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Article

Thermo-Economic Performance of an Organic Rankine Cycle System Recovering Waste Heat Onboard an Offshore Service Vessel

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Abstract: Recent regulatory developments in the global maritime industry have signalled an increased emphasis on the improvement of energy efficiency onboard ships. Among the various efficiency enhancement options, recovering waste heat using the organic Rankine cycle (ORC) has been studied and identified as a promising one in many earlier studies. In this paper, a marine application of ORC for waste heat recovery will be discussed by performing the first law thermodynamic analysis based on the operating profile and machinery design data of an offshore service vessel (OSV) and defining four standard cycle configurations that include simple, recuperated, dual heat source, and with intermediate heating. The use of five hydrocarbon working fluids that are suitable for shipboard usage comprising cyclopentane, n-heptane, n-octane, methanol and ethanol are examined. The economic analysis found that annual fuel saving between 5% and 9% is possible and estimated a specific installation cost of \$5000–8000 USD/kW. Among the various options, the methanol ORC in a simple cycle configuration is found to have the shortest payback time relatively balancing between annual fuel saving and total module cost. Finally, the simple cycle ORC running on methanol is further examined using the second law entropy generation analysis and it is found that the heat exchangers in the system accounted for nearly 95% of the overall entropy generation rate and further work is recommended to reduce this in the future.

Keywords: Organic Rankine cycle (ORC); marine application; thermo-economic analysis; entropy generation analysis; offshore service vessel

1. Introduction

Increasing awareness of the detrimental effects of climate change has led to recent regulatory developments in many countries and global industries like the maritime sector. In order to demonstrate that the shipping industry is taking steps to alleviate the situation, its global regulator, International Maritime Organisation (IMO) announced the initial strategy to cut shipping's total greenhouse gases (GHG) by at least 50% from 2008 levels by 2050 [1].

This seemingly ambitious target sent the shipping industry into deep reflection as the current preferred technology of switching over to liquefied natural gas (LNG) as fuel for propulsion could optimistically provide about 20% reduction in GHG emissions only. Hence, other measures will need to be considered holistically to meet the stated emission target imposed by IMO.

The international ship classification society, DNV GL summarised the three main options available to the shipping industry to cut its carbon footprint: energy efficiency, alternative fuels and speed reduction [2]. The same study estimated energy efficiency measures alone can lead to a 21–37% reduction in GHG emissions which can be extremely helpful to meet the strategic aim together with LNG fuel usage.

Waste heat recovery system (WHRS) is an essential aspect of improving overall energy efficiency onboard ships and the installation of a WHRS based on organic Rankine cycle (ORC) had been introduced in various research review papers [3–5] comparing ORC with other WHRS like steam Rankine cycle, Kalina cycle, power turbine systems, etc.

As shown on Figure 1, ORC is a thermodynamic cycle similar to a traditional steam Rankine cycle but uses an organic fluid like hydrocarbon or refrigerant as the working fluid. Compared to water, the selected organic fluid has lower boiling point with lower specific heat of vaporisation which makes it possible to exploit waste heat of lower temperatures sources like heated cooling water and lubricating oil thereby improving the overall energy efficiency of the power plant.

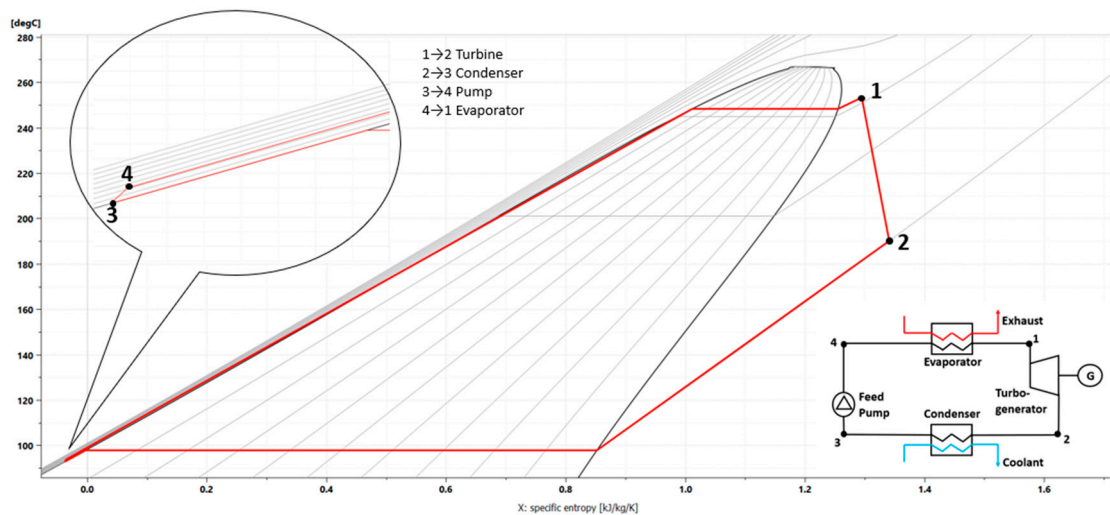


Figure 1. T-s diagram of a simple ORC with heptane (subcooled and superheated).

Over the past few years, ORC has attracted much attention and there are about six actual installations onboard ships under testing or operations stage as shown in Table 1:

Table 1. List of reported maritime applications of Organic Rankine Cycle plants.

Ship Name (Year)	MV Figaro (2012)	Viking Grace (2015)	Arnold Maersk (2016)	Asahi Maru (2017)	Orizzonte (2017)	Panerai I & II (2018)
Vessel type	PCTC	Cruise Ferry	Container	Bulk	Fishing Vessel	Fast ferry
ORC Maker	Opcon	Climeon	Calnetix	Kobe Steel	Enogia	Orcan Energy
Capacity	500 kW	150 kW	125 kW	125 kW	4.8 kW	154 kW
Expander type	Twin-screw	Turbine	Radial turbine	Semi-hermetic screw	Microturbine	Screw
Fuel savings	4–6%	Up to 5%	Up to 10–15%	3%	5%	6–9%
References	[6]	[7]	[8]	[9]	[10]	[11]

In addition to this list, there are several more marine ORC projects at the laboratory or earlier development phase [12–14].

The topic of ORC garnered consistent interest in maritime research over the recent years. An earlier example includes Choi and Kim [15] who had examined an ORC integrated with a trilateral cycle to recover waste heat from the exhaust from the main engines of a 6800 TEU (Twenty-foot Equivalent Unit) containership and found that the specific fuel oil consumption can be reduced by 6.06%. Larsen et al. [16] analysed 109 working fluids used in the simple and recuperated ORC cycle optimised by the principles of natural selection with varying heat source inlet temperature between

180 and 360 °C. Their results showed that dry fluids like hydrocarbons have a better thermodynamic performance compared to other wet and isentropic fluids in the recuperated process.

Yang and Yeh [17] studied six working fluids for a recuperated ORC with jacket cooling water from a marine engine and determined that R600a performed the best thermodynamically compared with the rest but this does not guarantee it performed as well as others in terms of the heat transfer process. Song et al. [18] applied an optimised ORC cycle where jacket cooling water is used in the preheating of cyclohexane as the working fluid followed by evaporating by the engine exhaust heat. They found that the optimised cycle has a lower net work output but this is outweighed by a more compact design and lower capital cost.

Several other ship specific ORC research were also presented in the literature, for example, Soffiato et al. [19] determined that a two-stage ORC provides the maximum net power output onboard an LNG carrier although it may lead to increased complexity in design and operations. Kalikatzarakis et al. [20] presented an ORC system techno-economic analysis for a 13,600 TEU containership with an optimized system that offers 2–3% of additional power and more than \$1 million in savings for the case ship, but indicated that this is dependent on operating profile of the main engine and price of fuel.

The more recent development in LNG fuel adoption by ships also led to novel cycles where the cold energy from LNG vaporisation is being used in addition to waste heat from the main engine. Tsougranis et al. [21] made a feasibility study on ORC using both thermal and cryogenic waste energy for a LNG passenger ferry and concluded that the two-stage ORC performs better and could save about 2.5% of the ferry's annual natural gas consumption. More recently, Baldasso et al. [22], Koo et al. [23], and Han et al. [24] made similar techno-economic studies for LNG-fuelled vessels with more advanced optimisation algorithms.

It is known for ORC operating in waste heat recovery applications, especially for the transportation sector like automotive and maritime, heat source temperature and mass flow rate varies with required loads on the internal combustion engine. Hence, ships' operational profiles will play a big role in the selection and design of ORC systems onboard ships. Lümmer et al. [25] performed a thermodynamic analysis for ORC on a short fast passenger ferry operating in Norway with an actual operating profile and found that the short transit time is insufficient to charge the batteries for full electric propulsion during manoeuvring in the ports. Mondejar et al. [26] in their comprehensive review of ORC in marine applications also looked at the engine load profile for containership, bulk carrier and oil tanker and they estimated that the yearly fuel savings could be 10–15%.

From the above works, it is clear that in order to encourage the adoption of more ORC onboard ships, techno-economic studies will be instrumental with the actual operational profile of a ship as a critical input. It is also observed that the application of ORC for an offshore service vessel (OSV) had not been examined in published literature. This is significant as OSV unlike other big cargo vessels like containerships or tankers covered in previous literature are smaller in size and have a very specialised role to support the offshore oil and gas industry and hence will not be operating at high engine load at most of the time.

This paper aims to present the feasibility of ORC application onboard OSVs in terms of thermo-economic performance by first using the first law of thermodynamics analysis, followed by an economic analysis to assess the techno-economic aspects and finally applying the second law of thermodynamic approach using entropy generation analysis for the best performing design. Four potential ORC configurations and five working fluids will be assessed and compared for their performance for onboard application. A one-dimensional, system simulation software, Siemens Simcenter Amesim [27] is used for this study to perform the thermodynamic analysis using the operating profile and design data of an OSV.

2. The Case Ship

Offshore service vessels (OSVs) refer to a class of ships that serves the offshore oil and gas industry and can include diverse operations like platform supply, anchor handling, construction, firefighting,

stand-by and rescue, diving support, accommodation, etc. Due to different operations, their load patterns can also differ widely unlike a conventional cargo vessel, e.g., containership or tanker.

As a case study to examine the thermodynamic performance of ORC, an 88.8m 5200 DWT multipurpose platform supply vessel (MPSV) which is designed to operate in offshore Malaysia with its main operations of carrying supplies like fuel oil, drilling material and chemicals to offshore rigs in the region. The general specifications of the case ship are as below in Table 2:

Table 2. General specifications of case ship.

Length Overall	88.8 m
Breadth	20.0 m
Depth	8.4 m
Deadweight	5200 ton
Class	American Bureau of Shipping +A1, (E) Offshore Support Vessel, Supply-HNLS, FFV 1, OSR-S1, +DPS-2, SPS, +AMS, HAB(WB), UWILD, CRC
Area of operations	Offshore Malaysia (seawater temperature 32 °C)

The MPSV is powered by four Wartsila main diesel generators with Wartsila electric propulsion & azimuthing stern thrusters to perform dynamic positioning operations (DynPos) in offshore locations in Malaysia. The main machinery specifications are as below in Table 3:

Table 3. Technical specifications of diesel generators.

Engine Make/Model	4 × Wartsila 6L26
Rated power	1950 kWm @ 900 rpm
No. of cylinders	6
Fuel consumption	184 g/kWh
Generator make/model	4 × AVK DSG 00 L1/8 W

2.1. Engine Loading

The operational profile of the MPSV will dictate the power consumption which will affect the diesel generator load and hence the temperature and quantity of the exhaust gas and cooling water.

The typical operation profile of a MPSV includes the following:

- loading of supplies from supply base at harbour mode,
- transit to offshore location at steaming mode
- unloading of supplies on DPS-2 mode
- loading of waste from rig on DPS-2 mode
- transit back to supply base at steaming mode
- unloading of waste to shore at harbour mode
- standby mode

The ship’s power requirement at each operational mode and the percentage of operating hours assuming 350 days of operating days per year is shown below in Figure 2.

In order to meet the operating profile shown above, the diesel generators can be operated with engine loads and numbers as shown in Table 4:

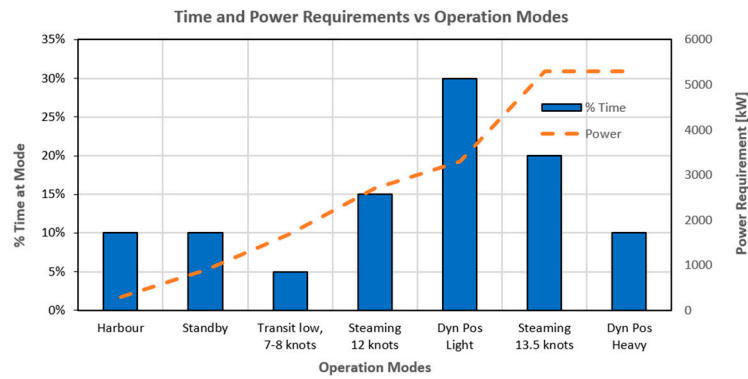


Figure 2. Operational time spent at each mode and resulting engine load.

Table 4. Number of operating engines and engine load for different operation modes.

Operation Modes	Steaming 13.5 Knots	Steaming 12 Knots	Transit Low, 7–8 Knots	DynPos Heavy	DynPos Light	Standby	Harbour
Number of engines	4	2	2	4	2	1	1
Engine load	75%	75%	50%	75%	85%	50%	25%

2.2. Engine Heat Sources

The possible heat sources from the diesel generators include exhaust gas, lubricating water, high temperature (HT) and low temperature (LT) cooling water etc. Values of the heat source temperature range as well as the flow rates can be derived from the engine maker information [28] and representative values are listed in Table 5.

Table 5. Engine heat source and flow rate.

Heat Source	Temperature Range Category	Temperature (°C)	Flow Rate (m ³ /h)
Exhaust gas	Medium	353–392	2.08–3.82 kg/s
Lub. oil	Low	66–67	60 m ³ /h
HT cooling water	Low	84–89	35 m ³ /h
LT cooling water	Low	42–57	42 m ³ /h

3. Thermodynamic Modelling of ORC Waste Heat Recovery System

3.1. Working Pressure and Temperature of ORC in Marine Applications

Application of ORC in the marine environment will impose some constraints on its design which will help to reduce the number of alternatives. For example, for the heat source using exhaust gas from diesel engines, its inlet temperature will typically range from 300–400 °C and the outlet temperature is typically kept at above 100 °C in order to reduce the effects of condensing corrosive sulphuric acids [18]. Lower exhaust outlet temperature can be accepted for MPSV as it burns low sulphur content marine gas oil (MGO) and there is no need for heating for any crude oil onboard, hence the exhaust outlet temperature will be limited to 150 °C.

For the heat sink onboard ships, seawater cooling for the condenser is used and this temperature varies base on the ship’s location. For a tropical region, seawater temperature can be taken as 32 °C and outlet temperature is set at 45 °C to prevent enhanced corrosion on seawater piping as per normal marine design.

The pinch point temperature or the smallest temperature difference between the two streams of fluid exchanging heat is set at 5 °C as a balance between performance and economy [25,29].

In terms of pressure limits for ORC system, the upper limit of evaporation pressure is selected at 30 barg for cost efficiency and a lower limit of condensation pressure of 1 barg is selected to prevent

atmospheric air to be sucked into the system. It is also recommended not to operate too close to the critical pressure and hence the evaporation pressure is limited to 30 barg or 80% of the working fluid’s critical pressure, whichever is lower.

Organic fluids also exhibit chemical instability when heated to higher temperatures and hence it is necessary to limit the maximum process temperature after heating at evaporator to 327 °C [30]. Considering the engine exhaust gas is at 300 °C, the thermal instability temperature of the organic fluids will not be reached.

For the safety aspects, working temperatures of organic fluids should not exceed its autoignition temperature which is the temperature at which the organic fluids will combust with contact with hot surfaces. The flashpoint of organic fluids should be kept in mind as the maritime regulatory regime will require additional safety features when handling fluids with low flashpoints. These limitations are especially relevant when hydrocarbons are selected as working fluids.

3.2. Selection of Working Fluids

To be suitable for installation onboard a ship, the working fluids need to meet the following requirements:

- Good performance given the heat sources from diesel engines and heat sink using seawater.
- Thermal stability at high temperature
- Safe to be used onboard, working fluid shall have low flammability and toxicity.
- Little impact on environment in terms of ozone depleting potential (ODP) and greenhouse warming potential (GWP).
- Availability and low cost.

Organic working fluids that meet the criteria above include hydrocarbons, refrigerants and some inorganic fluids. In terms of choosing between these three main classes, hydrocarbons will be advantageous due to their lower cost, ease of transport and familiarity with the marine industry. This is also highlighted by the research by de la Fuente [31] for the use of flammable organic fluids onboard ships.

Considering the above, a list of possible hydrocarbons as working fluids is compiled in Table 6 below.

Table 6. Main properties of selected organic fluids.

Fluid	Formula	-	T _{boiling} (°C)	T _{crit} (°C)	P _{crit} (bar)	GWP	ODP	Flash-Point (°C)	Autoignition Temperature (°C)
Cyclopentane	C ₅ H ₁₀	HC	49.25	238.55	45.1	Low	0	−37.2	361
n-Heptane	C ₇ H ₁₆	HC	98.43	267.05	27.4	Low	0	−4.1	203.9
n-Octane	C ₈ H ₁₈	HC	125.68	295.55	24.9	Low	0	12.9	205.9
Methanol	CH ₃ OH	HC	64.7	239.49	80.97	Low	0	10.9	463.9
Ethanol	C ₂ H ₅ OH	HC	78.29	240.77	61.48	Low	0	12.9	422.9

Source: [32].

3.3. ORC Cycle Configuration

Several ORC cycle configurations had been reviewed in the literature [31,33,34], among those discussed, the simple (sORC), recuperative (rORC), simple with dual heat source (sdhORC) and simple with intermediate heating (sORC/ih). These basic configurations can be considered as the most relevant for application of hydrocarbon working fluids in marine ORCs and can be used as a standard design module to reduce fabrication cost.

sORC is noted for their simplicity and low cost, while rORC with a recuperator between expander outlet and evaporator inlet improves the thermal efficiency. When heat source other than heat from the exhaust gas is considered like the HT cooling water from engine, sdhORC can be considered but the

complication lies in matching the temperature for the working fluid with the preheater and evaporator. As hydrocarbon working fluids that are flammable are used, the current marine classification rules require them to be located outside the main engine room, hence an intermediate heating circuit with a thermal oil with flashpoint higher than 60 °C may be used in sORC/ih cycle configuration. The various cycle configurations are shown in Figure 3:

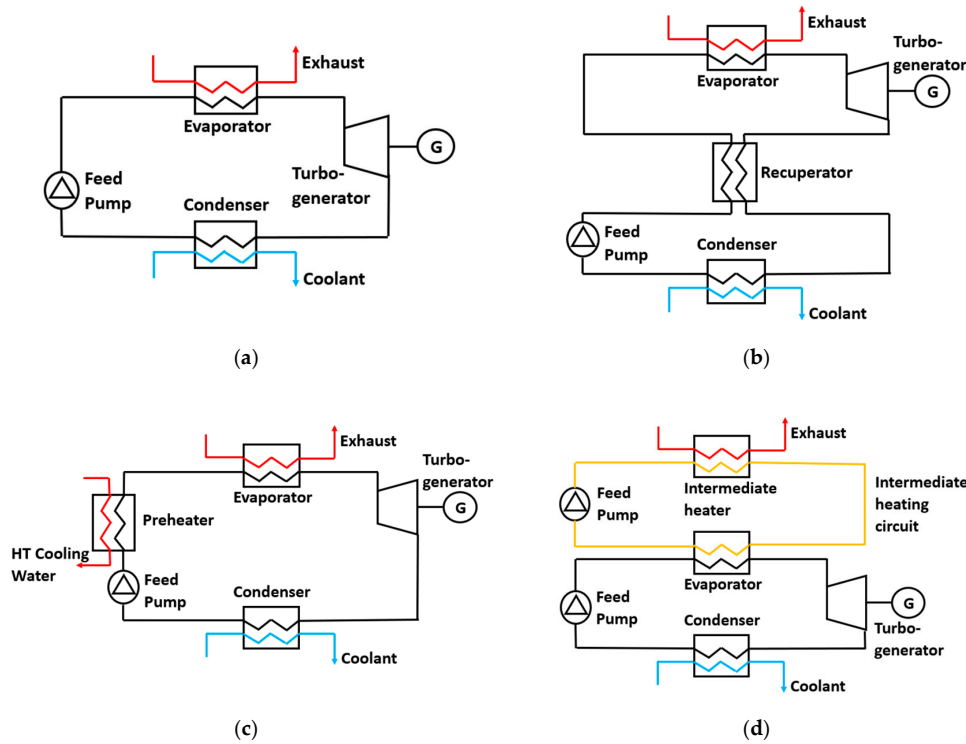


Figure 3. ORC basic cycle configurations: (a) Simple ORC (sORC); (b) Recuperative ORC (rORC); (c) Simple with dual heat source ORC (sdhORC); (d) Simple ORC with intermediate heating (sORC/ih).

3.4. Component Modelling

3.4.1. Modelling of Heat Exchangers

Heat exchangers include evaporators, condensers and preheaters based on the cycle configuration selected. As the purpose of the analysis is to identify the optimum ORC systems, the global heat transfer analysis method of using the effectiveness-number of transfer units (ϵ -NTU) method. The basic equations related to this method as explained in [35] are listed below:

$$\epsilon = \frac{\dot{Q}_{\text{actual}}}{\dot{Q}_{\text{max}}} \tag{1}$$

$$\dot{Q}_{\text{actual}} = \epsilon C_{\text{min}}(T_{h,i} - T_{c,i}) \tag{2}$$

$$NTU = \frac{UA}{C_{\text{min}}} \tag{3}$$

For the evaporators and condensers that involves phase change, the following effectiveness relationship is used:

$$\epsilon = 1 - e^{-NTU} \tag{4}$$

For the preheaters, the following will be used instead to model a shell-and-tube heat exchanger with one shell pass and two tube passes:

$$\varepsilon = 2 \left\{ 1 + C_r + (1 + C_r^2)^{\frac{1}{2}} \frac{1 + \exp \left[-(NTU)(1 + C_r^2)^{\frac{1}{2}} \right]}{1 - \exp \left[-(NTU)(1 + C_r^2)^{\frac{1}{2}} \right]} \right\}^{-1} \quad (5)$$

$$C_r = \frac{C_{\min}}{C_{\max}} \quad (6)$$

By definition, NTU is the ratio of the overall thermal conductance to the smaller heat capacity rate and is a non-dimensional heat transfer size or thermal size of a heat exchanger. Effectiveness of the heat exchanger or the ratio of the actual heat transferred to the maximum heat transferred in an ideal heat exchanger is affected by NTU value. According to Shah and Sekulić [36], when the cost is an important consideration, heat exchangers are designed with $NTU \leq 2$ or $\varepsilon \leq 60\%$.

The overall heat transfer coefficient is assumed based on representative values of a similar heat transfer phenomenon from Mills [37]. The area of heat transfer, A is then calculated from Equation (3).

3.4.2. Modelling of Fluid Machinery

Fluid machinery includes expanders and feed pumps and are modelled with efficiency characteristics of fixed positive displacement machines base on the following formula (points refer to Figure 1):

For expanders:

$$\eta_{is} = \frac{(h_2 - h_1)}{(h_{2s} - h_{1s})} \quad (7)$$

For feed pumps:

$$\eta_{is} = \frac{(h_{4s} - h_{3s})}{(h_4 - h_3)} \quad (8)$$

For both expanders and feed pumps:

$$\eta_{\text{mech}} = \frac{\dot{W}_{\text{mech}}}{\dot{W}_{\text{fluid}}} \quad (9)$$

$$\eta_{\text{elec}} = \frac{\dot{W}_{\text{elec}}}{\dot{W}_{\text{mech}}} \quad (10)$$

For this study, the volumetric, isentropic and mechanical efficiencies are assumed to be 80%, which is the same as recent prior research by Koo et al. [23] and Han et al. [24], while the electrical efficiency is at 95% and is close to that used by Akman and Ergin [29] and Han et al. [24].

3.4.3. Net Work and Thermal Efficiency

To assess the thermodynamic performance of the ORC system, net work and thermal efficiency is calculated as below:

$$\dot{W}_{\text{net}} = \dot{W}_{\text{exp}} - \sum \dot{W}_{\text{pumps}} \quad (11)$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} \quad (12)$$

4. Entropy Generation Modelling

The thermodynamic analysis presented so far has been based on the first law of thermodynamics which is primarily based on the conservation of energy. However, in order to further improve and optimise the thermodynamic performance of engineering systems, the second law of thermodynamics approaches like the entropy generation analysis can be employed [38]. Using this approach, components contributing to high irreversibilities can be identified in a system and efforts made to optimise their performance.

Based on equations developed in [39], for the control volume in steady-state conditions, the rate of work production, \dot{W} , maximum rate of work production under reversible conditions, \dot{W}_{rev} , the lost work production rate, \dot{W}_{lost} can be derived from the following from the first law of thermodynamics:

$$\dot{W} = \sum_{in} \dot{m} \left(h + \frac{1}{2} V^2 + gz \right) - \sum_{out} \dot{m} \left(h + \frac{1}{2} V^2 + gz \right) + \dot{Q} \quad (13)$$

$$\dot{W}_{rev} = \sum_{in} \dot{m} \left(h + \frac{1}{2} V^2 + gz - T_0 s \right) - \sum_{out} \dot{m} \left(h + \frac{1}{2} V^2 + gz - T_0 s \right) - \frac{\partial}{\partial x} (E - T_0 S) \quad (14)$$

$$\dot{W}_{lost} = \dot{W}_{rev} - \dot{W} \quad (15)$$

$$\dot{W}_{lost} = T_0 \left(-\frac{\dot{Q}}{T_0} - \sum_{in} \dot{m} s + \sum_{out} \dot{m} s \right) \quad (16)$$

Based on the second law of thermodynamics, entropy generation rate, $\dot{S}_{gen} \geq 0$ and defined as:

$$\dot{S}_{gen} = -\frac{\dot{Q}}{T_0} - \sum_{in} \dot{m} s + \sum_{out} \dot{m} s \quad (17)$$

$$\dot{W}_{lost} = T_0 \dot{S}_{gen} \quad (18)$$

This equation is referred to as the Gouy-Stodola theorem and describes the proportionality between the lost available work against entropy generation in that a system with lower entropy generation will lead to lower lost work and therefore more work generation.

When analysing the entropy generation rate for the whole system, this can be obtained by summing up the entropy generation by each individual component:

$$S_{gen} = \sum_{Components} S_{gen} \quad (19)$$

4.1. Entropy Generation Modelling for Heat Exchangers

From Equation (17) above, the entropy generation rate for heat exchangers assuming zero heat lost at the exchanger boundary can be derived as below:

$$\dot{S}_{gen} = \left[-\frac{\dot{Q}_h}{T_{avg,h}} + \dot{m}_h (s_{h,o} - s_{h,i}) \right] - \left[-\frac{\dot{Q}_c}{T_{avg,c}} + \dot{m}_c (s_{c,o} - s_{c,i}) \right] \quad (20)$$

4.2. Entropy Modelling for Fluid Machinery

From Equation (17) above, the entropy generation rate for fluid machinery assuming zero heat lost can be found by the following:

$$\dot{S}_{gen} = \dot{m}_{FM} (s_{FM,o} - s_{FM,i}) \quad (21)$$

5. Economic Modelling

Module cost of ORC will give added dimension for the assessment of different ORC designs for onboard applications. The cost modelling procedure described by Turton et al. [40] for the chemical and process industry will be used as a basis with the intention of comparing the relative cost of installation for different designs. Only direct and indirect capital costs related to the ORC module have been estimated, other costs like contingency fees, auxiliary facilities, operational and maintenance have been excluded.

For each option, the specific investment cost (SIC) in \$/kW can be derived as below:

$$SIC = \frac{C_{TM}}{W_{net}} \quad (22)$$

where C_{TM} is the total module capital cost of the ORC power system and W_{net} is the net work output from ORC system

To account for inflation for C_{TM} between the years 2001 and 2018, the chemical engineering plant cost index (CEPCI) is used as below:

$$C_{TM} = \frac{CEPCI(2018)}{CEPCI(2001)} C_{TM(2001)} \quad (23)$$

where CEPCI for year 2018 and 2001 is 603.1 [41] and 397 [40] respectively.

The total module cost for ORC can be derived from summing the various major components—heat exchanger, expander, pump bare module costs, C_{BM} as follows:

$$C_{TM} = \sum_{i=1}^n C_{BM,i} \quad (24)$$

The bare module cost, C_{BM} for heat exchangers and pumps is given as below:

$$C_{BM} = C_p^0 F_{BM} = C_p^0 (B_1 + B_2 F_M F_P) \quad (25)$$

where C_p^0 is the purchased equipment cost and calculated from the following:

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10}(A) + K_3 [\log_{10}(A)]^2 \quad (26)$$

where A is the area of heat transfer in m^2 of the heat exchanger or work input in kW for pump, F_{BM} is the bare module cost factor and is equal to $B_1 + B_2 F_M F_P$, F_M is the material factor, and F_P is the pressure factor calculated from following:

$$\log_{10} F_P = C_1 + C_2 \log_{10} P + C_3 (\log_{10} P)^2 \quad (27)$$

where P is the gauge pressure in barg.

For heat exchangers:

$K_1 = 4.3247$; $K_2 = -0.3030$; $K_3 = 0.1634$ for fixed tube $B_1 = 1.63$; $B_2 = 1.66$ for heat exchangers fabricated using fixed tube sheets F_M is 1.0 for carbon steel construction $C_1 = 0.03881$, $C_2 = -0.11272$, $C_3 = 0.08183$ for fixed tube sheet heat exchanger with pressure rating between 5 and 140 barg

For pumps:

$K_1 = 3.4771$; $K_2 = 0.1350$; $K_3 = 0.1438$ for positive displacement pumps $B_1 = 1.89$; $B_2 = 1.35$ $F_M = 1.4$ for carbon steel positive displacement pump $C_1 = -0.245382$, $C_2 = 0.259016$, $C_3 = -0.01363$ for positive displacement pump with pressure rating between 10 and 100 barg

The expander is assumed to be constructed with carbon steel in an axial flow arrangement. The bare module cost, C_{BM} has a simpler relation:

$$C_{BM} = C_p^0 F_{M2} \quad (28)$$

where C_p^0 is the purchase equipment cost calculated by the following:

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10}(W_{out}) + K_3 [\log_{10}(W_{out})]^2 \quad (29)$$

where $K_1 = 2.7051$; $K_2 = 1.4398$; $K_3 = -0.1776$ for W_{out} between 100 and 4000 kW, W_{out} is the work output by turbine in kW, and F_{M2} is the material factor for carbon steel of 3.5.

Shipowners will be interested in the benefits that can be generated by investing in an ORC system. The way to measure this will be the annual fuel savings calculated as follows taking into account the time spent at each engine load annually:

$$\text{Annual fuel savings} = \sum_{\text{load}=25, 50,75, 85,100\%} \dot{W}_{\text{net}} \times t_{\text{load}} \times \text{SFOC} \times \text{FC} \quad (30)$$

It is assumed that the additional power produced by ORC will reduce the power to be generated by the main diesel generator, the specific fuel oil consumption (SFOC) of the diesel generators will be used to calculate the tons of marine gas oil (MGO) that can be saved. The fuel cost per ton of MGO is assumed to be \$610/ton.

The payback time for the installation of ORC can be estimated by the following:

$$\text{Payback time} = \frac{C_{\text{TM}}}{\text{Annual fuel savings}} \quad (31)$$

6. Results and Discussions

6.1. Net Work and Thermal Efficiency

The net work, W_{net} is plotted against the engine loads for different working fluids as shown in Figure 4. As more waste heat is available at higher engine loads, the net work increases with engine load with the highest net work being consistently recorded for cyclopentane at 166 kW and 164 kW for sdhORC and rORC respectively for 100% engine load of 1950 kW. Other working fluids have also performed better both in rORC and sdhORC except for methanol and ethanol which have experienced negligible effects in rORC. This is due to both fluids are wet fluids meaning that lesser heat can be transferred to the recuperator due to the slope of its saturated vapour line and consequently leading to a smaller increase in working fluid mass flow rate and a lower increase in net work.

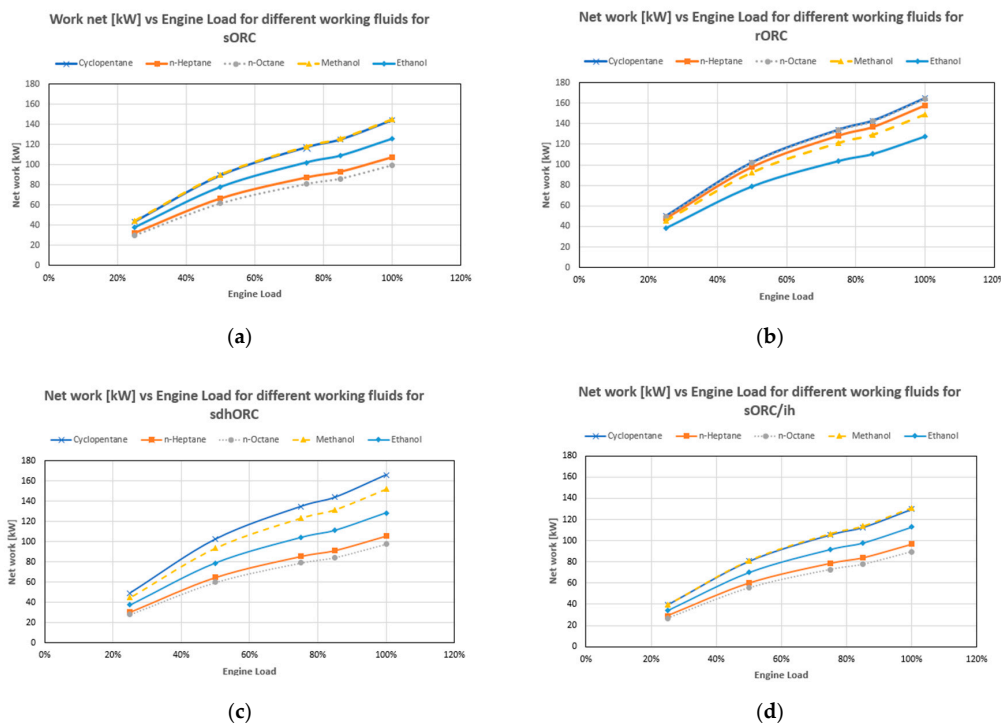


Figure 4. Net Work against Engine Load for different working fluids (a) sORC; (b) rORC; (c) sdhORC; (d) sORC/ih.

The thermal efficiency, η_{th} is plotted against engine loads for different working fluids as shown in Figure 5. Generally, thermal efficiency does not vary a lot with increasing engine load, this is because the additional net work is derived from the higher heat input from the engine.

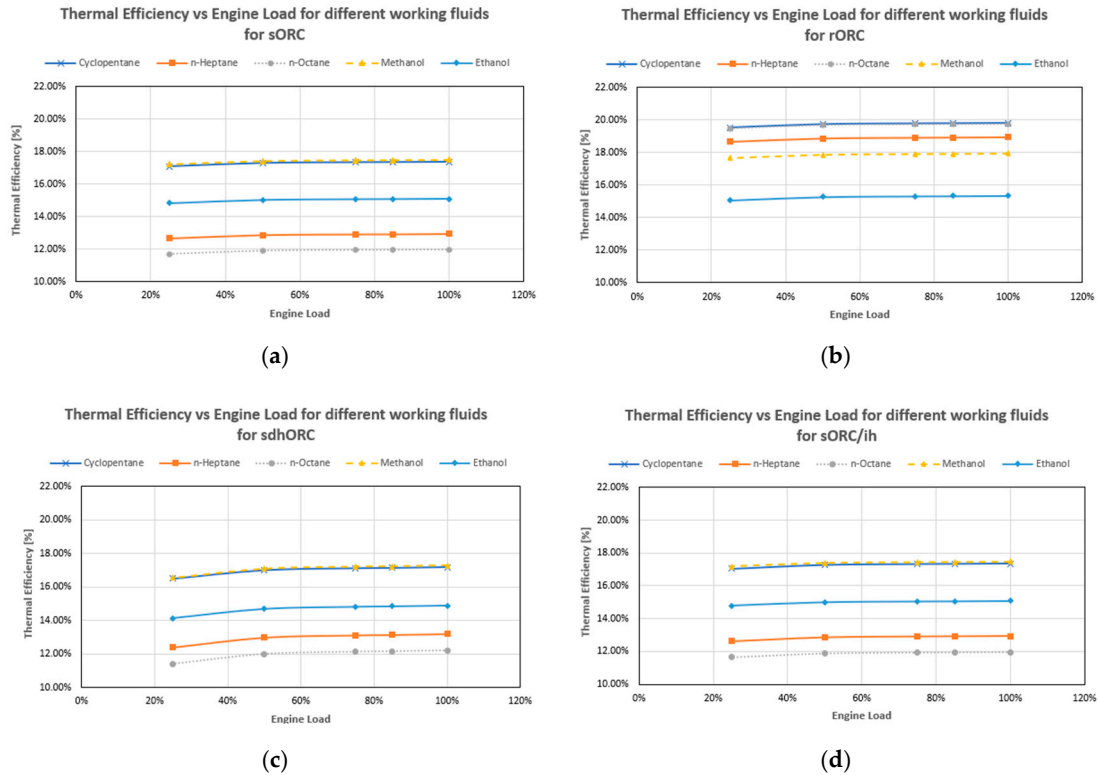


Figure 5. Thermal Efficiency vs Engine Load for different working fluids (a) sORC; (b) rORC; (c) sdhORC; (d) sORC/ih.

The rORC improves the thermal efficiency as compared with the sORC for cyclopentane, n-heptane and n-octane while for methanol and ethanol there is no improvement for rORC over sORC with similar reasons as for net work above.

The sdhORC performed with lower thermal efficiency compared with sORC as the heat input, Q_{in} has to account for the heat input from preheater in addition to evaporator. For sORC/ih, the thermal efficiency obtained is almost the same.

6.2. ORC Module Costing

The bare module costs, C_{BM} of the various components of ORC are illustrated in Figure 6 and compared across the different working fluids. It can be observed that the major cost component is due to the expander that can take up between 60% and 70% of the total module cost for ORC. Next, the heat exchangers: evaporator, condenser, preheater, intermediate heater take up about 10% each of total cost. The fluid pumps have lesser impact on total cost, generally accounting for less than 5% only.

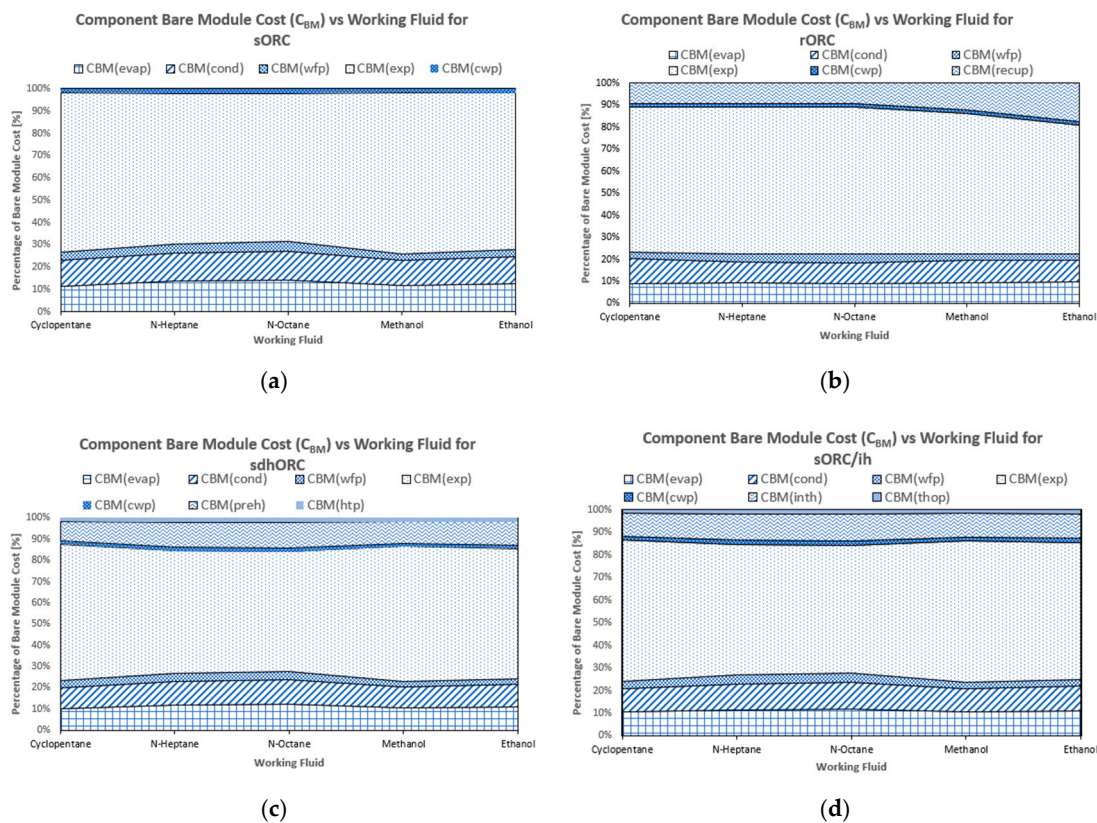


Figure 6. Percentage of total cost for ORC components (a) sORC; (b) rORC; (c) sdhORC; (d) sORC/ih.

This indicated that any cost optimisation for ORC will depend largely on the expander followed by heat exchangers.

It is interesting to find out how the different cycle configurations and working fluids will affect the total module cost, C_{TM} of ORC system. For the sake of comparing the relative total cost, the C_{TM} is plotted for different systems in Figure 7. It can be seen that sORC followed by sORC/ih and sdhORC provides the lowest C_{TM} , with the rORC being the most expensive system. The price difference between the cheapest and most expensive can be around 20%.

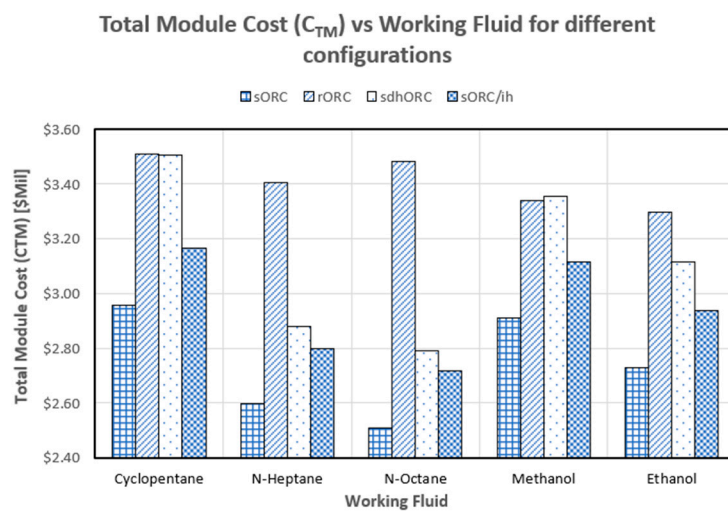


Figure 7. Total Module Cost vs working fluids and cycle configurations.

6.3. Annual Fuel Savings

Annual fuel saving is an indicator for return on investment for shipowners to adopt ORC. As seen on Figure 8, rORC will provide the highest annual fuel savings of around \$300,000 USD which is about 9% of the total fuel cost. Among the working fluids, cyclopentane and methanol are the best performers.

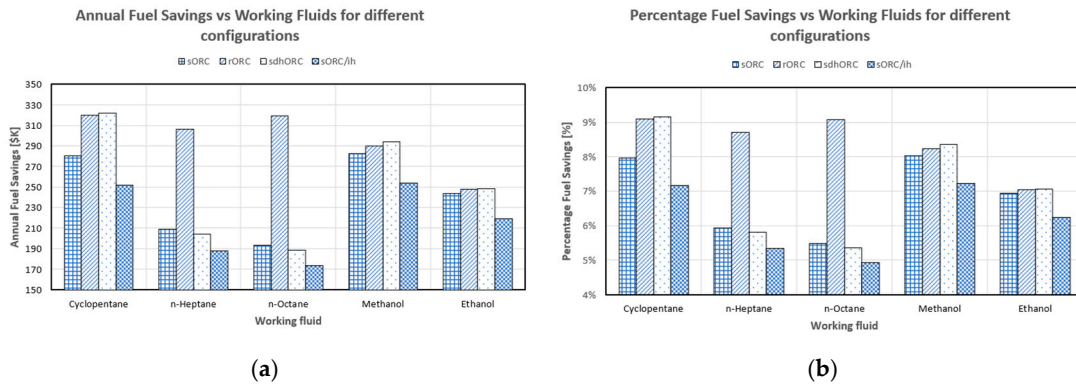


Figure 8. Annual Fuel Savings for different working fluids and cycle configurations (a) In Thousand dollars (\$) (b) Percentage fuel savings.

As the diesel engines of the case ship varies throughout its operations, the annual fuel savings that can be attributed to each engine load for various working fluids for sORC is shown in Figure 9. It can be seen that largest engine load contribution to annual savings comes from 75% load at around 60% followed by 85% load making up about 30% of total savings.

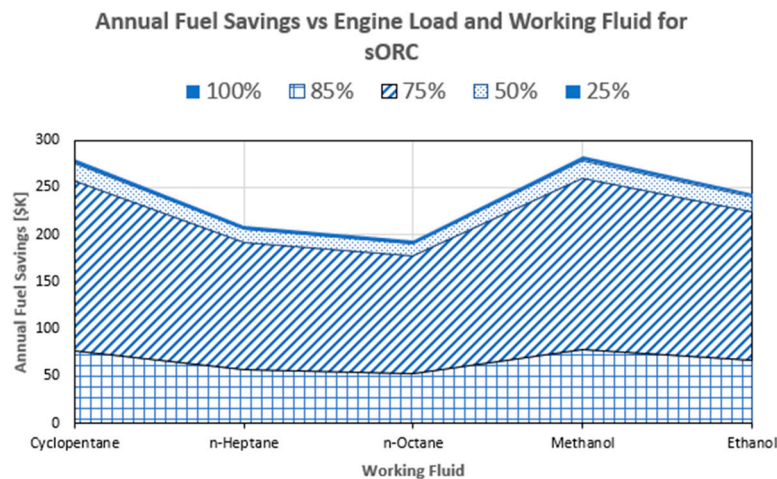


Figure 9. Annual fuel savings for different engine loads and working fluids.

6.4. Payback Time and Specific Installation Cost

Payback time is an indication of how fast the benefits of a capital expenditure can be recovered. With an assumed fuel cost of \$610/ton of MGO, the payback time of the investment is estimated to be as shown in Figure 10. Noting that the price of MGO had fluctuations between -40% and 20% over the past five years, payback time would be similarly affected. Hence, it is necessary to look at the relative payback time of different designs instead of the absolute figures.

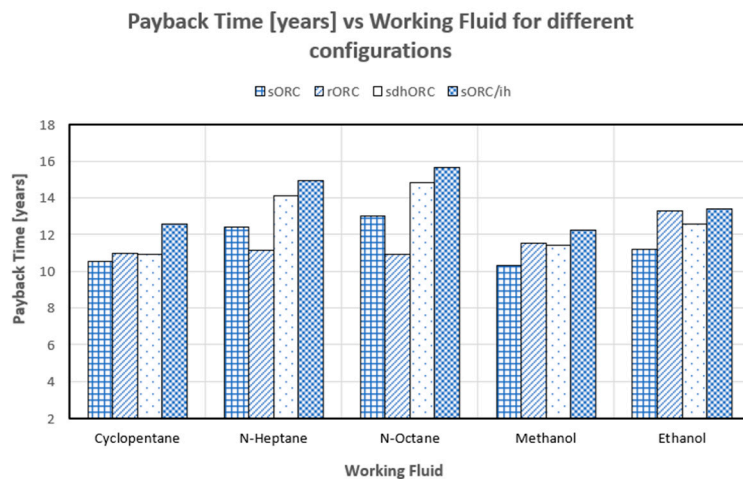


Figure 10. Payback time for ORC of different working fluids and cycle configurations.

From Figure 10, it can be seen that the design that provides the lowest payback time is the methanol sORC that has about 30% less payback time than the option with the longest payback time.

Another metric of interest is the specific installation cost (SIC) as shown in Figure 11 with the corresponding co-relations for SIC for the four cycle configurations and five hydrocarbon working fluids. When compared to earlier literature [42], the SIC found is higher and could be attributed to the numbers of years since the research was conducted and buffer being incorporated into the method by Turton et al. [40]. sORC still offers the lowest SIC among the different cycle configurations.

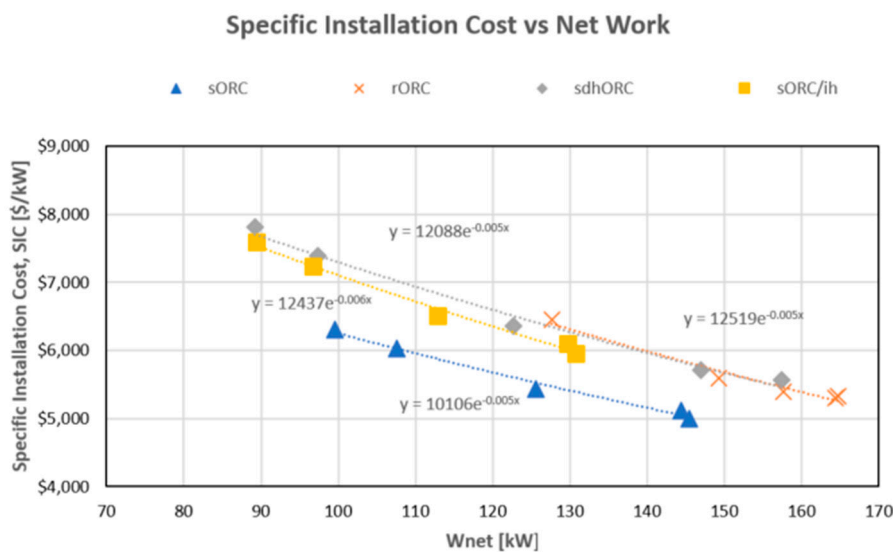


Figure 11. Specific installation cost for different cycle configuration.

6.5. Entropy Generation Analysis

As identified earlier, the methanol sORC has the fastest payback time and hence its rate of entropy generation is shown in Figure 12. It can be seen that evaporator and condenser contributed to the largest portion of the overall entropy generation rate at about 95%. This indicates the importance of properly designing heat exchangers for specific conditions to maximise work output in order to reduce entropy generation so that power output is optimised.

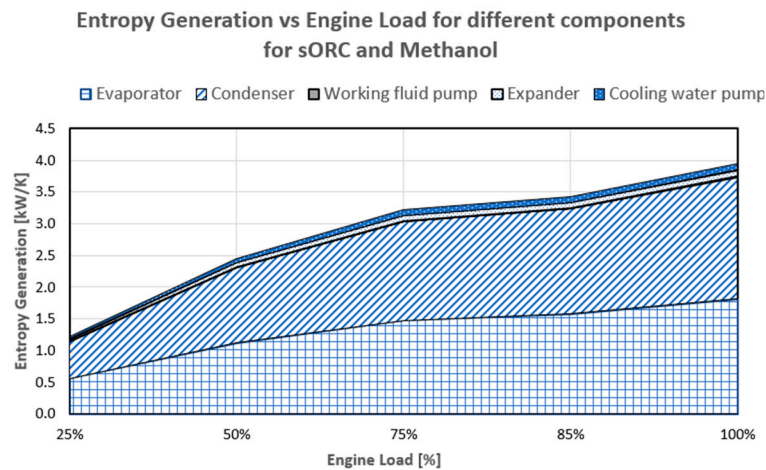


Figure 12. Entropy generation rate vs engine load for methanol sORC.

7. Conclusions

As part of the marine industry's drive to reduce the carbon footprints of the world's fleet, improving the energy efficiency of ships is one of the main options. Incorporating a modern advanced energy system onboard like organic Rankine cycle (ORC) as a waste heat recovery system (WHRS) for diesel engines will be interesting in the short to medium term while transiting to a zero-carbon fuel like hydrogen in the long term.

This paper starts off by looking at the current actual installations of ORC onboard ships and then develops a thermodynamic model for four potential cycle configurations and five hydrocarbon working fluids in a commercial-of-the-shelf system simulation software, Siemens Simcenter Amesim.

The operating profile and machinery design of a multi-purpose platform service vessel (MPSV) engaged in the offshore oil & gas industry in Southeast Asia is used as a case ship to understand the potential of installing an ORC system onboard ship.

The results from the thermodynamic analysis showed that net work output of about 160 kW is achievable for a diesel engine with rated output of 1950 kW with the thermal efficiency of 17–20%. Cyclopentane and methanol as working fluids and the recuperated ORC (rORC) configuration performed well.

The economic analysis showed that the simple ORC (sORC) and simple ORC with intermediate heating (sORC/ih) cycle configurations yield the lowest capital costs excluding contingency fees, auxiliary facilities, operation and maintenance costs. Among the components, the cost for expander is found to account for the majority for the total module cost. This means that additional considerations need to be made to refine the design and subsequent costing of expanders.

Annual fuel saving of 5–9% is estimated for various designs with the engine loads, 75% and 85% making up about 90% of the total annual fuel savings. The payback time for investment is derived for various designs and the best is the methanol in the sORC cycle configuration. The specific installation cost (SIC) is derived for different cycle configurations and found to be higher than that found in previous literature. This is attributed to the time that had elapsed since their publication and also buffers being used in the cost model, particularly in the expander costing.

Lastly, the second law of thermodynamics based entropy generation analysis is performed for the best performing design for a methanol sORC and revealed that majority of the entropy is generated in the heat exchangers: evaporator and condenser and more effort should be focused on optimising their designs.

In conclusion, from this research, the application of ORC in marine applications can be aided by the following: proper selection of cycle configuration and working fluid, expander design and costing to reduce cost, heat exchanger design to reduce entropy generation. In particular, detailed marine heat exchanger models will be developed to address the second observation.

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Nomenclature

ε	Effectiveness
\dot{Q}	Heat flow rate
C	Heat capacity rate
T	Temperature
U	Overall heat transfer coefficient
A	Area of heat transfer
η	Efficiency
h	Specific enthalpy
\dot{W}	Work rate
V	Velocity
\dot{m}	Mass flow rate
z	Height
s	Specific entropy
\dot{S}_{gen}	Entropy generation rate
C_{TM}	Total module cost
C_{BM}	Bare module cost
t_{load}	Time spent at engine load in the year

Abbreviations

h	hot
i	in
c	cold
min	minimum
max	maximum
is	isentropic
mech	mechanical
elec	electrical
th	thermal
rev	reversible
gen	generation
FM	fluid machine
avg	average
SIC	Specific Installation Cost
CEPCI	Chemical Engineering Plant Cost Index
1	expander inlet
2	expander outlet
2s	expander outlet (isentropic)
3	working fluid pump inlet
4	working fluid pump outlet
4s	working fluid pump outlet (isentropic)
SFOC	Specific fuel oil consumption
FC	Fuel cost per ton of MGO

References

1. IMO UN Body Adopts Climate Change Strategy for Shipping. Available online: <http://www.imo.org/en/MediaCentre/PressBriefings/Pages/06GHGinitialstrategy.aspx> (accessed on 27 April 2019).
2. DNV GL Low Carbon Shipping Towards 2050. Available online: <https://www.dnvgl.com/publications/low-carbon-shipping-towards-2050-93579> (accessed on 27 April 2020).
3. Shu, G.; Liang, Y.; Wei, H.; Tian, H.; Zhao, J.; Liu, L. A review of waste heat recovery on two-stroke IC engine aboard ships. *Renew. Sustain. Energy Rev.* **2013**, *19*, 385–401. [CrossRef]
4. Singh, D.V.; Pedersen, E. A review of waste heat recovery technologies for maritime applications. *Energy Convers. Manag.* **2016**, *111*, 315–328. [CrossRef]
5. Suárez de la Fuente, S. Marine Waste Heat Recovery System. *Encycl. Marit. Offshore Eng.* **2017**, 1–24. [CrossRef]
6. Opcon Energy Systems AB. *Commissioning and Testing of First Reference Installation of Opcon Technology for Ships*; Opcon Energy Systems AB: Stockholm, Sweden, 2012.
7. Viking Line Viking Grace First Ship to Install New Swedish Heat Recovery System. Available online: https://www.vikingline.com/globalassets/documents/market_specific/corporate/press/pressrelease-eng/2014/20141201-climeon-eng.pdf (accessed on 14 June 2019).
8. Marinelog Maersk Vessel First with New Waste Heat Recovery System—Marine Log. Available online: http://www.marinelog.com/index.php?option=com_k2&view=item&id=10842%253Amaersk-vessel-first-with-new-waste-heat-recovery-system&Itemid=231 (accessed on 4 December 2016).
9. Kobelco Binary Cycle Power Generation System for Ships Completes Sea Trials, Kobe Steel to Begin Sales of the New System in 2019. Available online: http://www.kobelco.co.jp/english/releases/1196609_15581.html (accessed on 23 June 2018).
10. Enogia EfficientShip. Available online: <http://www.efficientship.eu/> (accessed on 14 June 2019).
11. Orcan Energy AG Solutions—Applications. Available online: <https://www.orcan-energy.com/en/applications-marine.html> (accessed on 14 June 2019).
12. NTUA Marine ORC: A Fully Automated WHR ORC Prototype for Marine Applications. Available online: http://www.lsbtp.mech.ntua.gr/MarineORC_Video (accessed on 24 October 2019).
13. Bunkerports News Up to 8% Marine Fuel Savings with New Waste Heat Recovery System. Available online: <http://www.bunkerportsnews.com/News.aspx?ElementId=2892382a-518b-4183-8bb8-1a21050bb4a9> (accessed on 24 October 2019).
14. Peña, X.; Gabiña, G.; Aldecoa, S.; Ramirez, M.; Uriondo, Z. Modelling and design of a R1233ZD(E) multiexpander ORC prototype for WHR application in marine vessels. In Proceedings of the 5th Engine ORC Consortium Workshop, Lyon, France, 26–28 September 2018.
15. Choi, B.C.; Kim, Y.M. Thermodynamic analysis of a dual loop heat recovery system with trilateral cycle applied to exhaust gases of internal combustion engine for propulsion of the 6800 TEU container ship. *Energy* **2013**, *58*, 404–416. [CrossRef]
16. Larsen, U.; Pierobon, L.; Haglind, F.; Gabriellii, C. Design and optimisation of organic Rankine cycles for waste heat recovery in marine applications using the principles of natural selection. *Energy* **2013**, *55*, 803–812. [CrossRef]
17. Yang, M.H.; Yeh, R.H. Analyzing the optimization of an organic Rankine cycle system for recovering waste heat from a large marine engine containing a cooling water system. *Energy Convers. Manag.* **2014**, *88*, 999–1010. [CrossRef]
18. Song, J.; Song, Y.; Gu, C. Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines. *Energy* **2015**, *82*, 976–985. [CrossRef]
19. Soffiato, M.; Frangopoulos, C.A.; Manente, G.; Rech, S.; Lazzaretto, A. Design optimization of ORC systems for waste heat recovery on board a LNG carrier. *Energy Convers. Manag.* **2015**, *92*, 523–534. [CrossRef]
20. Kalikatzarakis, M.; Frangopoulos, C.A. Thermo-economic optimization of synthesis, design and operation of a marine organic Rankine cycle system. *Proc. Inst. Mech. Eng. Part M J. Eng. Marit. Environ.* **2016**. [CrossRef]
21. Tsougranis, E.-L.; Wu, D. A feasibility study of Organic Rankine Cycle (ORC) power generation using thermal and cryogenic waste energy on board an LNG passenger vessel. *Int. J. Energy Res.* **2018**. [CrossRef]
22. Baldasso, E.; Graa, J.; Mondejar, M.E.; Larsen, U.; Haglind, F. Technical and economic feasibility of organic Rankine cycle-based waste heat recovery systems on feeder ships: Impact of nitrogen oxides emission abatement technologies. *Energy Convers. Manag.* **2019**, *183*, 577–589. [CrossRef]

23. Koo, J.; Oh, S.R.; Choi, Y.U.; Jung, J.H.; Park, K. Optimization of an organic Rankine cycle system for an LNG-powered ship. *Energies* **2019**, *12*, 1933. [CrossRef]
24. Han, F.; Wang, Z.; Ji, Y.; Li, W.; Sundén, B. Energy analysis and multi-objective optimization of waste heat and cold energy recovery process in LNG-fueled vessels based on a triple organic Rankine cycle. *Energy Convers. Manag.* **2019**, *195*, 561–572. [CrossRef]
25. Lümmer, N.; Nygård, E.; Koch, P.E.; Nerheim, L.M. Comparison of organic Rankine cycle concepts for recovering waste heat in a hybrid powertrain on a fast passenger ferry. *Energy Convers. Manag.* **2018**, *163*, 371–383. [CrossRef]
26. Mondejar, M.E.; Andreasen, J.G.; Pierobon, L.; Larsen, U.; Thern, M.; Haglind, F. A review on the use of organic Rankine cycle power systems for marine applications. *Renew. Sustain. Energy Rev.* **2018**, *91*, 126–151. [CrossRef]
27. Siemens Industry Software NV Simcenter Amesim 2019. Available online: <https://www.plm.automation.siemens.com/global/en/products/simcenter/simcenter-amesim.html> (accessed on 30 April 2020).
28. Wärtsilä Marine Solutions Wärtsilä 26 Product Guide; Vaasa. 2018. Available online: https://www.wartsila.com/docs/default-source/product-files/engines/ms-engine/product-guide-o-e-w26.pdf?utm_source=engines&utm_medium=dieseleengines&utm_term=w26&utm_content=productguide&utm_campaign=msleadscoring (accessed on 30 April 2020).
29. Akman, M.; Ergin, S. An investigation of marine waste heat recovery system based on organic Rankine cycle under various engine operating conditions. *Proc. Inst. Mech. Eng. Part M J. Eng. Marit. Environ.* **2019**, *233*, 586–601. [CrossRef]
30. Drescher, U.; Bruggemann, D. Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants. *Appl. Therm. Eng.* **2007**, *27*, 223–228. [CrossRef]
31. Suarez de la Fuente, S. Reducing Shipping Carbon Emissions under Real Operative Conditions: A Study of Alternative Marine Waste Heat Recovery Systems based on the Organic Rankine Cycle. Ph.D. Thesis, University College London, London, UK, 2016.
32. Yaws, C.L. *Yaws' Handbook of Thermodynamic and Physical Properties of Chemical Compounds Physical, Thermodynamic and Transport Properties for 5000 Organic Chemical Compounds*; Knovel: Norwich, NY, USA, 2003.
33. Lecompte, S.; Huisseune, H.; Van Den Broek, M.; Vanslambrouck, B.; De Paepe, M. Review of organic Rankine cycle (ORC) architectures for waste heat recovery. *Renew. Sustain. Energy Rev.* **2015**, *47*, 448–461. [CrossRef]
34. Shi, L.; Shu, G.; Tian, H.; Deng, S. A review of modified Organic Rankine cycles (ORCs) for internal combustion engine waste heat recovery (ICE-WHR) Organic Rankine cycle. *Renew. Sustain. Energy Rev.* **2018**, *92*, 95–110. [CrossRef]
35. Incropera, F.P.; Dewitt, D.P.; Bergman, T.L.; Lavine, A.S. *Fundamentals of Heat and Mass Transfer*, 6th ed.; John Wiley & Sons (Asia) Pte Ltd.: Singapore, 2007.
36. Shah, R.K.; Sekulić, D.P. *Fundamentals of Heat Exchanger Design*; John Wiley & Sons, Inc.: Hoboken, NJ, USA, 2003.
37. Mills, A.F. *Basic Heat and Mass Transfer*, 2nd ed.; Pearson Education Limited: London, UK, 2014.
38. Sciacovelli, A.; Verda, V.; Sciubba, E. Entropy generation analysis as a design tool—A review. *Renew. Sustain. Energy Rev.* **2015**, *43*, 1167–1181. [CrossRef]
39. Bejan, A. *Entropy Generation Minimization*; CRC Press LLC: Boca Raton, FL, USA, 1996.
40. Turton, R.; Bailie, R.C.; Whiting, W.B.; Shaeiwitz, J.A.; Bhattacharya, D. *Analysis, Synthesis, and Design of Chemical Processes*, 4th ed.; Pearson Prentice Hall: Upper Saddle River, NJ, USA, 2012.
41. Access Intelligence LLC Chemical Engineering Plant Cost Index: 2018 Annual Value. Available online: <https://www.chemengonline.com/2019-cepci-updates-january-prelim-and-december-2018-final/> (accessed on 10 February 2020).
42. Quoilin, S.; Van Den Broek, M.; Declaye, S.; Dewallef, P.; Lemort, V. Techno-economic survey of Organic Rankine Cycle (ORC) systems. *Renew. Sustain. Energy Rev.* **2013**, *22*, 168–186. [CrossRef]

