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Using the forward movement of a container ship navigating in the Arctic to air-cool a marine organic Rankine cycle unit

Santiago Suárez de la Fuente[a]*, Ulrik Larsen[b], Rachel Pawling[a], Iván García Kerdan[c] and Alistair Greig[a]

[a]University College London, Department of Mechanical Engineering, Roberts Building, Torrington Place, London, WC1E 7JE, United Kingdom
[b]Chalmers University of Technology, Maritime Operations, 412 96 Gothenburg, Sweden
[c]Centre for Process Systems Engineering, Imperial College London, Roderic Hill Building South Kensington Campus, London, SW7 2AZ, UK

Abstract

Ice coverage in the Arctic is declining, allowing for new shipping routes. Navigating Rotterdam-Yokohama through the Arctic instead of going through the Suez Canal reduces the travel distance by about 60% thus potentially reducing fuel consumption, CO_2 emissions and other pollution factors. It is important to reduce the environmental impact further in the sensitive Artic, and this can be done with a waste heat recovery system (WHRS). Low heat sink temperatures increase the WHRS thermal efficiency substantially and the cold Arctic air presents an attractive opportunity at the cost of increased power consumption due to air moving through the condenser. This paper investigates the exploitation of the forward movement of a container ship navigating in the Arctic Circle and density-change induced flow as means of moving air through the condenser in an organic Rankine cycle (ORC) unit to reduce the fan power required. The ORC unit uses the available waste heat in the scavenge air system to produce electric power. The paper uses a two-step optimisation method with the objective of minimising the ship's annual CO_2 emissions. The results suggest that using the supportive cooling could reduce the fan power by up to 60%, depending on the ambient air temperature.

Keywords: Shipping; Arctic; Efficiency; CO_2 emission reductions; Air-cooling; free-flow; Waste heat recovery systems;

* Corresponding author. Tel.: +44-20-7679-7063.
E-mail address: santiago.fuente.11@ucl.ac.uk

1. Introduction

Recent studies show that ice coverage in the Arctic has been constantly declining [1,2] and suggest that it will continue to do so [3,4]. For shipping this means a possibility for greatly reduced traveling distances. Schøyen and Bråthen [5] indicated a distance reduction of 37% when switching the Yokahama-London route from via the Suez canal to via the Arctic.

Waste Heat Recovery Systems (WHRS) have been identified as one of the most important technologies for reducing ships' fuel consumption and CO_2 emission [6]. Heat from the main engine is normally used to cover the steam demand and, when possible, produce mechanical or electrical power [7]. However, in Arctic operations, the on board heating demand precludes the use of an exhaust gas WHRS [8]. The engine's charge air heat is an alternative source that represents about 17% of the fuel energy [9], at temperatures of 100˚C-200˚C [10], which is suitable for Organic Rankine Cycle (ORC) units [11].

The cold Arctic air represents an opportunity for reduced heat sink temperatures and increased WHRS efficiencies. Drawbacks are large air mass flow rates, fan power input and heat transfer areas due to air's low heat transfer coefficient and specific heat [8].

There exists potential to reduce the fan power consumption by using the ship's forward movement and passive ventilation designs, as seen in land-based buildings and road vehicles. The simplest (Fig. 1A) is to arrange heat exchangers on the open deck. Windscoops can rotate into the wind (Fig. 1B) and is an established method [12] of increasing ventilation below-decks on ships and in low-energy buildings. More sophisticated wind catchers and wind towers (Fig. 1C) can use the pressure differential on the windward and leeward sides of a structure to generate an internal airflow. Appropriately shaped devices can use external airflow to increase the updraft in chimneys (Fig. 1D). These can take the form of Venturi-shaped devices, which cause significant reductions in the local air pressure over the top of the chimney [13].
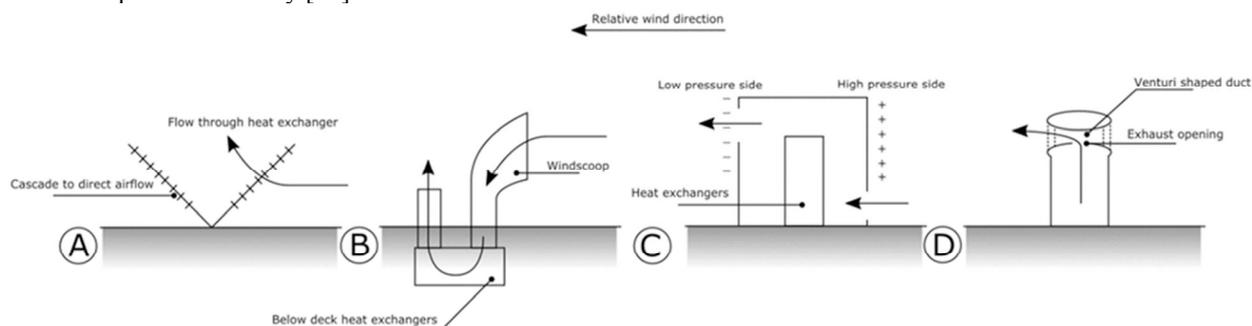
Fig. 1. Four possible options for using apparent airflow to cool the ORC unit condenser.

The aim of this paper is to investigate the reduction of the air condenser's power input of a marine ORC unit by using the ship's forward movement and passive ventilation to capture part of the air demand to reject the WHRS excess heat. The work contributes to the study of marine WHRS, Arctic shipping and hybrid cooling and ventilation.

2. Case Study

2.1. The ship

An hypothetical 4,130 twenty-foot equivalent unit (TEU) containership is used to study the performance of the WHRS and its power demand when sailing in the Arctic [14,15]. The propulsive power is calculated using UCL's Whole Ship Model using the ship characteristics described in Ref. [8]. The operational profile, shown in Table 1, was obtained using anonymised Automatic Identification System data from containerships navigating in the Arctic during 2012. Note that there is no impact of sea state. To reach a maximum speed of 25.2 knots, a 41,125 kW two-stroke low-speed Diesel engine is used [16]. The main and auxiliary engines consume heavy fuel oil (HFO) with an

assumed Carbon Factor of 3.1144 g CO_2/g fuel [17]. The on board electric demand, while at sea, is assumed constant at 1,390 kW_e [17] with an auxiliary engine specific fuel consumption (SFC) of 227 g/kWh.

Table 1. Operating profile for the hypothetical containership.

Speed (kn)	Power Required (kW)	Time (%)	Scavenge Air Temperature (˚C)	Scavenge Mass Flow Rate (kg/s)
<19.8	<16,810	47.4	79	39.6
21.1	20,925	20.3	98	44.9
22.1	25,210	15.5	118	55.6
23.3	30,844 (design point)	12.8	140	76.0
24.6	37,550	4.0	160	86.8

2.2. Route characteristics

The 1,980 km route starts from Reykjavik, Iceland ending in Ballstad, Norway and 4.3 round trips/month is assumed. The ambient air temperatures, from the CRUTEM4 data set [18,19], are shown in Table 2. For the wind data, the route is discretised into 44 waypoints. A random voyage date is chosen and for each waypoint along the route the wind data [20] for that date and time is polled. This process is repeated 500 times to have sufficient results to represent the entire 36 year period for which data is available. The resultant data is clustered from which probability distributions are obtained (see Table 3).

Table 2. Monthly ambient air temperature seen on the studied route.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Air Temperature (˚C)	-1.7	-1.1	-0.9	1.0	4.0	5.4	7.3	7.3	7.4	4.1	1.4	-0.5

Table 3. Probability distribution for True wind speed (TWS) and angle (TWA) seen from the ship's bow along the route.

TWS (m/s) / TWA (˚N)	0	2	4	6	8	10	12	14
0	0.00%	1.26%	0.21%	0.42%	0.00%	0.00%	0.00%	0.00%
45	1.26%	5.03%	6.92%	2.10%	1.89%	0.00%	0.00%	0.00%
90	0.21%	7.34%	12.58%	7.13%	7.55%	2.10%	0.00%	0.00%
135	0.84%	6.71%	10.69%	5.24%	3.98%	1.89%	0.84%	0.63%
180	0.21%	3.14%	4.61%	2.31%	1.26%	1.05%	0.42%	0.21%

2.3. Waste heat recovery system

The turbochargers are in use at engine loads above 35%, and the charge air temperatures are partly reduced via the ORC unit [16]. The ORC working fluid is R1233zd(E) which is suitable for the temperatures, 60-180˚C [21]. It is non-flammable [22] thus allowed in the machinery room [23], but also has a low Global Warming Potential. A simple plant layout [8] is used for the ORC unit generating electricity for consumption on board. The maximum turbine power output is limited to 600 kW and a maximum air condenser volume equivalent to the volume of one TEU, both with the objective of minimising the WHRS size and impact on board the ship.

2.4. Condenser cooling approach

A modular cross-flow finned tube heat exchanger is used, it offers large heat transfer area densities [24] thus improving the overall heat transfer coefficient. The finned tube condenser unit is made of aluminum, to obtain a high thermal conductivity, it has five tube rows with a constant 0.083 m transverse pitch. The condenser unit model

calculates the air demand to absorb the excess heat and sizes the fan power requirement. The fan air flow rate is complemented by hybrid air cooling. This approach is unique for its application in marine WHRS as it combines the air flow caused by stack effect, ship's forward movement and natural wind speed, and forced air due to the fan. The concept uses windscoops with a capture area of 2m^2 positioned either side of the ship's superstructure (Fig. 2). The chimneys can be integrated into the superstructure to minimise structural weight and wind drag. The chimney exhaust is assumed to be one deck, 2.8 m, above the bridge and having a superstructure with six decks, the chimney exhaust is 19.6 m above the upper deck. The chimney diameter is assumed to be 2 m. The ship's beam (width) is 32.2 m similarly the superstructure width. The Venturi-shaped roof is of an omnidirectional construction; has a contraction ratio of two with a final flow area of 2 m^2. The hybrid cooling system has a supportive role in covering the air flow demand and it is not intended as a substitute of the condenser's fan.
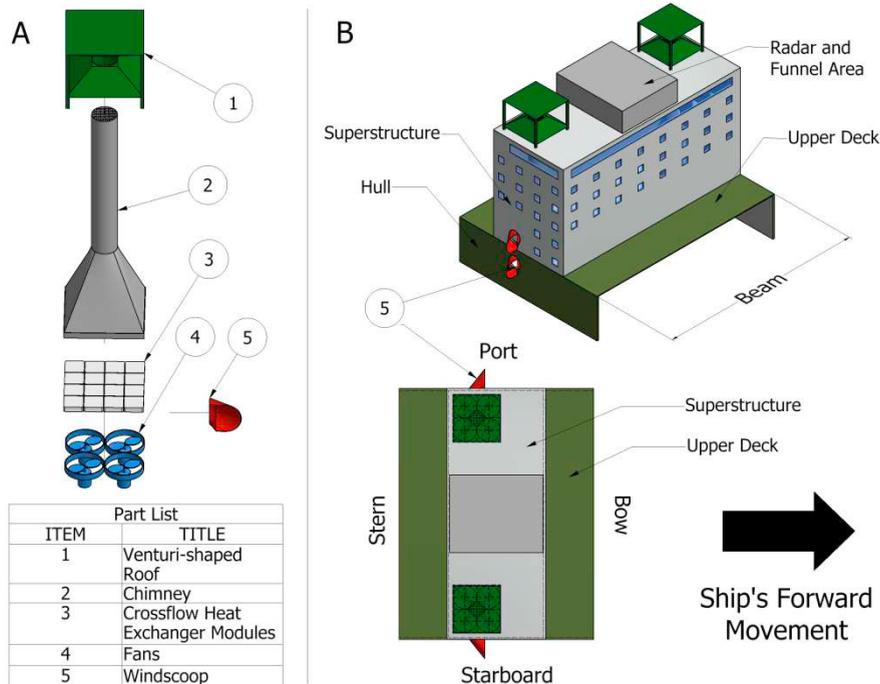
Fig. 2. A) Hybrid cooling system components for the ORC unit. B) Suggested location for the ORC unit cooling system with different parts of the ship as well as the direction of the ship movement.

3. Method

The ORC and passive cooling are modeled as shown in Ref. [8] and Ref. [25] respectively. In this section, air flow demand and how the systems support the fan are discussed. For apparent wind angles and speed see Ref. [26].

3.1. Passive cooling approaches

The condenser's saturation temperature and pinch point temperature difference are held constant. For the saturation temperature it was set at 25˚C while the pinch point temperature difference was determined by the optimisation process.

The windscoops opening effectiveness is assumed to be 0.55 due to the ability to rotate. For the stack flow ($\dot{Q}_s$) the height from the midpoint of lower opening to neutral pressure level is assumed to be 0.25 m above the chimney exit (i.e. 19.85 m). The pressure drop found at the middle of the chimney generated by the Venturi-shaped roof is found using the Bernoulli equation. This gives a pressure coefficient of -0.75, which is a conservative value [13], and it is assumed to be the same for any wind angle. The total air volume flow rate ($\dot{Q}_T$) going through the condenser is given by the following approximation [25,27]:

$$\dot{Q}_T = \sqrt{\dot{Q}_s^2 + \dot{Q}_{ws}^2 + \dot{Q}_f^2 + \dot{Q}_{vr}^2}$$

(1)

Where $\dot{Q}_{ws}$ is the windscoops flow rate; $\dot{Q}_f$ is the fan flow rate required with an assumed efficiency of 60%; and $\dot{Q}_{vr}$ is the Venturi-shaped roof volumetric flow rate. Pressure losses due to ducting and flow through the chimney were assumed insignificant. The implications of fan locations, ducting path and the energetic cost, drag and movement, of the rotating windscoops and Venturi-shaped roof is beyond the scope of this work.

3.2. Optimisation and cooling study

The models of both the WHRS and condenser unit were optimised to reduce the annual CO_2 emissions using the operating profile shown in Table 1 for a whole year and are calculated as shown in Ref. [8]. The optimisation of the 14-dimensional space is a single objective approach using Particle Swarm and Pattern Search algorithms [8]. The 14 independent variables are: ORC's design point with respect to the ship speed; ORC's maximum pressure; superheating temperature before the expansion process; scavenge air outlet temperature; percentage of waste heat bypass; three pressure points, as percentage of the ORC's maximum pressure, to determine the ORC high pressure at off-design ship's speeds; condenser's pinch point temperature difference, tube's length and internal diameter; and fins' height, thickness and pitch. The data obtained from the optimisation is post-processed to include the effects of wind speed, direction and air's temperature and its contribution to the condenser's fan power reduction is calculated.

3.3. Route

The probability of finding a wind with certain angle and speed as shown in Table 3 is assigned to any given operating speed and month. This means that any operating condition will experience, at some time, all wind conditions. In the case of twin chimneys, as shown in Fig. 2B, it is assumed that when the wind is captured in the port side then at starboard the wind pressure will be negative, see Fig. 1, and not working as an air inlet. Under these circumstances only the fan and stack flow rates will be operating. When the total passive flow rate is above the required, the wind cooling will be limited by rotating the windscoops to less favorable angles.

3.4. Model validation

The WHRS thermodynamic model was used in Larsen et al. [28]. There was a 1% difference on the heat exchanger overall heat transfer coefficient when compared to Richardson and Peacock [29]. The air condenser model gave an error of 0.3% and 0.5% for the logarithmic temperature difference and the outlet temperature respectively when compared to Gnielinski [30]. The wind and stack flows were validated against results of EnergyPlus[TM]. For a headwind of 11.4 m/s, at a temperature of -25.3°C with a TWA of 30° and a rejection of 1650 kW there is a fan power reduction of about 32.7% while when using it EnergyPlus[TM] the reduction was 28.5%.The difference could be caused by the chimney and duct frictional losses not considered in this work.

4. Results

4.1. Organic Rankine cycle unit

The ORC unit is capable of reducing the CO_2 emissions by 543 t/year without the hybrid cooling with some of the condenser characteristics shown in Table 4. The ORC unit operates when the engine load is above 90%. Below this speed the waste heat temperature is too low. During the summer when the air temperatures are high, a maximum fan consumption of 25 kW$_e$ is seen at design speed which results in the lowest ORC unit net power (see Fig. 3). Maximum net power, 451 kW$_e$, is found at the maximum ship speed, but at minimum ambient air temperatures, while the maximum fan power consumption happens during September.

4.2. Passive cooling

During the whole year of operation, the twin hybrid cooling system manages to save around 1.6 t of fuel and 4.9 t of CO_2, which is 0.9% less. The low savings are caused mainly by a low probability of encountering favorable winds, such as headwinds (see Table 3); low windscoop and Venturi-shaped roof capture areas for the air's flow rate required; and the fact that while one passive system is working the other will be only operating with fans and stack effects. But putting the results into perspective, if the ORC unit condensers were operating only with fans, the fuel consumption due to the condenser cooling is around 6.2 t which means that the passive cooling brings a reduction of about 25.7%.

Table 4. Some of the design characteristics for the hybrid cooling approach. The air requirement and cooling loads are given for design speed and ambient air temperature of 7.4˚C.

Modules (-)	Length (m)	Width (m)	Tube rows (-)	Frontal Area (m^2)	Heat Rejected (kW)	Air Requirement (m^3/s)	Windscoop Area (m^2)	Chimney Outlet Area (m^2)
43	7.30	6.34	5	46.28	1648.51	87.39	2.00	2.00
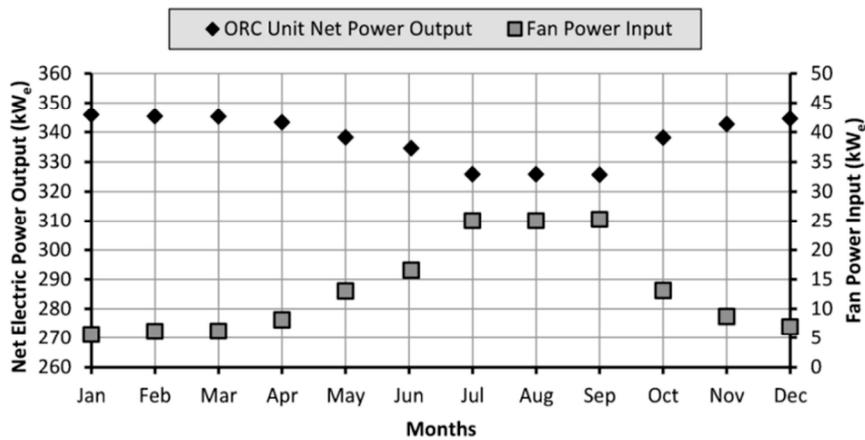
Fig. 3. Monthly ORC unit power output and fan's input at design speed (i.e. 23.3 knots).

Looking only into one leg of the trip and into the hybrid system fully operating, the fuel saving is about 152 kg. The passive system in January substitutes 33.5 kW$_e$h from the fan and in September it reaches 88.6 kW$_e$h, see Fig. 4. While the passive volumetric flow rate participation reduces during the warmer months, due to a reduction in air's density, it is the higher energetic cost of moving air through the condenser that increases the passive flow energy contribution. In January, a volumetric flow rate of 61.1 m^3/s and a power of 5.5 kW$_e$ is required while in September the volumetric flow rate and power increase to 111.8 m^3/s and 24.3 kW$_e$ respectively. Then, the power requirement to move a cubic meter of air in January is about 0.09 kW$_e$/(m^3/s) while in September is about 0.22 kW$_e$/(m^3/s).

The largest passive cooling contribution is from the windscoops, followed by the Venturi-shaped roof and the stack effect. The stack flow contributes to the net passive cooling by 3.6% to 8.1% of the total energy. As the ambient air temperature increases, the stack volumetric flow reduces due to a lower density difference between the ambient air and the air exiting the condenser. The Venturi-shaped roof suffers from the same issue and has a proportional contribution of around 21.1% in January and 12.3% in September. The wind volumetric flow rate stays the same during the different months, but changes with the ship and apparent wind speeds. In September, it supplies 19.5% of the energy requirement and 29.9% in January.

For the hybrid cooling system that is blocked by the superstructure, the fuel reduction achieved is around 32 kg in a single leg. Apart of the fan ventilation, the largest contributor to the condenser cooling comes from the stack flow rate which represent between 4.9% and 12.1%, depending on the month. Volumetric flow rates from the windscoops and Venturi-shaped roof are only active when there are headwinds or tailwinds reducing considerably their benefits. Wind and roof cooling represent 1.6% and 1.1% of the total energy required respectively in January.
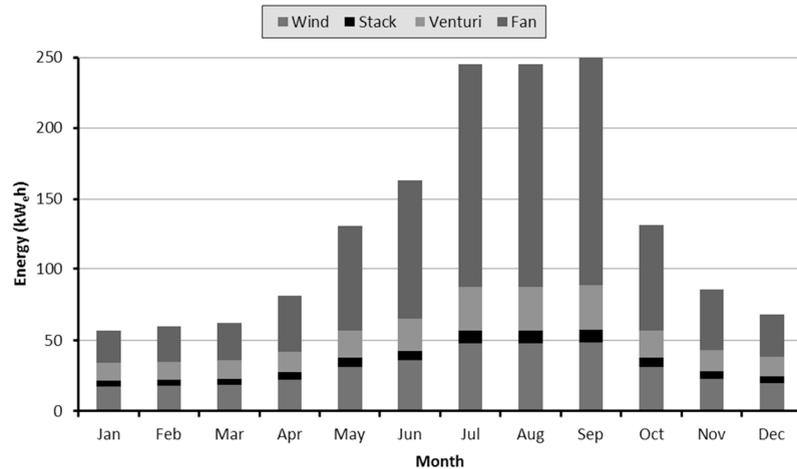
Fig. 4. Single monthly trip energy contribution for each cooling approach.

5. Limitations of the analysis

This analysis did not include an evaluation of a potential increase in wind resistance due to the chimneys, as wind resistance is typically 4.0% or less of the ships total resistance [26], and the indicative system described only increases the projected area by 4.5%. The windscoops have shown to be one of the largest contributors for the passive cooling system, but suffer from blockage due to the superstructure in their current location. Other locations, such as the superstructure roof, may entail much greater ducting losses. Similarly, the local airflow near windscoops and Venturi devices may be degraded by the presence of fittings and masts. The initial analysis presented here has also assumed straight and smooth ducts, with a minimum of losses.

Conclusions

The potential fuel and emissions savings for a containership navigating in the Arctic from the installation of a forced-air-cooled ORC unit assisted by a passive cooling system was investigated. It was found that the ORC unit could reduce the vessel's CO_2 emissions by 543 t/year. A passive cooling system with windscoops, Venturi-shaped roofs and chimneys provided CO_2 emission reductions of 0.9%, or 1.6 t fuel/year. Wind cooling was the largest source of fan power reductions followed by Venturi-induced cooling and stack-induced cooling. During the colder months and for an unblocked system, the passive cooling system had a larger contribution to the fan power reduction mainly caused by lower power demand at the condensers. For the summer months, the fan power requirement increases considerably while the volumetric flow rate of the roof and stack cooling reduces. However, it was found that the energy contributions are higher during the warmer months due to the fan power requirement from 0.09 kW$_e$/(m^3/s) in January to 0.22 kW$_e$/(m^3/s) in September. For the blocked passive system, the largest passive cooling contribution comes from the stack volumetric flow rate representing a maximum of 12.1%. Further work is required to evaluate pressure losses and leakages into the optimisation process.

Acknowledgement

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