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# Thermo-economic optimization of Organic Rankine Cycle CHP with low temperature waste heat

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## EXTENDED ABSTRACT

## INTRODUCTION

Combined heat and power (CHP) systems are able to decrease the total energy use of primary energy sources. In the CHP system studied, internal combustion engines produce electricity and the hot engine cooling water is used for building heating. However, there is still waste heat left which can be fed to an Organic Rankine Cycle (ORC) to produce electricity. The objective of this study is to develop a methodology to design an economically optimal ORC system, taking into account the variable load for heating and the change in ambient temperature during a year. Also the auxiliary equipment such as pumps and fans are considered. A thermodynamic steady-state part-load model is developed to simulate the changing behaviour hour-by-hour of the complete system in different operating conditions. The ORC efficiency varies strongly over a year. The methodology allows selecting the optimal size of the heat exchangers (condenser and evaporator), the optimal mass flow rates and the maximal power of fans and pumps needed for the considered application.

## DESCRIPTION OF THE CASE

Figure 1 shows the setup of the investigated system. Waste heat $Q_{ICE}$ is available from the closed loop cooling circuit of the internal combustion engines. Part of this heat is used for building heating $Q_v$. The heat $Q_{ICE} - Q_v = Q_{ORC}$ is supplied as heat input to the ORCs evaporator. A thermostatic valve keeps the water entering the ORC at 85°C. Table 1 summarizes the constant parameters in the setup. Three applicable working fluids were considered: R245fa, R152a and R1234yf.

| | |
|---|---|
| Hot water inlet temperature | 85°C |
| Cooling circuit medium | water |
| Condenser medium | air |
| $Q_{ICE}$ | 3452 kW |
| $Q_v$ | hour-by-hour profile |

**Table 1:** Model assumptions

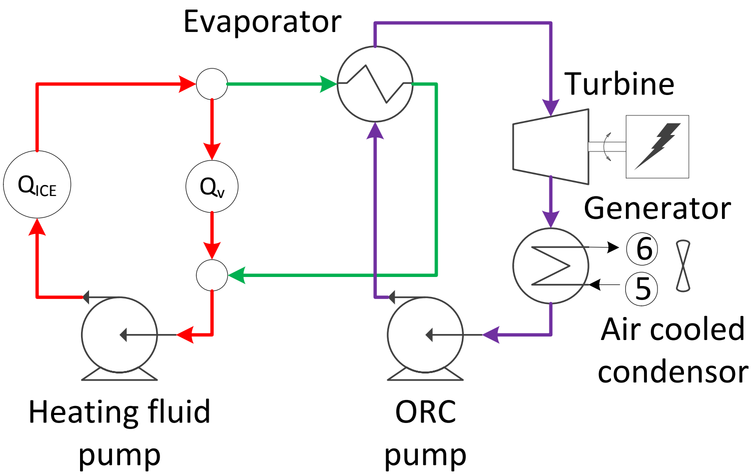

**Figure 1:** Simplified configuration of the ORC and CHP

## METHODOLOGY

### Design strategy

In this paper a design methodology is suggested based over the specific investment cost (SIC). The specific investment cost is defined as the ratio of the net power output on the investment cost. A thermo-economic quasi steady-state model is developed which allows simulating hour-by-hour variations in different operating conditions. The design strategy itself consists of four steps:

- An ORC is sized for fixed operating conditions
- With this sizing the part load regime is modelled
- A part load model is used in a one-year simulation to assess the actual power output
- The optimal sizing is selected

In the first step a design is made for fixed nominal operating conditions (heat input and ambient temperature). The optimization criterion is the minimization of the SIC. As a result the dependent design variables are fixed: the heat transfer area of the evaporator and condenser, the length and width of the evaporator, the number of tube rows in the condenser.

Next, a part load map (figure 2) is calculated from this design. The independent variables are the ambient temperature and the waste heat input. The optimization criterion is the maximisation of the net power output. Hung et al. [1] showed that there is an optimisation potential by controlling the evaporation pressure in operation. System efficiency increases with increased inlet turbine pressure while the availability of waste heat decreases. Therefore an optimal evaporation pressure exists whereby the output power is maximized. Furthermore the mass flow rate of heating fluid and air is optimized, offsetting the expander power output against the auxiliary power use of pumps and fans.

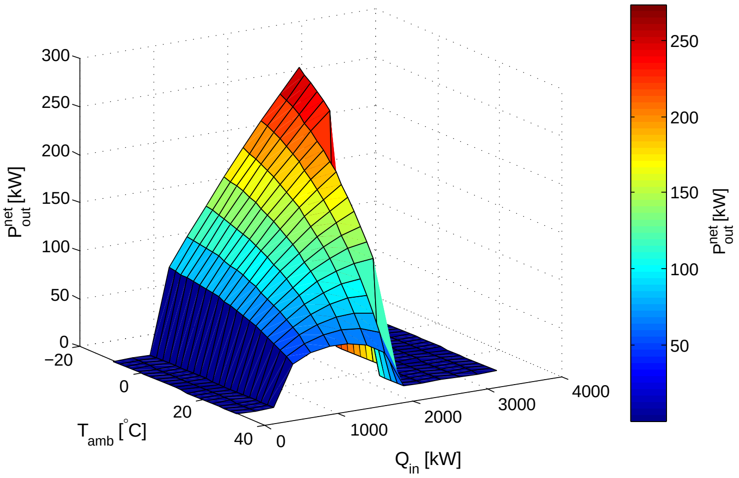

**Figure 2:** Example part load map, nominal parameters: ambient temperature = 6.5°C, heat input = 2236 kW (R245fa)

The third step is the use of these part load maps in a year simulation. The year simulation has a time step of one hour. In each time step the described steady state model of the ORC is solved. As a result the mean net power production throughout a reference year is calculated. From this result the SIC is recalculated.

The last step comprises of selecting the ORC with the lowest SIC according to the year simulations.

### Model and assumptions

The thermodynamic models and equations for the steady state system is presented in a previous paper [2]. The models in this work are extended with part load characteristics for the turbine, pump, evaporator and condenser. The part load model of the pump and expander are respectively adapted from Lippke [3] and Lemort [4]. The heat transfer coefficients used in the model of the evaporator and condenser are calculated from appropriate correlations. The model of the air cooled condenser is based on the work of [5]. The available waste heat profile and dry bulb ambient temperatures were measured during one year with a resolution of one hour. They are used as input to the model. The superheating of the working fluid after the evaporator is set at 5°C, the subcooling after the condenser at 3°C.

## RESULTS AND DISCUSSION

The results show that the relative cost of the turbine is the highest for R245fa (34 % of the total cost), followed by R152a (24 % of the total cost) and R1234yf (22 % of the total cost). In absolute terms, the design with R152a has the highest investment cost followed by R152a and R1234yf. On the other hand, R245fa has the highest thermal efficiency (7.41 %) followed by R152a (6.53 %) and R1234yf (5.85 %). The SIC is lowest for R152a.

Each design has fixed operation boundaries. The thermal efficiency decreases as much as 50 % between ambient temperatures from -9°C to 35°C.

It is noticed from the year simulations, that the SIC based on the mean net power production differs by more than 29 % from the SIC at nominal conditions. This is due to the large variation in operating conditions throughout a year. The optimal design has a SIC of 2210 euro/kWe.

## CONCLUSIONS

In this work, a design strategy is proposed to optimize an ORC for a CHP system. Because of the the large variations in operating conditions it is important to define a part-load model. It is observed that the SIC based on nominal conditions differs significantly from the SIC that accounts for changing operating conditions throughout a year. Furthermore, it is shown that the choice of working fluid has a notable effect on the cost of the system.

## ACKNOWLEDGEMENTS

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