An analysis of shipboard waste heat availability for ballast water treatment

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Heat treatment of ballast water is one of the many treatment options being explored. This analysis has tried to assess the heat availability from the cooling water, exhaust gases of the engines and steam condensers based on design and operational data obtained from an existing crude oil carrier. Time requirements for ballasting and treatment using the seawater and condenser circulating pumps are projected. Heat balance exercises were also carried out on a testbed engine to verify attainable heat recoveries. It is seen that, although considerable heat is available, a longer time than that available during ballasting and normal ballast passages will be required for the treatment process.

AUTHORS' BIOGRAPHIES

Rajoo Balaji is attached to the Marine Engineering Department of the Malaysian Maritime Academy as Head of Advanced Marine Engineering. He is currently pursuing doctoral research in the field of ballast water treatment systems focussed on cost-effective solutions for onboard treatment using shipboard resources with minimal dependence on new systems. Viability of a heat treatment system in combination with another physical method excluding use of chemicals is being researched.

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INTRODUCTION

The threat from transporting invasive species into waters of various regions of the globe has assumed top priority. Ships' hulls transport species in their sea chests and ballast water tanks.[1,2] Currently, to mitigate the problem, ships are following ballast water exchange regimes prescribed by guidelines, which are soon to give way to effective treatment systems. One of the cognizable treatment methods is based on sterilisation by heat. Research on heat treatment from laboratory to shipboard trials is on record.[3–6] A current status report on ballast water treatment (BWT) technologies projects six installations based on heat treatment.[7] Treatment by microwave heating[8] and by employing a heating medium such as steam[9] have shown good efficiencies in attaining satisfactory mortality rates of species at low temperatures below 45°C.[10]

From the laboratory tests to the shipboard trials, it has been established that significant mortality rates of species could be achieved in the temperature range 35–38°C. Research[11,12] has differentiated effective heat treatment — long-term heating involves heating for ≥ 16h at 36°C, medium-term for 10 mins to 16h at 35–40°C and short-term heating at ≤ 10 mins at ≥ 46°C. Effectively, heating to a temperature of 35°C for a period of at least 20h has been described as good.[9]

Although concerns over the mortality of bacteria pathogens prevail, such temperature ranges have recorded a biological efficiency of 100% on all zooplankton and most of phytoplankton.[13] These ranges of temperature rise are possible with purposeful heat taken from heat generators, such as boilers and engine cooling water. Cooling water apart, the potential for heat recovery also exists from other sources. If waste heat can be harnessed on board, it would not only optimise the treatment but also benefit operational economics. The economics of utilising engine waste heat, at US$0.056/t of treated water, fares well when compared with other heat treatments, including microwave applications, and this warrants further investigations into heat treatment from engine waste heat.[13]

The potential for increased recoveries has been demonstrated using availability analysis.[14] Using exhaust gas heat recovery to lower fuel consumption and improve efficiencies have been researched[15] and models have been proposed for optimised heat exchanger designs that enhance waste heat recoveries.[16]

Amongst the global merchant ship fleet, bulk carriers account for 39% of ballast water carried, oil tankers 37% and the rest 24% (general cargo vessels, container vessels, chemical tankers, LNG tankers, etc).[17] It has been noted[18] that heat treatment could be a cost-effective option on oil tankers having steam-driven pumps. Therefore, an operational, large crude oil carrier, MT *Bunga Kasturi*, owned by MISC Berhad, was chosen for the study. The vessel's particulars and machinery data are tabulated in Tables 1 and 2. The ballast capacity of the vessel works to 39.34% of the deadweight tonnage (dwt), close to the average of 40% for most tankers.[17] It may be said that the vessel is truly representative for the range of ships in its category. Based on the vessel's operational data, the heat quanta and hypothetical time requirements for achieving the temperatures have been calculated. In addition, heat recoveries were calculated while performing heat balance exercises on a testbed diesel engine. The testbed engine data are given in Table 3.

Vessel type	Crude oil carrier
Year	2008
DWT (MT)	257 418
Total ballast capacity (m³)	98 794
Number of water ballast tanks	11 (WBT 1–5 P&S + Aft Peak)

Table 1: Vessel particulars: MT 'Bunga Kasturi'

Type	Ford SD425TCM, 4-stroke, 4-cylinder, turbocharged
Bore	93.7 mm
Stroke	90.5 mm
Swept volume	2496 cc
Output	kW at 3800 rev/min

Table 3: Technical data: Testbed engine

METHODOLOGY

The methodology consists of two parts. Firstly, heat balance calculations were carried out with operational data from the ship and, secondly, from the testbed engine. From the ship trial and operational data, the heat balance of the low temperature system (LT) and high temperature system (HT) heat availability from the atmospheric condenser and the vacuum condenser were obtained. Additionally, heat from the exhaust gases of the main engine (ME) and auxiliary engine (AE) was considered. The heat availability from engine cooling water and exhaust gases were balanced as a percentage of input energy.

For a diesel engine, the heat input is from the fuel and the heat balance could be shown as:

$$Q_{in} = Q_{exhaust} + Q_{water} + Q_{odd\ losses} + W_{engine\ power} \qquad (1)$$

The three thermodynamic losses would include the heat lost to the exhaust gases, cooling water and odd losses comprising friction, radiation, convection, etc. The heat input for a certain output power can be computed otherwise from:

$$Q_{in} = W_{engine\ power} \cdot SFT \cdot LCV \qquad (2)$$

Main engine	Hitachi Zosen B&W 7S80MC 2-stroke marine diesel engine 25 090kW@78.6 rev/min, mcr 22 580kW@75.9 rev/min, CSO
Auxiliary engine (3 sets)	Yanmar 6N21AL-GV 4-stroke marine diesel engine 1020kW/900 rev/min
LT cooling freshwater pump x 3	570 m³/h
HT cooling freshwater pump x 2	193 m³/h
Cooling seawater & vacuum condenser circulating pump x 3	550 m³/h
Atmospheric condenser cooling seawater pump	120 m³/h
Vacuum condenser circulating pump	1550 m³/h
Ballast pump (electric drive)	3200 m³/h
Ballast pump (steam-turbine driven)	3200 m³/h
Boiler water circulating pump (EGE)	25 m³/h
LT central freshwater cooler x 2	13 697.78 kW (Designed)
HT freshwater cooler	3651.19 kW (Designed)
Exhaust gas economiser (EGE)	Forced circulation finned steel tube, 0.59 MPa, 2250 kg/h evaporation

Table 2: Vessel machinery data: MT 'Bunga Kasturi'

When the mass of fuel burnt and calorific value are known,

$$Q_{in} = m\mathrm{f} \cdot LCV \tag{3}$$

The heat energy in the exhaust will be the product of the mass flow and specific heat capacity of the gases. Assuming steady-state and neglecting heat losses due to radiation, the heat available would be:

$$Q_{avail\ g} = m_g \cdot C_g \cdot \Delta T_g \tag{4}$$

Only the temperature rise of freshwater or seawater through a heat exchanger has been considered. Since heat exchanger duties and efficiencies are not required for the analysis and for keeping the calculations simple, logarithmic temperature differences were not computed.

Fig 1 shows the energy distribution for a large diesel engine (B&W Type KMC), but of a different type to that fitted on the vessel (B&W Type SMC). Considering energy flows in various streams, heat rejected in exhaust gases for typical turbocharged engines is around 35–40%. Heat from these gases is recovered in turbochargers, used for production of steam and other purposes.[19] Allowing for such recoveries, heat for treatment of ballast water was calculated from a hypothetical heat exchanger design, assuming main engine (ME) operation at continuous service output (CSO).

In ports, the auxiliary engines (AEs) usually operate at 25–80% of output power as demands are generally not more than 50% of the installed power.[20] Crude oil carriers will have higher port loads, especially during discharge, but for stricter approximations, a lower average of 40% was chosen with only one AE in operation. Equation (2) was used for both ME and AE, based on individual SFC (specific fuel consumption) but assuming the same LCV (lower calorific value). This compensated for the irrecoverable heat in exhaust gases and also resulted in a lower value for heat availability. Although the heat availability from condensers would vary with steam demands, this heat was computed from the temperature rise of seawater from the vessel's trial data under typical situations.

Based on such calculations, the heat available while sailing at CSO and in port conditions was identified. In the first heat balance, heat and time projections were identified with the mass flow of seawater at 70% flow capacity, allowing for a drop in efficiencies due to wear and tear of pumps and also system losses. In the next computation, the total ballast water requirement during light condition to heavy weather conditions were obtained from the *Trim & Stability* booklet of the vessel and the heat requirements were calculated. The time required was then computed assuming a temperature rise of 25°C.

Time requirements were calculated for two recovery regimes. One requirement was computed for recovery from cooling water, while another requirement was computed, supposing additional heat recoveries from ME and AE exhaust. The major waste heat sources considered were the HT/LT coolers and exhaust gases of the ME and AEs. Table 4 projects the approximate values for heat balance computations for most of the auxiliaries with vessel at sea and ME at CSO. The next exercise involved the heat balance exercises on the diesel engine testbed to verify the assumed recoveries, using equation (3) and (4).

DESCRIPTION OF THE SYSTEM

Freshwater is the primary coolant which removes the heat from the diesel engine jacket and from various other auxiliaries, such as air compressors, air conditioning and refrigeration systems, etc. Fig 2 shows the general layout of the cooling systems including other auxiliaries. In the present vessel arrangement, only the engine jacket freshwater is passed through the HT coolers while the cooling freshwater from other auxiliaries passes through the LT coolers. The freshwater is circulated again to cool the engine and the auxiliaries. The heat exchangers are of the plate type and have provision for controlling freshwater flows and temperatures.

The main seawater (MSW) pumps draw from sea suction chests through coarse filters and pump through the LT and HT coolers, arranged in series, and discharge the seawater overboard. The EGE is of the forced circulation type with water flowing inside the tubes and gases on the outside. The tubes are finned on the outside to increase the heating surface area. The feedwater circulating pumps take suction from the drum and circulate the water through the exhaust gas economiser (EGE) and return it to the boiler drum, aiding evaporation.

The atmospheric condenser handles all the normal steam condensate returns. The vacuum condenser handles heavier steam returns from the cargo oil pump turbines, etc. As shown in Fig 2, the MSW pump can also supply circulating seawater for the vacuum condenser, but there are also individual pumps provided for vacuum and atmospheric condensers. These condensers are of the shell and tube type heat exchangers.

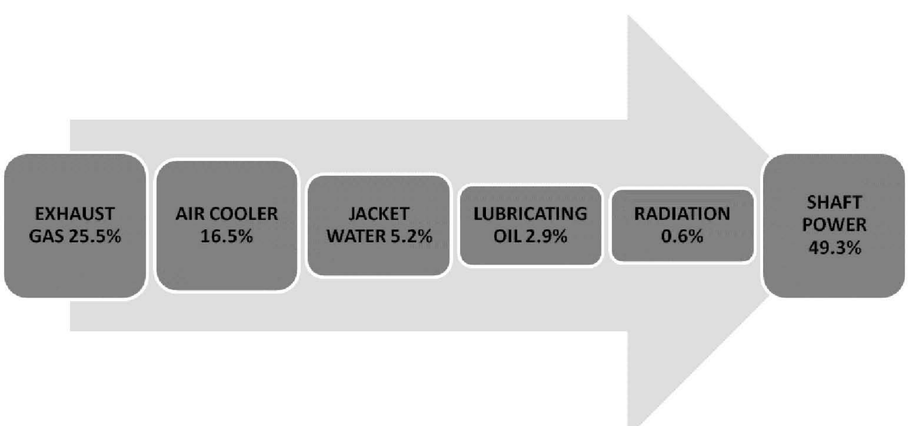

Fig 1: Heat balance of a modern large diesel engine

Machinery	Inlet temperature °C	Outlet temperature °C	Flow rate m³/h	Heat energy kW	Remarks
Freshwater:					
ME LO cooler	36	42.7	250	1944.9	
ME air cooler	36	54	350	7315	
Steady bearing (shaft bearing)	36	37	5	5.8	
Sterntube LO cooler	36	37	5	5.8	
Drain cooler	36	50	70	1137.9	
A/C & Ref. machinery	36	39	60	209	
Provision Ref. machinery	36	39	10	34.8	
Engine control room coolers x 2	36	39	5 through each cooler	34.8	
Workshop cooler	36	39	5	34.8	
Cargo oil pump (COP) turbine LO cooler x 3	36	36	10 through each cooler	-	25m³/h; 36–38°C during cargo discharge
Cargo ballast pump turbine LO cooler	36	36	10	-	36–40°C during cargo discharge
Boiler water circ. pump casing & bearings	36	38	2	4.6	
Main & topping-up air compressors	45	57	2	27.9	
AE LO cooler	36	41	30	174.2	
AE air cooler	41	48	30	243.8	
Seawater:					
LT cooler x 2	32	42	1100	(12772.2) 11409.39	Sea trial Calculated from operational data
HT cooler	41	47	550	3383.75	Calculated from operational data
Vacuum condenser	32	48	2160	(40128)	Only during cargo heating/discharge 1550 m³/h vacuum condenser p/p+ 550 m³/h MSW p/p
Atmospheric condenser	32	46	120	1950.7	Cargo heating/discharge/seagoing

Table 4: Heat balance of MT 'Bunga Kasturi': Approximate operating temperatures and flow rates

The seawater pumps draw water from the sea chests and, after passing through the condensers, discharge it overboard.

The water exiting from these heat exchangers will be at higher temperatures. If this water pumped in by the MSW pump and condenser circulating pumps is further heated by heat harvested from other sources and substituted for ballast, it would amount to heat treated seawater. The heat calculations are based on this premise.

DESCRIPTION OF THE TESTBED

The Ford diesel engine fitted with testbed features was used for the study. The engine has been installed in the marine engineering workshop spaces of the Malaysian Maritime Academy, located in Terengganu, Malaysia, on the shores of the South China Sea. The ambience may be assumed to be similar to that experienced by ships at sea. The room is fitted with supply fans and exhaust blowers, resembling any shipboard engine room. The testbed is used in training for calculating the power and efficiency of internal combustion engines. The set-up is fully wired and all system parameters are obtained through sensors and transducers routed through a control module interface. A desktop computer displays the parameters and the engine conditions are monitored from an adjacent control room. The control room houses the desktop computer and the control module, which are always

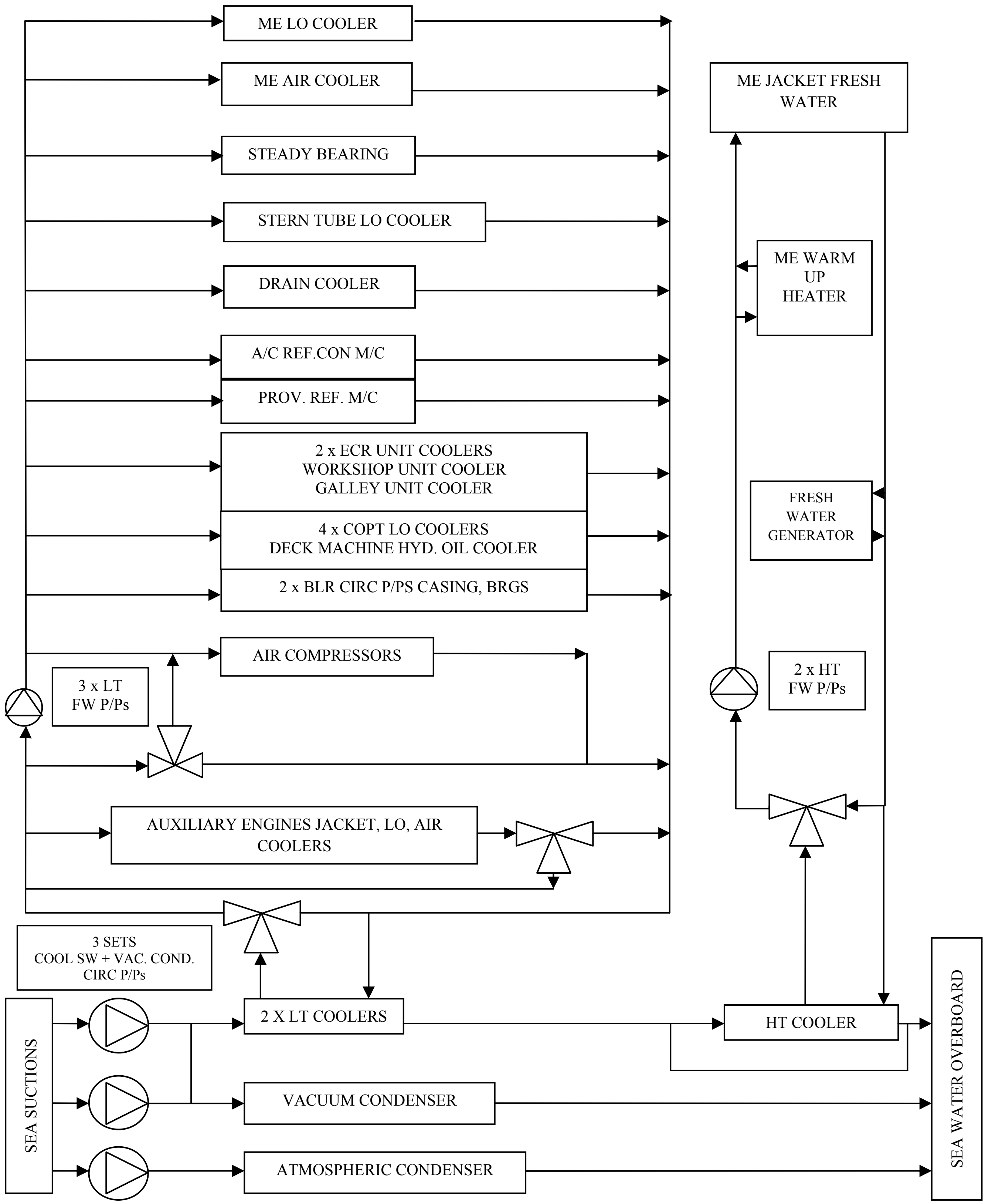

Fig 2: MT 'Bunga Kasturi': General layout of HT, LT systems

maintained in an air-conditioned ambience. The testbed engine can be viewed from the control station through large transparent windows fitted on the walls of the control room.

The four cylinder engine is permanently coupled to an eddy current dynamometer fitted with a load cell. The torque is computed from the load on the cell and the radius length from the dynamometer centre to the load cell. The engine has two cooling-water circuits, one circulating for the lubricating oil cooler and the engine jacket while another plant water circuit cools this water in turn, similar to marine engines. A part of the exhaust gas heat is extracted in the turbocharger and the remaining heat from the exiting gases has been estimated as recoverable for treatment. A heat exchanger (calorifier) placed in the path of the exhaust gases with water as the cooling medium estimated this heat using equation (4). Cooling freshwater for the calorifier fitted in the path of the exhaust gases has been branched off from the inlet of the plant water supplied from high storage tanks located in the premises. Therefore, the temperature of the water to the calorifier may be assumed to be similar to that of open seawater temperature. The exhaust pipe connection from the engine is of metal reinforced type and about 1–2m from the engine to the calorifier inlet. Fig 3 shows the testbed engine arrangement.

Since mass flow was considered for heat calculations, it was assumed that results obtained using freshwater would be similar to seawater. The freshwater leaving the calorifier was not returned to any containment. The ambient temperature was typically tropical during the experiment days, varying between 31–33°C. The test runs were spread over almost four months. Each effective period of run was not shorter than one hour, excluding warm up and cool down runs. The limiting factor was the dynamometer cooling water temperature which was never allowed to go beyond 50–52°C. A commercially available high-speed diesel (HSD) was used throughout. The lower calorific value (LCV) of the HSD was assumed at 42 000 kJ/kg and specific heat capacity of water at 4.2 kJ/kg °C. It is noted that the testbed engine experiments were intended for heat recovery assessments and no further similitude laws were applied while considering the large two-stroke shipboard engine.

DISCUSSION

Table 5 shows the heat utilisation in the main engine exhaust and cooling water. When the freshwater generator (FWG) is in operation, HT coolers are bypassed and engine freshwater heat is discarded in the FWG and this heat will not be available. Based on the full mass flow of exhaust gases projected in the design data, the heat utilised in two turbochargers and the EGE placed in the path of the exhaust gases in the uptakes, was computed. Assuming a hypothetical waste heat recovery, a ballast water heater of the shell and tube design was designed and heat duty was computed, as shown in Table 5. A maximum seawater flow of 100m^3/h targeting a treatment temperature of 55°C was assumed for the design. The details of heat exchanger calculations are not reflected herein. Table 6 shows the heat distribution as a percentage of the input energy at CSO.

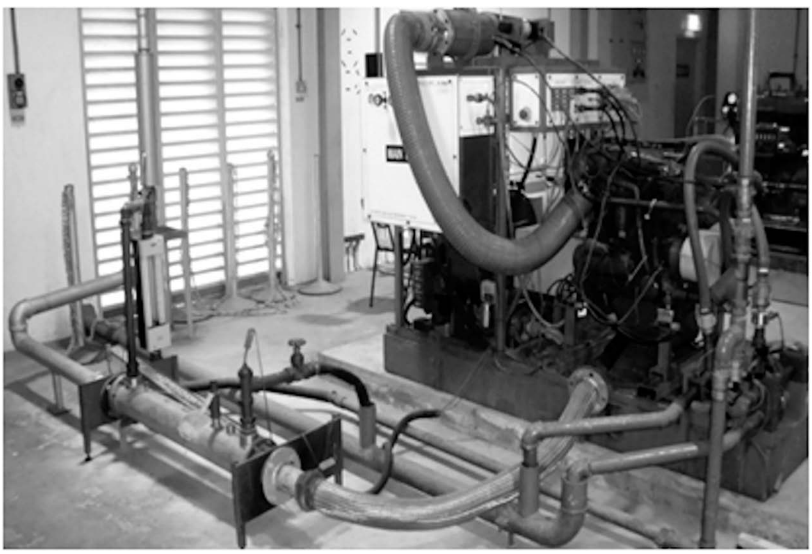

Fig 3: Testbed engine with exhaust gas calorifier arrangement

	Mass flow	Inlet temperature	Outlet temperature	Heat Energy
	kg/s	°C	°C	kW
FWG (Fresh water generator)	53.61	81	66	3377.5
Turbochargers (Exhaust gas)	60.81	301.9	245	3816.48
EGE (Exhaust gas)	60.81	249	205	3215
Ballast water heater (Exhaust gas)	60.81	200	132.5	3052

Table 5: Data for exhaust gas and FWG heat energy computations

	System energy kW	% of Input energy		System energy kW	% of input energy
HT system	3383.75	7.39	Fresh water generator	3377.5	7.33
Turbochargers	3816.48	8.28			
Exhaust gas economizer	3215	6.97			
Ballast water heater	3052	6.62			
Output	22 580	48.98			
Radiation, LO	4610.08	10			
Total accountable heat	**40657.31**	**88.24**			
Unaccountable heat	**5443.517**	**11.76**			
Input	**46100.83**	**100**			

Table 6: Heat balance at CSO: MT 'Bunga Kasturi'

Heat balance and recoveries

Referring to Table 6, the input energy was computed using equation (2), based on a SFC obtained from shop trial data. An upper rounded value of 175 g/kWh for the SFC has been applied, considering data from sea trials for maximum continuous rating (mcr) and CSO operations. The HT heat discarded in the HT coolers was computed from the trial/operational data of seawater which is in agreement to the heat utilised in the FWG. Accounting for the existing systems, radiation/lubrication oil (LO) losses at 10% and the hypothetical ballast water heater at 6.62%, the unaccountable heat is projected as 11.76%. This could be attributed to freshwater, exhaust gases and irrecoverable heat. If exhaust gas heat recovery alone is considered, it works out to 21.87%, which is still less than 25.5% projected in Fig 1. The freshwater heat accounts for 7.33%, or 7.39% (FWG or HT cooler), which is higher than the 5.2% projected in Fig 1. Considering approximations, the heat balance appears fairly distributed as applicable for large two-stroke turbocharged diesel engines.

Table 7 shows the recoverable heat obtained from the heat balance of the engine and the exhaust recovery from one auxiliary engine (AE). The HT heat of 2368 kW was computed from the actual temperature rise of seawater from the data, whereas the 3383.75 kW projected in Table 6 was obtained from calories rejected in the HT system at CSO. As the temperature rise of seawater objectively indicates the heat recovered, the lower value of 2368 kW was assumed. For AEs, an average SFC value of 200g/kW h was considered. Both ME and AEs are designed to burn heavy fuel oil (HFO) and the SFC values are applicable for such oils, although in actual practice the SFC of fuels may vary depending on the specification of fuels bunkered and engine loads. Furthermore, only 10% exhaust heat recovery for a single AE was considered, whereas in operation, depending on SFC and number of machines, this is bound to be nominally higher.

Table 8 has been drawn up summarising heat availability in sailing and in port conditions. While sailing at CSO, the LT heat was computed from the heat balance of the LT coolers considering the seawater temperature rise. It is to be noted that some heat from the LO Coolers and ME air coolers are realised in the LT system, as tabulated in Table 4. The HT heat accounted for in the Table was neglected, assuming FWG operation, and hence not available.

System	Heat kW
LT	11409.39
HT	2368
ME exhaust	3052
AE exhaust	95.20
Total heat recoverable	**16924.59**
Total heat recoverable if FWG is in operation	**14556.59**

Table 7: Heat recoverable with ME at CSO and one AE on load during sailing

Time requirements for heating

Table 9 has been based on LT, HT heat and heat from the condensers. The possible heat which could be absorbed by the seawater will be the heat rejected by the freshwater and this was computed using the designed operational temperatures, specific heat capacity and mass flow of seawater from the

	At port discharging kW	At port loading kW
LT	2984.91	2827.93
HT	0	0
ME exhaust gas	0	0
AE exhaust gas	95.2	95.2
Vacuum condenser	38938.68	2704.67
Atmospheric condenser	1825.6	1825.6
Total heat available	**43844.39**	**7453.4**

Table 8: Heat available in port

Condition	Engine output power	Heat absorption in LT cooler	Heat absorption in HT cooler	Heat absorption in atmospheric condenser	Heat absorption in vacuum condenser	Flow rate	Time for complete ballast water to be treated once	Possible treatment protocol
	kW	kW	kW	kW	kW	m^3/h	Hours	
Designed	25 130	13 697.78	3880	0	0	770	128.3	Sailing
Normal seagoing (CSO)	22 580	11 409.39	2368	0	0	770	128.3	Sailing
Normal seagoing (CSO) + cargo heating on	22 580	11 409.39	2368	1825.60	0	854	0	Not applicable
At port, cargo heating and full discharge	0	2984.91	0	1825.60	38938.68	1981	49.87	At port partly
At port, cargo loading	0	2827.93	0	1825.60	2704.67	854	115.7	At port partly

Table 9: Time requirements with heat absorption from HT, LT and steam condenser systems: MT 'Bunga Kasturi'

Ballast condition	Bunkers	Ballast recommended	Heat required for 25°C rise	Total heat available at CSO	Total heat available while discharging cargo	Time required (treatment protocol: sailing)	Time required (treatment protocol: sailing)	Time required (treatment protocol: In port, discharging)	Time required (treatment protocol: In port, discharging)
		m^3	kJ	kW	kW	Hours	Days	Hours	Days
Light	Full	77210.73	8517550125	14556.59	43844.39	162.54	6.77	53.96	2.25
	50%	81548.29	8996050875	14556.59	43844.39	171.67	7.15	56.99	2.37
	10%	85712.20	9455394375	14556.59	43844.39	180.43	7.52	59.91	2.50
Heavy	Full	91753.17	10121808375	14556.59	43844.39	193.15	8.05	64.13	2.67
	50%	95917.07	10581151875	14556.59	43844.39	201.92	8.41	67.04	2.79
	10%	98793.17	10898430375	14556.59	43844.39	207.97	8.67	69.05	2.88
Gale	Full	112421.46	12401844000	14556.59	43844.39	236.66	9.86	78.57	3.27
	50%	116759.02	12880344750	14556.59	43844.39	245.79	10.24	81.60	3.40
	10%	121096.59	13358845500	14556.59	43844.39	254.92	10.62	84.64	3.53

Table 10: Time requirements for varied ballast conditions during sailing and during cargo discharge in port

MSW pump. The temperature rise of water if all this heat was to be absorbed at the maximum possible pump rate was then computed using equation (4). For example, if the engine were to produce an output power of 22 580 kW, then the heat from LT coolers alone would elevate the seawater temperature by 8.7°C and HT coolers by 3.6°C in a single circulation; whereas the design data from the vessel's manuals project a temperature rise of 9.3°C for LT seawater flow and 5.4°C for the HT. Assuming this heat transfer for a flow rate equal to 70% of the total rated, the time taken to raise the temperature was calculated. In the first normal seagoing scenario at CSO, the time taken is projected to be 128.3h, or about 5.4 days.

In the next situation, cargo heating operation implies the vessel is in the loaded condition with no need for ballast, so the treatment protocol is shown as not applicable. Therefore, although heat is available, the time for heat treatment is shown to be nil. But while discharging oil cargoes, heating is gradually shut-off and a good amount of heat is available from the condensers and some from the LT system. Assuming ballasting of seawater is started at an appropriate condition after some amount of cargo is discharged, the seawater removing this heat can be directed to the ballast tanks. The time taken for this is projected as 49.87h. A similar scenario has been tested onboard an oil tanker with steam resources with a target temperature range of 65–75°C.[21]

In the next scenario, deballasting is taking place as the cargo is loaded. Assuming a treatment protocol in port, the ballast water can be circulated through the LT and atmospheric condenser system and discharged overboard. This would amount to heat treating the water just prior to discharging. A maximum ballast loading time of 115.7h is projected in this last scenario when little heat from LT coolers, atmospheric and vacuum condensers is available. The time projections in Table 9 are simplistic for nominal temperature rises, as mentioned earlier. The temperature rise and time for the rise to occur are projected for a single circulation. This computation assures a seawater temperature rise and also a possibility for recirculation of the ballast water from tanks while the ship is sailing. This time is not to be construed as the time required for a rise of 25°C.

In the actual situation, a higher number of water circulations will be needed to achieve a high temperature rise. This is due to stratification and heat losses through the tank walls. In the design and operational conditions of the vessel data, seawater inlet temperatures were in the range of 31–32°C. For the calculations, instead of assuming these higher values, the amount of temperature rise has been considered at standard 25°C throughout. Other research[10] had considered a 50°C rise in temperature, assuming average seawater temperatures in the range of 10–20°C.

Time was also computed assuming that the complete water is raised through 25°C, as projected in Table 10. For this temperature rise, LT heat, AE and ME exhaust heat were considered. Table 10 refers to one normal light condition and two heavy weather conditions. One projection has been done based on heat availability at CSO and another at port while discharging cargo. These are the two realistic scenarios when available heat could be used for treatment. ME exhaust will be available only while the ship is sailing, but AE exhaust is available at all times. Also, a high quantum of LT heat is available only while sailing. LT heat availability varies in port around an average of 3000 kW and, while sailing, is in the range 11 000–12 000 kW.

Ballast requirements were considered for three bunker conditions, as shown in Table 10. Primarily, fuel oils constitute the major portion of bunker quantities. This is a variable depending on amount loaded and consumed and it may be assumed that the three considered situations (full, 50% and 10%) widely represent any given condition of the vessel. The treatment protocol is during sailing but the ballast would have been loaded in port. It is to be mentioned that heavy weather ballast requires additional tank space other than the segregated ballast tanks (SBT) on tankers and one or more cargo tanks will be assigned for this carriage. The treatment time computed includes this amount also, while regarding the heavy weather/gale conditions.

The time computations show 6.77 days of heat treatment for a normal ballast (light) passage and 8.67 days for full ballast in designated ballast tanks (~98 794 m^3). For treatment in port, the time projected for full ballast is 69.05h (2.88 days) but the treatment protocol has to be while ballasting. The ballasting of the complete amount is possible using the main ballast pumps, but with MSW pumps being used, this time might not be enough. Invariably, port stays for oil tankers are usually shorter than five days, whereas voyage periods depend on the terminals the vessel is operating between. For normal voyage considerations, a voyage longer than two weeks is termed as a long voyage.[22]

These projections are simplistic and do not consider heat losses from pipelines, from tank walls while the ship is moving and from stratification, etc. An analysis approximately accounting for such losses is projected in Table 11, where only LT heat availability is considered, and accounting for heat from ME and AE exhaust gases and HT systems towards these losses. The treatment time of 12–16 days indicates that heat treatment is only possible during long voyages.

	Bunkers	Total ballast	Time required for one complete circulation of ballast water	Temperature rise realised with one circulation with LT heat alone	Time required for 25°C rise	Time required for 25°C rise
		m^3	Hours	°C	Hours	Days
Light	Full	77210.73	100.3	8.7	288.14	12.01
Heavy	10%	98793.17	128.3	8.7	368.68	15.36

Table 11: Time requirements assuming heat losses from tanks

Output	Output at 40% mcr	Energy in (Q_{in})	Heat Available in exhaust gas (1 AE)	Heat Available in exhaust gases (2 AEs)	SW flow capacity	Temperature difference achieved (1 AE)	Temperature difference achieved (2 AEs)
kW	kW	kW	kW	kW	m³/h	°C	°C
1020	408	952	95.20	190.4	100	0.8	1.59
					70	1.14	2.27
					50	1.59	3.18
					25	3.18	6.37
					10	7.96	15.92
					5	15.92	31.84
					4	19.9	39.8
					3	26.54	53.07
					2	39.8	79.61
					1	79.61	159.22

SFC 0.2 kg/kW h; LCV 42000 kJ/kg

Table 12: Hypothetical heat recovery from AE exhaust on board: Flow rates and temperature rise

Heat balance exercises for a steam system were not carried out. Since the heat rejections in condensers will only augment heat for treatment and reduce treatment periods, the omission of such an exercise is not expected to negatively affect the proposal for a heat-based treatment system.

Testbed exercises

Table 12 projects the possible heat recoveries from the ship's AE exhaust gases, based on results from the testbed exercises. On vessels, exhaust gas heat recovery units are normally fitted to the ME exhaust uptakes for the production of steam, etc. If such an arrangement is fitted in the line of exhaust uptakes from the AEs, then heat recoveries might be possible. The auxiliary engines on board vessels are usually four-stroke diesels and the heat quanta and temperatures from their exhausts are lower than from two-stroke engines due to a greater amount of air mixing with the exhaust. Yet the temperature ranges and heat could be sufficient for heating ballast water. The projections are based on these possibilities and calculations were made to analyse if tangible temperature rises could be achieved.

Although it is common on ships to have three AEs, normally only two are in operation for any given high load situation. Therefore, computations at 40% output power for one AE and two AEs were done, as shown in Table 12. Flow rate is a limiting factor for recoveries. With higher outputs, say with two AEs, a temperature rise of 10°C appears attainable with 10–15 m³/h flow rates. After-peak tanks on tankers require filling from a non-hazardous zone and the water must be treated prior to discharge.[23] The heat from the AE exhaust could suffice for short-term heating of ballast water from such isolated tanks (aft peak and fore peak) because their capacities are relatively small.

Testbed data were collected over a period. The initial runs were at 60–70% of the maximum rev/min, at low loads. Then one set of data was obtained at a higher than normal rev/min but at around 28% load. Data were obtained for operations at a low load of 22% to almost a high 88% load, matching with the rated rev/min. With all such operations, the temperature rise of jacket water and heat recovered in the calorifier were observed. The water flow rate was maintained to obtain a difference of 50–80°C between exhaust gas inlet and exit temperatures. Then the flow rate of water was increased to test if more heat could be recovered from the exhaust gases, increasing the temperature difference to >100–200°C. Heat taken up by the jacket cooling water and exhaust gases were then computed.

Assuming heat available is the heat recovered by the calorifier, equation (4) becomes:

$$Q_{rec} = mg \cdot Cg \cdot \Delta Tg = mw \cdot Cw \cdot \Delta Tw \tag{5}$$

Assuming steady-state and neglecting radiation losses, heat recoveries were calculated. Input energy was calculated from equation (3). Data for runs at 76% loads only are projected in Table 13.

Exhaust gas recoveries were higher at lower loads as a percentage of fuel energy input, indicating that more heat is wasted in the exhaust gas. LCV values of marine distillates used on board are bound to be higher and so increased recoveries may be expected at even low loads on shipboard machines. As flow rates were increased, recoveries improved but stack condensation showed an increase. This is a matter of concern as corrosion effects will be enhanced at lower exit temperatures.

From a wide set of heat balance exercises, the lowest exhaust gas heat recovery recorded was 13.9% and the highest 33.36%. With data projected in Table 13, exhaust heat recovery amounts to 14.87% of the input power. Such heat recoveries from diesel engine exhaust gases have been demonstrated.[24] But considering optimum brake power, radiation losses, pinch point, condensation and corrosion, recoveries

Rev/min	Load	Fuel flow	Water flow	Calorifier water in	Calorifier water out	Jacket water in	Jacket water out	Heat lost to jacket water	Output BP	Q_{in}	Heat lost in Exhaust gases (Q_{rec})
	Nm	kg/s	kg/s	°C	°C	°C	°C	kW	kW	kW	kW
3002.73	232.54	0.00578	0.28	27.64	57.32	27.64	36.43	10.46	73.11	242.92	35.15
3012.24	225.62	0.00554	0.28	27.52	58.46	27.73	36.56	10.51	71.16	232.68	36.64
3004.62	230.73	0.00558	0.28	27.54	57.43	28.52	37.75	10.98	72.58	234.31	35.40
3483.8	204.50	0.00575	0.28	28.42	57.28	28.63	36.53	9.40	74.59	241.50	34.18

Table 13: Testbed engine: Heat recoveries from cooling water and exhaust gas

Installed output power	Power output-in port	Input energy	Heat available for recovery	Possible flow rate (20°C rise)	Possible flow rate (25°C rise)	Possible flow rate (30°C rise)
kW	kW	kW	Q_{avail}, kW	m³/h	m³/h	m³/h
	25%		14%			
1000	250	583.33	81.67	3.43	2.74	2.29
800	200	466.67	65.33	2.74	2.20	1.83
600	150	350.00	49.00	2.06	1.65	1.37
500	125	291.67	40.83	1.72	1.37	1.14
400	100	233.33	32.67	1.37	1.10	0.91
200	50	116.67	16.33	0.69	0.55	0.46
	43%					
1000	430	1003.33	140.47	5.90	4.72	3.93
800	344	802.67	112.37	4.72	3.78	3.15
600	258	602.00	84.28	3.54	2.83	2.36
500	215	501.67	70.23	2.95	2.36	1.97
400	172	401.33	56.19	2.36	1.89	1.57
200	86	200.67	28.09	1.18	0.94	0.79

Table 14: Shipboard AE: Hypothetical heat recovery and flow rates

near to 15% seem imprudent. Noting this and allowing for further losses, exhaust gas recoveries have been assumed to be 10% for all hypothetical projections in Tables 12, 13 and 14. From the heat balance calculations on the vessel's ME projected in Table 6, recoveries amount to 15.25% of the input energy (turbocharger 8.28% + EGE 6.97%) which is close to the testbed recovery of 14.87%. With the testbed arrangement being similar to the vessel's arrangement, the recovery computations appear validated.

In general, AE average outputs in ports are 25–60% of installed powers.[20] Therefore, the projections were calculated for 25% and a mid-line average of 43% of rated power operations, as shown in Table 14. These calculations with shipboard powers show effective but low flow rates. This is a crucial factor for ballast water treatment. A highest flow rate of 5.6m³/h is projected for a typical 1MW installation operating at 43% of the rated power, assuming a temperature rise of 20°C. This is sufficient for a ship operating in tropical waters where seawater temperatures are >25°C. Researchers[9] had heated seawater using steam at similar flow rates. With projections in Table 12 for similar powers and operation, the hypothetical temperature rise projected is around 15.92°C. With two-stroke engines of higher powers, the recoveries could be significantly higher. Fig 4 shows projections of possible heat recoveries from ME exhaust gases which appears considerable, from 50% mcr output itself. So, it appears certain that at such low flow rates, heat can be recovered to raise temperatures to effective levels. But for treating huge quantities of seawater these flow rates are extremely low, given the limited port stay periods. Heat treatment protocols might have to be extended during voyages and, if required, during deballasting also.

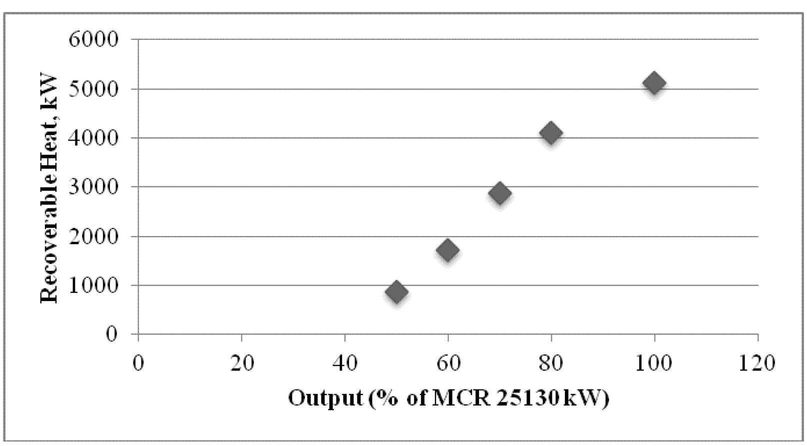

Fig 4: Possible heat recoveries from ME after turbocharger and EGE

Heat balance validation

In heat balance exercises on the testbed engine, heat losses in cooling water and exhaust gases were treated as potential heat recovery sources and indexed as a percentage of the input energies. Heat lost to jacket water amounts to an average of 4.35% of the input energy. In the study from the vessel's engine data and calculations, cooling water heat carriage was found to be about 7.3% of the input energy for CSO. For heat recovery calculations, possible heat recovery from freshwater by seawater in HT coolers (2368 kW) amounts to 5.14%, which is close to the testbed result of 4.35%. The heat balance percentages were found to be reasonably within the range projected by the energy distribution diagram (Fig 1). This shows conformance to conventional heat balance projections for turbocharged diesel engines and confirms the possibility for heat recoveries projected.

CONCLUSION

With realistic assumptions, a good approximation of heat availability has been obtained. But shipboard treatment entirely based on this waste heat will require a long time to attain effective species mortality temperatures, time which will not be available on all voyages. In terms of treatment efficacy, a comprehensive elimination of all species appears improbable with heat treatment alone. Furthermore, the corrosion potential due to low gas exit temperatures and scaling needs to be considered. It would be encouraging to assume that yearly ballast voyage periods are going to be less and hence corrosion effects will be minimal. For example, the vessel considered for the study had only clocked approximately 1824h of ballast period for the last year. This works out to 25–30% of engine operating hours, assuming vessel movement for 75% of the year. Additionally, non-corrosive and resistant materials have to be employed in the systems.

The limitations of flow and time availability do not augur favourably for treatment by harvesting waste heat. But it may be surmised that if such a high waste heat potential is utilised, it will translate to better returns over the life-time of the vessel. A combination system using less power consuming physical treatment such as filtration and waste heat-based heat treatment may prove cost effective and also overcome these limitations. In the light of emerging BWT systems being very high in terms of capital and operational costs, such investigations are worthwhile. Waste heat recovery potential from other types of ships, heat losses from the hulls, suitable treatment methods complementing heat treatment, etc, are worth probing. Since treatment technologies are still nascent, further research will prove beneficial in developing a less expensive, adequately effective BWT system.

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REFERENCES

1. Ruiz GM, Rawlings TK, Dobbs FC, Drake LA, Mullady T, Huq A and Colwell RR. 2000. *Global spread of microorganisms by ships. Ballast water discharged from vessels harbours a cocktail of potential pathogens*. Nature 408:49–50.

2. Hewitt CL, Campbell ML, Thresher RE, Martin RB, Boyd S, Cohen BF, Currie DR, Gomon MF, Keough MJ, Lewis JA, Lockett MM, Mays N, McArthur MA, O'Hara TD, Poore GCB, Ross DJ, Sotrey MJ, Watson JE and Wilson RS. 2004. *Introduced and cryptogenic species in Port Philip Bay, Victoria, Australia.*, Marine Biology, **144**:183–202.

3. Sobol Z, Wladyslaw K and Bohdan W. 1995. *System for destruction of microorganisms occurring in ballast waters*. Paper presented at IMO MEPC 38.

4. Ballast Trial Report, Ballast Water Trial on MV *Sandra Marie*, A trial of Prentice-Thornton heat treatment system for ballast water. 1997. (Report provided by Manufacturer, Hi Tech Marine International).

5. Rigby G, Hallegraeff G and Sutton C. 1999. *Novel ballast water heating technique offers cost-effective treatment to reduce the risk of global transport of harmful marine organisms*. Mar. Ecol. Prog. Ser., 191, 289–293 (Inter-Research (IR), Germany.

6. Thornton G. 2000. *Ballast water decontamination-using heat as a biocide*. Sea Australia Conference, Sydney, Australia, 1–3 Feb 2000.

7. Lloyd's Register Report, Ballast Water Treatment Technology, Current Status. June 2011. Fourth Edition. pp. 14–16.

8. Boldor D, Balasubramanian S, Purohit S and Rusch KA. 2008. *Design and implementation of a continuous microwave heating system for ballast water treatment*, Environmental Science and Technology **42**:4121–4127.

9. Quilez-Badia G, McCollin T, Josefson K, Vourdachas A, Gill ME, Mesbahi E and Frid CLJ. 2008. *On board short-time high temperature heat treatment of ballast water: A field trail under operational conditions*. Marine Pollution Bulletin **56**:127–135.

10. Mesbahi E, Norman RA, Vourdachas A and Quilez-Badia G. 2007. *Design of a high-temperature thermal ballast water treatment system*. Proc. of the IMechE, Part M: J. Engineering for Maritime Environment **221**:31–42.

11. Mountfort DO, Hay C, Taylor M, Buchanan S and Gibbs W. 1999. *Heat treatment of ships' ballast water: development and application of a model based on laboratory studies*. J. Marine Env. Eng. **5**:193–206.

12. Mountfort DO, Dodgshun T and Taylor M. 2001. *Ballast water treatment by heat – New Zealand laboratory and shipboard trials*. In: 1st International Ballast Water Treatment R&D Symposium, 26–27 March, 2001, No. 5. IMO, London, pp. 45–50.

13. Gregg M, Rigby G and Hallegraeff GM. 2009. *Review of two decades of progress in the development of management options for reducing or eradicating phytoplankton, zooplankton and bacteria in ship's ballast water*. Technical Report, Aquatic Invasions, Vol. 4, **3**: 521–565.

14. Rakopoulos CD and Giakoumis EG. 2004. *Availability analysis of a turbocharged diesel engine operating under transient load conditions*. Energy, **29**:1085–1104.

15. Hountalas DT, Katsonas CO and Kouremenos DA. 2007. *Study of available exhaust gas heat recovery technologies for HD diesel engine applications*. Int. J of Alternative Propulsion, **1**(2/3): 228–249.

16. Söylemez MS. 2003. *On the thermoeconomical optimization of heat pipe heat exchanger HPHE for waste heat recovery*. Energy Conservation and Management, **44**: 2509–2517.

17. Endersen Ø, Behrens HL, Brynested S, Andersen AB and SkJong R. 2004. *Challenges in global ballast water management*. Marine Pollution Bulletin **48**:615–623.

18. Rigby GR, Hallegraeff GM and Taylor AH. 2004. *Ballast water heating offers a superior treatment option*. Journal of Marine and Environmental Engineering, **7**: 217–230.

19. Woodyard D. 2004. Theory and General Principles. Pounder's marine diesel engines and gas turbines, Eighth Edition, p6, Elsevier Butterworth-Heinemann, Burlington, UK. 884pp.

20. Cooper DA. 2003. *Exhaust emissions from ships at berth*. Atmospheric Environment **37**:3825.

21. Zhou P and Lagogiannis V. 2001. *On board treatment of ballast water (technologies development and applications*. Proceedings of 1st International Ballast Water Treatment R&D Symposium, London, pp.35–42.

22. Suban V, Vidmar P and Perkovič M. 2010. *Ballast water replacement with fresh water—Why not? Emerging ballast water management Systems*. Proceedings of the IMO-WMU Research and Development Forum, Malmö. pp.53–76.

23. DNV Ballast Water Convention, General Information-Status, October 2010, pp.12.

24. Pandiyarajan V, Chinna Pandian M, Malan E, Velraj R and Seeniraj RV. 2011. *Experimental investigation on heat recovery from diesel engine exhaust using finned shell and tube heat exchanger and thermal storage system*. Applied Energy, **88**: 77–87 .

NOMENCLATURE

ΔTg	Temperature difference between gas inlet and outlet, °C
ΔTw	Temperature difference between water inlet and outlet, °C
Cg	Specific heat capacity of gas, kJ/kg K
Cw	Specific heat capacity of water, kJ/kg K
LCV	Lower calorific value, kJ/kg
$Q_{exhaust}$	Heat lost to exhaust gases, kW
$Q_{avail\ g}$	Energy available in exhaust gas, kW
Q_{in}	Input energy, kW
$Q_{odd\ losses}$	Heat lost in radiation, convection, etc, including non-availability, kW
Q_{rec}	Energy recovered, kW
Q_{water}	Heat lost to cooling water, kW
SFC	Specific fuel consumption, g/kW h
$W_{engine\ power}$	Productive output power of the engine, kW
$m\mathrm{f}$	Mass Rate of fuel burnt, kg/s
m_g	Mass rate of gas flow, kg/s
m_w	Mass rate of water flow, kg/s