PERFORMANCE AND ECONOMIC EVALUATION OF A REVERSIBLE HIGH-TEMPERATURE HEAT PUMP/ORC FOR WASTE HEAT RECOVERY IN THE MARINE SECTOR

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ABSTRACT

The prevailing technology for waste heat recovery in ships is the ORC, which can exploit a variety of waste heat sources to produce electricity, and thus reduce the fuel consumption of the generator engines. The main challenge is to increase the capacity factor and exploit the highest possible amount of the various waste heat sources on-board, preferably of elevated temperature, in order to reach a high electric power output. The exploitation of the thermal content of the exhaust gases usually introduces severe constraints, since heat is extracted from this gas flow first from the engine's turbocharger and then by the exhaust gas boiler (economizer) for steam production in the case of the main engine. Moreover, the main engine in a typical merchant ship operates for about two thirds of the year, which might not be enough to secure the ORC cost-effectiveness when only its low-grade heat source is exploited. On the other hand, the generator engines operate for an even larger duration, but with a much lower capacity.

With the aim to increase the capacity factor (operating days/year), an advancement of the ORC technology is examined here, which exploits low-temperature waste heat sources. Such configuration is based on a reversible cycle, operating either as an ORC or a high-temperature heat pump (HTHP). The heat pump mode produces steam, replacing the use of auxiliary boilers, which mostly operate when the ship is at port. When the ship is at sea, no steam is needed by those boilers, because the steam generation is accomplished by the exhaust gas economizer (exhaust gas boiler), and then the unit reverses its operation to ORC mode for electricity production.

This flexible unit is examined in terms of performance for producing saturated steam at 6 bar/158 ºC (heat pump mode) and electricity (ORC mode), when using an ultra-low global warming potential (GWP) refrigerant. A detailed techno-economic study has then been conducted for a variety of long-distance ships, according to their typical boiler capacity that is matched to the heat pump capacity and days at sea/port, leading to the estimation of the net fuel savings and discounted payback period. The latter becomes short in the range of 3-4 years for the reversible unit when the fuel price is increased.

1 INTRODUCTION

ORC units are being increasingly applied in industry for the exploitation of waste heat sources and their conversion to electricity. The main parameters that greatly decide their cost-effectiveness are their size and heat source temperature level, assuming that integration aspects are resolved. The ORC size should be as high as possible (Braimakis and Karellas, 2017), in order to reduce the specific cost, in €/kW, and bring a high impact to the industry with a short return on investment of less than 5 years. This is usually interpreted as the available heat source amount to be exploited, preferably over 1-2 MW. At the same time, efforts are made on exploiting high-grade heat sources of over 100-150 ºC, increasing the ORC thermal efficiency and leading to the reduction of the specific cost. The combination of these two aspects, i.e. large size and high temperatures, ensures to a great extent the success of a heat recovery project, with a large potential in EU industries (Papapetrou *et al.*, 2018).

The use of ORC units in ships has recently gained lots of interest, due to the strict fuel efficiency and emissions regulations that are coming into force, and the increasing fuel prices. However, the size of ORC units is restricted by the available waste heat sources, with a maximum electrical capacity in the range of 100-200 kW_e in medium/large merchant ships. This capacity is much lower than the one at industrial settings, and combined with the moderate capacity factor, which is decided by the annual operational hours of the vessel's main engine (Lion *et al.*, 2019), imposes uncertainties on the wide applicability of this technology in marine settings, due to the long payback period (PBP). On the other hand, the easy access to the waste heat of the on-board engine, mainly of the jacket water (typical outlet/inlet temperature of 90/75 ºC) and cooling water of the scavenge air intercooler (typical outlet/inlet temperature of 57/32 ºC), which is currently rejected in the cooling circuit (Shu *et al.*, 2013), reduces the infrastructure needs for such integration with the main challenge being the low available space in the engine room. Another challenge that is ship-specific is the exploitation of high-grade waste heat from the engine's exhaust gases, with its thermal content partly recovered by the turbocharger and the exhaust gas economizer, thereby leaving only few quantities at a reduced temperature. Therefore, the cost-effectiveness of marine ORC units is not secured, requiring either a higher performance by better cycle design, or recovering high-grade heat, or even reducing the capital cost of the ORC unit.

An alternative waste heat recovery technology is gaining attention mostly for industrial applications, having many similarities to the ORC. This concerns the high-temperature heat pump (HTHP) for waste heat upgrading. Its primary function is for low/medium temperature heat production (Kosmadakis, 2019) with many research efforts directed to expanding the supply temperatures up to 150 ºC (Kosmadakis *et al.*, 2020).

The two previous technological solutions, a heat pump (HP) and an ORC, can be combined into a single unit, forming a reversible unit for either heat or electricity production, reaching high savings and a short PBP in industry (Kosmadakis and Neofytou, 2019). Such solution has also been proposed for domestic applications combined with solar thermal collectors (Dumont *et al.*, 2015), showing that it is possible to use the same volumetric machine for the compressor/expander and keep a high isentropic efficiency, although one mode will not be as efficient as the other. A similar set-up is also proposed for energy storage, by using excess electricity to drive the heat pump and store the produced (high-temperature) heat. This heat supplies the ORC for power production, when needed, resulting in a pumped thermal energy storage (Dumont and Lemort, 2020).

The current work explores the performance and cost effects of the reversible HTHP/ORC unit in several ship types, in order to identify the expected benefits. For that purpose, a detailed thermodynamic analysis has been conducted to identify the performance of the two modes, when sharing the same components, such as the heat exchangers (HEXs) and the compressor/expander. The costs depend on the system scale and are evaluated via appropriate cost correlations, estimating the capital cost in a large variety of ships. The main scope is to identify the potential of such technology in the marine sector, and examine whether it is superior to a separate HTHP or ORC for the same conditions.

2 REVERSIBLE HTHP/ORC FOR MARINE SETTINGS

2.1 Reversible HTHP/ORC configuration

The reversible unit is driven by the waste heat of the engines' jacket water at a temperature of 90 ºC, which is supplied to the evaporator, as shown in Figure 1. The two modes of the reversible unit share the same components for enhanced compactness, which is crucial for marine applications, but more importantly to reduce the capital cost compared to individual systems. An economizer cycle is selected for the HP mode to reduce the discharge temperature (Mateu-Royo *et al.*, 2021), requiring the use of an economizer, which evaporates a fraction of the total refrigerant mass flow rate, with the whole compression split into two stages. This economizer is not used at ORC mode with no working fluid flowing through the eco port of the compression-expansion machine. A generic cycle in a pressure-enthalpy chart is shown in the right Figure 1.

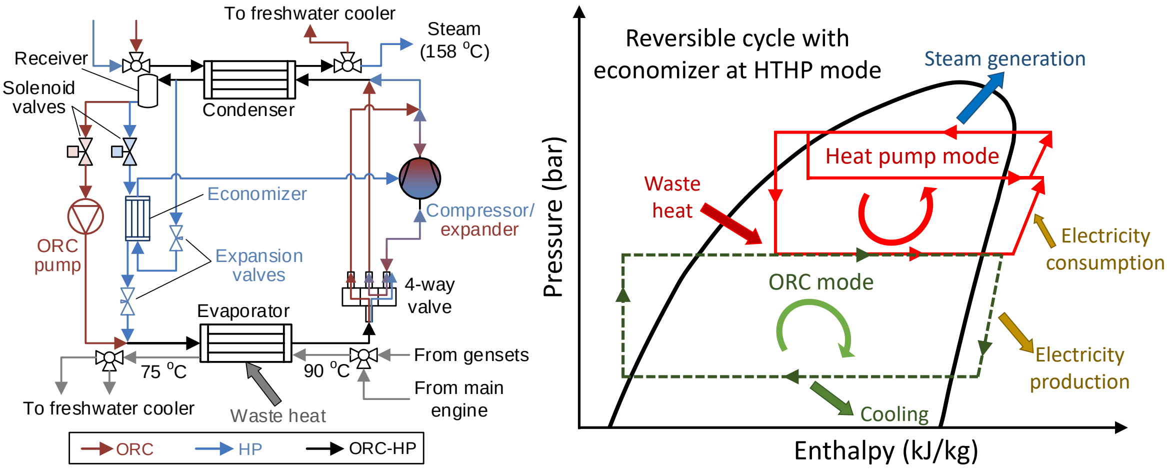

Figure 1: Cycle design of the reversible high-temperature heat pump/ORC unit

The ORC mode requires an additional pump and parallel piping with three-way valves that are actuated according to the operating mode. On the other hand, the economizer and the expansion valve at heat pump mode are by-passed at ORC mode. At heat pump mode, the waste heat is from the cooling water of the gensets that is upgraded to almost 160 ºC for steam generation, replacing the use of the auxiliary boiler (steam is used to preheat the fuel oil and lubricant, for cargo heating, etc.). At ORC mode, the waste heat of either the auxiliary or the main engine is used to produce electricity, reducing the fuel consumption of the gensets. The selection of either heat source depends on their relative magnitude, in order to ensure the efficient operation of the HTHP mode (max. compressor speed reduction of 40%).

Therefore, the reversible unit is operating at heat pump mode when the ship is at port, and at ORC mode at sea, when the main engine is operating and its exhaust gas economizer covers the steam needs. This temporal distinction of the two modes increases the capacity factor and the fuel savings, and leads to a superior cost-effectiveness compared to having individual systems (Kosmadakis and Neofytou, 2019). The relative duration of each mode depends on the ship type and size examined later in this work.

2.2 Operating conditions and design parameters

The supplied waste heat at both modes is hot water at 90 ºC from the high-temperature cooling circuit of the vessel. The temperature glide of this water flow is always 15 K. The heat pump generates saturated steam at a pressure of 6 bar (158 ºC) at the condenser side. At ORC mode, the condenser heat is rejected to the central cooler, with the cooling water at a temperature of 25 ºC, as shown in Figure 1.

A screening of ship types and sizes has been accomplished based on data reported by the 3rd IMO GHG study (IMO, 2015) to identify the waste heat amounts. Table 1 summarizes the main parameters and waste heat sources.

Table 1: Main parameters and waste heat sources in medium (M), large (L) and very large (VL) ships

Ship class / size	Capacity bin (dwt, TEU or gt)	Auxiliary loads at port (kW)		Operation (days/ year)		Waste heat (kW)		Thermal/electrical capacity (kW)	
		Boiler	Engine	At sea	At port	At sea	At port	HTHP	ORC
Bulk carrier / L	35,000–59,999	100	315	173	192	678	84	100	37.3
Container / M-L-VL	3,000–7,999	450	1,165	241	124	3,380	311	450	185.9
	8,000–11,999	520	1,315	256	109	5,032	351	520	276.8
	14,500+	700	1,740	251	114	6,061	464	700	333.4
General cargo / M	5,000–9,999	75	250	166	199	249	67	75	13.7
Oil tanker / M-L-VL	20,000–79,999	300	750	173	192	751	200	300	41.3
	120,000–199,999	500	1,250	206	159	1,412	333	500	77.6
	200,000+	600	1,500	233	132	2,079	400	600	114.4
Cruise / L	10,000–59,999	1,000	3,500	217	148	1,476	933	1,000	81.2

The average/typical power capacities of auxiliary and main engines are reported in the IMO study, with the waste heat amounts estimated considering a 5.2% and 12% of the fuel energy wasted in the jacket water cooling for the main and auxiliary engine respectively.

The capacity of the reversible unit at both modes (last column of Table 1) has been estimated based on typical efficiency values considering the available waste heat (kept constant at sea or at port), which will be refined later in this work. The available waste heat at port is lower by about 4-15 times than the one at sea depending on the ship type/size, not allowing the same ORC unit to operate efficiently. However, this is resolved by reversing the operation to HTHP mode, recovering a fraction of the waste heat that is adequate to cover the full demand of the auxiliary boiler at these moments. The reversible unit is then sized according to the needs of the ORC mode setting the size of the compressor/expander and the surface of the HEXs, while respecting the minimum capacity of the HTHP.

Finally, the working fluid of the reversible unit is R1233zd(E), an hydrochlorofluoroolefin (HCFO) refrigerant with an extremely low ODP and GWP that shows a good performance at both ORC (Eyerer *et al.*, 2019) and HTHP modes (Kosmadakis *et al.*, 2020).

3 MODELING AND COST-BENEFIT ANALYSIS

3.1 HTHP and ORC modeling

A simulation model has been developed in Engineering Equation Solver (EES) software (Klein, 2020), which has built-in libraries of the refrigerant properties and optimisation features. The model solves the refrigerant properties at the different state points of the cycles. Moreover, it considers the pressure drop at the different parts of the cycle (e.g. heat exchangers, piping) and calculates the surface area of the plate HEXs, based on recent correlations for the heat transfer coefficients. Further details of the numerical approach for the plate HEXs and pressure drop as well as its validation are provided in recent works (Kosmadakis *et al.*, 2020; Kosmadakis and Neofytou, 2020). The reference values of the cycle parameters that have been selected are as follows:

- A pinch point temperature difference (PPTD) of 5 K is imposed on all HEXs at ORC mode, leading to an even lower value at HTHP mode, which maximises the cost-effectiveness (Kosmadakis *et al.*, 2020).
- The superheat temperature is set to a typical value of 5 K, while a reduced value for the subcooling at HTHP mode is selected to reduce the condensation temperature, due to the steam generation.
- The superheat at the economizer outlet at heat pump mode is set to 3 K.

The key performance indicator at HP mode is the COP, which expresses the ratio of the heating capacity at the condenser to the electricity consumption of the compressor. On the other hand at ORC mode, the thermal efficiency gives the ratio of the net power production to the waste heat supply. The design indicators are related to the HEXs surface of the evaporator, condenser and economizer, calculated by the numerical model and then used in the cost analysis. These are then adjusted for each ship type/size, based on the sizing of the ORC mode, fixing the HEX surface.

The compressor's displacement is 535 m^3/h for the reference case with a built-in volume ratio (BVR) of 3, in order to identify the main specifications based on the manufacturer's data (Bitzer, type HSK8591). At heat pump mode, a correlation of the volumetric efficiency (n_{vol}) is used (Fu *et al.*, 2002), as a function of the pressure ratio (PR) given by Equation (1). The same expression is also used at ORC mode, but reversed, with this parameter also referred to as the filling factor ($n_{vol,ORC}=1/n_{vol}$).

$$n_{vol} = 0.95 - 0.0125\,PR \qquad (1)$$

The isentropic efficiency ($n_{c,is}$) has been estimated based on the processing of a large number of operating conditions of the screw compressor (evaporation and condensation temperatures, superheat, subcooling), in which all properties have been converted to volume flow rates. This has been done in order to make such correlation applicable for any refrigerant. After that, a regression analysis concluded to a third-order polynomial with independent variable the volume ratio (*VR*) and the suction volume flow rate ($\dot{v}_{in}$, in m^3/h), given by Equation (2).

$$n_{c,is} = 1.4401525E + 01 + 2.8881657E - 01\,VR - 6.9329510E - 02\,VR^2 + 4.6973646E - 03\,VR^3 - 9.2028924E - 02\,\dot{v}_{in} + 1.9589580E - 04\,\dot{v}_{in}^2 - 1.3638752E - 07\,\dot{v}_{in}^3 \quad (2)$$

The above expression is an improved version of a similar one that has been previously used (Kosmadakis *et al.*, 2020) based on a recent work (Astolfi, 2015), with the main improvement for low volume ratios (typically from 1.5 to 2), capturing with a more realistic way the reduction of the isentropic efficiency in that range. The isentropic efficiency of the expander is calculated with the same correlation, but the inlet volume flow rate is corrected with the use of the built-in volume ratio.

3.2 Cost-benefit analysis

The first step of the cost-benefit analysis is the estimation of the equipment cost with the use of appropriate cost correlations that consider the component size (Kosmadakis *et al.*, 2020). Appropriate correlations for the main cost items of the reversible unit are included, such as the HEXs and the compressor/expander (Ommen *et al.*, 2015), while all other auxiliaries have been summed into two main categories. These cost correlations are given in Table 2 for estimating the cost of the reversible unit, which is slightly higher than the cost of the HTHP-only or ORC-only.

Table 2: Cost correlations as a function of the component size

Component	Correlation	Sizing parameter
Plate HEX (evaporator, condenser, economizer)	$C_{HEX} = 15{,}526\left(\frac{A}{42}\right)^{0.80}$	*A*: HEX surface in m^2
Compressor / expander	$C_c = 19{,}850\left(\frac{\dot{v}_{in}}{279.8}\right)^{0.73}$	$\dot{v}_{in}$: displacement at compressor mode in m^3/h
Receiver	$C_{rec} = 1{,}444\left(\frac{V_{rec}}{0.089}\right)^{0.63}$	V_{rec}: receiver volume in m^3
Piping, tanks and auxiliaries	$C_{p,t,a} = 0.10\left(\sum C_{HEX} + C_c + C_{rec}\right)$	-
Pump and electrical parts	$C_{p,e} = 0.10\left(\sum C_{HEX} + C_c + C_{rec}\right)$	-
Refrigerant	$C_{ref} = 50\,m_{ref}$	m_{ref}: mass of refrigerant in kg (specific cost of 50 EUR/kg)

Then, a multiplication factor is used to convert the equipment cost to the total capital cost, which considers additional costs, such as the labour cost for the system design and production, engineering and on-site integration. The typical value of this factor is 4.16 (Lemmens, 2016), with the heat extraction cost approaching the ORC cost in industry. However, a much lower value of 1.5 is used here, since the waste heat of the jacket water is already available with minor additional equipment needs, and the heat rejection at ORC mode requires small modifications to the piping of the engine room.

The operating costs include the electricity/fuel for running the heat pump calculated by the electricity consumption, the fuel price that corresponds to a very low sulphur fuel oil equal to 500 USD/ton (about 415 EUR/ton), a typical engine efficiency of 45%, and a fixed annual operation and maintenance cost equal to 2% of the capital cost (Kosmadakis *et al.*, 2020). The annual savings refer to the fuel savings of a steam auxiliary boiler with an efficiency of 82.6% to produce the same amount of steam, depending on the fuel price, as well as electricity/fuel savings due to the ORC operation. The difference of the above cash flows decides the net annual savings of the reversible unit.

The annual cash flows are assumed constant each year and are obtained with a different capacity factor among the two modes, as given in Table 1. The discounted PBP is then calculated, using a discount rate of 6%, which is an average value for waste heat recovery projects in ships (Olaniyi and Prause, 2020).

4 RESULTS AND DISCUSSION

The first part of the results concerns the investigation of the performance of the two modes, considering the main cycle parameters, for reaching a high performance and a short PBP. Then, the outcomes are extrapolated to match the design parameters of the screened ships presented previously.

4.1 Performance and cost-effectiveness of the two modes for the reference case

For the reference case with a 535 m^3/h compressor, the main interest is on the surface of the evaporator and condenser that greatly affect the performance of the reversible unit. The waste heat supply is in the range of 250-350 kW and is mainly affected by the evaporator surface, while the heat capacity of the HTHP is about 500 kW and the net electrical capacity of the ORC in the range of 12-20 kWe. The effect of the evaporator and condenser surface on the COP and thermal efficiency is presented in Figure 2, indicating the reference surfaces that have been obtained with the cycle parameters provided previously.

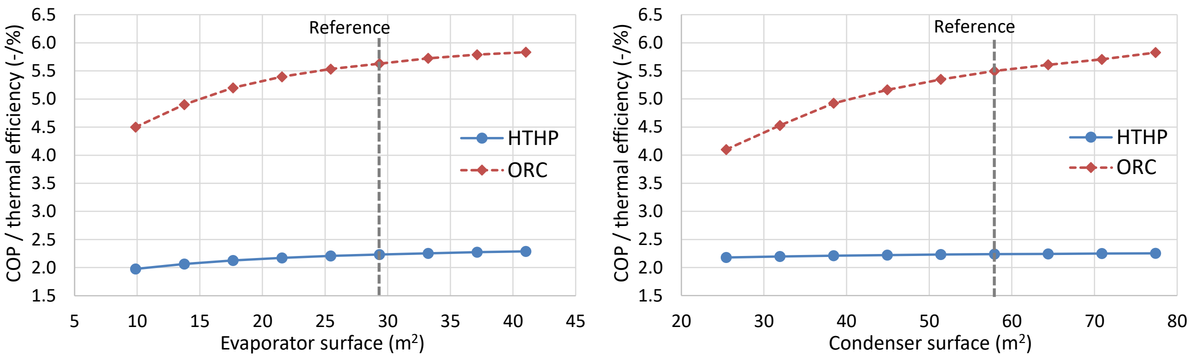

Figure 2: Effect of the evaporator and condenser surface on the COP and thermal efficiency of the HTHP and ORC modes of the reversible unit

The variation of the HEX surfaces introduces a large difference to the PPTD in the range of 3-10 K at ORC mode, greatly affecting the thermal efficiency that is reduced by as much as 25% compared to the reference. The maximum ORC efficiency that can be reached is almost 6%. On the other hand, the HEX surfaces are already large for the HTHP mode resulting to a low PPTD in the range of 0.8-3 K at both HEXs, and thus having a very small effect on the COP, being about 2.3.

The main properties of the cycles of the two modes are given in Table 3 for the reference conditions with an ORC efficiency of about 5.5% and a COP of 2.34, while in Figure 3 is shown the pressure-enthalpy chart of the HTHP and ORC cycle. Although there is a large difference of the mass flow rate at the evaporator (twice at HTHP mode), the ratio of the inlet volume flow rates of the two modes is about the same as the built-in volume ratio, and thus no speed regulation is needed at either mode.

Table 3: Properties of the HTHP and ORC modes

Mode	P_{high} (bar)	P_{low} (bar)	$T_{ev,out}$ (°C)	m_{ev} (kg/s)	$\dot{V}_{in}$ (m^3/h)	Q_{ev} (kW)	n_{is} (%)	$P_{el,c/exp}$ (kW)	Q_{cd} (kW)	COP/n_{th} (-/%)
HTHP	33.26	5.65	76.6	3.16	488.9	344.4	66.3	257.5	601.9	2.34
ORC	5.56	2.01	78.1	1.54	194.8	353.3	67.0	20.4	333.8	5.52

Based on the above sizing and performance values, it is possible to calculate the capital and operating costs, using a typical capacity factor of 33% at HTHP mode and 67% at ORC mode. The resulting net annual savings are then used to calculate the discounted PBP of the reversible unit, which is shown in Figure 4 for variable HEX surfaces. Moreover, the PBP of the HTHP-only and ORC-only units are also shown with the above capacity factors for comparison purposes aiming to identify the benefits of the reversible configuration. It should be stressed here that the PBP of the reversible unit is always shorter than the one of the HTHP and the ORC, since it sums up the savings of both modes by reaching a capacity factor of 100%.

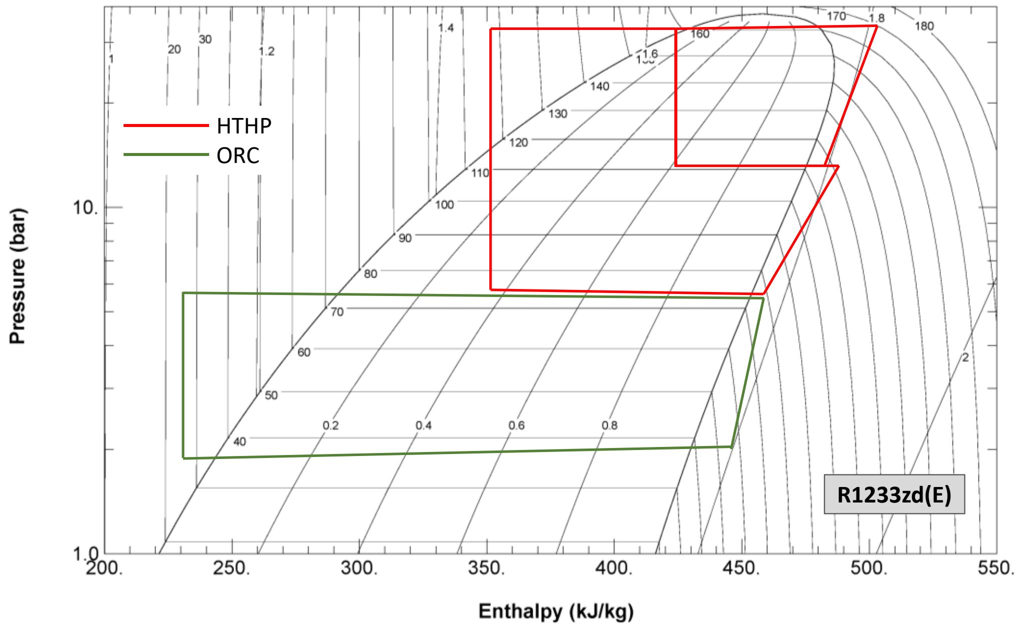

Figure 3: Pressure-enthalpy chart of the two cycles for the reference conditions

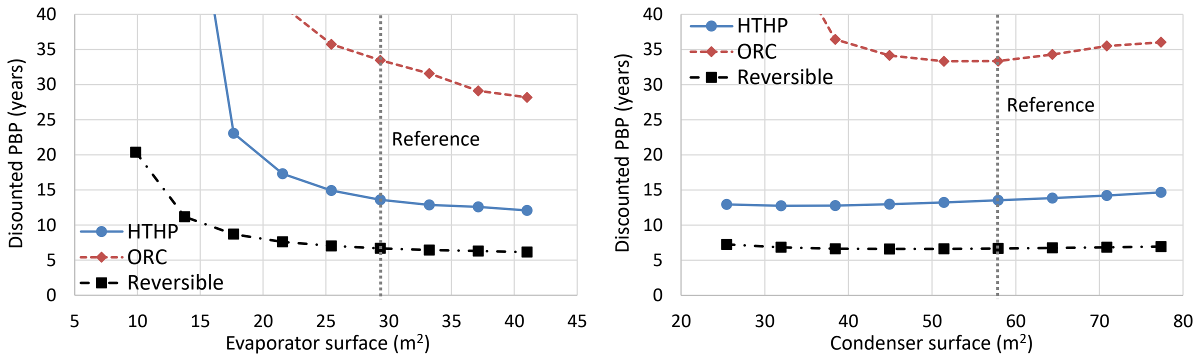

Figure 4: Effect of the evaporator and condenser surface on the discounted PBP of the HTHP, ORC and the reversible unit

The evaporator surface should be large enough to reach the reference PPTD at ORC mode and thus minimize the PBP of the reversible unit. On the other hand, the variation of the condenser surface has a minor effect on the PBP, showing a minimum for a moderate surface of 50 m^2. The PBP of the ORC-only is always longer than 28 years, due to the small scale of 20 kW$_e$ having a high specific capital cost of over 4,200 EUR/kW$_e$. The main reasons for this high specific cost is the low thermal efficiency of the ORC and the small scale, with the latter examined next.

4.2 Cost-effectiveness of the up-scaled units

The previous results correspond to a relatively small system for waste heat recovery of about 350 kW. As the scale increases, the specific cost decreases, leading to an enhanced cost-effectiveness that is expressed through the discounted PBP shown in Figure 5 for a variable waste heat source, once both modes operate at the design conditions. The surfaces of the evaporator and condenser are 40 and 50 m^2 respectively, which minimize the PBP, as concluded previously, and they are appropriately scaled to account for the higher or lower capacity.

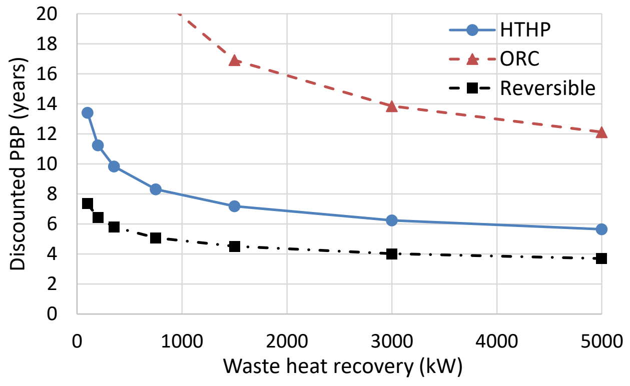

Figure 5: Effect of the system scale on the discounted PBP of the HTHP, ORC and the reversible unit

The discounted PBP of the reversible unit is much shorter than the ones of the HTHP and ORC for the same waste heat, reaching a PBP of 4 years for a waste heat source of over 3,000 kW. The PBP of the ORC is always longer than 12 years, indicating that waste heat at higher temperature should be recovered for increasing its thermal efficiency and reducing its specific cost. For the operating conditions assumed, the ORC specific capital cost is reduced up to 2,900 EUR/kW$_e$ for the maximum waste heat source considered (including integration work). Moreover, the PBP of the HTHP can be shortened to 6 years for a large scale system. However this scale-assumption is extreme due to the fact that the steam requirements of even large ships are up to 1,000 kW, as shown in Table 1. By respecting this threshold, the PBP of the HTHP becomes longer than 8 years.

4.3 Performance and payback period in various ships

The previous results for the reversible unit have been obtained, considering that both modes operate at design conditions with a capacity factor of 33/67% for the HTHP and ORC modes respectively, and exploiting the same amount of waste heat. However, restrictions are introduced for the HTHP mode, since the heating demand is lower than its nominal capacity in all ship types/sizes of Table 1, exploiting a fraction of the available waste heat of the auxiliary engines at port. This brings lower annual savings than the actual potential of the HTHP. Considering the lower limit of the part-load operation of the HTHP mode, the ORC capacity of the reversible unit is restricted as well, since they share the same compressor/expander. Moreover, the system sizing (e.g. compressor displacement, HEXs surface) is the one that minimizes the PBP (after appropriately scaled), as described in the previous section.

The resulting (net) fuel savings for the HTHP and ORC modes of the reversible unit are presented in Figure 6, based on a fuel specific thermal content of 40.4 MJ/kg. An ORC-only solution (no reversible operation) is also examined and its savings presented in the same figure. This solution exploits the whole waste heat of the main engine in the different ships based on their capacity factor of each mode.

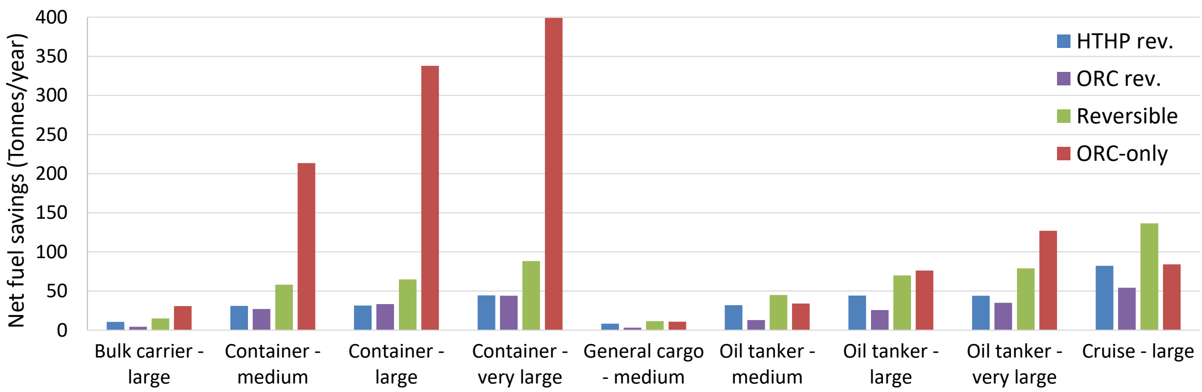

Figure 6: Net annual fuel savings of the HTHP and ORC modes of the reversible unit and of an ORC-only solution in different ships

The fuel savings of the ORC-only solution are significant in all containerships (over 1% of total fuel consumption), due to the large capacity of the ORC and the many operating days at sea (about 70%), while the reversible unit brings much lower savings equally shared between its two modes. In oil tankers and cruise ships the fuel savings of the reversible unit and the ORC-only are comparable (relative savings in the range of 0.8-1.7%). Finally, in bulk carriers and cargo ships the savings are very low, due to their small size, with a low nominal capacity of both the reversible and the ORC-only solution, and on top of that their operating days at sea are just 50%, resulting to a relative fuel saving below 0.6%. This is also reflected in the discounted PBP shown in Figure 7, based on a fuel oil price of 415 EUR/ton.

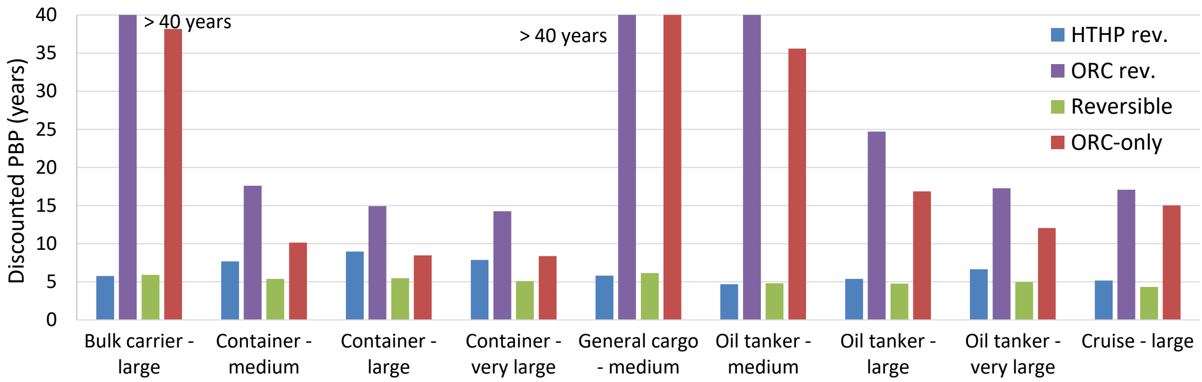

Figure 7: Discounted payback period of the HTHP and ORC modes of the reversible unit and of an ORC-only solution in different ships

The discounted PBP of the reversible unit is always lower than the one of the HTHP and ORC modes, since the fuel savings are higher due to the high capacity factor. In all ship types it is shorter than 6 years, although its fuel savings are lower than the ones of the ORC-only solution in most of the cases. The latter seems very cost-competitive in containerships, achieving a PBP of 8-10 years.

The above exhibit the merits of the reversible unit that even if it does not save more fuel than the ORC-only solution in all ship types, it always shows a shorter discounted PBP. It should be stressed that in case the fuel cost is increased in the future or CO_2 emissions costs are introduced, the cost-effectiveness of all solutions will be greatly improved. Indicatively, the increase of the fuel oil price to 600 EUR/ton shortens the PBP of the reversible unit to less than 4 years in all ships and just 2.8 years in large cruises.

5 CONCLUSIONS

An alternative waste heat recovery configuration has been examined, reversing its operation between ORC and HTHP mode for electricity (at sea) and steam production (at port) respectively. Both modes contribute to fuel savings, while their combination into a single reversible unit increases the annual operational hours and the total fuel savings. The first part of the results focused on identifying the performance of each mode in relation to the sizing of the evaporator and condenser, which is mostly decided by the ORC mode needs, and reaches an efficiency of 5.5%. At the same time, the discounted PBP in relation to the HEX surface has been presented, showing that the reversible unit enhances the economics of such application, especially for large system scales with waste heat above 1,000 kW.

All solutions have been examined in different ship types/sizes, using typical indicators and capacity factors, showing that the ORC and HTHP modes save about the same fuel. An ORC-only solution has been also examined, whose capacity is not restricted by the HTHP mode, and exploits larger amounts of waste heat. The reversible unit adds the benefits of both modes due to the increase of the operational days per year, achieving comparable savings as the ORC-only in most of the ship types, and reaching a much shorter discounted PBP in the range of 4 to 6 years, which can be even below 3 years in case the fuel oil price increases. This is expected to happen in future scenarios with the increase of the number of ships fueled with very low-sulphur fuel oil and the introduction of alternative fuels in shipping.

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