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EVALUATION OF ORC SYSTEM FOR HARVESTING REJECTED HEAT OF IC ENGINE

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ABSTRACT
A bottoming Organic Rankine Cycle (ORC) system for harvesting rejected energy of Internal Combustion (IC) engine exhaust gases is presented in the paper. A simple model was developed to estimate the ORC system operating parameters, as well as system investment cost and electricity production cost. A parametric study was performed to study the influence of the main system parameters, such as evaporating pressure, heat exchangers pressure drop, and IC engine exhaust gas temperature and mass flow rate on thermal efficiency and electricity production cost. It was found that the economic viability did not only depend on the thermal efficiency of the ORC system but it is influenced highly by the number and size of applied heat exchangers, and by the pressure increase at the exhaust side of the IC engine caused by the pressure drop in the heat exchangers.
Keywords: Organic Rankine Cycle, IC engine exhaust heat recovery, techno-economic model

1. INTRODUCTION
Diverse technologies exist to recover energy in the form of power from waste heat using waste heat as the primary energy source. Examples of technologies for work generation include thermoelectric generators, phase change materials, Organic Rankine Cycles (ORC), Kalina cycles and trilateral flash cycles. The ORC is the most mature and tested technology when compared to Kalina cycles and thermoelectric generators [1]. It has a higher conversion efficiency and longer technical life. The aim of this paper is to show the potential for power generation from available heat in diesel engine exhaust by ORC. A simple ORC simulation model was developed in order to perform a parametric study on the thermal efficiency and economic viability of ORC under different thermal parameters (temperature and heat rate) of available waste heat.

2. ORC MODEL
The system layout is presented in Fig. 1. Exhaust gases leaving the IC engine flow through the super-heater (SH), evaporator (E) and preheater (PH), and reject their heat to the working fluid before being released to the atmosphere at approximately 120 ^{0}C, which is set as the lower limit in order to avoid any water condensation within the exhaust. High pressure working fluid vapour expands in the turbine and then enters the regenerator, where the exhausted vapour rejects heat to the vapour cooler (VC) integrated within the water cooled condenser (C), where it finally condenses to the liquid phase. The condensate is then pumped to the working pressure and fed to the system of heat exchangers to produce fresh high pressure superheated vapour.
A simple model written in Excel was developed to determine the main system operational parameters. ORC operational points 1 through 7 are first calculated, where the fresh vapour thermodynamic state (p_1 and T_1) and condensation temperature $T_3 = T_4 = T_c$ are set as input data. Using the REFPROP database as an Excel Add-in, it was possible to find all other thermodynamic states, specific turbine and specific pump work as well as thermal efficiency where turbine and pump isentropic efficiency, was set to $\eta_{T,s} = 0.7$ and $\eta_{P,s} = 0.8$, respectively.
Working fluid mass flow rate is calculated next from Eq. (1) where the $T_{g,out}$ is set to 120 ^{o}C as mentioned before:

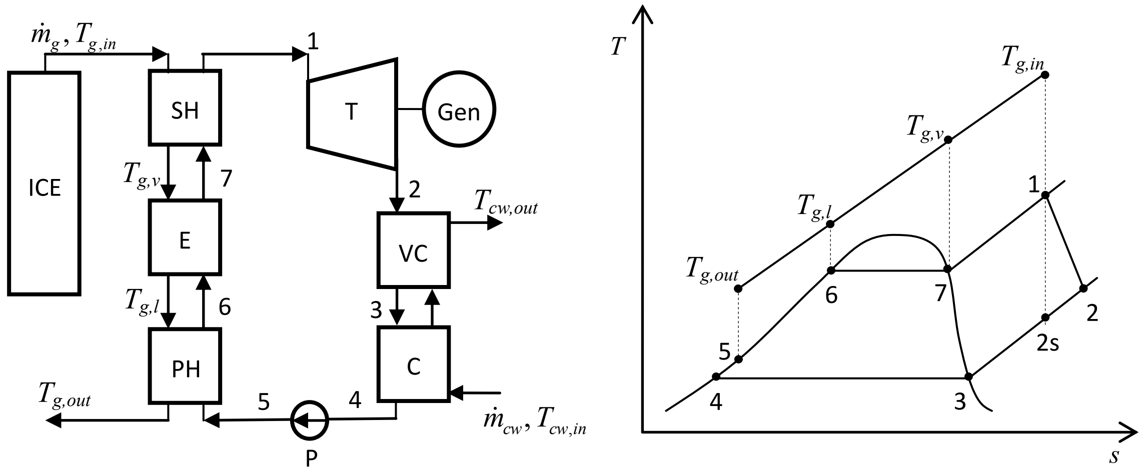

Fig. 1. ORC model

$$\dot{m}_{wf} = \frac{\dot{m}_g c_{p,g}\left(T_{g,in} - T_{g,out}\right)}{h_1 - h_5} \tag{1}$$

With known working fluid mass flow rate, it is possible to calculate the exhaust gas temperature at the evaporator inlet $T_{g,v}$ and the exhaust gas temperature at the preheater inlet $T_{g,l}$ by applying corresponding energy conservation equations.
Once all the temperatures and mass flow rates are known, it is possible to estimate the heat exchanger parameters. The plate type heat exchangers are applied in preheater, evaporator, superheater, vapour cooler and condenser in this study, due to their compactness and high heat transfer coefficients. The heat transfer area of the individual heat exchanger is calculated from the heat flow rate divided by the logarithmic mean temperature difference ΔT_m and overall heat transfer coefficient U defined as:

$$\frac{1}{U} = \frac{1}{\alpha_h} + \frac{l}{k} + \frac{1}{\alpha_c} \tag{2}$$

α_h and α_c are the heat transfer coefficient at the hot side and at the cold side, respectively, l is plate thickness and k is plate conductivity. Empirical correlations were used to predict both heat transfer coefficients α_h and α_c. For the single phase fluid flow, a correlation proposed by Kumar [2] has been applied. The Cooper correlation [3] has been used for the two phase flow conditions prevail at the working fluid side of the evaporator and the correlation proposed by Hsieh et al. [4] was used for the condenser. The total pressure drop is composed of the frictional channel pressure drop and the port pressure drop. Both were estimated by the model proposed by Kakac et al. [5].

3. ECONOMIC MODEL

The so-called Electricity Production Cost (*EPC*) can be estimated using the following formula:

$$EPC = \frac{C \cdot R + M}{E} \tag{3}$$

Where C is capital cost, R is capital recovery factor, E is annual electricity output and M is operating and maintaining the ORC system annual output. 5% interest rate and 15 years life time were used in this study to calculate capital recovery factor R. Annual electricity output was estimated under the assumption that the ORC system annually operates 7500 hours at full load. Operating and maintaining annual cost were estimated to 1.65% fraction of capital cost. Capital cost of the ORC system is the sum of the capital cost of each system component (turbine T, pump P, and the whole set of heat exchangers PH through C), including the cost of assembling. Equations for evaluating the cost of system components are adopted from [6], which is used widely for analysis, synthesis and design of chemical processes. Turbine cost depends on turbine power P_T:

$$C_T = 3.5 \cdot 10^{3.514+0.589\log(P_T)} \tag{4}$$

Pump cost is estimated by:

$$C_P = \left\{1.8 + 2.718\left[0.168 + 0.348\log\Delta p_P + 0.488(\log\Delta p_P)^2\right]\right\} \cdot 10^{3.579+0.321\log(P_P)+0.003\log(P_P^2)} \tag{5}$$

Where P_P is pump power and Δp_P is pump pressure rise. Any heat exchanger cost depends on heat transfer area A_{HE} and is predicted as:

$$C_{HE} = 5.086 \cdot 10^{3.853+0.424\log(A_{HE})} \tag{6}$$

The investment cost predicted by Eq. (4) to (6) is in US dollars and is valid for the year 1996. The actual investment cost may be predicted by multiplying estimated cost in year 1996 by the ratio of the Chemical Engineering Plant Cost Index (CEPCI) for the year 2018 and 1996. In order to express it in EUR, the actual exchange rate was used.

4. RESULTS AND DISCUSSION

Both ORC and the economic model were applied in a parametric study to investigate the parameters that influence thermodynamic and economic effectiveness of the bottoming ORC system. A commercial diesel generator set is considered as a topping system. The engine is an inline 6 cylinder 4 stroke supercharged diesel engine. The main parameters of the engine are presented in Table 1. All applied heat exchangers were plate type with heat transfer plates 0.5 m long, 0.2 m wide and 0.002 m thick. The chevron angle of heat transfer plates was $\beta = 45\,^0$. Mean channel spacing was 6 mm at the working fluid and cooling water side, while 11 mm mean channel spacing was applied at the gas side.

Table 1: Main parameters of the commercial diesel generator set

Parameter	Value	Parameter	Value
Electrical power output (kW)	235.8	Engine speed (rpm)	1501
Torque (N m)	1500	Fuel consumption (kg/h)	47.79
Exhaust temperature (^{0}C)	519	Exhaust mass flow (kg/h)	990.79

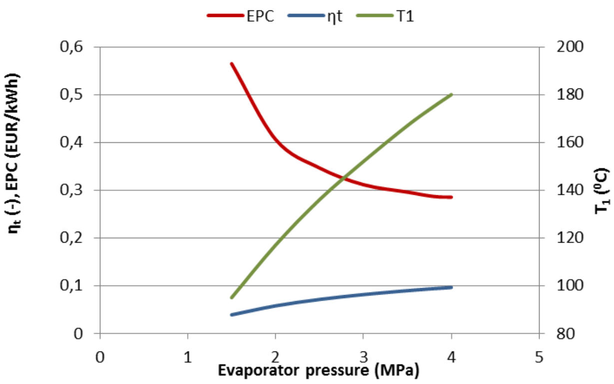

Fig. 2. Optimal fresh vapour temperature T_1, thermal efficiency ηt and EPC

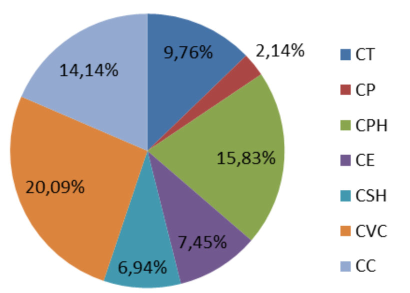

Fig. 3. ORC investment cost structure at the evaporator pressure 3.0 MPa

R134a is used as ORC working fluid, although any other refrigerant may be applied too. Condensation temperature was set constant at 35 ^{0}C in all cases. All other parameters, such as evaporator pressure and fresh vapour temperature T_1 were changing. When the evaporator pressure was set constant, a simple trial and error procedure was used to find the optimal fresh vapour temperature T_1 at which the thermal efficiency is the highest. Results are presented in Fig. 2. Both temperature T_1 and thermal efficiency increase with evaporator pressure. The maximum obtained thermal efficiency is 0.0963 at 4.0 MPa, which is only slightly under the R134a critical pressure. *EPC*

is also shown in Fig. 2. It reduces with evaporator pressure, however, it remains very high even under maximum thermal efficiency conditions. The main reason for this is the very high investment cost of the ORC system. Figure 3 shows the ORC investment cost structure at the evaporator pressure 3.0 MPa. The low pressure vapour cooler represents 20% and super heater 7% of the total cost. Both can be omitted if the ORC operates with saturated vapour. Figure 4 shows how this affects the thermal efficiency and *EPC*. Although the thermal efficiency reduces by approximately 1.7 percentage points, the *EPC* reduces by almost 0.050 EUR/kWh. Both thermal efficiency and *EPC* reach their extreme values between 3.6 MPa and 3.7 MPa.
Figures 2 and 4 show the efficiency of an isolated ORC where no influence of bottoming ORC on topping IC engine is considered. It may be shown that the set of heat exchangers at the exhaust side of the IC engine increases its back pressure, thus reducing the IC engine power and influences overall efficiency. Corrected efficiency was introduced to consider this. Fig. 5 shows the IC engine back pressure increase due to pressure drop in the preheater and evaporator. Pressure drop is small when the evaporator pressure is low to moderate, and increases substantially when the evaporator pressure is high. This moves the corrected efficiency maximum to the lower evaporator pressure at 3.2 MPa and, similarly, shifts the minimum *EPC* to 3.2 MPa too. The latter thus increases from 0.245 EUR/kWh to 0.265 EUR/kWh.

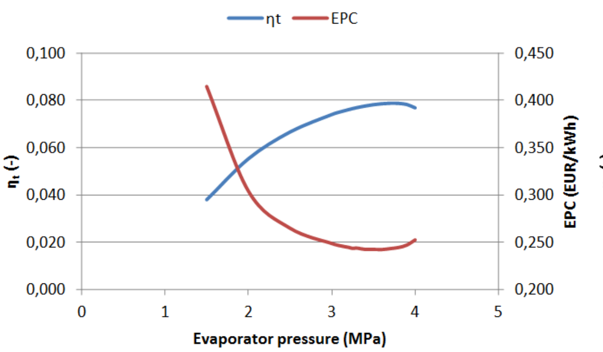

Fig. 4. Thermal efficiency and EPC of an ORC working with saturated vapour

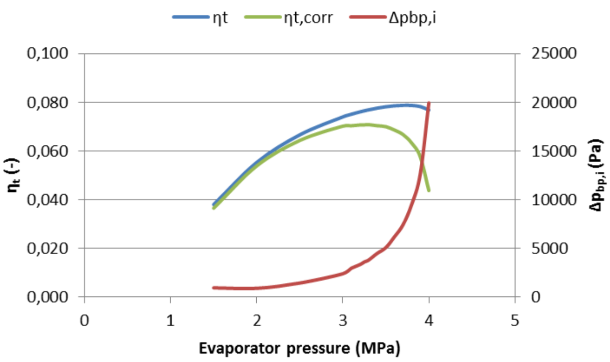

Fig. 5. IC engine back pressure increase due to pressure drop in heat exchangers and its influence on system thermal efficiency

5. CONCLUSION

It was proven by the presented parametric study that *EPC* does not correlate proportionally with the thermal efficiency. A thermodynamically more efficient ORC working with superheated vapour does not attain higher economic efficiency than a simple ORC working with saturated vapour; moreover, the estimated *EPC* was more than 15% higher. Further, it was found that pressure drop at the exhaust gas side of heat exchanger can reduce the topping IC engine output power substantially, therefore, special attention has to be paid to hold pressure drop low even at the cost of increased investment cost of the heat exchanger.

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