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Research Note

Optimisation of a waste heat exchanger for ballast water treatment

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Abstract. Ballast Water Treatment systems, which are type approved and commercially available, require improvements to meet stricter standards, and heat treatment could be a viable additional option. Considering the waste heat potential on a ship, a system harvesting the engine exhaust heat may be envisaged for which a heat exchanger could be vital. Design optimisation of a heater, employing the exhaust gases of an engine as utility fluid, and ballast sea water as the process fluid, was achieved using Lagrangian methods, keeping the annual cost as the objective function. Limiting the number of variables, optimal values were calculated with cost considerations for utility fluid and also pumping costs for utility and process fluids. In all, four optimum designs and three comparative designs were developed. Heat balance data from an operational tanker, specific fuel consumption values and fuel costs were considered for the design. The thermodynamic and geometric designs were worked out using computer based software for a comparison. Designs were compared on the basis of annual cost, optimum exit temperature of shell side fluid, optimum mass flow of tube side fluid and heat exchanger effectiveness. It is demonstrated that an optimal heat exchanger design can be obtained with simple optimisation procedures.

1. Introduction

Transportation of non-native species in the ballast water of ships is an environmental issue for which industry has yet to find a suitable solution in terms of efficiency and economy of cost. While the ballast water convention is nearing full ratification, Ballast Water Treatment (BWT) systems based on various technologies are also emerging. All available systems are compliant to requirements and yet are not geared up to meet the stricter standards of some administrative regulations. The Environmental Protection Agency of the United States [1] assessed and reported that only five combination systems show promise of meeting stricter standards, and has suggested that improvements may be achieved with innovative combinations. Heat treatment has been researched and some heat-based systems are already on offer. Though apprehension regarding sufficient waste heat availability for treatment prevails, research into harvesting shipboard heat is warranted, considering the high cost of other BWT options. Research on heat treatment has shown species mortality at low to medium temperature ranges [2-5]. Also, Mesbahi et al. [6] modelled and tested a thermal system at high temperature ranges for heat treatment of ballast water.

Typical shipboard waste heat sources include cooling water and steam system rejections. The sea water receiving all these rejections can be further heated using the engine exhaust heat. Sea water can be routed from the main sea water circuit to the ballast

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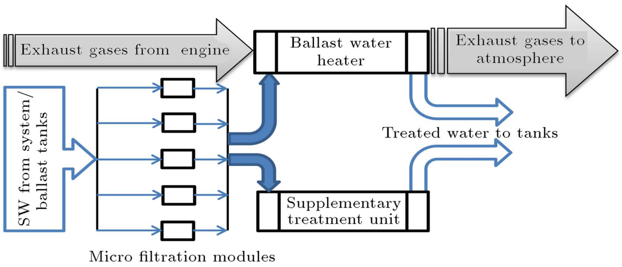

Figure 1. Schematic diagram of the treatment system.

tanks, as also re-circulated from the tanks. A simple layout of such a system is shown in Figure 1.

With respect to waste heat recoveries, the potential for increased recovery has been demonstrated using availability analysis [7]. Heat recovery from exhaust gases to lower fuel consumption and improved efficiency has been researched [8]. Models have been proposed for optimised heat exchanger designs that enhance waste heat recovery [9]. The efficacy of harvesting this heat requires a well-designed heat exchanger. Various methods for optimising heat exchanger designs have been suggested, including selections based on genetic algorithms [10] and multi objective optimisation [11] etc.

Many optimisation methods are oriented towards processing plants involving a network of heat exchangers and other components. Marine heat exchangers are mostly singular and independent of other processes, and so, simpler optimisation techniques can be employed. Similar rational approaches with engineer defined parameters and constraints have shown satisfactory results [12]. The objective of this exercise is to design a heater for heating ballast sea water, employing waste heat from exhaust gases. The design values were obtained using Lagrangian equations, keeping the annual cost as the objective function. The choice of final design was based on minimum annual cost and scope for temperature and mass flow improvement.

2. Methodologies

2.1. Basic approaches to optimisation

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

$$Q_{\text{in}}=Q_{\text{exhaust}}+Q_{\text{water}}+Q_{\text{odd losses}}+W_{\text{engine power}}. \tag{1}$$

The three thermodynamic losses would include the heat lost to the exhaust gases, cooling water, and odd losses comprised of friction, radiation, and convection etc. The heat input for a certain output power can be computed, otherwise, from the engine power output, Specific Fuel Consumption (SFC) and Lower Calorific Value (LCV).

$$Q_{\text{in}} = W_{\text{engine power}} \bullet \text{SFC} \bullet \text{LCV}. \tag{2}$$

On board ships, typical recovery from exhaust gases is from turbochargers and exhaust gas boilers. The ballast water heater is envisaged after these recoveries and a conservative 10% of recovery is assumed. For the specified duty, a single pass, shell and tube heat exchanger, with baffles, having a counter flow pattern, is considered. The fluids were assumed to undergo no phase change. Other assumptions included steady state operation, constant specific heat for the fluids, constant over all heat transfer coefficient and negligible heat losses [13]. The heat duty, inlet temperature of the shell side, cold fluid, and the tube diameters were considered known. The objective function was the annual cost. The objective function can be written as follows [14,15]:

$$C_T = A_o K_F C_{Ao} + m_u H_y C_u + A_o E_i H_y C_i + A_o E_o H_y C_o. \tag{3}$$

The relationship for the thermal design is based on the enthalpy rate equations for single phase fluids, where $j = i, o$ denotes each of the fluids inside and outside the tubes [16]:

$$q = q_j = \dot{m}_j \Delta \mathbf{h}_j = (\dot{m} c_p)_j \, \Delta T_j = (\dot{m} c_p)_j \, |T_{j,i} - T_{j,o}| \,. \tag{4}$$

The heat balance of hot and cold streams can be protracted from Eq. (4) as follows:

$$Q = m_c \bullet C_{pc}(t_2 - t_1) = m_h \bullet C_{ph}(T_1 - T_2). \tag{5}$$

The mass flow of the fluids can be obtained from:

$$m_u = \frac{Q}{C_{ph}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} \qquad \text{for exhaust gases,}$$

or:

$$m_u = \frac{Q}{C_{pc}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} \qquad \text{for sea water,} \tag{6}$$

where $\Delta t_1 = T_2 - t_1$ and $\Delta t_2 = T_1 - t_2$ are the respective temperature differences between fluids in the counter flow pattern at entry and exit.

Then, the fundamental equation for heat transfer is given by:

$$Q = UA\Delta T_{lm}. \tag{7}$$

The optimisation exercise was treated as both a rating and sizing problem. Determining the area, A, and the overall conductance, UA, were necessary. Of the variables identified from the enthalpy rate equations, the area and overall heat transfer coefficients were treated as unknowns. The overall heat transfer coefficient can be calculated from:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_i} \bullet \frac{D_o}{D_i} + R_{fo} + R_{fi}. \tag{8}$$

The overall heat transfer coefficient equation is further simplified combining the fouling factors:

$$U_o = \left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)^{-1}. \quad (9)$$

The Logarithmic Mean Temperature Difference, LMTD, is calculated from:

$$\Delta T_{lm} = \frac{F(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\frac{T_1 - t_2}{T_2 - t_1}\right]}. \quad (10)$$

The heat duty is then:

$$Q = F U_o A_o \frac{(\Delta t_2 - \Delta t_1)}{\ln(\Delta t_2/\Delta t_1)}. \quad (11)$$

A correction factor, F, is applied for counter current heat exchangers, depending on the number of tube and shell passes of the process fluids. Though the tube side fluid flow may be assumed unidirectional, the shell side flow is rather mixed, due to the guided flow of the baffles. But, for counter flow heat exchangers, this can be assumed as unity.

Eq. (11) can be written as:

$$\frac{1}{U_o A_o} = \frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)}. \quad (12)$$

Substituting for U_o from Eq. (9):

$$\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} = \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right). \quad (13)$$

This can also be expressed as:

$$\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} - \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right) = 0. \quad (14)$$

Substituting Eq. (6) for m_u in Eq. (3), the objective function is written as follows, where Eq. (15a) is for exhaust gas and Eq. (15b) for sea water:

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} + A_o E_i H_y C_i + A_o E_o H_y C_o, \quad (15a)$$

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} + A_o E_i H_y C_i + A_o E_o H_y C_o. \quad (15b)$$

The power losses inside and outside the tubes, E_i and E_o, are related to the friction factors and respective heat transfer coefficients. They are represented as follows [17,14]:

$$E_i = \psi_i h_i^{3.5}, \quad (16)$$

$$E_o = \psi_o h_o^{4.75}. \quad (17)$$

Substituting these, the objective function may be written as Eq. (18a) or (18b), respectively, for exhaust gas or sea water:

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o, \quad (18a)$$

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o. \quad (18b)$$

The objective function is assumed to have been structured on four variables of Δt_2, A_o, h_i and h_o of which only three can be independent. If three of the variables, say A_o, h_i and h_o, are known, the temperature difference, Δt_2, can be found.

2.2. Optimisation by calculus method

In optimisation techniques, calculus methods can be conveniently employed, where the expressions are continuous and differentiable [18]. With the use of Lagrangian multipliers, optimal candidate points may be obtained, where the problem is equality constrained [19]. With Eq. (14), the objective function (18a) or (18b) can be expressed as an unconstrained problem with λ, the Lagrangian multiplier. Eqs. (19a) and (19b) represent exhaust gases and sea water, respectively:

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o + \lambda\left[\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} - \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)\right], \quad (19a)$$

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o + \lambda\left[\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} - \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)\right]. \quad (19b)$$

The obtained expressions are differentiable, with respect to the four chosen variables, resulting in the following simultaneous equations. Solving the equations and eliminating λ, the optimum values can be obtained:

$$\frac{\partial C_T}{\partial h_i} = 3.5 A_{o\ \mathrm{opt}} \psi_i h_{i\ \mathrm{opt}}^{2.5} H_y C_i + \frac{\lambda D_o}{A_{o\ \mathrm{opt}} D_i h_{i\ \mathrm{opt}}^2} = 0, \tag{20}$$

$$\frac{\partial C_T}{\partial h_o} = 4.75 A_{o\ \mathrm{opt}} \psi_o h_{o\ \mathrm{opt}}^{3.75} H_y C_o + \frac{\lambda D_o}{A_{o\ \mathrm{opt}} D_i h_{o\ \mathrm{opt}}^2} = 0, \tag{21}$$

$$\frac{\partial C_T}{\partial A_o} = K_F C_{Ao} + \psi_i h_{i\ \mathrm{opt}}^{3.5} H_y C_i + \psi_o h_{o\ \mathrm{opt}}^{4.75} H_y C_o + \frac{\lambda}{A_{o\ \mathrm{opt}}^2} \left(\frac{D_o}{D_i h_{i\ \mathrm{opt}}} + \frac{1}{h_{o\ \mathrm{opt}}} + R_{dw} \right) = 0, \tag{22}$$

$$\frac{\partial C_T}{\partial \Delta t_2} = \left(\frac{\lambda F}{Q \ln(\Delta t_2/\Delta t_1)} \right) + \left(\frac{F(\Delta t_1 - \Delta t_2)}{Q \Delta t_2 \ln(\Delta t_2/\Delta t_1)^2} \right) + \frac{C_u H_y Q}{C_{pu}(\Delta t_1 - \Delta t_2 + t_1 - t_2)^2} = 0, \tag{23a}$$

$$\frac{\partial C_T}{\partial \Delta t_2} = \left(\frac{\lambda F}{Q \ln(\Delta t_2/\Delta t_1)} \right) + \left(\frac{F(\Delta t_2 - \Delta t_1)}{Q \Delta t_2 \ln(\Delta t_2/\Delta t_1)^2} \right) + \frac{C_u H_y Q}{C_{pu}(\Delta t_1 - \Delta t_2 + t_1 - t_2)^2} = 0. \tag{23b}$$

Eqs. (23a) and (23b) represent the derivations for exhaust gas and sea water, respectively. The values of respective variables will be the optimum values. The sequences of further calculations are as follows. The system of five equations, Eqs. (14) and (20) to (23a) or (23b), is solved for five unknowns, i.e. the four variables (h_i, h_o, A_o and Δt_2) and the Lagrangian multiplier, λ. Substituting these values in Eq. (9), the optimum value for overall heat transfer coefficient, $U_{o\ \mathrm{opt}}$, is obtained. With $h_{i\ \mathrm{opt}}$ and $h_{o\ \mathrm{opt}}$ values, the friction power losses, E_i and E_o, are calculated from Eqs. (16) and (17). The dimensionless factors are obtained from the following equations:

$$\psi_i = B_i \left[\frac{12200 D_i^{1.5} \mu_i^{1.83} (\mu_{wi}/\mu_i)^{0.63}}{D_o \rho_i^2 k_i^{2.33} c_{pi}^{1.17}} \right], \tag{24}$$

$$\psi_o = \frac{B_o}{n_b} \frac{N_r N_c}{N_t} \left(\frac{2 b_o D_c D_o^{0.75} F_s^{4.75} \mu_{fo}^{1.42}}{\pi a_o^{4.75} \rho_o^2 k_{fo}^{3.17} c_{pfo}^{1.58}} \right). \tag{25}$$

The next step is to calculate the optimum temperature difference, $\Delta t_{2\ \mathrm{opt}}$, at the warm end. Then, the optimum area, $A_{o\ \mathrm{opt}}$, is calculated from Eq. (11). The other values are obtained from the following equations.

The optimum cross sectional area of the tubes is obtained from:

$$S_{i\ \mathrm{opt}} = \frac{w_i}{G_{i\ \mathrm{opt}}}. \tag{26}$$

The optimum mass velocity of fluid flow in the tubes, $G_{i\ \mathrm{opt}}$, is obtained from [15]:

$$G_i = \left[\frac{h_i D_i^{0.2} \mu_i^{0.8}}{0.023 k_i} \left(\frac{k_i}{c_{pi} \mu_i} \right)^{1/3} \left(\frac{\mu_{wi}}{\mu_i} \right)^{0.14} \right]^{1.25}. \tag{27}$$

Similarly, the optimum values for the shell side parameters are calculated.

The optimum value of the number of tubes is obtained from:

$$N_{t\ \mathrm{opt}} = 4 \frac{n_p S_{i\ \mathrm{opt}}}{\pi D_i^2}. \tag{28}$$

Then, the optimum value of the length of the tube is found from:

$$L_{\ \mathrm{opt}} = \frac{A_{o\ \mathrm{opt}}}{\pi D_o N_{t\ \mathrm{opt}}}. \tag{29}$$

The optimum value for the shell-side free flow area is found from:

$$S_{\ \mathrm{opt}} = \frac{w_o}{G_{s\ \mathrm{opt}}}. \tag{30}$$

The power losses due to friction are then calculated. For turbulent flow and flows in a uniform cross section with no sudden contraction or enlargement, the losses can be obtained as follows [15]:

$$\Delta p_i = \frac{\beta_i 2 f_i G_i^2 L n_p}{\rho_i D_i \phi_i}, \tag{31}$$

where the correction factor is:

$$\phi_i = 1.02 \left(\frac{\mu_i}{\mu_{wi}} \right)^{0.14},$$

and for turbulent flow in the tubes, the friction factor is obtained from:

$$f_i = \frac{0.046}{(\mathrm{Re}_t)^{0.2}} = \frac{0.046}{(D_i G/\mu_i)^{0.2}}.$$

Power losses in the shell side are calculated from:

$$\Delta p_o = \frac{B_o 2 f' N_r G_s^2}{\rho_o}. \tag{32}$$

The friction coefficient for turbulent flow across the tubes is obtained from:

$$f' = b_o \left(\frac{D_o G_o}{\mu f_o} \right)^{-0.15},$$

and for staggered tubes,

$$b_o = 0.23 + \frac{0.11}{(X_T - 1)^{1.08}}.$$

2.3. Calculations using computer based software

The calculations using software were based only on thermodynamic properties, and the results were oriented towards the geometric design of the heat exchanger. These results, which involve no cost function, were compared to those obtained from optimisation, based on cost as the objective function. Values of important variables were verified with those obtained from Lagrangian equations. The principal approaches are explained below.

The heat transfer coefficient for tube side, h_i, is calculated from:

$$h_i = \frac{k_t}{D_i}\mathrm{Nu}_t. \tag{33}$$

This is based on the general Sieder-Tate equation, h_iD/k [15]. The value will depend on the flow characteristic being turbulent, viscous or in transition. For determining the nature of the flow and the Nusselt number, the Reynolds number is calculated. A correction for the Nusselt number may be applied, depending on if the fluid is liquid or gas.

$$\mathrm{Re}_t = \frac{\rho_t w_t D_i}{\mu_t}. \tag{34}$$

The shell side heat transfer coefficient is obtained similarly from the Nusselt number and thermal conductivity.

$$h_o = \frac{k_s}{l'}\mathrm{Nu}_s. \tag{35}$$

The fluid stream on the shell side flows over half the circumference of the tube. The characteristic length of this stream flow, l', is taken as $\pi D_o/2$. Then, the equation becomes:

$$h_o = \frac{k_s}{(\pi/2)D_o}\mathrm{Nu}_s. \tag{36}$$

The calculation of shell side Nusselt number involves a series of corrections. The mean Nusselt number is obtained by applying correction factors to the ideal value of Nusselt number for the tube bundle. The ideal value is obtained by applying correction factors for the tube rows and for change in physical properties of the fluid's boundary layer while flowing over the tube surface. The ideal value depends on the values calculated for laminar and turbulent flows and applying further corrections for tube arrangement being staggered or in-line. To determine the nature of the flow, the Reynolds number needs to be calculated. Due to the exhaustive procedure, the equations used in the software are not explained herein.

Since it was possible to give values to a number of geometric parameters while using the software,

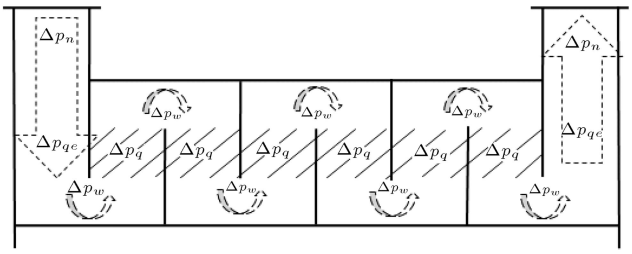

Figure 2. Shell side pressure drop regions.

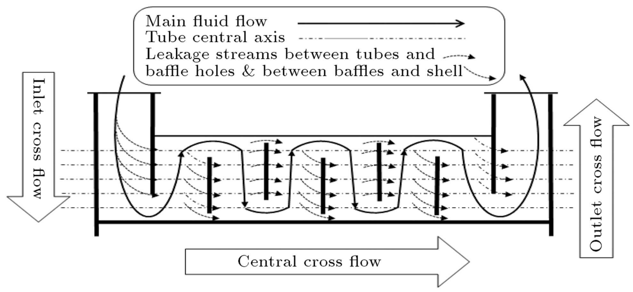

Figure 3. Path of leakage streams in shell side cross flow sections.

the comprehensive values for pressure drop could be obtained. For the tube side, the pressure drop was calculated from adding the drops in nozzle sections in the entry and exit sections, and due to friction.

$$\Delta p_t = \Delta p_{\mathrm{noz}} + \Delta p_{\mathrm{in\ out}} + \Delta p_{\mathrm{friction}}. \tag{37}$$

The shell side pressure drop was also an addition of drops in the central section, end sections, the window section and the nozzles. Figure 2 shows the shell side pressure drop regions. The calculation of central and end sections involved a number of correction factors based on Bell-Delaware approaches [20]. The losses, due to various leakage streams, are included in the computation. Figure 3 illustrates the flow of the leakage streams.

$$\Delta p_s = (N_b - 1)\Delta p_q + 2\Delta p_{qe} + N_b\Delta p_w + \Delta p_n. \tag{38}$$

3. Discussion

3.1. Approaches to methodology steps

The various steps of the methodology are tabulated in Figure 4. With the objective of heating the sea water and sterilising it, a shell and tube type heat exchanger was chosen. An in-line arrangement, with the exhaust gases exiting from the economiser/exhaust gas boiler section, was assumed. A single pass arrangement was chosen for the gases, so that the resulting backpressure on the turbocharger would be less and a silencer effect may also be realised. The mass flow and temperature of the exhaust gases were adopted from actual ship data. For the exercise, an operational crude oil carrier was considered. The relevant vessel design data and

Initial steps	**Optimisation steps**	**Calculations with software**
• Identify heat exchanger duty. • Determine temperatures, pressures and flow rates. • Choose the heat exchanger type. • Allot shell and tube circuits. (Consider fluid properties of heat transfer coefficient and fouling nature.) • Choose materials of construction. • Assume reasonable values of heat transfer coefficient based on material and fluids. • Determine tube arrangement (square, triangular pitch etc.) • Determine number of tube passes. (Consider mechanical cleaning.)	• Estimate purchase/installation costs for heat exchanger. • Estimate pumping costs for the fluids. • Estimate purchase costs for fluids. • Obtain values of various constants. • Obtain partial differential equations with Lagrangian multiplier. • Obtain partial differential equations neglecting cost of sea water. • Obtain partial differential equations neglecting cost of exhaust gases. • Solve the equations • Estimate h_i, h_o, U • Calculate all related formulae and estimate power losses of E_i and E_o. • Estimate area A_o. • Calculate heat exchanger effectiveness. • Calculate annual costs. • Calculate installation costs. • Estimate the number of tubes. • Calculate values for other variables such as length of tubes, flow across sections etc. • Calculate pressure drops. • Compare optimum values. • Identify heat exchanger suitable for ballast water treatment.	• Calculate overall heat transfer coefficient. • If calculated value is too small or too large, recalculate. • Calculate geometric values for variables of area, number of tubes, tube length, nozzle diameter etc. • Determine the shell diameter. • Estimate tube bundle clearances. • Estimate the number of baffles and baffle spacing. • Calculate pressure drops.

Figure 4. Summary of methodology steps.

Table 1. Vessel design data and operational data.

Main engine	Hitachi Zosen B&W 7S80 MC
	2 stroke marine diesel engine
	25 090 kW@78.6 rev/min, MCR
	22 580 kW@75.9 rev/min, CSO
Mass flow of exhaust gas at MCR	218900 kg/h
For year 2011:	
Total period in ballast mode	1824 h
Average output power during ballast passages	62% MCR
Possible waste heat recovery at 62%MCR output	4214.65 kW
Average sea water exit temperature from LT/HT coolers	34°C
Average exhaust gas exit temperature from EGB	274°C

operational data for the heat exchanger design are shown in Table 1. The heat duty was maintained within the possible recovery range matching approximately to heat recoveries at 60%MCR (Maximum Continuous Rating). The gas flow was assumed lower at 150000 kg/h, since part of the gas was assumed to bypass the ballast water heater. The heat duty was computed, matching with 3176 kW, the recovery calculated for 60%MCR operation. The period of vessel engagement in ballast mode 1824 h was rounded to a slightly higher value of 2000 h for H_y, which, effectively, is the number of hours the heat exchanger would be in operation.

The sea water temperature was assumed to have increased while removing heat from the LT/HT (Low Temperature/High Temperature) coolers. Considering the high temperature of gases, steel was chosen as the tube material, with a thermal conductivity of 52 W/mK. Standard values of 30 mm and 2 mm were chosen for the tube outside diameter and tube wall thickness [21]. The tubes were arranged in a staggered, triangular pattern, so that the heating phase for the sea water could be enhanced and more tubes could be accommodated in the shell. Sea water flowed through the shell after being sucked through coarse filters and passing through coolers and micro filters. The fouling effects were thus minimised. Fouling coefficients were adopted from typical data available for plain tubes used in shell and tube design [21]. Furthermore, the winding path through the shell side increased the heating effect during the heating phase. The sea water flow was maintained at 100 m^3/hr to achieve better temperature rise for sterilisation.

Table 2 shows the primary data assumed for the design.

The purchase cost was calculated assuming rea-

Table 2. Primary data for heat exchanger design.

	Tube side	Shell side
Fluid	Diesel engine exhaust gas	Sea water
Flow rate	150000 kg/hr	100 m^3/hr
Inlet temperature	200°C	28°C

Table 3. Costs derived for design.

Costs US$	
Cost of purchase C_{pur}	123/m^2
Cost of installation C_{Ao}	141.45/m^2
Cost of utility fluid, exhaust gas C_u	0.04/kg
Cost to pump exhaust gas C_i	0.2/kWh
Cost to pump sea water C_o	0.2/kWh

sonable values for heat transfer coefficient and approximating the area. The purchase cost was computed based on carbon steel for tubing for a conventional shell and tube design. The value was further adjusted considering the effects of tube diameter, tube length and operating pressures [15]. The installation cost was 115% of the purchase cost. The annual maintenance cost was fixed at 20% (K_F) of the installation cost.

Since the waste heat of exhaust gases is to be harvested, no direct costs need be imposed for the utility fluid. Yet, as cost is the objective function and for studying design variations, cost was considered for the utility fluid. Exhaust gas generation was estimated based on the stoichiometric analysis of fuel combustion [22,23]. The cost of the unit mass of fuel was equated as the cost incurred for the generation of exhaust gases, and the cost for the unit mass of exhaust gas was derived. The values of C_u were determined for reasonable variations in excess air for combustion. For the design, the value at around 50% excess air was assumed.

The costs to pump the sea water and exhaust gas were obtained assuming the SFC to be 200 grams/kWh. While pumping costs for sea water are well justified, costs for pumping exhaust gases are also considered, thus, apportioning a cost for the power spent by the engine in pumping the gases. Estimation of the energy expended by the engine and the turbocharger for pumping the gas will be a complex process. So, irrespective of the power spent to pump the fluids, the cost incurred for SFC was equated as the cost for generating unit power. The cost of Marine Diesel Oil (MDO) varied between US$810-1090 per tonne, while the cost of Heavy Fuel Oils (HFO) varied between US$490-835 per tonne, depending on the grade and sulphur content [24]. A flat value of US$1000 per tonne of fuel was assumed for both MDO and HFO. The derived costs are tabulated in Table 3. Various constants and thermodynamic values were obtained by interpolations and approximations [25,26]. The calculated values assumed for design are shown in Table 4.

3.2. Approaches to optimal values

From Eq. (6), it can be seen that the mass flow of the utility fluid, m_u, depends on the temperature difference at the warm end, Δt_2, while the other values are fixed. The value of Δt_2 is obtained by solving Eqs. (22) and (23a) or (23b). The optimum value for the temperature difference, $\Delta t_{2\ \text{opt}}$, is obtained from the following equation:

$$\frac{FU_{o\ \text{opt}}H_Y C_u}{c_{pu}(K_F C_{Ao} + E_{i\ \text{opt}}H_Y C_i + E_{o\ \text{opt}}H_Y C_o)} = \left(1 + \frac{T_1 - T_2}{\Delta t_1 - \Delta t_{2\ \text{opt}}}\right)^2 \left(\ln\frac{\Delta t_{2\ \text{opt}}}{\Delta t_1} - 1 + \frac{\Delta t_1}{\Delta t_{2\ \text{opt}}}\right). \tag{39}$$

Referring to Eq. (39), the optimum value of the overall heat transfer coefficient, $U_{o\ \text{opt}}$, and the power losses having been ascertained from $h_{i\ \text{opt}}$ and $h_{o\ \text{opt}}$, the only unknown will be $\Delta t_{2\ \text{opt}}$. The value of this variable will depend upon the cost of the utility fluid, C_u.

Case 1 was treated assuming no cost for the utility fluid, only pumping costs. For Case 2, all costs were considered. The value of C_u was considered for both Cases 3 and 4. But, the pumping cost for the exhaust gas only was considered for Case 3, and, for Case 4, only the pumping cost for sea water was considered. Case 5 represents a parallel design, where no costs were allocated for materials or fluids. The cost considerations and optimal values of the primary and a few other variables are tabulated in Table 5 for all cases.

Although Case 1 has the least annual cost and area, the outlet temperature of sea water for the calculated $\Delta t_{2\ \text{opt}}$ was 30.06°C, which is quite below the target temperature. The next three cases, which showed a good increase in sea water temperature and mass flow of utility fluid, were favourably considered. The temperature rise of sea water and mass flow increase of exhaust gas indicated that a nominal increase in the flow of shell side fluid is possible. If the actual inlet temperature averages 274°C, as obtained from operational data, the extra heat available can be realised by increasing the mass flow of the sea water in the shell side. Furthermore, with the higher areas of other designs, better recoveries are also possible, considering that the inlet temperature of the sea water obtained from

Table 4. Data assumed for design.

	Tube side	**Shell side**
Mass flow m_h, m_c	41.67 kg/s	28.25kg/s
Inlet temperature T_1, t_1	200°C	28°C
Outlet temperature T_2, t_2	132.5°C	55°C
Wall temperature	66.45°C	47.74°C
Density ρ_i, ρ_o	0.8767 kg/m^3	1017 kg/m^3
Specific heat capacity C_{ph}, C_{pc}	1086 J/kg K	4001 J/kg K
Thermal conductivity k_i, k_o	0.034232 W/m K	0.636 W/mK

Table 5. Optimum values of important variables.

Cost consideration	C_T **(US\$/year)**	**Installed cost (US\$)**	**LMTD** Δp_o ΔT_{lm} **(°C)**	**Heat transfer coefficient, tube side** $h_{i\ \mathrm{opt}}$ **(W/m²K)**	**Heat transfer coefficient, shell side** $h_{o\ \mathrm{opt}}$ **(W/m²K)**	**Overall heat transfer coefficient** $U_{o\ \mathrm{opt}}$ **(W/m²K)**	**Area** $A_{o\ \mathrm{opt}}$ **(m²)**	**Number of tubes** N_t	**Pressure drop, tube side** Δp_i **(Pa)**	**Pressure drop, shell side** Δp_o **(Pa)**
Case 1 $C_u = 0$ $C_i = 0.2$ $C_o = 0.2$	14153.16	51391.96	148.26	81.10	1315.43	56.70	363.32	3213	182.16	427.33
Case 2 $C_u = 0.04$ $C_i = 0.2$ $C_o = 0.2$	22124.65	68233.27	111.67	81.10	1315.43	56.70	482.38	4266	241.85	567.37
Case 3 $C_u = 0.04$ $C_i = 0.2$ $C_o = 0$	22077.93	67138.94	111.64	82.77	1315.43	57.64	474.65	4198	238.78	565.26
Case 4 $C_u = 0.04$ $C_i = 0$ $C_o = 0.2$	17477.46	70011.78	108.64	81.10	1370.80	56.80	494.96	4377	248.88	583.92
Case 5 No costs; Software design	n.a	54274.37	123.70	94.97	1418.00	64.50	383.70	3250	423.69	415.49

Table 6. Heat exchanger effectiveness.

	$t_{2\ \mathrm{opt}}$ **(°C)**	$m_{u\ \mathrm{opt}}$ **(kg/s)**	ε
Case 2	80.85	53.21	0.55
Case 3	80.9	53.17	0.55
Case 4	87.1	47.16	0.71
Case 5	55	41.67	0.32

operational conditions is 34°C. Table 6 shows the calculated values of the outlet temperature of sea water, mass flow of exhaust gas and the heat exchanger effectiveness of the considered cases, and that of the software design. Of these, Case 4 has the minimum annual cost and maximum temperature rise. For the optimum outlet temperature and mass flow, the effectiveness of Case 4 is highest amongst the three cases. In a ballast water treatment system incorporating heat treatment, mass flow of fluids is a crucial factor [27]. With this perspective, Case 4 was preferred.

The effectiveness of the heat exchanger was computed based on the following equations [25]:

$$\varepsilon \equiv \frac{\text{Actual heat transferred}}{\text{Maximum heat that could have been transferred}},$$

$$\varepsilon = \frac{C_h(T_{h\ \mathrm{in}} - T_{h\ \mathrm{out}})}{C_{\min}(T_{h\ \mathrm{in}} - T_{c\ \mathrm{in}})} = \frac{C_c(T_{c\ \mathrm{out}} - T_{c\ \mathrm{in}})}{C_{\min}(T_{h\ \mathrm{in}} - T_{c\ \mathrm{in}})}, \tag{40}$$

Table 7. Values of important variables (t_2 fixed).

Cost consideration	C_T (US$/year)	Installed cost (US$)	LMTD ΔT_{lm} (°C)	Heat transfer coefficient, tube side $h_{i\ \mathrm{opt}}$ (W/m^{2}K)	Heat transfer coefficient, shell side $h_{o\ \mathrm{opt}}$ (W/m^{2}K)	Overall heat transfer coefficient $U_{o\ \mathrm{opt}}$ (W/m^{2}K)	Area $A_{o\ \mathrm{opt}}$ (m^2)	Number of tubes N_t	Pressure drop, tube side Δp_i (Pa)	Pressure drop, shell side Δp_o (Pa)
Case 6 $C_u = 0.04$ $C_i = 0.2$ $C_o = 0.2$	20302.73	61622.85	123.67	81.10	1315.43	56.70	435.61	4220	239.24	561.25
Case7 $C_u = 0$ $C_i = 0.2$ $C_o = 0$	16924.19	60620.26	123.67	82.77	1315.43	57.64	428.56	4100	233.23	552.11
Case 8 $C_u = 0$ $C_o = 0$ $C_o = 0.2$	12427.60	61515.90	123.67	81.10	1370.80	56.80	434.90	4200	238.80	560.28

where $C_{\min}$ is the lesser of $C_h = m_h C_{ph}$ and $C_c = m_c C_{pc}$.

In the optimisation cases, although the number of optimum baffles worked to two, they were increased to three to give better support for the tube bundle and counter vibration effects. This increased the pressure drops nominally. For tube side pressure drop computations, the values of optimal tubes were adjusted to the first decimal, thus accommodating tube sheet thickness. The projected pressure drop values are attributed to frictional losses only. But, with the software calculations, losses could be computed for other factors also. The total losses on the tube side were 1347 Pa and 52505 Pa for the shell side. However, effects due to changes in kinetic energy and vertical head etc. were not considered.

For comparison, three additional designs were derived assuming fixed target temperature and mass flow, but for varying cost considerations. Table 7 shows the cases. The heat exchanger effectiveness worked to 0.39 in these cases. While the areas are almost the same, Case 8 has the least annual cost. Compared to this, Case 4 works to an extra area of 60.06 m^2 with an additional installed cost of US$8495.88. If Case 5, having the least area, is to be taken as the reference, Case 4 works to an extra 111.26 m^2, with an increased installed cost of US$15737.42. If the annual cost consideration in Eq. (3) for the exhaust gas in Case 4 is presumed to be a saving, since only waste heat is harvested, the recovery period for this extra increase in installation cost works out to be 4.7 years. If all costs are neglected, Case 5 has the most compact design. For similar thermodynamic variables, if annual cost is considered, Case 8 is preferable. While other cases, except Case 1, are also viable, maximum scope for better heat recovery is sighted with Case 4 only for the purposeful heat treatment of ballast water.

Process streams involving many components in the network may require time consuming complex techniques such as genetic algorithms [28,29], simulated annealing [30], nonlinear approaches [31] and various economics based approaches [32,33]. Without such methods, a rational approach to optimisation using calculus methods has been demonstrated to obtain a few viable designs.

4. Conclusions

From the exercises, the practical design of a heater for treating ballast sea water using exhaust gases has been realised. For the present purpose of harvesting waste heat for ballast water treatment, scope for further improvements include switching the shell and tube side fluids and finned tubular heat exchanger designs. Limitations of heat availability and flow depend on ship type, and designs may be worked upon for other ship types. Increased realisation of waste heat will give a competitive edge to heat treatment methods, not only in costs but also in enhancing the treatment potential of combination type systems.

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Nomenclature

$A_{o\ \mathrm{opt}}$ Optimum area, m^2

A_o	Outside area of tubes available for heat transfer, m^2
B_o	Friction correction factor for change in cross section and flow reversal
C_{Ao}	Installed cost of heat exchanger, US\$/m^2
C_T	Total annual variable costs for heat exchanger including operational costs, US\$
C_i	Cost to pump fluid inside the tubes (exhaust gas), US\$/kWh
C_o	Cost to pump fluid on the shell side (sea water), US\$/kWh
C_h	Heat capacity of hot stream, W/K
C_c	Heat capacity of cold stream, W/K
C_{ph}	Specific heat of hot fluid (exhaust gas), J/kg K
C_{pc}	Specific heat of cold fluid (sea water), J/kg K
C_{pur}	Cost of purchase, US\$/m^2
C_u	Cost of utility fluid, US\$/kg
D_c	Clearance between tubes giving smallest area for shell side fluid flow
D_i	Inside diameter of tube, mm
D_o	Outside diameter of tube, mm
E_i	Power loss inside tubes/m^2
E_o	Power loss outside tubes/m^2
F_s	Bypass factor, shell side
$G_{i\ \text{opt}}$	Mass velocity (optimum) inside tubes, kg/m^2 s
G_i	Mass velocity inside tubes, kg/m^{2}s
G_o	Mass velocity shell side, kg/m^{2}s
$G_{s\ \text{opt}}$	Mass velocity (optimum) shell side, kg/m^{2}s
H_y	Number of hours of heat exchanger operation/year
K_F	Fixed charges including maintenance/year as a fraction of installed cost (percent)
L_{opt}	Optimum length of tube, m
N_b	Number of baffles
N_c	Number of clearances between tubes for shell side fluid flow
N_r	Tube rows across which shell side fluid flows
$N_{t\ \text{opt}}$	Optimum number of tubes
N_t	Number of tubes
Nu_s	Nusselt number, shell side
Nu_t	Nusselt number, tube side
Q_{exhaust}	Heat energy lost to exhaust gases, kW
Q_{in}	Heat energy input, kW
$Q_{\text{odd losses}}$	Heat energy lost due to other factors, kW
Q_{water}	Heat energy lost to cooling water, kW
R_{dw}	Fouling resistance, combined (tube, scale & dirt), m^{2}K/W
Re_t	Reynolds number, tube side fluid
R_{fi}	Inside fouling resistance (tube side), m^{2}K/W
R_{fo}	Outside fouling resistance (shell side), m^{2}K/W
$S_{i\ \text{opt}}$	Optimum cross sectional area inside tubes/pass, m^2
T_1	Inlet temperature of tube side fluid (exhaust gas), °C
T_2	Outlet temperature of tube side fluid (exhaust gas), °C
$U_{o\ \text{opt}}$	Optimum overall heat transfer coefficient
U_o	Overall heat transfer coefficient, W/m^{2}K
$W_{\text{engine power}}$	Useful energy output, kW
X_T	Ratio of transverse pitch to tube diameter
a_0	Constant for evaluating outside heat transfer coefficient
b_0	Constant for calculating shell side friction factor
c_{pi}	Specific heat, inside tube, J/kg K
f'	Friction factor, shell side flow
f_i	Fanning friction factor, tube side flow
$h_{i\ \text{opt}}$	Optimum heat transfer coefficient, tube side, W/m^{2}K
h_i	Heat transfer coefficient, W/m^{2}K
$h_{o\ \text{opt}}$	Optimum heat transfer coefficient, shell side, W/m^{2}K
h_o	Heat transfer coefficient, W/m^{2}K
k_{fo}	Thermal conductivity at film temperature, W/mK
k_i	Thermal conductivity, inside tube, W/mK
k_o	Thermal conductivity, outside tube, W/mK
k_s	Thermal conductivity, shell side, W/mK
k_t	Thermal conductivity, tube side, W/mK
l'	Characteristic length of stream flow, m

m_h	Mass flow of hot fluid (exhaust gas), kg/s
m_c	Mass flow of cold fluid (sea water), kg/s
m_u	Mass flow of utility fluid, kg/s
n_b	Number of baffle spaces (number of baffles + 1)
n_p	Number of tube passes
t_1	Inlet temperature of shell side fluid (sea water), °C
t_2	Outlet temperature of shell side fluid (sea water), °C
w_i	Mass flow inside tubes, kg/s
w_o	Mass flow outside tubes kg/s
w_t	Mass flow, tube side, kg/s
Δt_1	$(T_2 - t_1)$ temperature difference at tube exit/shell entry, °C
Δt_2	$(T_1 - t_2)$ temperature difference at tube entry/shell exit, °C
β_i	Friction correction factor for sudden change in tube section and flow reversal
μ_{fo}	Absolute viscosity of shell side fluid at film temperature, Pas
μ_i, μ_t	Absolute viscosity of tube side fluid, Pas
μ_{wi}	Absolute viscosity of tube side fluid at wall temperature, Pas
ρ_i	Density, inside tube, kg/m^3
ρ_o	Density, outside tube, kg/m^3
ρ_t	Density, tube side, kg/m^3
ψ_i	Dimensional factor for estimating power loss inside tubes
ψ_o	Dimensional factor for estimating power loss outside tubes
ϕ_i	Correction factor for friction, and tube side
ε	Heat exchanger effectiveness
ΔT_{lm}	Logarithmic mean temperature difference, °C
Δp_i	Pressure drop at tube side, Pa
Δp_o	Pressure drop at shell side, Pa
$\Delta p_{\text{friction}}$	Pressure drop due to friction in tubes, Pa
Δp_{inout}	Pressure drop at inlet and outlet sections, Pa
Δp_n	Pressure drop in nozzles on shell side, Pa
Δp_{noz}	Pressure drop in nozzles on tube side, Pa
Δp_q	Pressure drop in central sections of shell, Pa
Δp_{qe}	Pressure drop in end sections of shell, Pa
Δp_w	Pressure drop in window sections of shell, Pa
A	Total surface area, m^2
F	LMTD correction factor
L	Length of tube, m
LCV	Lower Calorific Value of fuel, MJ/kg
Q	Heat duty of heat exchanger, W
SFC	Specific Fuel Consumption, grams/kWh
U	Overall heat transfer coefficient, W/m^{2}K
λ	Lagrangian multiplier

Subscript

i, o	Inside (tube); Outside (shell)
t, s	Tube side; shell side
h, c	Hot fluid; cold fluid
opt	Optimum value

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