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COMPARISON OF HEAT EXCHANGER DESIGNS FOR SHIP BALLAST WATER HEAT TREATMENT SYSTEM

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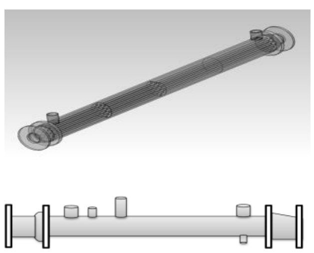
Graphical abstract



Abstract

Sterilisation by heat can be a capital treatment for ballast water and waste heat from ship's engines will be a good resource. Based on the waste heat availability on an operational tanker, a ballast water treatment system was envisaged including a shipboard heat exchanger for waste heat recovery. To verify the heat availability and the species' mortalities, test rigs were arranged similar to shipboard arrangement. For assessing the smaller heat exchangers for the tests, designs were developed using Bell-Delaware approaches based on the shipboard heat exchanger design. The thermodynamic and geometric values were computed and the features of the commercially available and fitted heat exchangers were compared with the developed designs. Two commercially procured heat exchangers fitted on two separate engine test rigs were used for tests. The designs of commercially procured heat exchangers were close to the developed designs and were found to be suitable for the tests planned.

Keywords: Ballast water treatment, exhaust gases, waste heat recovery, heat exchanger design

1.0 INTRODUCTION

The International Convention for the Control and Management of Ships' Ballast Water and Sediments is yet to be ratified. As of 30 June 2015, 44 countries totalling 32.86 % of the global tonnage have become signatories, whereas it requires 30 countries totalling 35% of the global tonnage to bring the Convention into enforcement [1]. Apart from other issues, the management of ballast water by treatment is yet to be resolved.

Treatment of ballast water implies getting rid of the microorganisms present in the ballast water. Many of these microorganisms turn invasive when they are released in waters which are not native to them. They are generally referred to as 'invasive species'. Ballast water treatment systems aim to kill the species or render them to a non-viable state.

Current ship ballast water treatment technologies employ physical and chemical methods in combination [2, 3] and are mostly adopted from land-based water treatment methods. Heat treatment has been considered as a candidate method [4] and has been researched upon [5 - 10]. The major issues with heat treatment are the heat availability for treating large quantities and efficacies for treating certain types of bacteria. Other concerns include discharge of heated water, effect on tank coatings etc. Heat treatment in combination with another proven system might be able to overcome these problems. Gregg *et al*. [11] had suggested heat treatment could be used in combination with cavitation, UV and ultrasonic methods. Steam [12], microwave [13] and engine heat have been the suggested heat sources. One commercially available thermal system is based on engine jacket water heat [2, 3]. In addition, if heat from engine exhaust gases and other sources could be harnessed, the option would be more economical.

Balaji and Yaakob [14] had analysed heat availability on board an operational tanker, considering all available waste heat. The range of temperatures on shipboard diesel engines for propulsion and auxiliary purposes would be around 200-450º C, depending upon the loads.

Motor ships have greater waste heat potential in the engine exhaust gases. For harvesting this heat, a heat exchanger design was optimised using the Lagrangean methods [15].

A combination treatment system was envisaged including the optimised heat exchanger. The envisaged system was developed based on the existing system on board an operational vessel.

The working of the system was checked on two separate engine test rig arrangements. Two heat exchanger designs were developed to suit the engines and the identified heat duties. Heat exchangers suitable for the duties were commercially procured and fitted. The paper reports on the development of the heat exchanger designs for the tests and comparison with the fitted heat exchangers.

2.0 METHODOLOGY

The designs of heat exchangers for experiments were developed on the basic characteristics of this optimised design. The optimised shipboard design was developed for harvesting exhaust heat from a 2-stroke engine.

However, due to non-availability of marine 2-stroke diesel engines at laboratory levels, 4-stroke engines were chosen. Moreover, the control and measurement systems (Dynamometer etc.) available in workshops were well suited for 4-stroke engines only. The data for design were based on the parameters of 4-stroke engines employed for experiments. Since only temperatures and heat recovery percentages (of input energies) were significant for the experiments, the use of 4-stroke engines was justified.

Commercially available heat exchangers were based on the duties computed for these engines. The process of selection and procurement of the heat exchangers is shown as a flowchart in Figure 1.

The approach to the optimised shipboard design is as follows. The heat input (input energy) for a certain output power can be computed from

$$Q_{in} = W_{engine\ power} \cdot SFC \cdot LCV \qquad (1)$$

The heat absorbed by the engine cooling water systems and the structure (engine parts), losses in radiation, lubricating oil and exhaust gases, as also the useful work realised on the shaft are accountable in this input energy.

The heat energy absorbed by various systems will be the product of the mass flow rate and specific heat capacity and temperature rise. The heat balance and hence the heat availability for treatment was computed based on this.

Heat lost to exhaust gases were higher indicating a greater potential for waste heat recovery.

The analyses based on test bed heat recovery experiments by Balaji and Yaakob [14] had recorded a maximum recovery of 33 % from exhaust gases. Cooling water had accounted for 5.38 % and shaft power accounted for 43.86 % of the input energies.

Assuming steady state and neglecting heat losses due to radiation, the heat available in exhaust gases would be

$$Q_{exg} = m_g \cdot C_g \cdot \Delta T_g \qquad (2)$$

In the shipboard arrangement, heat from exhaust gases would be utilised partly by the economiser before entering the ballast water heater. Allocating for such recoveries and the unrecoverable energy, a lower potential was envisaged for the heat exchanger.

Considering these, the heat exchanger intended for shipboard use was designed for maximising this recovery to 6-10 % of the input energy.

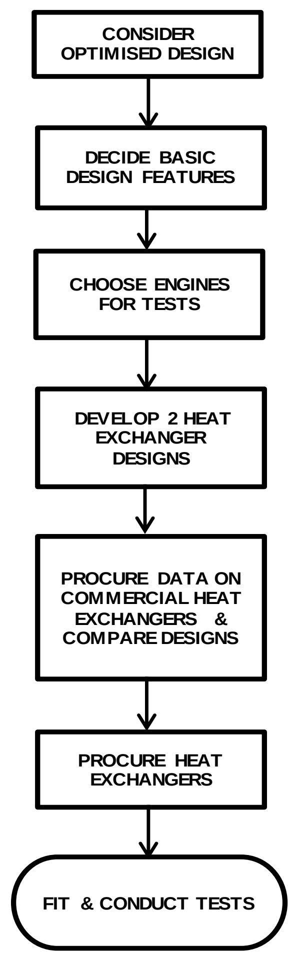


Figure 1 Flowchart of process

The chosen design was optimised and design verifications were done keeping cost as the objective function [15, 16]. Based on this design two more designs were developed using Bell-Delaware [17] approaches.

The steps of approaches to the designs are shown in Figure 2. The geometric values were also computed applying TEMA [18] standards.

The equations used for the design calculations were used from References [19 - 21]. The dimensionless factors of Reynolds Number, Prandtl Number, Nusselt Number etc., were derived for the test rig design based on the engine data.

The suitability for the tests was determined by comparing the developed designs with the design data obtained for the commercially available heat exchangers. The main features of the shell and tube design were in agreement. Exhaust gases passed through the tubes and water flowed through the shell circuit.

The general scheme of the commercially procured and fitted heat exchangers is shown in Figure 3. Further, the designs developed were geometrically created using SolidWorks® software. Figures 4, 5 and 6 show the wire diagrams of various designs developed using SolidWorks®.

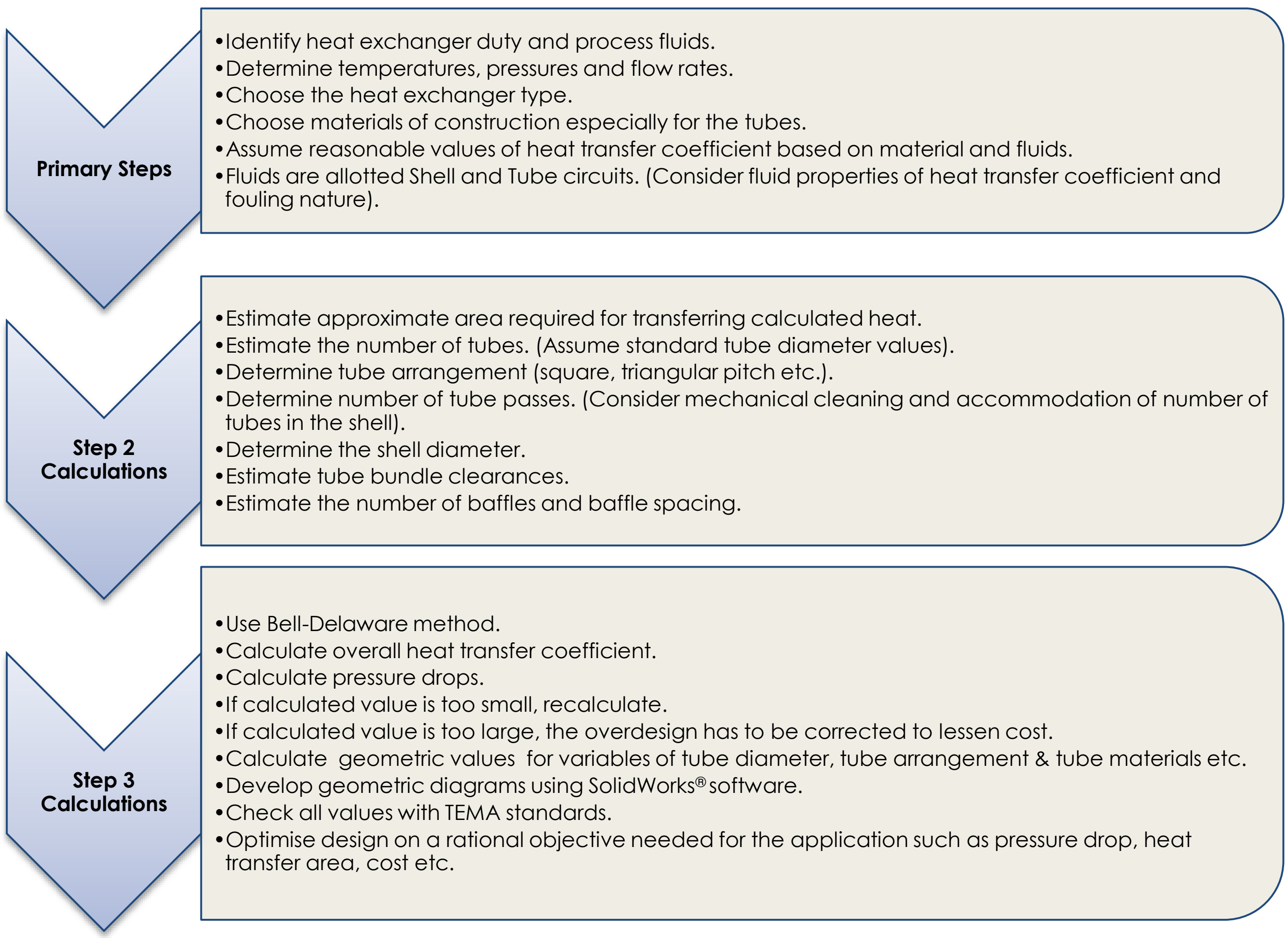


Figure 2 Methodology steps for heat exchanger designs

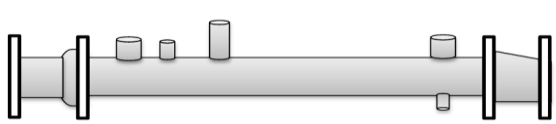


Figure 3 Commercially procured heat exchangers: General scheme

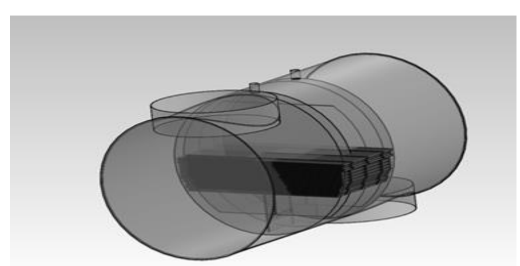


Figure 4 Optimised heat exchanger for shipboard use (part tube stack shown)

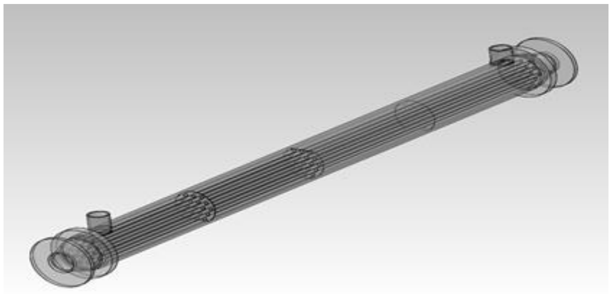


Figure 5 Developed design: 3 baffle arrangement

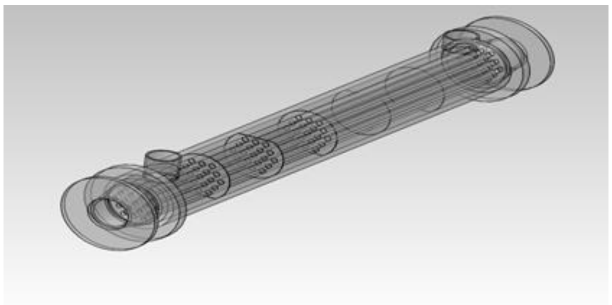


Figure 6 Developed design: 5 baffle arrangement

3.0 RESULTS AND DISCUSSION

3.1 Heat Exchanger Design for Verification of Heat Availability

The first commercially procured heat exchanger was fitted in the path of the exhaust gases of a test bed diesel engine. The data calculated for developing the design are shown in Table 1.

Table 2 shows the comparison between the developed design and the commercially procured and fitted heat exchanger.

Table 1 Heat exchanger for heat availability tests: Data calculated for developing the design

	Tube Side	Shell Side
Mass flow	254.5 kg/h	423.2kg/h
Mean temperature	368° C	57.53° C
Density	1.203 kg/m³	984.9 kg/m³
Specific heat capacity	1147 J/kg K	4181 J/kg K
Thermal conductivity	0.04812 W/m K	0.648 W/m K
Dynamic viscosity	0.029993 mPa s	0.4841 mPa s
Velocity	21.25 m/s	0.02051 m/s

There were few cognizable differences between the developed design and the heat exchanger commercially procured and fitted in place.

The heat duty computed for the designed exchanger was for 14- 20 % of input energy. The temperature ranges assumed for the developed design were from the maximum and minimum values obtained from the engine manuals. Based on the input energy and recovery calculations of the engine used in the experiment, the heat duty was found to be about 5% greater than required.

Table 2 Heat availability tests: Comparison of selected features of designed and fitted heat exchangers

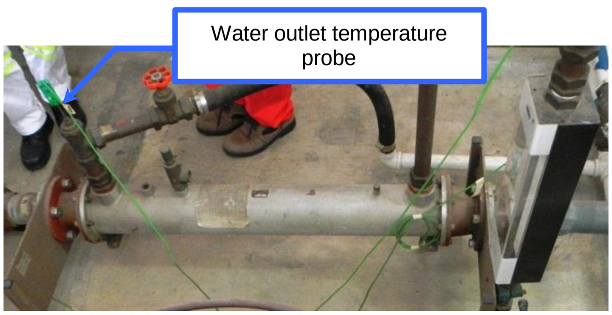
	Designed	Fitted (commercially procured)
	Counter flow	Counter flow
Heat duty	46.72 kW	58.3kW
Exhaust gas in/out	656/80° C	600/166° C
Water in/out	10/105.1° C	80/85.7° C
Overall heat transfer coefficient	89.49 W/m²K	121 W/m²K
LMTD	233.1° C	239.51° C
Pressure drop (shell side)	178.41 Pa	4180 Pa
Pressure drop (tube side)	2606 Pa	1370 Pa
Mass flow (shell side)	423.2 kg/h	9225 kg/h
Mass flow (tube side)	254.5 kg/h	402 kg/h
Area	2.24 m²	2.09 m²
Nozzle inside diameter (shell side)	54.5 mm	50.8 mm
Nozzle inside diameter (tube side)	260.4 mm	82 mm
Tube dimensions	1169 x 10 x 7.6 mm	1028 x 12 x 10.6 mm
Number of tubes	61	54
Tube arrangement	21Triangular 60°	21Triangular 60°
Tube material	Steel	316 L SS
Number of baffles	3	5
Shell outside diameter	219.1 mm	141.3 mm
Shell inside diameter	206.5 mm	131 mm

In the heat recovery experiments, a maximum of 33% recoveries were recorded with increased condensation in the exhaust gases. The overall heat transfer coefficient was lesser in the developed design. This was due to assumption of greater fouling factor. The commercially procured and fitted exchanger had about 35% greater heat transfer coefficient primarily due to the choice of stainless steel for tube material and lesser tube thickness of 0.7mm.

The figures for pressure drops provided by the manufacturer were assumed to be for the heat exchanger only and excluding the pressure drops in the nozzles and other fittings. Comparatively, the shell side pressure drop in the commercially procured and fitted design was higher due to more number of baffles and higher mass flows. The tube side pressure drop was higher for the developed design as wider nozzles were designed for entry and exit of the gases to lessen back pressure on the turbocharger. Also, the tube side mass flow range was high allowing for slightly higher velocities. The developed design had assumed the tube side velocity at 21.25m/s still within the range for tubular exchangers (30m/s for flue gases) [22].

The designed area was found to be greater by 7.2% than the commercially procured and fitted design. The choice of stainless steel, tube thickness and improved heat transfer coefficient and hence lesser number of tubes are reasons which can be attributed to the compactness of the commercially procured design. Comparing the area densities of the heat exchangers, the commercially procured and fitted design had a compactness of about 2.63 times than the designed exchanger.

Figures 7(a) and (b) show the commercially procured heat exchanger fitted in place and a view of tube ends. The overall length of the heat exchanger was 1232mm. The sensor probes and the flow-meter for measurement of the temperatures and flow rates are seen. The fitted design was well suited for the planned tests.



(a)

(b)

Figure 7 (a) Fitted heat exchanger with flow meter and fluid connections and (b) end view of tubes (Malaysian Maritime Academy, Marine Engineering Workshop)

3.2 Heat Exchanger Design For Verification Of Species' Mortalities

The second heat exchanger was commercially procured and fitted in the path of gases of a naturally aspirated diesel engine mounted on the test bed for testing the mortalities of species. Table 3 shows the data computed for the developed design and Table 4 shows the comparison between the commercially procured and the developed designs. Referring to Table 4, the values for the commercially procured and fitted design were obtained from the manufacturer's data whereas the values for the developed design were based on the operational values. The heat duty based on heat recoveries for the developed design was calculated for approximately 30-40% of the input energies to give wider scope for recoveries during the experiments.

Table 3 Data calculated for developing the design

	Tube Side	Shell Side
Mass Flow	270kg/h	1500kg/h
Mean Temperature	350°C	42.84°C
Density	0.2812kg/m³	1017kg/m³
Specific Heat capacity	1142J/kg K	4001J/kg K
Thermal Conductivity	0.04691W/m K	0.6314W/m K
Dynamic Viscosity	0.02934mPa s	0.67mPa s
Velocity	28.36 m/s	0.03562m/s

Table 4 Heat exchanger for species mortality tests: Comparison of developed and fitted heat exchanger designs

	Developed	Fitted (commercially procured)
	Counter flow	Counter flow
Heat Duty	42.8kW	43kW
Exhaust Gas in/out	600/113.3°C	600/120°C
Water in/out	30/65°C	80/86°C
Overall Heat transfer coefficient	15.01W/m² K	12W/m² K
LMTD	229.7°C	239.5°C
Pressure drop (shell side)	72Pa	697Pa
Pressure drop (tube side)	648Pa	920Pa
Mass Flow (shell side)	1500kg/h	16605kg/h
Mass Flow (tube side)	270kg/h	270kg/h
Area	1.79m²	1.68m²
Nozzle Inside diameter (shell side)	54.5mm	50mm
Nozzle Inside diameter (tube side)	70.5mm	66mm
Tube Dimensions	1580 x 12 x 8mm	1530 x 10 x 8mm
Number of Tubes	31	35
Tube arrangement	21 Triangular 60°	Triangular 60°
Tube material	Stainless Steel	316L Stainless Steel
Number of baffles	3	4
Shell outside Diameter	115.3mm	114.3mm
Shell Inside Diameter	105.3mm	104mm

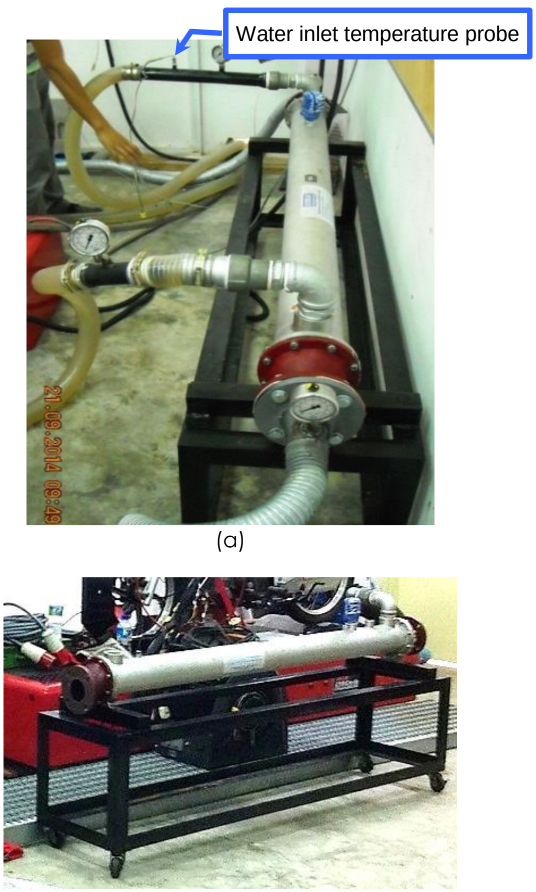
The commercially procured and fitted heat exchanger had similar rating as the developed design. The overall heat transfer coefficient, area, tube dimensions and few other geometric parameters were close to the developed design values.

The figures for pressure drop provided by the manufacturer were assumed for the heat exchange section only, excluding the drops in the nozzles and other fittings. Comparatively, the shell side pressure drop in the commercially procured and fitted design was higher due to larger number of baffles and high mass flows. The tube side pressure drop was higher for the developed design as wider nozzles were designed for entry and exit of the gases to lessen back pressure on the turbocharger. For the experiments, a naturally aspirated engine was employed.

The designed area was greater by 6.5% than the commercially procured design. The choice of superior grade stainless steel improved the compactness of the fitted design.

Comparing the area densities of the heat exchangers, the commercially procured heat exchanger had a compactness of about 0.83 (~1) times than the designed exchanger.

Figures 8(a) and (b) show the commercially procured heat exchanger fitted in place. The overall length of the heat exchanger was 1700mm. The pressure gauges and temperature probes (labelled) fitted on the branch lines are seen. The fitted design had the similarity to simulate effects similar to the shipboard optimised design. It was found suitable for the intended tests.



(a)

(b)

Figure 8 Heat exchanger (fitted) for testing species' mortalities with (a) connections and (b) without connections on bench (Universiti Teknologi Malaysia, Automotive Development Centre)

4.0 CONCLUSION

In marine applications with defined parameters, the sizing and rating of heat exchangers can be determined with simpler techniques. Bell-Delaware approaches consider all flow losses, especially on the shell circuit where the flows are mixed in a shell and tube heat exchanger. Applying same, with numerous simple iterations the optimum values can be found for various constraints. In these exercises heat exchanger designs have been developed based on such conventional methods. The fitted designs were found suitable for the thermodynamic requirements while analysing the heat availability and the species mortalities due to heat effects. The studies illustrate a practical approach to verification of heat exchanger designs.

NOMENCLATURE

C_g	Specific Heat, Exhaust gas	kJ/kg K
Q_{exg}	Heat lost to exhaust gases	kW
Q_{in}	Input Power (Input energy)	kW
$W_{engine\ power}$	Output Power	kW
m_g	Mass flow of exhaust gas	kg/s
ΔT_g	Temperature Difference	°C
LCV	Lower Calorific Value	kJ/kg
SFC	Specific Fuel Consumption	g/kWh

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