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Study of the on-route operation of a waste heat recovery system in a passenger vessel

Maria E. Mondejar [a,*], Fredrik Ahlgren [b], Marcus Thern [b], Magnus Genrup [a,*]

[a]Lund University, PO Box 118, Lund SE-22100, Sweden
[b]Linnaeus University, Kalmar Maritime Academy, Kalmar SE-39182, Sweden

Abstract

Waste heat recovery systems for power generation are gaining interest among the marine transport sector as a solution to accomplish the upcoming more restrictive regulations on emissions, and to reduce the total fuel consumption. In this paper we evaluate how a waste heat recovery system based on a regenerative organic Rankine cycle (rORC) could improve the performance of a passenger vessel. The case study is based on the M/S Birka Stockholm cruise ship, which covers a daily route between Stockholm (Sweden) and Mariehamn (Finland). Experimental data on exhaust gas temperatures, fuel consumption and electricity demand on board were logged for a period of four weeks. Based on the results of a fluid and configuration optimization performed in a previous work, an off-design model of a rORC working with benzene was used to estimate the net power production of the rORC at the different load conditions during a port-to-port trip of the vessel. The power generation curve of the rORC over time was compared to that of the electricity demand of the ship. Results showed that the rORC could provide up to 16 % of the total power demand. However, this value should be corrected if the auxiliary engines load is reduced as a consequence of the partial coverage of the electricity demand by the ORC.

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Keywords: cruise vessel; waste heat recovery; organic Rankine cycle; off-design.

1. Introduction

The international ship bunker fuel was in 2012 responsible for 602 million tonnes of CO_2 emissions worldwide, which accounts for about 1.90 % of the total CO_2 emissions from fuel combustion. Although this figure seems low, the international shipping sector has experienced a 28 % rise of the total CO_2 emissions from fuel combustion during the period from 1990 to 2012 [1]. This value accounts only for the international shipping, but it gives a good picture of how the shipping industry CO_2 emissions are rising at a higher degree than the world total fossil fuel combustion. In fact, according to the International

* Corresponding author. Tel.: +46 46 222 14 09; fax: +46 46 222 47 17.
E-mail address: maria.mondejar@energy.lth.se.

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Nomenclature

UA	overall heat transfer coefficient	**Subscripts**	
c_p	specific heat at constant pressure	0	design conditions
m	mass flow rate	cond	condenser
N	turbine rotational speed	evap	evaporator
p	pressure	ORC	cycle parameter
T	temperature		
V	volume flow rate	HFO	heavy fuel oil
W_{net}	net power output	ORC	organic Rankine cycle
η	energy efficiency		

Maritime Organisation (IMO), all shipping, both including international and domestic, stands for approximately 949 million tonnes of emissions CO_2 in 2012 [2]. Although the shipping sector plays still a minor role in the total world CO_2 emissions, its contribution is expected to rise even more in a foreseeable future [3]. Moreover, besides the global warming effect, there is a direct negative effect of ship emissions on the coastal population health, as shipping stands for a large fraction of sulphur dioxides (SO_x) and nitrogen oxides (NO_x) in these areas. This is due to the use of high sulphur residual fuels, which most ships use today because of their low price and unregulated emissions in the open seas [4]. From an analysis of the ship transport patterns, it is estimated that about 70 % of the ships routes are within 400 km from the coast. As a consequence, a great amount of the airborne emissions are displaced to dense populated areas. As a matter of fact, it is estimated that shipping accounts for up to 60000 deaths globally each year [5], which evidences the urgency of reducing ship emissions.

The IMO stipulates several emissions control areas (ECA), where the emissions of SO_x and NO_x [6] are under control. The Baltic Sea will be, after 1st January 2015, a sulphur emission control area (SECA), which implies that the sulphur mass in the fuel should not exceed 0.1 %. In order to comply with this new regulation, sulphur emissions must be reduced by either using a fuel with a lower content in sulphur (mainly a distillate fuel) or by cleaning the exhaust gases, implying both ways an extra cost for the ship-owner [7].

About 98 % of all ships use diesel engines as the main propulsion, which have approximately a thermal efficiency of (49 - 51) %. This means that a great part of the fuel energy is dissipated to both the exhaust gases and the engine cooling water. When a heavy fuel oil (HFO) is used, the bunker oil needs to be heated up to (65 - 75) °C before it is combusted in the engine, which requires a significant amount of energy. Normally, this energy is supplied through heat recovery from the exhaust gases. If, in the near future, the fuel is shifted to a distillate fuel instead of a high viscosity fuel, the demand for low temperature heat will be lower and therefore the total system efficiency will be lower, increasing the heat wasted through the exhaust gases.

Organic Rankine cycles (ORC) are recently gaining interest for their use in waste heat recovery systems from industrial exhaust gases. In this sense, on-going research is focusing on the use of ORC for waste heat recovery in ships. Although the integration of this technology in vessels is still at an early stage, ORC have a great potential to reduce their fuel consumption and therefore, the CO_2 emissions.

The main objective of this paper is to estimate the potential for power production of an ORC installed in a passenger vessel, that uses the exhaust gases as heat source, and evaluate how the ORC production curve fits the electricity demand along the route. For this purpose, experimental data from the cruise ship MS Birka Stockholm were logged on the route between Stockholm (Sweden) and Mariehamn (Finland) during a four-week period. An ORC model was built, in which the exhaust gases experimental temperatures and estimated mass flow rates were used as inputs, in order to evaluate the performance of the waste heat recovery.

2. Simulation of the waste heat recovery system

On-board data acquisition and analysis

The ship of our case study has four main engines Wärtsilä 6L46B with a total power of 23560 kW for the propulsion, and four auxiliary engines Wärtsilä 6L32 with a total power of 10720 kW to satisfy the electricity demand of the ship. Except for two of the main engines, the rest of the engines have exhaust gas boilers, which produce steam to heat up the bunker fuel. For this study the exhaust gas temperatures were measured at each engine outlet, or after the existing exhaust gas boilers where applicable, in one-minute intervals. The exhaust mass flow rates for each engine were estimated as a function of the measured engine load, by using a linear regression with least squares fit of the Wärtsilä project manual data [8,9]. During the logging time-frame the ship used a heavy fuel oil as bunker fuel, and therefore, the exhaust temperatures were lower for those engines where a boiler was integrated.

In this study, it was decided to use the mixture of all exhaust streams as the input source for the waste heat recovery system. This decision, that could appear initially inefficient due to the different temperature levels of the streams, was adopted as a result of a previous study [10], where it was found that process integration of the streams did not improve substantially the efficiency of the system, due to the limitation of the cooling temperature of the exhaust gases to 150 °C. Figure 1.b shows the estimated mass flow rates for several trips as a function of time. It can be observed that the operating conditions differ significantly with time. Reasons for this are that, along the route, the MS Birka Stockholm sails through different zones where special operational conditions are needed (i.e. the Stockholm Archipelago, where the speed of the vessel is lowered, the proximities of Mariehamn, where the speed of the vessel is highly unstable, and the open sea, where the vessel sails at a higher and steady speed). Although it is appreciable that there are variations on the logged data between different days, it is observed that there is a common operational pattern, so that average trip conditions can be calculated (in red in Figure 1.b). It can be also seen that the conditions for the initial and return trip are not symmetric, which can be a result of different marine currents and winds. A similar behavior can be observed on the exhaust temperature logged values. As a consequence of this pattern, we decided that it was of interest to analyze the performance and operation of an ORC along the route, and compare its net power production with the electricity demand on board.

In order to perform the study, the logged temperatures and mass flow rate estimates of the exhaust gases were averaged per trip, as a function of time, so that the curve for an *average trip* was generated (see red line in Figure 1.b.). Using the average route values for both temperature and mass flow rate, ten time intervals where conditions were roughly stable were identified (which are represented in Fig.1.c., for the exhaust mass flow rate). The ORC performance was analyzed at the mean conditions for each interval, so that the operating curve along the route could be drawn.

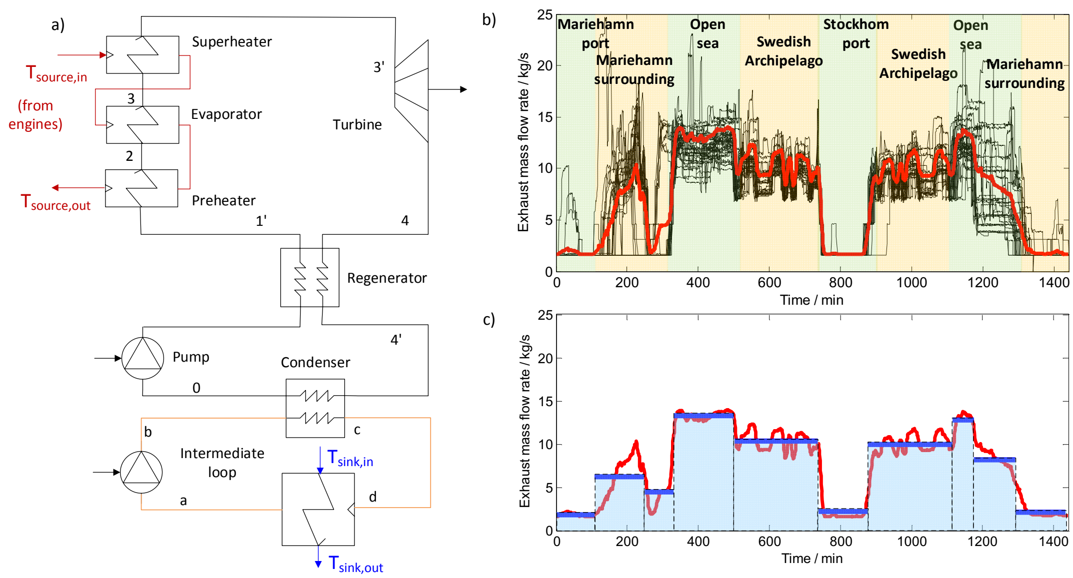

Fig. 1.a. Schema of the rORC model used in this work; b. Total estimated exhaust mass flow rate for different trips covering the same route as a function of time (black). The red line represents the average values from all the trips considered during the logged time. c. Approximation to the ship operation conditions in a return trip, where averaged values have been estimated to be used in the ORC model

Table 1. Simulation parameters of the ORC at the design conditions.

Parameter	Value	Parameter	Value
Heat source temperature	319.6 °C	Turbine/pump mechanical efficiency	0.99
Heat source mass flow rate	10.372 kg/s	Pump isentropic efficiency	0.80
Pressure drops in heat exchangers (liquid)	0.02 bar	Turbine isentropic efficiency	0.763
Pressure drops in heat exchangers (gas)	0.006 bar	Turbine volume ratio	133.8

ORC model simulation

The waste heat recovery system studied in this work consists of a regenerated organic Rankine cycle (rORC) with an internal loop of water between the condenser and the seawater, which is used as the heat sink. The heat source consists of the mixed exhaust gases from the main and auxiliary engines of the vessel. In a previous work, in which we studied the optimization of the fluid and configuration of the ORC for this vessel [10], we found that benzene was the working fluid that yielded the greatest net power output for the regenerated configuration under the logged conditions. Therefore this work is based on this optimal cycle, which schema is shown in Figure 1.a. Benzene is a hydrocarbon with high critical temperature suitable for waste heat recovery from medium-temperature heat sources. However it presents the inconvenience of its high flammability and toxicity, which have to be considered in detail for before a final design decision. Table 1 collects the main simulation parameters for the rORC, as well as the design conditions used during the analysis (which were chosen as those present during a extended time over the trip). The values of the pressure drops in heat exchangers and the isentropic efficiency of the pump were adopted based on experience. The isentropic efficiency of the 4-stages axial turbine was calculated according to the loss model presented in our previous work [11], that accounts for the effect of the thermophysical properties of each working fluid. A superheating of 5 K was applied at the inlet of the turbine in order to avoid wet expansion during the operation of the cycle.

As mentioned in the previous section, the exhaust temperatures and mass flow rates, averaged for the 27 trips for which data were available on each defined time interval, were used as inputs for the ORC model. For each operation interval, the performance of the ORC was evaluated by using an off-design model of the rORC, so that the overall conversion into power of the system along the route could be predicted.

The off-design model used in the simulations integrates the characteristic curve of the heat exchangers and regenerator, and accounts for the variation in the pump and turbine efficiencies, according to (1), (2) and (3)[12], respectively.

$$UA = UA_0 \left(\dot{m} / \dot{m}_0 \right)^{0.6} \tag{1}$$

$$\eta_p = \eta_{p0} \left(c_1 \left(\dot{V} / \dot{V}_0 \right)^3 + c_2 \left(\dot{V} / \dot{V}_0 \right)^2 + c_3 \left(\dot{V} / \dot{V}_0 \right) + c_4 \right) \tag{2}$$

$$\eta_t = \eta_{t0} \left(N/N_0 \sqrt{\Delta h_{s,o} / \Delta h_s} \left(2 - N/N_0 \sqrt{\Delta h_{s,o} / \Delta h_s} \right) \right) \tag{3}$$

Shell-tube heat exchangers were considered in (1). The parameters for (2) used in this work were taken from the work of Larsen [12], given the similitude in the order of magnitude of the required pump power, and are c_1=-0.168, c_2=-0.0336, c_3=0.6317 and c_4=0.5699. In addition, a swallowing capacity model based on Beckmann's equation was used for the turbine, so that the mass flow rate in the cycle was set by the evaporator pressure, specific volume and the turbine size, as expressed in (4).

$$\dot{m} = \left[C_{T,B} + K \left(\mu - \mu_0 \right) \right] \left(1 + \mu \right) \sqrt{p_3 F / v_3 \mu} \tag{4}$$

In this expression, C_{TB}=3.4241E-2 is a constant that was fitted at the design conditions, and F is a function of the number of stages of the turbine and the inlet and outlet pressures. The coefficient K was assumed to be null.

3. Results

On-route ORC power production

Table 1 collects the power production and operation parameters of the rORC for each time period considered in Figure 1.c. Figure 2 depicts the estimated ORC net power production over the round trip, together with the average electrical power demand measured on board. This curve was calculated by averaging the logged values of the power consumption of the ship during the different trips considered. It can be observed, that the power demand is minimal during the stop of the vessel in Mariehamn's port, and reaches the maximum values during the sailing in the open sea and the stop in Stockholm's port.

The power production of the ORC is maximal when the ship is sailing in the open sea, for which both the main engines and auxiliary engines are running, and therefore the exhaust mass flow rates and temperatures are greater. This is an advantage since the greatest production periods of the ORC happen when the electricity demand in the ship is maximal. As it can be observed in Figure 2, there are steep variations of the power demand along the trip, especially in the arrivals and departures from the port. In order to evaluate if an ORC would be capable of following such variations it is necessary to perform a dynamic simulation of its operation. However, evaluating the rORC power production in the specified intervals gives a preliminary estimation of its potential for waste heat conversion. In this sense, it was estimated that the average power production of the ORC could be of 281.03 kW over the entire trip, while the average power demand is of 1784.6 kW. According to these values about 16 % of the total power demand on board could be provided by the waste heat recovery system. However, it must be mentioned that, if the ORC is integrated, the power demand for the auxiliary engines would be lower, thus affecting the exhaust mass flow rates and temperatures from which waste heat is recovered. This consideration is not included here as it is out of the scope of this work. However it is our will to analyze the reduction of waste heat energy so that the potential production of the ORC can be better estimated.

Table 1. Simulation results for the design and off-design operation of the rORC in the different zones considered along the round trip.

Zone	p_{evap}	p_{cond}	m_{ORC}	η_{ORC}	W_{net}
	bar	bar	kg/s	%	kW
I	8.7	0.16	0.781	0.160	69.07
II	17.3	0.16	1.762	0.208	214.91
III	14.3	0.16	1.398	0.194	156.16
IV	26.5	0.16	2.981	0.237	428.78
V (Design)	26.7	0.16	3.027	0.239	440.55
VI	7.7	0.16	0.700	0.141	53.11
VII	25.5	0.16	2.852	0.237	411.25
VIII	27.2	0.16	3.093	0.240	452.27
IX	19.8	0.16	2.078	0.217	267.51
X	9.1	0.16	0.829	0.163	74.70

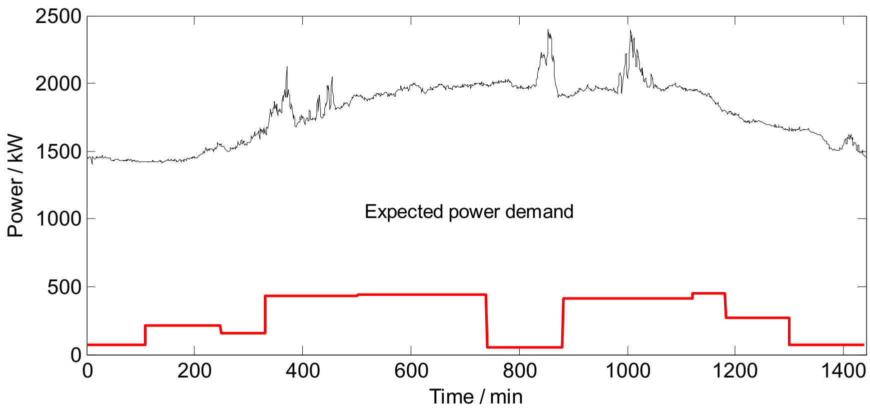

Fig. 2. Estimated net power production for the rORC (red line) and average logged values for the power demand on the ship (black line) during the average round trip.

4. Conclusions

In this work we analyze the performance of a regenerated organic Rankine cycle (rORC) installed on a passenger vessel for the waste heat recovery of the engines exhaust gases. Logged temperature data from the engines exhaust and estimated mass flow rates are used as input values for the simulation model, which consists of a rORC using benzene. The logged data were grouped in ten time periods, for which conditions were roughly stable, and average values were used for the simulations. The conclusions drawn from this work are as follows:

- The estimated net ORC power production over a round trip rises up to 281.03 kW, which represents approximately 16 % of the total power consumption on board over the round trip.
- The power production of the ORC follows the same pattern as the electricity demand of the ship, as it is greater while the vessel is in the open sea and the power demand is higher.
- If an ORC is installed on the ship, the power demand would be lower for the auxiliary engines, thus reducing their load and exhaust gases energy. This will be considered in our future work, for a better estimation of the potential of the ORC.

Copyright

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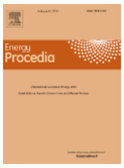

Biography

Maria E. Mondejar defended her Ph.D. thesis on Fluids Thermodynamics in 2012 at the University of Valladolid (Spain). Since 2013 she performs research on waste heat recovery systems as a postdoctoral researcher at Lund University (Sweden), with an special emphasis of working media and expander design.