Safety and CO_2 emissions: Implications of using organic fluids in a ship's waste heat recovery system

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Abstract

Current Marine Policies and regulations greatly favour the use of efficiency enhancing technologies such as the Organic Rankine Cycle (ORC) waste heat recovery systems (WHRS), through the entry into force of International Maritime Organisation (IMO) Energy Efficiency Design Index (EEDI). However, safety regulations such as IMO Safety Of Life At Sea (SOLAS), International Gas Code and Classification Societies still consider the use of highly flammable organic fluids on board ships as hazardous and undesirable, requiring special Administration approval. The benefits of organic fluids in emerging technologies will likely increase their usefulness on board in the near future. Furthermore, current ship safety systems and integrated platform management systems greatly reduce the risks associated with their low flash point making them acceptable for marine use given specific design considerations.

This paper studies the case of an Aframax tanker navigating the route North Sea – Naantali, Finland using a slow speed diesel engine. A code with a multi-objective optimization approach generated explicitly for this purpose produces different optimal WHRS designs for the vessel's operating profile. The WHRS is installed after the turbo compressors in the exhaust gas system, where it absorbs part of the available waste heat and converts it to electricity using a generator. This results in a reduction in fuel consumption, hence decreasing the emission of greenhouse gases. The different optimal designs are compared with a steam WHRS to show the strengths and weaknesses of using an ORC WHRS on board.

The ORC technology is at its early stages of development in the marine field, it is important that safety policies follow the evolution of the technology and its associated safety equipment. This paper will serve to recognize the specific safety considerations associated with the ORC and highlight the advantages of carrying organic fluids on board as a solution to increasing CO_2 emission restrictions and other environmental concerns.

Keywords: Waste heat recovery system, organic Rankine cycle, organic fluids, CO_2 emission reduction, safety.

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Nomenclature

Symbol	Name	Units	Symbol	Name	Units
C_F	Carbon Factor	t of CO_2/t of fuel	N	Rotational speed	RPM
C_p	Specific heat	kJ/kg-K	$\dot{Q}$	Heat transfer rate	kW
CS	Carbon Savings	t of CO_2/h	$SFOC$	Specific Fuel Oil Consumption	g/kWh
$EEDI$	Energy Efficiency Design Index	g CO_2/t-nm	T	Temperature	K
FP	Fuel Price	€/t	U	overall heat transfer coefficient	kW/m^2-K
FS	Fuel savings	€/h	$\dot{V}$	Volumetric flow	m^3/s
h	Enthalpy	kJ/kg	w	Weight factors	-
P	Pressure	kPa	$\dot{W}$	Power	kW
$\dot{m}$	Mass flow rate	kg/s	**Greek Symbol**	**Name**	**Units**
			η	Efficiency	%

Subscripts and superscripts	Name	Subscripts and superscripts	Name
a	Approach	p	Pump
$cond$	Condenser	pp	Pinch Point
cr	Critical	sat	Saturation
C	Sink/Cold	$size$	Size variable
d	Design condition	S	Seawater
e	Electrical	t	Net
exp	Expander	th	Thermal
g	Generator	to	Thermal oil
H	Source/Hot	wf	Working Fluid
i	Inlet/In	W	Work output variable
o	Outlet/Out	η	Efficiency variable
off	Off-design condition		

1. Introduction

While shipping is already very energy efficient compared to other modes of bulk transport, its level of activity makes it responsible for approximately 3.3% of global CO_2. By 2050 this is likely to increase by 150 to 200% [1,2]. Growing environmental concerns drove the International Maritime Organisation (IMO), through the Internal Convention for the Prevention of Pollution from Ships (MARPOL), to enforce a number of measures aimed at regulating emissions, particularly SO_x, NO_x and more recently CO_2, with the adoption of the Energy Efficiency Design Index (EEDI). A number of target areas considered particularly sensible to marine emissions have been designated as Emission Controlled Areas (ECAs) and are subject to stricter regulations for example the Baltic Sea. The number of ECAs is expected to increase in the near future requiring ship owners to comply with more stringent regulations to continue global shipping. Diesel engines are typically the prime mover of choice for high efficiency shipping vessels. Their efficiency having plateaued in recent years, waste heat recovery systems (WHRS) are now a likely solution for reducing greenhouse gas (GHG) emissions while still using Marine Diesel Oil (MDO) as fuel.

Marine WHRS use available ship's waste heat (e.g. exhaust gas or scavenge air) via a heat exchanger process (e.g. boiler), which evaporates a working fluid such as water or organic fluid. The fluid is then expanded (e.g. inside a turbine) and usually converted into electricity. The remaining energy is rejected to a heat sink, normally seawater. A marine WHRS has to face different challenges and restrictions when compared to a land based system:

a) high sulphur content in marine fuels, limiting the amount of absorbed heat due to sulphuric acid corrosion [3–6];
b) ships are much more sensitive to a WHRS' weight and the consequent impact on stability and cargo capacity;
c) diverse operating profiles and engine speeds with the engine often operating at part load; and
d) sea water is used as a cooling medium this can vary in salinity and from -3˚C to +30˚C as the ship navigates around the globe [7].

The most common marine WHRS is the steam cycle or Rankine cycle (RC). This cycle has been extensively researched [8–13] and is available for the marine industry [5,14]. MAN predicts that the use of a steam WHRS combined with a power turbine can reduce the EEDI by more than 9% [14]. With engine modifications to meet more stringent emission regulations, higher marine engine's thermal efficiencies, and speed reduction strategies, it can be expected that the waste heat quality (i.e. temperature) and availability will be reduced. This produces an important decrement in power production and thermal efficiency for the RC system.

An alternative option to the RC when facing low to medium quality waste heat (i.e. between 30˚C to 650˚C) is the use of organic fluids as a working fluid. Organic fluids contain carbon molecules in their structures. The advantages of these fluids over water are [15–17] [1]:

a) the vaporization heat, which is lower, allowing a better match for the waste heat at low temperatures;
b) enables the use of turbines with less pressure stages and due to a larger density than water the size of the expander can be reduced;
c) low drop in specific enthalpy along the expansion process which increases the mass flow rate and affecting directly the WHRS power output; and
d) a large catalogue of fluids that facilitates the creation of a tailored WHRS for the heat source [15].

On the other hand is important to consider also when selecting an organic fluid the fluid's instability due to chemical decomposition at high temperatures and pressures [18], hazard levels and flammability.

ORC WHRS is a mature technology in land base systems [19,20], and they are starting to gain momentum in the marine industry for low grade waste heat (i.e. up to a temperature of 160˚C) such as in the vessel's excess steam (e.g. OPCON Energy Systems AB [21]). From the academic point of view the vessel's low grade waste heat recovery via an ORC has been studied [22–26] showing promising results from different WHRS layouts and waste heat management approaches. Important work using the available heat from the marine engine's exhaust gas is found in Larsen et al. [27] and Nilsen et al. [6]. Both studies use the multi-objective optimization with the genetic algorithm (MOGA) in order to find marine ORC WHRS designs, which can reduce CO_2 emissions while remaining highly efficient. This work will focus on using the available waste heat in the exhaust system after the gas has gone through the turbo compressors without considering scrubbers in the exhaust gas system. The work presented here differs from Larsen et al. in the sense that the ORC will be coupled with a thermal oil in order to transport the waste heat from the exhaust gas to the organic fluid, avoiding the presence of flammable – but highly efficient – fluids in the engine room.

While good quality technical work has been done in the area of marine ORC WHRS and MOGA, this paper also considers practical applications and system integration. A case study is presented using an Aframax [2] tanker, which considers different speeds and loading conditions (e.g. in ballast) giving a clearer understanding of the WHRS capabilities and requirements at sea. The case study allows for a better comparison between RC and the different ORC systems which highlights the strengths and weaknesses of the ORC WHRS. This paper also considers system integration and the hazards organic fluids present. Current policies regulating the use of highly flammable fluids on board ships are discussed as well as the state of safety and monitoring technologies to render their use at sea safe. While emissions-regulating policies evolve, increasingly restricting emissions and favouring energy efficient technologies, safety regulations appear to remain

[1] The impact of the benefits has to be determined in a fluid by fluid basis.
[2] An Aframax tanker is a type of liquid carrier vessel with a deadweight between 80,000 to 120,000 tonnes [77].

conservative and static, they do not yet fully consider technological advancements in energy efficiency enhancing devices and safety equipment.

2. Case Study

2.1 Route characteristics, vessel's speed and operational time

A platform located in North Sea extracts oil from deep waters to supply the energy demand in Finland. The distance between the platform and the sea port in Naantali (Figure 1) is 2,235 km. This route is inside Europe's North Sea and Baltic Sea ECA [28], and is assumed that throughout the year it will be free of ice with a yearly average seawater temperature - including anomalies- of 5˚C [29].

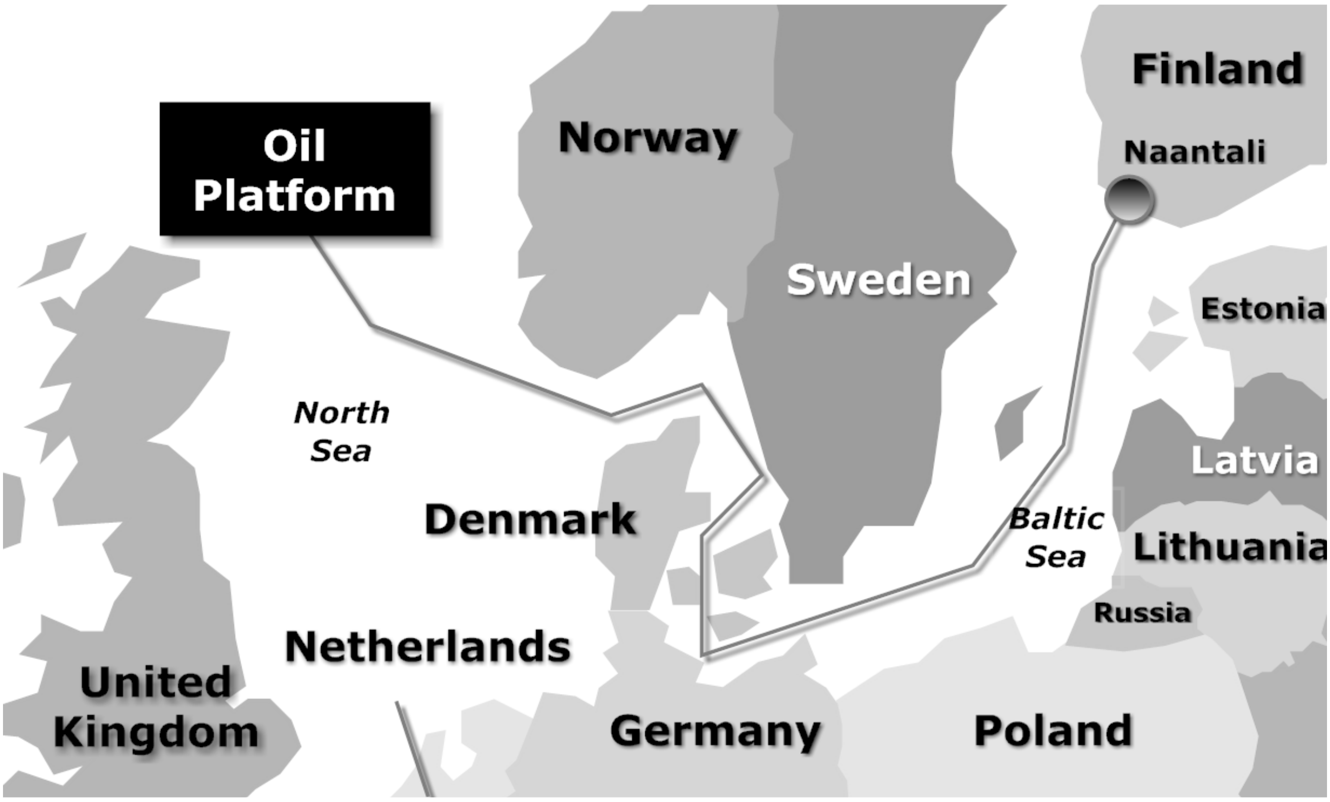

Figure 1 – Route covered by the Aframax tanker route [30]**.**

An Aframax tanker, with the characteristics shown in Table 1, is commissioned to transport oil between the platform and port. The tanker's design speed is 14.0 kt requiring 16.4 MW of power. Maximum speed is 15.4 kt demanding 21.8 MW (assuming the usual cubic power-speed relationship).

Table 1 - Tanker main characteristics.

Deadweight (t)	*Length (m)*	*Beam (m)*	*Draught (m)*
107,113	247	42	15

The duration of a single voyage is about 94 hours [30] using the operating profile shown in Figure 2. Assuming that the vessel will operate for 58% of the year (i.e. around 5,100 hours) it will complete 54 single voyages [31]. Of these half will be fully loaded and the other half in ballast conditions.

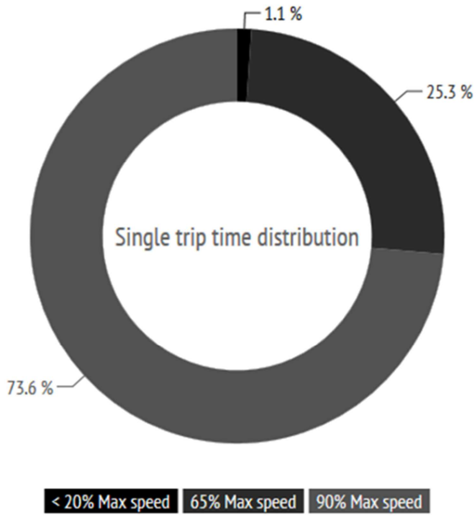

Figure 2 - Aframax tanker percentage of time spent at the different speeds in a single trip.

The engine conditions at the design speed and fully loaded will be used as the reference point for this study. It was assumed that the ballast speed profile will be the same as in the fully loaded condition, changing only the ship's power requirement due to a draught reduction. The influence of weather is neglected.

Table 2 - Power required by the Aframax tanker assuming that the power requirement is related to the cube of the vessel's speed.

Normalized Speed (%)	*Power required fully loaded (MW)*	*Power required ballast (MW)*
<20	0.2	0.1
65	5.9	4.8
90	16.4	13.0

2.2 Engine performance

The vessel has an installed power output of 21,840 kW delivered by MAN 6G70ME-C9.5-TII. The engine is set with a high load tuning and uses marine diesel oil (MDO) [32].

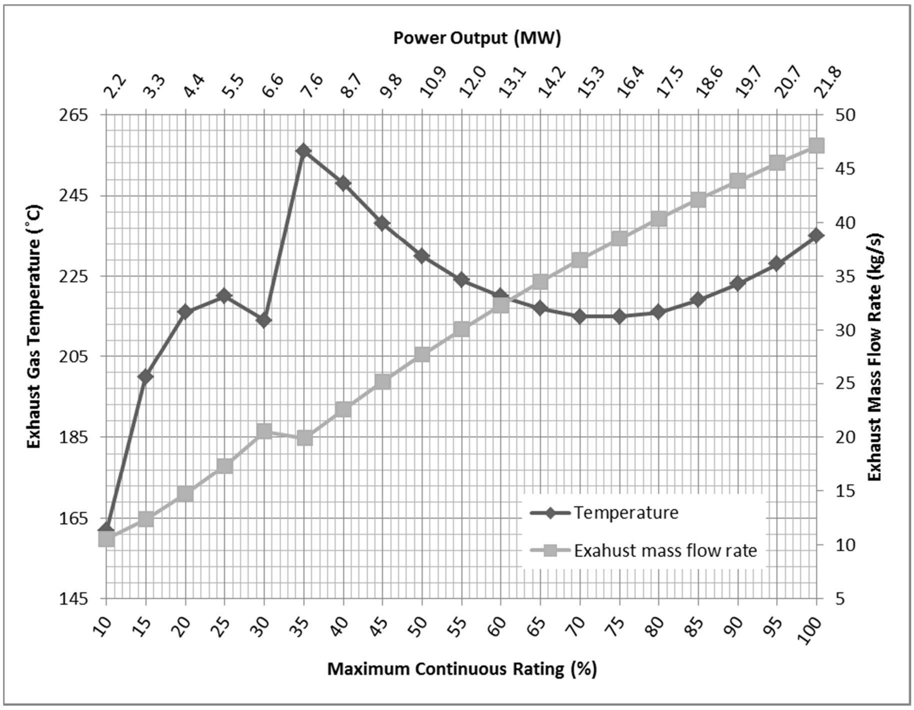

Figure 3 - Engine's power output, exhaust gas temperature after the turbos and mass flow rate for different loadings [32]**.**

2.3 *WHRS layout and location in the exhaust gas system*

The WHRS uses the heat available from the exhaust gas - assumed to behave as air - after it has passed the turbo-compressors (see Figure 4). With this layout it is assumed that the ship's steam demand would be covered by an auxiliary boiler and that it will not have an impact on the performance of the ship (i.e. it does not reduce the engine performance due to a pressure drop in the exhaust or increase the resistance of the ship due to its added mass).

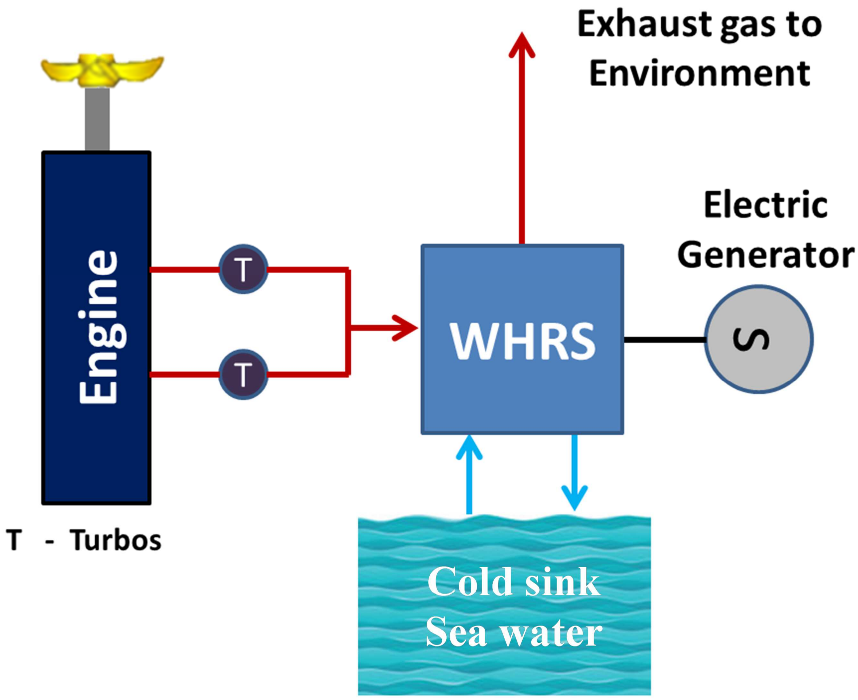

Figure 4 - WHRS location in the exhaust gas system.

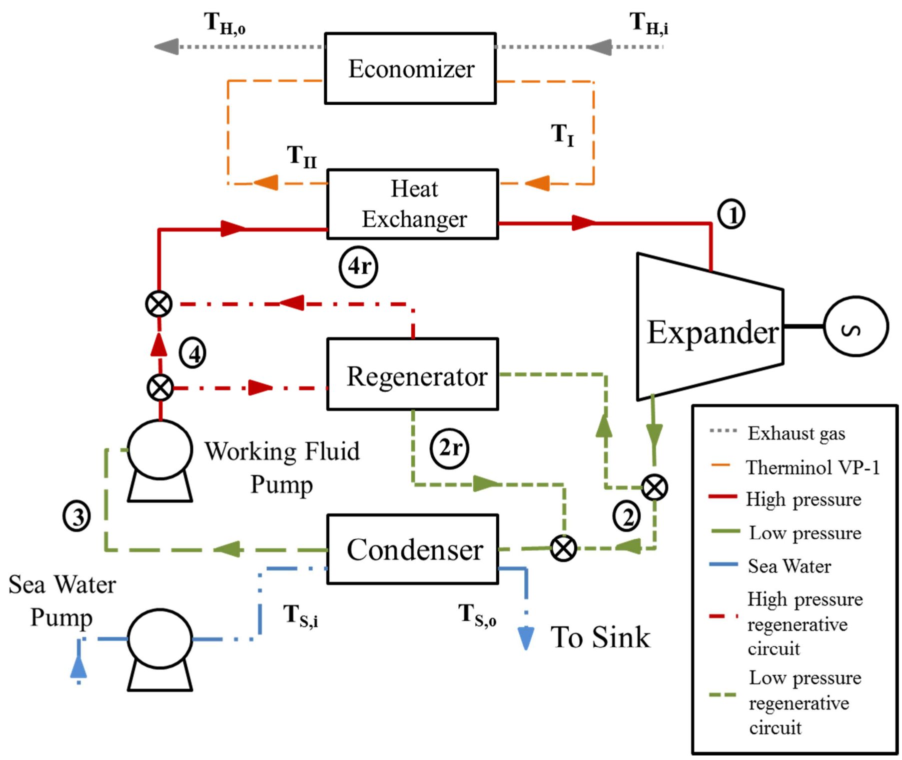

Figure 5 - General layout of the WHRS.

Since the fuel contains sulphur, the minimum exhaust gas temperature in the outtakes was assumed to be 165˚C in order to avoid soot and acid corrosion [5,33]. This means that at design condition there is around 2 MW of waste heat available.

The general thermodynamic system is composed of five sections (Figure 5): Exhaust gas, thermal oil, working fluid high pressure, working fluid low pressure and sea water. Depending on the working fluid analysed some of the sections may not be included in the analysis.

2.4 Working fluids

The working fluids used in this work are only analysed in their subcritical state, meaning that the WHRS evaporating condition cannot go beyond the critical temperature (T_{cr}) and pressure (P_{cr}) values shown in Table 3. From water to hexamethyldisiloxane, the fluids were selected due to their good performance in Saavedra et al. [16]. R245fa is a refrigerant selected as it is not flammable, it has low hazard levels and performed well in Ulrik et al. [27,34]. On the other hand, the global warming potential (GWP) of R245fa is 950, which means that a unit of mass of R245fa, should it escape to the atmosphere, will store 950 more heat energy in a period of 100 years than CO_2 with the same unit of mass.

Table 3 - Working fluids selected with their GWP value in a time interval of 100 years, auto-ignition and decomposition temperatures, and flash points.

Working Fluid	GWP_{100} [a]	Auto-ignition Temperature (˚C) [b]	Decomposition Temperature (˚C) [c]	Flash Point (˚C) [d]	T_{cr} (˚C) [e]	P_{cr} (MPa) [e]
Water	N/A	-	2000	-	374	22.06
Benzene	N/A	562	760	-11	289	4.91
Toluene	2.7	536	399	4	318	4.13
Heptane	3.0	223	550	-4	267	2.74
Hexamethyldisiloxane (MM)	<10	341	300	-2	246	1.94
R245fa	950	412	250	-	154	3.65

[a] Toluene: [35]; Heptane: [36] ; R245 fa:[37] , MM: [38].
[b] Taken from [39]. For R245fa:[34].
[c] Water: [40], Benzene: [41], Toluene: [42], heptane:[43] , R245fa: [34],MM: [44].
[d] Taken from [39]. For R245fa:[34].
[e] Taken from [45].

Most of the selected working fluids being hydrocarbons, higher priority was assigned to environmental concerns and efficiency than to fire safety, it is the authors' point of view that while SOLAS prescribes a flash point of no less than 60˚C, the limited amount of working fluid required for the system and today's leak detection and firefighting technology allow for the risks associated with such fluids to be adequately reduced.

2.5 Thermal oil

Therminol VP-1 was used as the thermal oil, which enables the heat transfer between the heat source and the working fluid [46]. A thermal oil is required for two main reasons:

a) to preserve the working fluid's operational life, specially the organic fluids, by not reaching their decomposition temperature; and
b) to increase safety on board by removing flammable fluids from inside the engine room [47] and permitting the ORC system to be contained in a relatively compact and separate enclosure where leaks or fires are more easily contained.

A system not using a thermal loop would require approximately 3,500 kg of toluene due to the very large coil boiler housed inside the exhaust path. This quantity of working fluid can be reduced to approximately 850 kg when using a thermal loop [48]. Further reductions could be possible by using modern compact heat exchangers [49].

The algorithm added the thermal oil circuit when the temperature of the exhaust gas along the economizer and at any of the vessel MCR was higher than the decomposition or auto-ignition temperatures; or if the working fluid flash point was below 43˚C [47]. In this paper all the organic fluids required the thermal circuit.

3. Theory and calculations

3.1 Expander, Pumps and Electrical Generator

The thermal efficiency is defined as follows:

$$\eta_{th} = \frac{\dot{W}_t}{\dot{Q}_i} = \frac{\dot{W}_o + \dot{W}_i}{\dot{m}_{wf}[(h_1 - h_4)]} \tag{1}$$

The power output of the expander ($\dot{W}_o$) is given by:

$$\dot{W}_o = \dot{m}_{wf}(h_1 - h_{2s})\eta_{exp} \tag{2}$$

$\dot{m}_{wf}$ represents the working fluid mass flow rate, h_1 is the enthalpy before the expansion process, h_{2s} is the enthalpy at low pressure after an isentropic expansion and η_{exp} is the expander's isentropic efficiency.

The pump's power input ($\dot{W}_i$) is defined by:

$$\dot{W}_i = \frac{\dot{m}_{wf}(h_3 - h_{4s})}{\eta_p} \tag{3}$$

h_3 is the enthalpy after the condenser, h_{4s} is the enthalpy at high pressure after an isentropic compression and η_p is the pump's isentropic efficiency.

The electric power delivered by the WHRS is defined as follows:

$$\dot{W}_e = \dot{W}_o\eta_e + \frac{(\dot{W}_i + \dot{W}_{S,i})}{\eta_e} \tag{4}$$

Where η_e is the electrical efficiency assumed constant at 90%. $\dot{W}_{S,i}$ is the power input from the seawater pump which follow the power curve of a Grundfos CR-150-4-2 pump which has a minimum head of around 110 meters and an efficiency of around 70%.[50]:

$$\dot{W}_{S,i} = 0.117\dot{m}_S + 38.60 \tag{5}$$

$\dot{m}_S$ is the seawater mass flow rate which had an assumed salinity of 35,000 ppm, it is accepted that it will be less than this for some parts of the voyage but the resulting change in mass flow rate is less than 3%:

$$\dot{m}_S = \frac{\dot{Q}_{wf,cond}}{C_{p\_S}(T_{S,i} - T_{sat,l} + \Delta T_{pp,C})} \tag{6}$$

3.2 Marine Diesel Generator

It is assumed that the WHRS will be covering part of the electrical demand inside the tanker. In order to quantify the fuel savings a model of a marine electrical generator was used. The generator model uses data from a Wärtsilä 6L20DF diesel generator which has an MDO specific fuel consumption ($SOFC_g$) of 187 g/kWh and delivers around 1 MW_e of electrical power at an electrical efficiency of 96% (included in the fuel consumption) [51].

$$FS = \frac{\dot{W}_e * SFOC_g * FP}{10^6} \tag{7}$$

FP is the price of MDO, which was €703 per metric tonne between the months of July and October 2014 [52,53]. For the CO_2 savings, a carbon conversion factor (C_F) of 3.206 metric tonnes of CO_2 per metric tonne of MDO is used [1]. The CO_2 savings are given by:

$$CS = \frac{\dot{W}_e * SFOC_g * C_F}{10^6} \tag{8}$$

3.3 Heat exchangers

The heat exchangers were assumed to be counter-flow. A regenerator was used for all organic fluids in order to take advantage of the excess temperature at the exit of the expander. In the case of water this was not possible as T_2 was almost the same as at the pump outlet (T_4), preventing the heat transfer process happening.

The regenerator's and condenser's approach temperature (T_a) was set to a minimum of 5˚C, being inside the capabilities of different types of heat exchangers [54,55]. For the economizer's exit, the T_a was set to a constant 5˚C or 10˚C when there is a thermal fluid circuit. A minimum of 5˚C was set as the pinch point temperature difference (ΔT_{pp}).

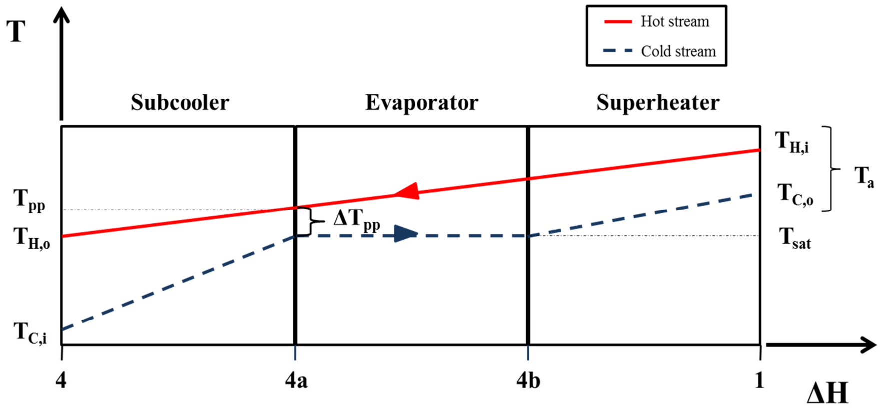

Figure 6 - Counter-flow heat exchanger diagram showing relevant temperatures and definitions.

The area of the counter-flow heat exchanger was calculated from the following formula:

$$A = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{\{\dot{Q} * ln[(T_{H,i} - T_{C,o})/(T_{H,o} - T_{C,i})]\}}{\{U * [(T_{H,i} - T_{C,o}) - (T_{H,o} - T_{C,i})]\}} \quad (9)$$

Where A is the heat exchanger area, q is the heat transferred in the process, U is the overall heat transfer coefficient, for which values were taken from Sinnott [56], and ΔT_{lm} is the log mean temperature difference between the heat exchanger's hot (H) and cold (C) streams.

3.4 WHRS Constraints

Table 4 - Constraints and conditions imposed on the different parts of the WHRS model. These values stay constant unless the contrary is stated.

Equipment	Variable	Value
Expander		
	Inlet temperature, T_1 (˚C)	$T_{H,i}$ -5(water)or T_I -5(organic)
	η_{exp}	75%
Condenser		
	T_a (˚C)	≥ 5
	$\Delta T_{pp,C}$ (˚C)	5 - 30
	Pressure, P_2 or P_{2r} (kPa)	10- 100
	Inlet Sink Temperature, $T_{S,i}$ (˚C)	5
Pump		
	Inlet temperature, T_3 (˚C)	Liquid saturation temperature at low pressure
	η_p	75%
	η_e	90%
Heat Exchanger/Economizer		
	T_a (˚C)	≥ 5
	$\Delta T_{pp,H}$ (˚C)	5 – 30 [a]
	Pressure, P_1 (kPa)	100 - P_{cr}
	Inlet Source Temperature, $T_{H,i}$ (˚C)	Per Figure 3
	Minimum exit Source Temperature (˚C)	165
WHRS Generator		
	η_e	90%

[a] For R245fa's high pressure pinch point was set between 30˚C to 45˚C in order to being able to operate under the conditions and constraints set.

3.5 Off-design conditions

From Table 2 it can be seen that there would be 5 different off-design conditions. At very low speeds (<20% the maximum speed) the WHRS will be off-line.

While the heat transfer area for all heat exchangers and design constraints from the system's low pressure side are kept constant, the WHRS was set to absorb all the heat available. The code modifies the working fluid's mass flow rate and the WHRS high pressure with the affinity laws assuming that the pump isentropic efficiencies stay constant as it changes its rotational speed (*N*):

$$\frac{\dot{V}_d}{\dot{V}_{off}} = \frac{N_d}{N_{off}} = \left(\frac{P_{1,d}}{P_{1,off}}\right)^2 \qquad (10)$$

3.6 Multi-objective optimization

When designing a vessel's WHRS, it is important to consider several aspects of the problem such as WHRS thermal efficiency and sizing (e.g. piping and heat exchangers). The "best" solution will not always be the one that has the highest thermal efficiency or the most compact unit; it will be a trade-off between the "ideal" properties of each characteristic considered. A multi-objective optimization allows finding different optimal solutions with different trade-offs between the desired characteristics. The set of optimal solutions is known as the pareto front and it indicates that no other solution can be better than the one found with a specific trade-off [57]. The multi-objective methodology is coupled with an evolutionary optimization, in this case the genetic algorithm, which allows more than one solution (i.e. population) per iteration (i.e. generation) and creates a new population based on a previous generations [57,58].

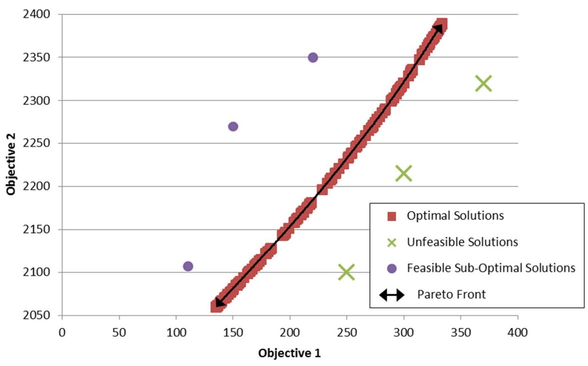

Figure 7 - Representation of the pareto front.

The objectives selected for the optimisation are the electrical power output of the marine WHRS - related to CO_2 emission -, working fluid mass flow rate - piping and pump sizing -, and heat exchanger area. The multi-objective with the genetic algorithm (MOGA) had four variables: high pressure (P_1), low pressure (P_3), boiler pinch point temperature difference ($\Delta T_{pp,H}$), and condenser pinch point temperature difference ($\Delta T_{pp,C}$).

In order to assure that the solutions are repeatable some MOGA parameters had to be set as shown in Table 5.

Table 5- MOGA setting used to solve the different marine WHRS.

Parameter	*Value*
Population size	1,000
Max. Generations	800
Tolerance	10^{-6}
Crossover percentage (%)	70
Migration percentage (%)	35

For the case of the Aframax tanker a single solution is required from the whole pareto front. A scoring system was used to find which of the solutions given by MOGA was the best suited for the case study. The WHRS design score was found by considering other WHRS characteristics as shown in the following equation:

$$Score = w_W * power - w_{size} * size + w_\eta * \eta_{th} \qquad (11)$$

Where the *power* and *size* variables group together different characteristics of the design such as working fluid mass flow rate or seawater pump power consumption. The values of these variables were normalized, in a scale from 0 to 1, with the extreme values found in the pareto front. w_w, w_{size} and w_η are the total *power*, *size* and *thermal efficiency* weight factors. These factors were determined using an Analytic Hierarchy Process (AHP) [59,60] using the authors' preferences between the three variables shown in Equation (11). The inconsistency of the preferences was below 10% making the scores reliable.

The normalized highest score achieved by the optimal design was the WHRS selected for the Aframax tanker. The scores for the optimal solutions ranged from 0.5 to 0.8 approximately where 0.0 was the minimum and 1.0 the maximum.

It was also assumed that no pressure or heat transfer losses are present; there are no leakages; constant efficiencies for the expander, pump and heat exchangers; and is solved as a steady-state problem.

3.7 Software and databases

The model is programmed in Matlab. In particular, tools from NIST Refprop 9.0 [45] for the working fluids, CoolProp [61] for the thermal oil and the code developed by Sharqawy and Mistry [62,63] was used for seawater properties.

3.8 Model Validation

The code was validated against results presented by Saavedra et al.[16], which studied a regenerative ORC and simple RC WHRS for a compressor gas plant. The code used in this paper adapted the boundary conditions, T_1 and regenerator efficiency given in Saavedra et al. The comparison was made only with heptane since it is the only fluid with conditions at all stages of the process.

Table 6 - Some of the important WHRS characteristics for both methodologies, Saavedra et al. and Suarez et al., are shown and compared in order to assess the level of accuracy.

Properties	**Heptane (ORC)**		
	Saavedra et al.	Suarez et al.	Difference (%)
$\dot{m}_{wf}$ **(kg/s)**	7.30	7.43	-1.8
$\dot{m}_{to}$ **(kg/s)**	10.39	10.75	3.5
W_t **(kW)**	1,083	1,067	-1.5
η_{th} **(%)**	26.8	26.6	-0.7
P_1 **(kPa)**	2,594	2,189	-15.6

From Table 6 it can be seen that the results show good correspondence with Saavedra et al. with the exception of the system's high pressure P_1. The main reasons why this discrepancy appears is the calculation of the thermodynamic properties: Saavedra et al. used the Peng-Robison Stryek Vera equation while in this work the data was extracted from NIST Refprop 9.0. When using the data points in each stage of the WHRS (Figure 5) there was a reduction of around 1.5% for the expander power output and 3.0% for the heat absorption when using NIST's software. By correcting the data and assuming that all the heat has to be absorbed and relaxing some of Saavedra's restrictions the heptane's new high pressure would be 2,410 kPa.

Other reasons for differences include the setting of the heat transfer area and thermal efficiency objectives designate Saavedra's solution as suboptimal and slightly different approaches to reconfiguring the system when the regenerator is absent.

4. Results and discussion

4.1 WHRS optimal design

Figure 8 compares the performance and characteristics of the different optimal WHRS found by MOGA.

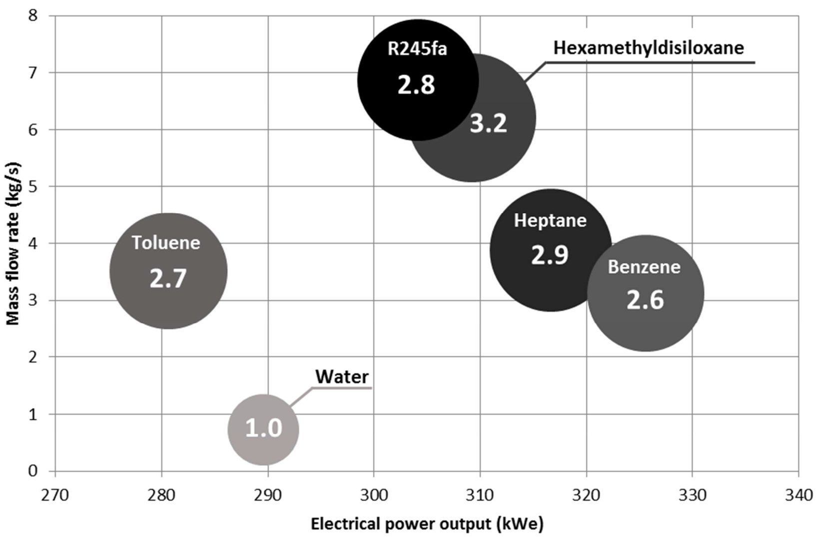

Figure 8 - The different WHRS optimal designs per their MOGA objectives. The bubble diameters represent the heat transfer area and are normalized to the RC system which has an area of 823 m^2.

From Figure 8 it is clear that all the ORC WHRS, except for toluene, deliver larger power outputs than the steam cycle. The largest power output is given by benzene at 326 kW$_e$ which is around 12.4% more than the RC system and represents 30% of the power output from the diesel generator. Toluene's low high pressure (P_1) and slow mass flow rate cannot compensate for the energy drop caused by the thermal oil and it is the WHRS with the lowest power output. R245fa is a contrasting example because it takes advantage from the thermal loop lower temperature and with its large pinch points ($\Delta T_{pp,H}$) manages to operate close to the critical point (i.e. T_{cr} and P_{cr}) increasing the energy entering the expander thanks to a large mass flow rate and high h_1 (see Equation (2)). Water's and benzene's ratio of returned mechanical power output to vessel's installed power at design point is of 2.0% and 2.2% respectively. These values are in a similar region than those found by Theotokatos and Livanos when doing the same comparison between a steam plant and a smaller two-stroke marine diesel engine [10].

Heat exchangers are the main drives for plant size. The approach used to calculate the heat transfer area is simple but it is useful for comparisons between the systems, a more accurate approach would be to design the heat exchangers such as in the case of Ulrik et al. [27] and Pierobon et al. [64]. The RC WHRS offers the most compact design with only 823 m^2 of heat transfer area. The closest organic system is benzene with an area of around 2.6 times larger. The increment in heat transfer area is caused by the inclusion of the regenerator and the intermediate heat exchanger for the thermal oil.

Water also has by far the smallest mass flow rate at 0.7 kg/s, while benzene is 4.3 times greater. The increase in mass flow rate is caused by the organic fluid's low evaporative energy forcing larger mass flow rates to absorb the available waste heat [16]. On a final point regarding the system compactness, ORC turbines tend to have less expansion stages which reduces their size [17].

4.2 Green technology: Fuel savings, CO_2 emissions and EEDI Impact

Figure 9 tells that by installing a WHRS using the available heat from the vessel's exhaust gas could save at least €130k per year. If it is assumed that the vessel will be in operation for 20 years, then the savings could be around €2.6M. The benzene system has the largest savings of approximately €186k. The first non-flammable fluid is R245fa saving around €21k more in fuel than what the RC system achieves, this represents an increase of around 14%. The ORC' fuel savings are an advantage over a traditional marine steam cycle and it may cover the initial costs due to larger heat exchangers and more expensive working fluids. The cost analysis only considers the money saved on fuel by using Equation (7), it does not include purchase, installation or running costs. A deeper cost analysis should be done to assess the equipment cost but it is out of the scope of this paper.

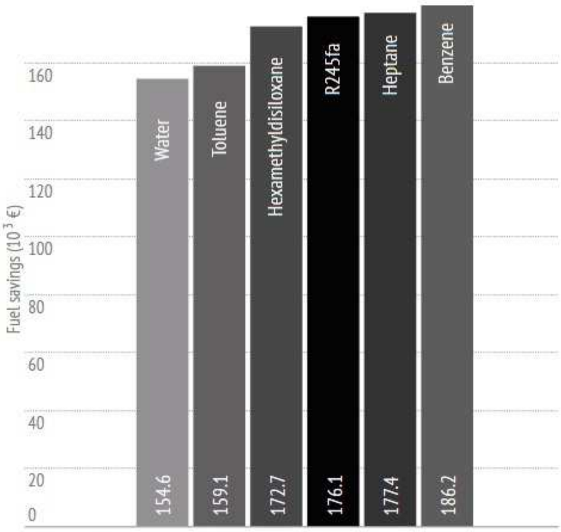

Figure 9 - Fuel savings achieved by the different WHRS during one year of operation on board the Aframax tanker.

Regarding CO_2 emission reductions benzene WHRS delivers the highest emission reductions at 849 metric tonnes after one year in operation. Next is heptane with around 5% lower emission reductions while the RC system achieves around 17% less than benzene for the same period of time.

When comparing the results from Figure 8 and Figure 10 it is seen that the extra performance that the RC system had over toluene at design point is eliminated in a single round trip. All systems at off-design conditions are required to absorb all the available energy; hence the performance increase is given by organic fluids' low evaporative energy forcing a larger mass flow rate. In the case of toluene, its mass flow rate is sufficiently large to outperform the RC cycle in every off-design condition. This example shows that ORC WHRS have superior adaptability over a simple RC when facing lower heat availability.

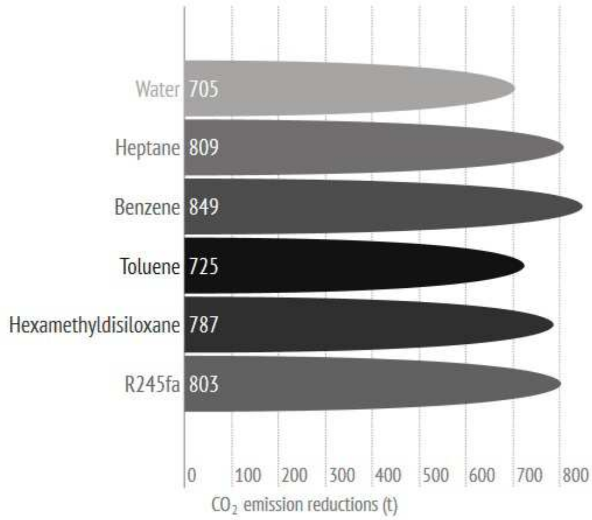

Figure 10- CO_2 emission reductions achieved by the different Aframax's WHRS for a single year of operation.

The Aframax's EEDI was calculated using BIMCO's EEDI calculator [65] having a score of 6.030 g CO_2/t-nm [3]. When installing a WHRS in the exhaust gas system the EEDI was reduced by at least 0.116 g CO_2/t-nm which represents a 1.9% drop in the index. The RC WHRS impact on the EEDI is similar to the one obtained by Theotokatos and Livanos [10] which achieved a reduction of around 1.8% for a two stroke engine. The reduction appears small because the WHRS CO_2 reductions are swamped by the amount of CO_2 emitted by the main engines on board the tanker. However, while the EEDI shows only modest improvements with the ORC WHRS, Figure 9 and Figure 10 show considerable fuel savings and CO_2 emissions reductions when using an ORC WHRS throughout a year of operation, which is a better indication of the system's relevance.

[3] Please refer to IMO MEPC 212(63) for the EEDI formula [78].

4.3 System integration

4.3.1 Safety regulations

The ORC WHRS has clear advantages over the RC, especially when fitted to high efficiency diesel engines. However, current regulations make the use of hydrocarbon-based organic working fluids challenging. Since 1974, IMO SOLAS has considered any fuel having a flash point of less than 60˚C hazardous and not normally permitted on board ships without specific Administration approval or for certain emergency generators not located inside the engine room [47].

Most classification societies base their regulations on IMO SOLAS, however, their situation allows for a more dynamic evolution of the rules, enabling for exemptions such as heating up fuel above its flash point for viscosity purposes or the Tentative Rules for Low Flashpoint Liquid Fuelled (LFL) Ships released by DNV GL, which defines classification requirements for the use of methanol as fuel (12˚C flash point) [66,67].

The flash point has historically been considered a major factor in assessing fuel safety. A review of marine accidents showed however that the vast majority of incidents relating to fuel were not triggered by the ignition of vapours at a temperature above its flash point but by spilled fuel coming into contact with a surface at a temperature exceeding its auto-ignition temperature, making the latter an important factor when assessing a fluid's safety [68]. All hydrocarbons considered for this research except heptane have significantly higher auto-ignition temperatures than MDO (at 250˚C [69]) as shown in Table 3, making them safer in most hazardous situations encountered in the marine environment [48].

The use of hydrocarbons in ORC systems carries numerous similar hazards as liquefied gas except for the cryogenic state. The International Gas Code regulates liquefied gas fuel carried in bulk, however it prohibits its use as a fuel or in the vicinity of machinery spaces due to its low flash point, which has meant special Administration approval for LNG carriers intending to use LNG boil off as fuel [70]. IMO is currently developing the IGF [4] code, a code that has the short-term aim to regulate the use of LNG as fuel for carriers, and later amendments will incorporate other LFL fuels. The IGF is currently being reviewed before implementation [71]. In the near future ocean going ships fuelled partly or entirely by LNG will be built; plans are already on the drawing board. LNG fuelled ferries are already operating successfully in Norwegian coastal waters.

4.3.2 Making ORC safe

While no current code applies directly to the use of highly flammable fluids in an ORC, a parallel analysis of requirements from the IGF draft code [72] and the DNV GL Tentative Regulation for LFL Fuelled Ships [67] allows for an understanding of the necessary measures required to ensure a potential system's compliance with relevant regulations.

The system could be partly housed within the machinery space (exhaust path) should no oil loop be used, or kept separate by the thermal oil. The IGF code requires complete separation between the gas and the machinery space. This can be achieved by the use of double walled piping or an enclosure vented to atmosphere fitted with appropriate gas detection and purging capabilities. There is also a requirement to have an inherently safe space or use an emergency shutdown procedure that will power off all equipment not rated safe should the explosive vapour concentration reach 40% of the Lower Explosion Level (LEL). Double walled separation would not be possible within the boiler in a system not using a thermal loop. While the exhaust gas flow would prevent the accumulation of explosive vapours, an undetected leak could result in a hazardous situation following the shutdown of the plant.

Due to the volatile nature of LFLs, additional leak detection and firefighting systems are required. While a leak of hydrocarbon working fluid causes explosion risks, its higher volatility when compared to diesel fuel oil introduces an additional health hazard for personnel exposed to an undetected leak. A modern platform management and firefighting system with targeted foam and chemical extinguishing agents would ensure that the system meets all relevant safety requirements of the above codes. Today's platform management systems can monitor thousands of sensors making the incorporation of the additional sensors and firefighting systems easily feasible [73]. The use of thermal oil would restrict the working fluid to a discrete location making an enclosure fitted with safety systems ideal to ensure personnel safety.

Fitting an emergency relief and purging system as well as a cooling system in the form of fine water spray to the enclosure would also ensure system protection from other ship-borne emergencies.

5. Conclusions

This paper demonstrates, with the aid of a case study, the advantages of installing a WHRS on board an Aframax tanker navigating in the Baltic and North Sea achieving yearly fuel savings of around €154k and CO_2 emission reductions of around 705 metric tonnes, which achieves an EEDI reduction of around 1.9%. It also shows that a marine ORC WHRS, with thermal oil separating the waste heat and the working fluid, can outperform a simple steam cycle by producing up to 36 kW_e more at design point representing an increase in power output of around 12%. In conjunction with the Aframax's operating profile, it achieves up to 17% savings in both fuel consumption and CO_2 emissions. Furthermore, thanks to the

[4] International Code of Safety for Ships using Gases or other Low Flashpoint Fuels

organic fluids adaptability to lower heat quality and availability, it was seen that during the tanker's operating profile toluene WHRS could deliver better performance than the RC equivalent, despite a lower performance from the organic system at design conditions.
However, it is also important to keep in mind some of the drawbacks of the ORC marine systems, which requires mass flow rates and heat transfer areas of up to 8.6 times and 3.2 times larger than a RC system. This increase in size could have an impact in the initial cost of the ORC system and space requirement inside the vessel.
The flammability of hydrocarbons as organic fluids is also an important aspect of the ORC. While they have the significant advantage of having no ozone depleting potential (ODP) and very low GWP when compared to refrigerants, their flammability introduces additional safety equipment requirements. It was however found that the limited amount of fluid carried on board and the compactness of the system allowed for this additional risk to be easily mitigated, even in the case of a retrofit to an existing vessel.
Long standing policies regulating minimum flash point on board ships introduce a cultural challenge when introducing the technology. The main regulation for flammable fluids as we know it today dates from 1974 [74] and was written for ships and technology of that age. Since then requirements and priorities of the shipping industry have changed and technology and practices have greatly altered. Unmanned machinery spaces are now common and sensor and firefighting technology has evolved dramatically. In light of this, a regulation revision would be desired so that ships can take full advantage of ORC technology.

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Glossary

Auto-ignition temperature: Is the lowest temperature at which a fluid will auto-ignite and combust under normal atmospheric conditions without the help of any external source.

Decomposition temperature: Describes when the fluid starts to decompose chemically. If the conditions in the system exceed this temperature then the useful life of the fluid will be reduced [16].

Critical Point: Is the point where the conditions of a saturated liquid and a saturated vapour are the same [75]. When observing a temperature-entropy diagram is the maximum point in the saturated curve.

Flash point: Is the lowest temperature at which a liquid can vaporize to combust with the introduction of an ignition source [76].