Validation of Engine Performance for Tests on Ballast Water Heat Treatment Using Engine Waste Heat

Rajoo Balaji[1*], Omar Yaakob[2], Kho King Koh[3], Faizul Amri bin Adnan[4], Nasrudin bin Ismail[5], Badruzzaman bin Ahmad[6], Mohd Arif bin Ismail[7]

***Abstract*-heat treatment has been considered as a suitable option for treatment of ballast water. Utilising the waste heat from the diesel engine fresh water and exhaust gases would be an economic option. For recovering the heat from the exhaust gases, heat exchangers are required to be placed in their flow path. The sea water coolant after recovering heat from fresh water has to be directed to this heat exchanger for sterilisation. For testing the effectiveness of these heat recoveries on species' mortalities, a mini-scale system was arranged and tests were carried out. The engine output and other flow rates were maintained to achieve a temperature range of 55 to 80°C. Data was obtained from the sensors and probes fitted at relevant points. The engine performance was monitored with computerised control equipment. Operational data from five test runs were analysed and verified by two approaches. In the first approach, the heat recovered by the water was compared with the heat lost by the exhaust gases and the maximum variation was observed to be 3.4%. In the second approach, the input energies were computed using two different methods using data values of brake power, thermal efficiency, mass flows, calorific value and specific fuel consumption. A maximum variation of -11% was seen for only one test run, while for other tests the variation was between -0.7% to -1.7%. The values obtained from the connected probes and the computed results were thus validated and further tests on species were carried out.**

Keywords—Ballast water; heat treatment; diesel engine; heat recoveries .

I. Introduction

The International Convention for the Control and Management of Ships' Ballast Water and Sediments (hereafter, the Convention) is an initiative of the International Maritime Organisation to mitigate the harm of invasive species which was ratified in 2016 and awaits implementation [1]. Treatment of the ballast water will be the primary measure once the Convention is in full effect. Ballast water treatment systems are on offer and have been well analysed for their effectiveness and economics [2-5]. The costs of the emerging systems are high [6] and there are issues with efficiencies of the systems [7]. The physical and chemical treatment methods are employed in combination and are mostly designed on water treatment methods.

One of the earliest treatment methods advocated was by heat [8]. Heat treatment has issues of heat quanta for treating large volumes, effectiveness on bacteria types and disposal of heated waters etc. [9]. If engine waste heat is considered, the method might be economical in terms of capital and operating costs. A treatment system based on engine waste heat was proposed using the ship's waste heat [10, 11] earlier.

Waste heat recoveries from internal combustion engines have been demonstrated by various studies [12-16]. There are waste heat recoveries for many shipboard applications [17, 18].

Considering the potential and the economic advantages, in this study, the heat recovered from ship's engines and other sources was utilised for treatment of ballast water. Effectiveness of heat treatment has been tested on board using steam and species' mortalities were demonstrated by Quilez-Badia et al. [19]. A combination system using the waste heat from engine exhaust gases and other shipboard sources was proposed by Balaji and Yaakob [20].

Heat availability for such a system was carried out based on operational data of a petroleum tanker [21]. It was established that though heat was available, treatment of high volumes will require more time and extended protocols. A combination system with heat and another proven method might overcome these shortcomings.

However, the effectiveness of heat recoveries especially from the high temperature source of exhaust gases depend on a heat exchanger arranged on the path of the gases. Studies on waste heat recoveries from exhaust gases of internal combustion engines using heat exchanger placed in the gas-path are on record [22-24].

Rajoo Balaji[1*], Advanced Marine Engineering Department, Malaysian Maritime Academy, Melaka, 78300, Malaysia.
Tel: +6 06 3882214; Fax: +6 06 3876700
E-mail: rajoobalaji@alam.edu.my
Omar Yaakob[2] Marine Technology Centre, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia.
E-mail: omar@fkm.utm.my
Kho King Koh[3] Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia.
E-mail: koh@fkm.utm.my
Faizul Amri bin Adnan[4] Marine Technology Centre, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia. E-mail: faizul@fkm.utm.my
Nasrudin bin Ismail[5] Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia.
E-mail: nasrudin@fkm.utm.my
Badruzzaman bin Ahmad[6] Marine Technology Centre, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia. E-mail: badrul_man@yahoo.com
Mohd Arif bin Ismail[7] Marine Technology Centre, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, Skudai, 81310, Malaysia. E-mail: mohd_arif.ismail@yahoo.com

To verify the working of the treatment method, a mini-scale arrangement was developed. Experiments were carried out on this mini-scale system similar to the ship's arrangements which included a diesel engine and a heat exchanger [25]. The development of the system included the design and optimisation of heat exchangers. Amongst various optimisation methods [26-33], a simple approach based on Lagrangean [34] was adopted. Based on shipboard engines and heat availability, heat exchangers were designed and optimised [35-37]. Heat exchanger designs were also developed for testing at a mini-scale level. Based on the designs, a suitable heat exchanger was commercially procured for the conduct of the tests. The tests on species' mortalities were completed at the laboratory levels which included the test on the mini-scale diesel engine-heat exchanger arrangement also [25]. This paper projects the verification of the thermodynamic results of the engine used for the tests on the mini-scale system.

II. Method

A. Description of the Engine-Heat exchanger Arrangement

The engine was chosen from the available units in Universiti Teknologi Malaysia. The technical data of the engine data is shown in Table1. The engine was then mounted on the test rig facility in the Automotive Development Centre. The test rig is regularly used for testing of commercial engines and was completely wired for precise measurements of engine parameters. The calibrations of the sensors and the input points to the computerized test rig were checked prior to the start of the tests. Where required, corrections were applied to thermodynamic values as appropriate from the properties for fluids [38].

The heat exchanger was procured commercially ensuring closeness to the design values. The tanks were fabricated with mild steel. They were constructed with stiffening arrangements and given anti-corrosion protection similar to ship's tanks. The tanks were located outside the shop room and were exposed to the open ambient conditions. All other components were arranged inside.

This arrangement was similar to the physical realism of on board arrangement where the ballast tanks form part of the hull structure while engines and heat exchanger are located inside in the machinery spaces. The schematic arrangement of the engine-heat exchanger set-up is shown in Figure 1 and the major components of the arrangement are shown in Figure 2 (a) to 2 (g).

A small container was attached to the over flow point on the header/cooler arrangement to allow for any over flow due to physical expansion of fresh water as steady state conditions are reached. The arrangement of the pipelines and the components were checked for vibrations and looseness. For safety, a movable safety cage was placed around the engine arrangement so that any detached components during test runs will be restricted.

B. Tests on Mini-scale System (Engine-Heat exchanger Arrangement) Units

Prior to actual experiments, the engine was run a number of times to confirm heat availability and temperature rise of water. Fresh water was used for all such initial runs. The engine was run at various loads and speeds. Five test runs, each for a minimum 30 minutes, were considered for selecting the RPM and the load range. A torque range between 25 to 60 Nm in the range of 2800 to 3200 RPM was chosen. At these steady loads, the temperatures of fresh water, exhaust gases, lubricating oils and other parameters were checked.

The next stage involved tests employing typical species. Artificial sea water was prepared and employed to represent shipboard ballast water. For the tests which will involve the species and sea water, the capacities and flow rates were determined. Firstly, the distance traversed by water in the shell circuit was evaluated based on the tube length and shell diameter of the heat exchanger. Time phase in the heat exchanger was estimated for possible velocities. These plots are shown in Figure 3. Various flow rates were then checked for attaining the temperature range of 55 to 75ºC, which was the range determined from the design calculations.

Secondly, the total distance the water has to flow to complete the circuit from one ballast tank to another was estimated to be around 20 to 22m. With the engine running, the approximate time for the full sample volume of 100 litres to complete this circuit was estimated.

The flow rate was adjusted to 20 litres per minute so that 100 litres of sample was transferred in five minutes. Considering the dimensions of the heat exchanger and the pipeline diameter (50.8mm), the average residence time was computed to be minimum 17.38 seconds and a maximum of 43.45 seconds.

The flow velocity was estimated to be in the range of 0.04 to 0.1m/s for the maximum and minimum time periods respectively from the plots in Figure 3. The minimum time of 17.38 seconds was slightly higher than 15.1 seconds, which was obtained from the calculations for optimising the shipboard heat exchanger.

After the initial warming up of the engine and checking the system for leaks etc., the tests were performed. The tests involved introduction of species in the tanks, circulating the water and collecting samples to study the species' mortalities. These tests are not reflected herein.

For a constant speed of 3000 RPM, engine parameters were considered for progressive loads of 30, 40, 50 and 60Nm and averaged for each test run. The engine data was recorded automatically in the computer at every 20 second intervals. For computations, data during the five minutes of each test run was considered.

The heat apportioned to the exhaust gases, fresh water and sea water were computed as the product of the mass flow, specific heat and the temperature difference as,

$$Q = m \cdot C_p \cdot \Delta t \qquad (1)$$

The brake power was obtained from the computer as also verified with the fundamental equation using the RPM and the torque.

$$Brake\ Power = \frac{2\pi NT}{1000} \quad (2)$$

The input energy to which all the allocations were indexed with was computed from the product of the fuel flow and the lower calorific value.

$$Q_{in} = m_f \cdot LCV \quad (3)$$

III. Results and Discussion

Data was averaged from four to five sets of the recorded values and are tabulated in Table 2. The heat recoveries (energy allocations) from fresh water (header/cooler) and exhaust gas (heat exchanger) were computed based on mass flow of salt water at 0.34 kg/s. All other values in Table 2 were averaged from those recorded in the computer.

The recoveries and other significant parameters are plotted in Figure 4(a) to Figure 4(d). The figures show that the recovery/energy distributions, thermal efficiencies and Brake Specific Fuel Consumptions (BSFC) are quite consistent for all the tests. Further, the averages of all tests were computed and the energy distribution as a percentage of input energy was calculated. The distribution is shown in Figure 4(d). After accounting for recoveries from fresh water (13%), exhaust gas (59%) and the shaft power (20%), the remaining (8%) was attributed to losses in lubricating oils and radiation. The fresh water and radiation losses confirm with representative figures for naturally aspirated engines.

As seen from Figure 4(a), the thermal efficiencies have been low but the fuel consumed for generating unit power has been quite consistent. This is due to the low engine loads. Since targeted temperature ranges were obtained with such levels, the loads were not increased. Consequently, higher heat discards in the exhaust gases were seen as shown in Figure 4(d). Waste heat from combustion processes burning gasoline could be in the range of 30 to 40% [39] and heat discards could be in the range of 25 to 45% of the input energies [40-42]. The heat discards in exhaust gases are comparatively higher considering 4-Stroke naturally aspirated compression ignition engines.

On ship board engines with higher volumes and flow rates, the engine loads will be proportionally higher and recoveries would be considerable. Further, the shipboard engines would be turbocharged and the shaft power and thermal efficiencies also would be higher. For the tests, the shaft power and heat recoveries were also consistent as shown in Figure 4(b) and Figure 4(c).

The correctness of these computations was verified by two approaches. In the first approach the heat recovered by salt water was compared with heat discarded by the exhaust gases. The heat recovered by the salt water was computed from the mass flow, specific heat and temperature increase from inlet to outlet of the heat exchanger. Appropriate corrections were applied to the specific heat values.

The heat discarded by the gases was computed from the mass flow, specific heat capacity and the temperature change across the heat exchanger. The mass flow of gases was based on the volume displacement of the engine. The density and specific heat capacities were interpolated for the average temperature. Then the mass flows and discarded heat quanta were calculated. This was done for each of the tests. The variations between heat recovered and discarded are negligible as shown in the fourth column of Table 3.

In the second approach the input energy values obtained from three different calculations were compared as shown in Table 3. The input energies shown in Table 3 (5th column) were computed from the recorded values of brake thermal efficiency and the brake power and these were taken as the reference. These were compared with input energy values computed (Table 3/ 6th column) from fuel flow and the LCV for the fuel obtained from BS5514 Part I. These were also compared with input energies obtained (Table 3/ 8th column) as a product of output shaft power (BP), BSFC and LCV.

As seen from Table 3, maximum variations are recorded for Test Run 2. But the averages of the variations (4th, 7th and 9th columns) were quite low at 0.1%, -0.4% and -3.3% respectively. Hence the values were considered valid and true. The trueness of the temperature values recorded by the electronic probes was also confirmed with these exercises.

IV. Conclusion

The data from engine test runs intended for testing the mortality of species due to the high temperature effects of exhaust gases have been verified. The variations in heat balance values of input, brake power and specific fuel consumption were negligible. The trueness of the collected data and so the reliability of the equipment were verified using by cross checking the values with different approaches.

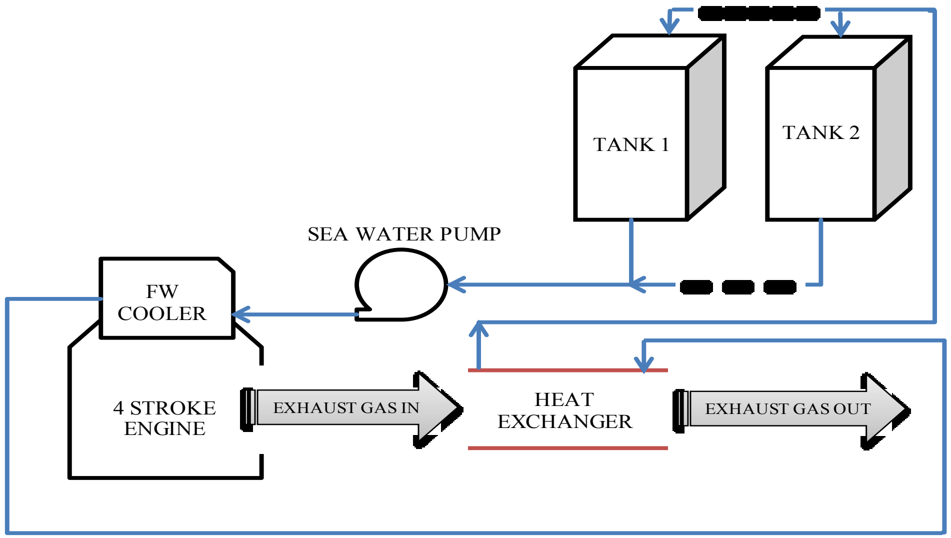
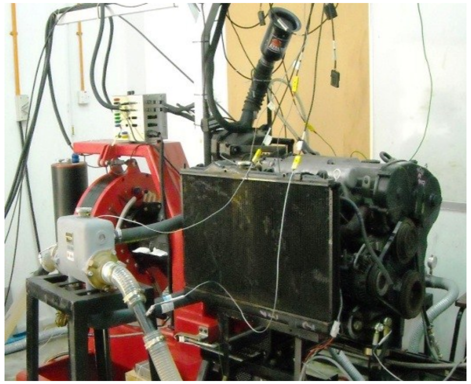


Figure 1. Schematic layout of the test arrangement



(a) Engine, dynamometer, FW cooler

(b) Engine with safety cage

(c) Engine control panel

(d) Ballast water tanks

(e) Fresh water header/cooler

(f) Centrifugal pump

(g) Heat exchanger

Figure 2. Major equipment used for the experiments

TABLE 1.
TECHNICAL DATA OF ENGINE

ITEM	SPECIFICATION/DATA
Type	4-Stroke, Mitsubishi 4D 68, Naturally aspirated
Bore	82.7 mm
Stroke	93 mm
Swept volume	1.998 cc
Output	69 kW at 4500 RPM

TABLE 2.
TEMPERATURES AND HEAT RECOVERIES

TEST RUN	SALT WATER			EXHAUST GAS		ENERGY ALLOCATIONS					
	INLET, FW COOLER (°C)	OUTLET, FW COOLER (°C)	OUTLET, HEAT EXCHANGER (°C)	INLET, HEAT EXCHANGER (°C)	OUTLET, HEAT EXCHANGER (°C)	FW (KW)	EG (KW)	BP (KW)	INPUT (KW)	BSFC (KG/KWH)	$\eta_{THERMAL}$
1	28.7	34.2	62.8	314.3	71.5	7.5	38.9	14.1	69.7	0.421	20.3
2	29.2	35	62	309	71.75	7.9	36.8	12.6	63.3	0.471	19.8
3	28.2	34.3	59.3	298.8	69.8	8.3	34.0	14.1	63.7	0.383	22.2
4	28.3	34.4	64.3	280.3	73.5	8.3	40.7	11.8	61.8	0.450	19.1
5	28.2	34.3	63	277.5	75.3	8.3	39.1	11.8	60.9	0.442	19.4

NOTES:
1. AMBIENT CONDITIONS:
TESTS 1 & 2: P_{ATM} 1.0063 BAR; T_{ATM} 27 TO 31.2°C; RELATIVE HUMIDITY 77%
TESTS 3, 4 & 5: P_{ATM} 1.0049 BAR; T_{ATM} 26.7 TO 31°C; RELATIVE HUMIDITY 67%
2. FUEL (HSD) LCV: 42700KJ/KG; DENSITY: 0.86KG/LITRE
3. SALT WATER DENSITY: 1.022; SALINITY: 30000 TO 31000PPM; SPECIFIC HEAT: 4.001KJ/KG K

TABLE 3.
HEAT BALANCE VERIFICATION

TEST RUN	HEAT RECOVERED BY SALT WATER (KW)	HEAT DISCARDED BY EXHAUST GAS (KW)	% VARIATION	INPUT ENERGY				
				(BP/ $H_{THERMAL}$)	(MASS FLOW x LCV)	% VARIATION	(BP x SFC x LCV)	% VARIATION
1	38.9	37.3	-4.3	69.7	68.6	1.6	70.6	-1.4
2	36.8	36.5	-0.9	63.3	67.4	-6.4	70.3	-11.0
3	34.0	35.2	3.4	63.7	63.2	0.9	64.2	-0.7
4	40.7	40.8	0.3	61.8	61.2	1.1	62.9	-1.7
5	39.1	39.9	2.0	60.9	60.4	0.8	61.8	-1.6

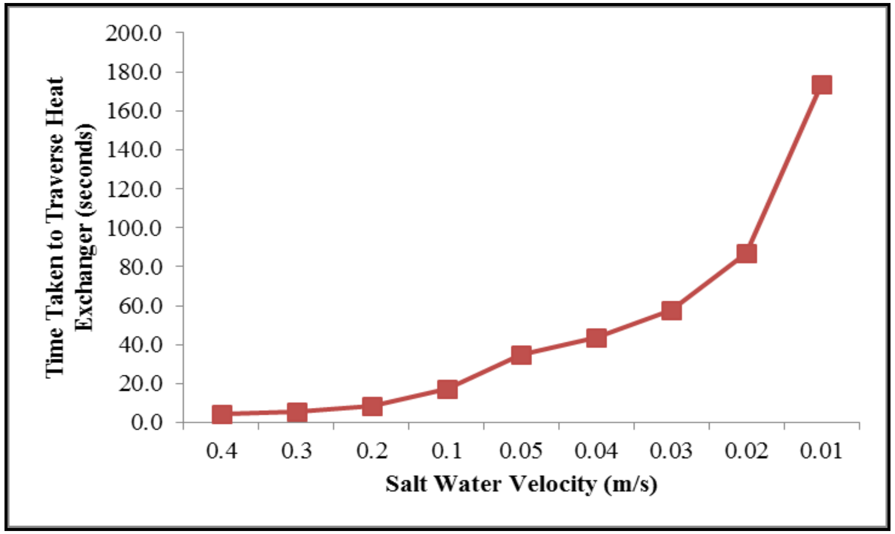
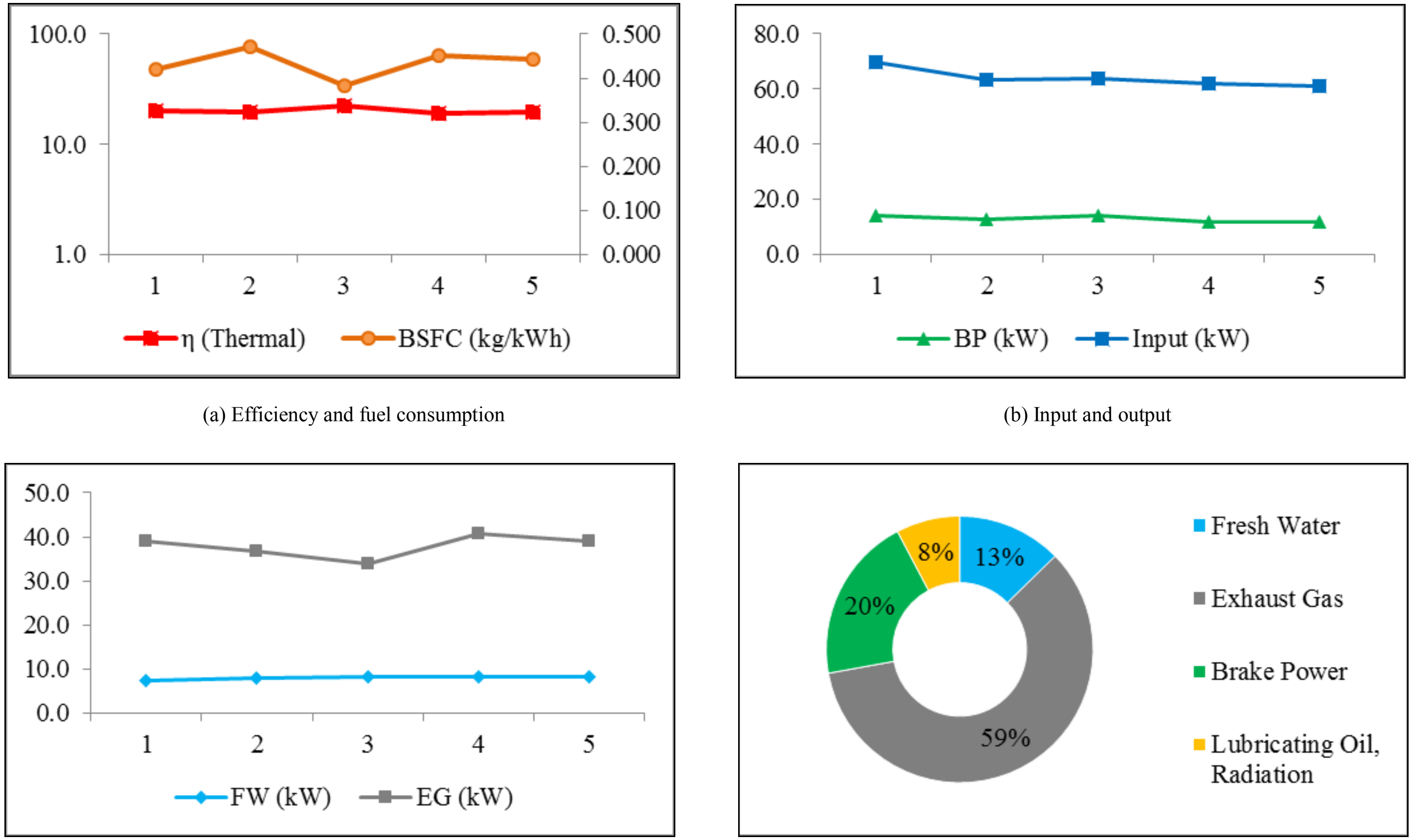


Figure 3. Heat exchanger time phase for different flow velocities



(a) Efficiency and fuel consumption

(b) Input and output

(c) Heat recoveries

(d) Energy distribution

Figure 4. Engine test runs: Heat distribution characteristics and performance

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NOMENCLATURE

BP	Brake Power
BSFC	Brake Specific Fuel Consumption
EG	Exhaust Gas
FW	Fresh Water
HSD	High Speed Diesel
LCV	Lower Calorific Value
RPM	Revolutions per Minute
SFC	Specific Fuel Consumption

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