10% increase in total power from a light duty S.I. engine, at typical highway loads by harnessing the waste-heat of both the exhaust and the coolant. Their study included an extensive analytical portion that included the development of a model that was subsequently validated with a less extensive experiment [4]. Behr, an automotive systems manufacturer and supplier, found fuel economy improvements of 2.2% harnessing only the tailpipe exhaust and 6.9% harnessing the tailpipe exhaust and the exhaust gas being recirculated (EGR) [12].

Zhang et al. developed an experimental system to validate previous theoretical work. Their experiment simulated a Toyota 8A-FE 4-cylinder gasoline engine, and yielded an ORC thermal efficiency of around 14% and a power output of approximately 0.7 kW, which compared favorably with their previous calculations. Interestingly, this group used a refitted Wankel engine to expand the working fluid instead of a turbine [1]. This could perhaps allow for a wider range of appropriate working temperatures. In general, when using a turbine as an expander, the working fluid should not be near its saturation point, but the use of the Wankel engine as the expander may perhaps allow for the working fluid to be run in a saturated or near-saturated state without risk of system degradation.

In similar fashion, the authors of the present study have developed a model of an ORC. The model includes the capability of including a recuperator/regenerator in the cycle, which serves the purpose of preheating the working fluid before it enters the exhaust/refrigerant heat exchanger, increasing cycle efficiency. The heat source for this regenerative process is the expanded working fluid (after it passes through the turbine but before it has reached the condenser). The model was created and validated based upon an experimental analysis performed by one of the authors (Briggs) and others at Oak Ridge National Laboratory (ORNL). Figure 2 illustrates the cycle. Figure 2a is a schematic of the ORNL experimental setup; Fig 2b illustrates the cycle as modeled, showing thermodynamic state points and how the experiment's preheater, evaporator, and superheater were modeled as one heat exchanger, referred to collectively as the heat exchanger.

In the experiment, the regenerative ORC captured a portion of the exhaust flow enthalpy of a 1.9L 4-cylinder automotive turbo-diesel to produce up to 4 kW of net electrical power resulting in an increase in overall thermal efficiency of 2.6% [5]. The engine was run without EGR.

ORNL EXPERIMENTAL SYSTEM

Results of ORNL ORC Experiments

At the single baseline operating condition at which the combined engine and ORC system was operated, an overall thermal efficiency of 45% was achieved, and the net electrical output of the ORC system was 4.0 kW; this accounted for the parasitic loss of the refrigerant pump. The heat flux from the exhaust to the ORC was 31.4 kW which gave an ORC thermal efficiency of 12.7%. The operating conditions and results are summarized in [Table 1](#). [Table 2](#) provides the measured temperatures and pressures at the various state points around the ORC. [Figure 3](#) presents a Temperature/Entropy (T-s) diagram of the thermodynamic states of the baseline conditions. An energy balance, based on these values, applied to the preheater, evaporator, and superheater, was used to determine the baseline experimental mass flow rate of the refrigerant; this was determined to be approximately 0.086 kg/s, assuming negligible heat loss from the well-insulated system.



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

Table 1. Operating conditions and output of combined engine and ORC system.

Engine speed	2250 rpm
Brake torque	279.5 N-m
Brake power	65.9 kW
Fuel consumption	12.96 kg/hr
Brake thermal efficiency	42.4%
Turbine outlet temperature	436 °C
Exhaust mass flow	313.2 kg/hr
Generator output	4.3 kW
Pump input	0.3 kW
Net ORC power	4.0 kW
Engine+ORC power	69.9 kW
Combined efficiency	45%

Table 2. Temperatures and Pressures at the various thermodynamic state points of the ORC.

State	Temp. [C]	Pressure [MPa]
Heat Exchanger Refrigerant out (1)	260	2.06
Turbine out (2)	216	0.15
Recuperator out - low pressure (3)	50	0.12
Condenser out (4)	10	0.13
Pump out (5)	13	2.02
Recuperator out - high pressure (6)	51	2.08
Heat Exchanger Engine Exh. in (7)	436	
Heat Exchanger Engine Exh. out (8)	99*	

*Not reported by ORNL, acquired independently

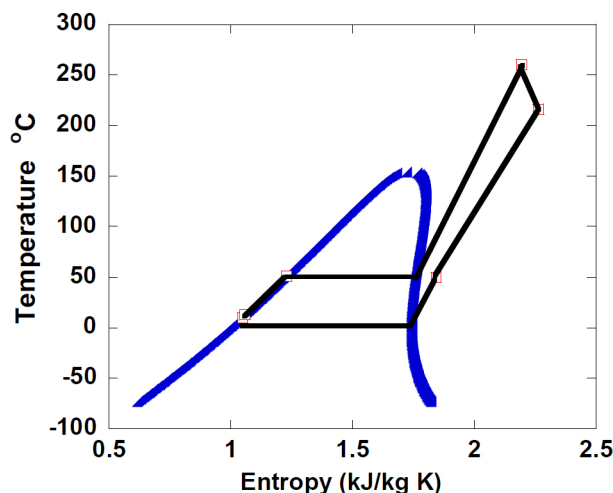


Figure 3. T-s diagram of the thermodynamics states of the R245fa refrigerant for baseline conditions.

DESCRIPTION OF MODEL

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. 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) [13]. The refrigerant preheater, evaporator, and superheater were modeled, collectively, as a counter-flow shell-and-tube heat exchanger using the effectiveness-NTU method. Thermodynamic properties 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 ($C_{12}H_{26}$).

For the baseline operating conditions the recuperator efficiency was derived from an energy balance on the device. The efficiency was 13% and was assumed constant when modeling other operating conditions. For the baseline conditions the heat exchanger effectiveness (ϵ), NTU, and heat exchanger capacity ratio (Cr) were also determined. The results of these analyses are shown in Table 3. In the model, heat transfer losses from the ORC components were assumed negligible except of the recuperator, for which an energy balance indicated a non-negligible heat loss. The heat loss from the regenerator in the model was scaled with refrigerant flow rate. From the baseline experimental data, the efficiency of the combined turbine/generators was calculated as 65%, and the refrigerant pump efficiency was 39%. These values were used for the other cases examined.

Table 3. Heat exchanger parameters derived for the baseline operating conditions for the recuperator and heat exchanger.

Baseline Heat Transfer Parameters	
Heat exchanger Effectiveness	0.88
Recuperator Efficiency*	0.13
Heat exchanger NTU	4.1
Heat Exchanger Capacity Ratio	0.73

*Taken as constant throughout analysis

The effectiveness of the refrigerant heating units, combined as the heat exchanger in the model, is a function of the flow rates of both the refrigerant and the engine exhaust flow. A model for the effectiveness was derived to account for this. For a counter flow heat exchanger, the effectiveness (ϵ) is a function of NTU and the capacity ratio Cr [14].

$$\epsilon = \frac{1 - e^{(-NTU(1-Cr))}}{1 - Cr * e^{(-NTU(1-Cr))}}$$

(1)

where: