

A Model Approach to the Sizing of an ORC Unit for WHR in Transportation Sector

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ABSTRACT

Internal combustion engines are actually one of the most important source of pollutants and greenhouse gases emissions. In particular, on-the-road transportation sector has taken the environmental challenge of reducing greenhouse gases emissions and worldwide governments set up regulations in order to limit them and fuel consumption from vehicles. Among the several technologies under development, an ORC unit bottomed exhaust gas seems to be very promising, but it still has several complications when it is applied on board of a vehicle (weight, encumbrances, backpressure effect on the engine, safety, reliability). In this paper, a comprehensive mathematical model of an ORC unit bottomed a heavy duty engine, used for commercial vehicle, has been developed. The model is completed with the sizing of the two exchangers involved in the ORC plant: the heat recovery vapor generator (HRVG), which affects the backpressure at the engine exhaust, and the condenser, which is realized with the similar technology of a radiator, and it affects vehicle frontal area occupation. Also weights have been calculated. Thanks to the model, a wide analysis of the recoverable power and the exchanger sizing has been performed on the ESC-13 engine working points. For each engine working point the backpressure effect due to the presence of the HRVG has been evaluated and overall ORC unit weight has been estimated. The extra propulsive power related to weight increase has been estimated considering the ETC cycle (which is the correspondent transient cycle of the ESC-13 modes) and it is compared to the ORC mechanical power recovered and the overconsumption due the backpressure effect. A real net power recovered is, therefore, assessed.

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INTRODUCTION

The energy recovery from internal combustion engines is one of the most important technologies suitable to reduce emissions and fuel consumption in the transportation sector. In particular, this recovery is huge when it is applied at the exhaust gases, which waste more than 30% of the chemical energy of the fuel. This recovery can be done directly, using for instance a secondary turbine that converts exhaust enthalpy in useful work [1, 2, 3], or by thermoelectric conversion (TEG), which recently reaches greater efficiencies and very low engine impact [4, 5], or making use of a thermodynamic cycle that utilizes the exhaust gases as hot source [6].

The waste heat recovery by thermodynamic cycles has been studied from several decades [7] and different cycles have been proposed [8, 9, 10], but the most suitable is surely the Rankine one [11]. This can be done with working fluids that have suitable critical temperatures for the matching with the hot and cold source ones and characteristics of non-flammability and non-toxicity. These fluids are organic ones and the research of the best one involves also its global warming and ozone depletion potentials. At the moment, for small scale Organic Rankine Cycle (ORC) recovery, R245fa seems to be the most utilized one [12, 13], but new fluids (more environmental friendly) are going to substitute it in next years [14, 15] and the potentiality of mixtures have also been demonstrated [16].

Although ORC for waste heat recovery in large plants is widely utilized and technological proven, it still meets several difficulties in small scale plants, in particular when it is applied on board vehicle. In fact, there are some limiting factors that actually prevent the market application of the ORC recovery in transportation sector:

1. energy conversion and utilization on board: the mechanical power recovered by ORC can be used as it is and reported in crankshaft (with not straightforward complications) or converted in DC electrical power and stored in batteries (with higher energy losses in the mechanical/electrical conversion [17]);
2. the weight of the ORC unit is an important issue, in particular for light duty applications, where the extra-weight present on board brings to a not-negligible overconsumption of the vehicle. On the other hand, this can be not so significant in high heavy duty (HHD) applications [18], in particular for long-hauling vehicles [19];
3. the presence of a heat exchanger (so called Heat Recovery Vapor Generator, HRVG) on the exhaust brings to an increase backpressure at the exhaust manifold (also in turbocharged engines [20]) that leads to an overconsumption of the engine itself;
4. cold source on board is still not clearly defined: it can be represented by external air or a cooling fluid (water or ethylene glycol). In the first case, the condenser must be a secondary

radiator and its dimensions and frontal area occupation is not always negligible. In the second case, the condenser can be cooled by the engine cooling fluid (which brings to a significant oversizing of the radiator) or a secondary cooling circuit (which, definitely, requires a secondary radiator) [21, 22].

Hence, great importance is represented by the sizing of the components not only for weights and encumbrances, but also for their effects on the fuel consumption of the vehicle. In this paper, the Authors present a mathematical model of an ORC recovery plant bottomed a heavy duty engine, particularly focused on the sizing of the condenser (considered as a radiator) and the heat recovery vapor generator (HRVG). The model was built considering the real technological limits related to a small size ORC plant: pressure levels, temperatures, fluid affordability [23, 24]. Thanks also to data coming from previous experimental activity [25], the model has been applied on an IVECO F1C 3.0L turbocharged diesel engine running a steady ESC-13 modes homologation test. Once the thermodynamic ORC has been reconstructed for each steady engine working point (i.e. hot source characterization), both heat exchangers have been designed and the effect of their dimensions on board were introduced in order to assess the real overall performances of the powertrain system (ICE + ORC). Weight increase was studied considering the reference vehicle on the ETC homologation transient cycle, which is the correspondent of the steady ESC-13. On the other hand, backpressure effect, due to the presence of evaporator on the exhaust line, has been calculated thanks to experimental data on the same engine and a final net power recovered has been estimated.

HOT SOURCE CHARACTERIZATION

The energy recovery analysis via ORC power plant has been proposed on a commercial vehicle (Iveco Daily) propelled by F1C 3.0 L heavy duty engine. Characteristic data of the vehicle and the engine has been reported in Table 1.

Table 1. engine and vehicle data

Kerb weight	2284 kg
Gross vehicle weight	6500 kg
Wheelbase	4100 mm
Tires and wheels	225/75 R 16
Drag coefficient	0.35
Frontal area	4.1 m ²
Engine cooling radiator size	649x440x36 mm
Engine displacement	2998 cm ³
Cylinders	4 in line
Max torque	400 Nm@2000 RPM
Max power	130 kW@3500 RPM

The considered engine has been fully tested on a test bench and, in particular, exhaust gas has been characterized when the ESC-13 homologation test is performed. ESC number points have been reported in Figure 1 considering that idle mode has been neglected being useless for waste heat recovery purposes. It shows also exhaust

mass flow rate and temperature (measured after catalyst section). These values represent the working conditions of the hot source on which the ORC plant is performed.

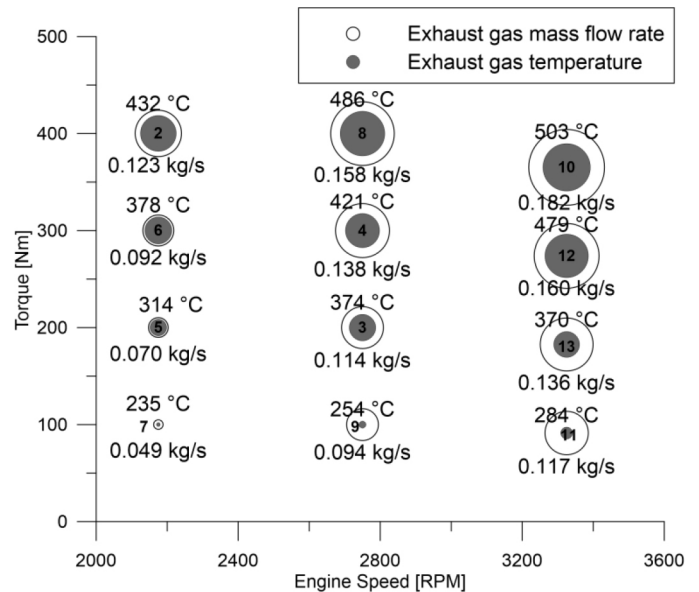


Figure 1. exhaust gas characterization of the IVECO F1C on the steady ESC-13 modes cycle (idle mode was neglected)

ORC THERMODYNAMIC MODEL

The thermodynamic model of the ORC developed by the Authors [26] manages the thermodynamic parameters in order to maximize the mechanical power of the cycle (i.e the product between fluid mass flow rate and useful work), for each different hot source working point considered.

Making reference to Figure 2, each thermodynamic state of the fluid can be defined knowing two independent thermodynamic variables (pressure, temperature or quality). In particular, the enthalpy values are evaluated thanks to a fluid properties database [27].

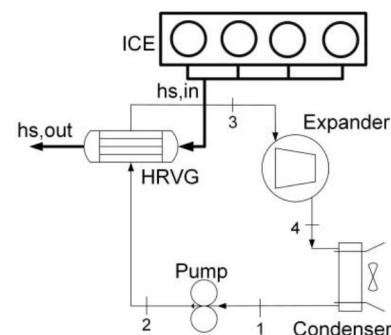


Figure 2. considered ORC cycle scheme

First, the fluid mass flow rate can be calculated as in eq. 1, according to a thermal balance in the heat recovery vapour generator.

$$m_{wf} = \frac{m_{hs} (H_{hs,in} - H_{hs,out})}{H_3 - H_2}$$

Where the exhaust enthalpies are evaluated knowing the gas temperature downstream the catalyst and considering mean value of the specific heat. The temperature at the outlet of the HRVG is imposed to be equal to 150 °C. The useful work is defined as in eq. 2:

$$L_u = (H_3 - H_4) - (H_2 - H_1) \quad (2)$$

The model iterates the values of evaporating and condensing pressure and that one of the superheating temperature in order to find the maximum of the power produced, as in eq. 3:

$$P_u = m_{wf} \eta_m L_u \quad (3)$$

In the model, some technological and thermodynamic constraints have been taken into account: a) maximum pressure is limited to 25 bar due to preserve the functionality of the machines (in particular volumetric ones) and avoid the use of special sealing; b) superheating value is fixed to 15 K: a slight superheating is suitable to avoid liquid phase in expansion transformation, but high superheating does not allow higher efficiency [28]; c) condensing temperature is fixed to 60 °C because the cold source on board is represented by external air in summer condition (i.e. 40 °C); d) mechanical efficiency η_m is equal to 0.9, while volumetric efficiency is considered equal to 1, although it is well known how it can be very low in particular for volumetric expanders [29, 30]. After a wide selection of fluid, the chosen one is R245fa, for its good thermodynamic performances and best actual affordability. Hence, once a specific steady ESC point is assumed, the sizing of the two heat exchangers must be re-calculated (and also the other components). Once a specific design has been assumed, the ORC based power unit will behave in design conditions only for that point. In all the other operating points of the engine, the unit will be in off design condition. So, the management of the unit should be done according to a map (previously defined) which will give the setup of the various control variables (fluid flow rate via pump speed, gas flow rate via an eventual by pass, expander speed of revolution via a four quadrants inverter, cooling fluid at the condenser, etc.) which will reduce the off design degree of the main components.

HRVG SIZING

The Heat Recovery Vapour Generator is a heat exchanger that has the role to extract heat from the exhaust gases for completely evaporating the working fluid and bring it to the desired superheated thermodynamic state. Among the different technologies studied, the shell and tubes one seems the most suitable, in particular if tubes are properly finned. This ensures a high thermal efficiency and low pressure drops. In this work, exhaust gases are considered to pass in the tubes side, while working fluid is in the shell.

The sizing procedure makes use of the LMDT method, dividing the heat exchange into three sections in counterflow [31]: the first where the working fluid is in the liquid phase, the second is the phase change one (vaporization) and the third is where the fluid is superheated (Figure 3).

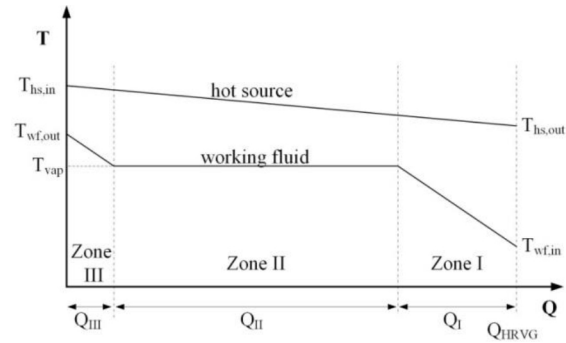


Figure 3. three-zones model for the HRVG sizing

For each i -th section, heat exchange area has been evaluated by knowing mass flow rates, pressures, inlet and outlet temperatures of both cold and hot fluid in the heat exchanger:

$$A_{HRVG,i} = \frac{Q_{HRVG,i}}{U_i \cdot LMDT_i \cdot F} \quad (4)$$

Fouling factor F is considered equal to 0.8. Knowing the inlet and outlet temperatures (which come from the ORC model) the logarithmic mean temperatures can be evaluated.

The overall heat transfer coefficient U is calculated according to:

$$\frac{1}{U_i} = \frac{1}{h_{cold,i}} + \frac{t_{wall}}{k_{wall}} + \frac{1}{h_{hot,i}} \quad (5)$$

where t_{wall} and k_{wall} are the thickness and the thermal conductivity of the exchanger's walls, while convective heat transfer coefficient h are calculated according to proper empirical correlations. In particular, for exhaust gas cooling (tubes side) it is calculated according to Dittus-Boelter correlation:

$$h = 0.0243 \text{Re}^{0.8} \text{Pr}^{0.3} \frac{k_{exh}}{D_{tube}} \quad (6)$$

k is the thermal conductivity of the fluid in a mean thermodynamic condition in the zone considered. For the working fluid a different single-phase correlation has been used [32]:

$$h = 1.04 \text{Re}^{0.4} \text{Pr}^{0.36} \frac{k_{wf}}{D_{shell}} \quad (7)$$

In two-phase region, instead, the Klimenko correlation [33, 34] has been considered:

$$h = 0.087 \text{Re}^{0.8} \text{Pr}^{1/6} \left(\frac{\rho_{wf,sv}}{\rho_{wf,sl}} \right)^{0.2} \left(\frac{k_{wall}}{k_{wf,sl}} \right)^{0.09} \frac{k_{wf,sl}}{D_{shell}} \quad (8)$$