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Design Verification of Heat Exchanger for Ballast Water Treatment

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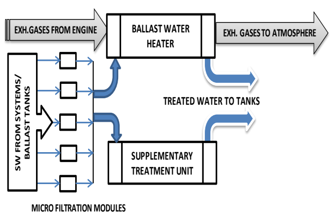
Graphical abstract



Abstract

Using waste heat from ship's engines is one of the methods considered for heat treatment of ballast water. For such a system harvesting the engine exhaust heat, a heat exchanger will be vital. Design optimisation of a heater employing exhaust gases of the 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. Costs for installation, maintenance as also costs for the utility and process fluids were considered. Heat balance data, specific fuel consumption values from a typical operational ship and current fuel costs were considered for the design. The thermodynamic and geometric designs were worked out using computer based software for comparing the designs. Costs were also computed using a different approach for all the designs. Since the amount of heat transferred was specified and the application was limited to a single process, direct cost method was used for the computation. The objective function values obtained from Lagrangian equations were compared with the values obtained from direct cost computations. From the optimal designs, choice was justified based on annual cost, optimum exit temperature of shell side fluid and optimum mass flow of tube side fluid..

Key words: Ballast water treatment; waste heat recovery; heat exchanger; optimization; costs

1.0 INTRODUCTION

International Convention for the Control and Management of Ships' Ballast Water and Sediments (BWM Convention) will come into force one year after the full ratification. As of 30th September 2013, 38 countries totalling 30.38% of global tonnage have agreed while the requirement for full ratification is 30 countries and 35% of world tonnage[1]. Subsequent to the ratification, ships have to comply with stricter performance standards. This is possible only by treatment of the ballast water replacing the current practice of ballast water exchange (BWE). Many ballast water treatment (BWT) systems are at commercial readiness after approvals but none of the systems are efficient enough to meet the stricter standards proposed by US Administration[2]. Research on BWT systems continues and heat treatment is one of the physical methods which have been probed into. Balaji and Yaakob[3] had shown heat availability on board though observing that heat treatment alone may not suffice for treating large quantities of ballast water. Though issues remain with heat treatment, the waste heat potential on board makes it an attractive option.

Most of available BWT systems are designed as a combination of two to five methods[4,5]. A ship board heat resource based system harvesting heat from engine exhaust gases in combination with another method could be a viable treatment system. Figure 1 shows a simple layout of such a BWT combination system[6]. For this heat-reliant BWT system, a well-designed heat exchanger is essential.

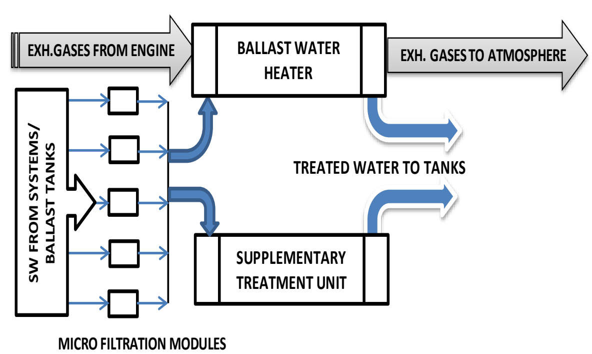


Figure 1 Ballast water treatment system

2.0 METHODOLOGY

Models have been proposed for optimising heat exchanger designs to enhance waste heat recoveries[7]. Heat exchanger design selection based on genetic algorithms[8] and multi objective optimisation[9] etc. have been proposed but these methods are suitable for processing plants involving a network of heat exchangers and other components. Since marine heat exchangers are mostly singular, simpler optimisation techniques with engineer defined parameters and constraints can be employed.

2.1 Optimisation of Heat Exchanger

The heat duty for the heat exchanger was fixed considering a recovery of 10% from the input energy. A single pass, shell and tube heat exchanger having a counter flow pattern with baffles was designed. The fluids were assumed to undergo no phase change. Other assumptions included steady state operation, constant specific heats for the fluids, constant over all heat transfer coefficient and negligible heat losses[10]. The heat duty, inlet temperature of the shell side cold fluid and the tube diameters were treated as known variables.

2.1.1 Basic Equations for Optimisation

The objective function of annual cost can be written as an addition of annual variable costs, costs incurred for utility fluid and the power losses on the tube and shell circuits[11,12].

$$C_T = A_o K_F C_{Ao} + m_u H_y C_u + A_o E_i H_y C_i + A_o E_o H_y C_o \tag{1}$$

C_T is the total annual variable cost including operational costs. A_o (m^2) is the area for heat transfer and K_F is the factor applied for computing fixed charges including maintenance (20% of the installation charges, C_{Ao}). m_u (kg/s) is the mass flow of utility fluid, H_y denotes the hours of heat exchanger operation (taken as 2000 from vessel data) and C_u is the cost of utility fluid. E_i and E_o are the power losses incurred per unit area on the inside and outside of the tubes whereas C_i and C_o are the costs to pump the fluids.

The relationship for the thermal design is based on the enthalpy rate equations for single phase fluids where $j = i, o$ denoting each of the fluids inside the tubes and outside respectively and $\dot{m}_j$ representing the mass flow[13].

$$q = q_j = \dot{m}_j \, \Delta h_j = \left(\dot{m}c_p\right)_j \Delta T_j = \left(\dot{m}c_p\right)_j \left|T_{j,i} - T_{j,o}\right| \tag{2}$$

For heat balance of hot and cold streams, the heat absorbed will be the product of the mass flows, specific heats and the temperature differences.

$$Q = m_c \cdot C_{pc} \, (t_2 - t_1) = m_h \cdot C_{ph} \, (T_1 - T_2) \tag{3}$$

Where m_c and m_h (kg/s) are the mass flow of cold (sea water) and hot fluid (exhaust gas), and C_{ph} and C_{pc} (J/kg K) are the specific heats of hot (exhaust gas) and cold fluid (sea water).

The inlet and outlet temperatures (°C) of the shell side fluid (sea water) are t_1 and t_2. The inlet and outlet temperatures (°C) of tube side fluid are T_1 and T_2 .

The mass flow of the fluids can be obtained from,

$$m_u = \frac{Q}{C_{ph}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} \tag{4a}$$

$$m_u = \frac{Q}{C_{pc}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} \tag{4b}$$

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 exits. Equations (4a) and (4b) represent exhaust gases and sea water respectively.

The fundamental equation for heat transfer in the heat exchanger is,

$$Q = UA \, \Delta T_{lm} \tag{5}$$

U (W/m^2 K) is the Overall Heat Transfer Coefficient. A represents the surface area and ΔT_{lm} (°C) is the Logarithmic Mean Temperature Difference (LMTD).

The optimum overall heat transfer coefficient can be calculated from,

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_i} \cdot \frac{D_o}{D_i} + R_{fo} + R_{fi} \tag{6}$$

Where h_i and h_o (W/m^2 K) are the Heat transfer coefficients for the inside and outside of the tubes respectively, D_i and D_o are the inside and outside diameters of the tube and R_{fi} and R_{fo} (m^{2}K/W) are the fouling resistances on the tube and shell sides.

The overall heat transfer coefficient equation is further simplified by combining the fouling factors. R_{dw}(m^{2}K/W) is the combined fouling resistance (tubes, scale & dirt).

$$U_o = \left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)^{-1} \tag{7}$$

The LMTD is calculated from,

$$\Delta T_{lm} = \frac{F \, (T_1 - t_2) - (T_2 - t_1)}{ln\left[\frac{T_1 - t_2}{T_2 - t_1}\right]} \tag{8}$$

The heat duty is then written as,

$$Q = F \, U_o \, A_o \, \frac{(\Delta t_2 - \Delta t_1)}{ln(\Delta t_2 / \Delta t_1)} \tag{9}$$

The subscript 'o' represents the optimum value. A correction factor F is applied for counter current heat exchangers depending on the number of tube and shell passes of the process fluids. Equation (9) can be rewritten as,

$$\frac{1}{U_o \, A_o} = \frac{F(\Delta t_2 - \Delta t_1)}{Q \, ln(\Delta t_2 / \Delta t_1)} \tag{10}$$

Substituting for U_o from equation (7),

$$\frac{F(\Delta t_2 - \Delta t_1)}{Q \, ln(\Delta t_2 / \Delta t_1)} = \frac{1}{A_o} \left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right) \tag{11}$$

The power losses inside and outside of the tubes, E_i and E_o are represented as follows[11]. ψ_i and ψ_o are the dimensional factors for estimating power losses in the tube and shell circuits.

$$E_i = \psi_i h_i^{3.5} \tag{12}$$

$$E_o = \psi_o h_o^{4.75} \tag{13}$$

These derivations were inserted in Equation (1) appropriately after substituting Equation (4) for m_u in (1). The objective function equation may be written for exhaust gases or sea water. Thus the objective function equations were structured on four variables $\Delta t_2, A_o, h_i$ 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.

2.1.2 Cost Computation using Lagrangian Multiplier

With the use of Lagrangian multipliers, optimal candidate points may be obtained where the problem is equality constrained[14]. With Equations (11), (12) and (13), the objective function can be expressed as an unconstrained problem with λ, the Lagrangian multiplier. Substituting Equations (4a), (4b), (11), (12) and (13), the Equation (1) can be expressed in the unconstrained form with the Lagrangian multiplier. Equations (14a) and (14b) represent exhaust gases and sea water respectively. C_{pu} is the specific heat of utility fluid (exhaust gas).

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_2 - \Delta t_1 + t_1 - t_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o + \lambda\left[\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} - \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)\right] \tag{14a}$$

$$C_T = A_o K_F C_{Ao} + \frac{Q H_y C_u}{C_{pu}(\Delta t_1 - \Delta t_2 + T_1 - T_2)} + A_o \psi_i h_i^{3.5} H_y C_i + A_o \psi_o h_o^{4.75} H_y C_o + \lambda\left[\frac{F(\Delta t_2 - \Delta t_1)}{Q \ln(\Delta t_2/\Delta t_1)} - \frac{1}{A_o}\left(\frac{D_o}{D_i h_i} + \frac{1}{h_o} + R_{dw}\right)\right] \tag{14b}$$

The obtained expressions are differentiable with respect to the four chosen variables resulting in following simultaneous equations. Equations (18a) and (18b) represent exhaust gases and sea water respectively. Solving the equations and eliminating λ, the optimum values can be obtained. The variables for optimum values are subscripted as '*opt*'.

$$\frac{\partial C_T}{\partial h_i} = 3.5 A_{o\,opt} \psi_i h_{i\,opt}^{2.5} H_y C_i + \frac{\lambda D_o}{A_{o\,opt} D_i h_{i\,opt}^2} = 0 \tag{15}$$

$$\frac{\partial C_T}{\partial h_o} = 4.75 A_{o\,opt} \psi_o h_{o\,opt}^{3.75} H_y C_o + \frac{\lambda D_o}{A_{o\,opt} D_i h_{o\,opt}^2} = 0 \tag{16}$$

$$\frac{\partial C_T}{\partial A_o} = K_F C_{Ao} + \psi_i h_{i\,opt}^{3.5} H_y C_i + \psi_o h_{o\,opt}^{4.75} H_y C_o + \frac{\lambda}{A_{o\,opt}^2}\left(\frac{D_o}{D_i h_{i\,opt}} + \frac{1}{h_{o\,opt}} + R_{dw}\right) = 0 \tag{17}$$

$$\frac{\partial C_T}{\partial \Delta t_2} = \left(\frac{\lambda F}{Q \ln(\Delta t_2/\Delta t_1)}\right) + \left(\frac{F(\Delta t_1 - \Delta t_2)}{Q \Delta t_2 \ln(\Delta t_2/\Delta t_1)^2}\right) + \frac{C_u H_y Q}{C_{pu}(\Delta t_1 - \Delta t_2 + t_1 - t_2)^2} = 0 \tag{18a}$$

$$\frac{\partial C_T}{\partial \Delta t_2} = \left(\frac{\lambda F}{Q \ln(\Delta t_2/\Delta t_1)}\right) + \left(\frac{F(\Delta t_2 - \Delta t_1)}{Q \Delta t_2 \ln(\Delta t_2/\Delta t_1)^2}\right) + \frac{C_u H_y Q}{C_{pu}(\Delta t_1 - \Delta t_2 + t_1 - t_2)^2} = 0 \tag{18b}$$

From equations (4a) and (4b) 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 optimum value for the temperature difference at the warm end $\Delta t_{2\,opt}$ is obtained from the following equation.

$$\frac{F U_{o\,opt} H_Y C_u}{c_{pu}\left(K_F C_{Ao} + E_{i\,opt} H_Y C_i + E_{o\,opt} H_Y C_o\right)} = \left(1 + \frac{T_1 - T_2}{\Delta t_1 - \Delta t_{2\,opt}}\right)^2 \left(\ln\frac{\Delta t_{2\,opt}}{\Delta t_1} - 1 + \frac{\Delta t_1}{\Delta t_{2\,opt}}\right) \tag{19}$$

The objective function values were computed for eight cases using the above equations. Four optimal design cases were developed with various cost considerations and for each design, optimum outlet temperature for sea water was calculated.

Three more cases were computed by keeping the outlet temperature of sea water constant but for similar cost considerations. One extra design was developed using software where Bell-Delaware approaches were employed with no cost considerations.

2.1.3 Computation of Direct Costs

The total annual costs will be an addition of capital cost C_{CA}, costs incurred in energy consumption C_E, and the operating costs C_s. Costs were computed for two considerations, one applying an interest rate on capital and a payback period and the other without considering both[15].

$$C_{tot} = C_{CA} + C_E + C_s \tag{20}$$

Capital costs were computed by assuming two different values for energy costs. The first energy cost was obtained for ship board generation of unit power (US$0.2/kWh) and the second one based on average local (ashore) cost (US$0.06/kWh).

Capital Costs assuming these two energy costs were computed for two cases. In one case, costs for sea water pump (for process fluid) were considered and in another the pump costs were neglected. The cost of turbocharger was neglected as the cost would have been included with the main engine costs.

Standard values for all reference costs and indexes were obtained from relevant handbooks[10,15] as also from the Chemical Engineering Plant Cost Index (CEPCI) published by Chemical Engineering.

The capital cost was calculated from the following.

$$C_{CA} = \left(\frac{1}{n} + \frac{z}{2}\right)\left(I_{EX} + I_{P\,tch} + I_{Ppump}\right) = a\left(I_{EX} + I_{P\,tch} + I_{Ppump}\right) \tag{21}$$

$$I_{EX} = I_{EXo\,ref}\left(\frac{A_{EX}}{A_{EXo\,ref}}\right)^{m_{EX}} \tag{22}$$

$$I_{P\,tch/pump} = I_{P\,tch/pump\,ref}\left(\frac{L_{P\,tch/pump}}{L_{P\,tch/pump\,ref}}\right)^{m_P} \tag{23}$$

Where a the payback coefficient is equal to 0.125 and s is a coefficient based on intensity of maintenance. Assuming a medium intensity for cleaning and maintenance, a value of 0.025 was obtained from the Cost Index tables. The number of years of cost recovery period n was taken as 10 and the interest rate z was assumed to be 5%.

I_{EX}, $I_{P\,tch}$ and I_{Ppump} are the purchase costs of the heat exchanger, turbocharger and sea water pump computed from the reference costs $I_{EXo\ ref}$ and $I_{P\,tch/pump\ ref}$ obtained from the industry indexes.

For the reference costs, the power (kW) referred to was denoted by $L_{P\,tch/pump\,ref}$. For the pump, the reference power was taken as 100 kW. The values of exponents m_{EX} and m_P were 0.59 and 0.67 as obtained from VDI Atlas [15]. The heat exchanger area and reference area are represented by A_{EX} and $A_{EXo\,ref}$.

The energy costs and operating expenses were calculated as follows.

$$C_E = C_{EL}H_y\left(\frac{M_{ex\,gas}\,\Delta p_{tube}}{\rho_{eg}\,\eta_{tch}} + \frac{M_{pump}\,\Delta p_{shell}}{\rho_{sw}\,\eta_{pump}}\right) + C_{\Delta T} + C_M \qquad (24)$$

$$C_s = sI_{EX} \qquad (25)$$

The mass flow of exhaust gas $M_{ex\,gas}$ was taken as 41.67 kg/s and sea water M_{pump} as 28.25 kg/s. The densities of the exhaust gas ρ_{eg} and sea water ρ_{sw} were 0.8767 kg/m^3 and 1017 kg/m^3 respectively. These values and the pressure drops Δp_{shell} and Δp_{tube} (Pa) were taken from the heat exchanger design parameters.

The sea water pump efficiency η_{pump} was taken as 0.7 and the turbocharger pump efficiencyη_{tch} as unity. The additional energy costs for increasing temperature $C_{\Delta T}$ and cost of supplies C_M were not considered as the heat exchanger was independent and not part of a network.

The objective function values and direct cost values were compared and variations observed. The least values of objective functions and the least variation from direct cost values were considered for selecting the area of the heat exchanger. The final design was determined by applying two other factors of mass flow and outlet temperature of sea water which are significant for the ballast water treatment.

3.0 RESULTS AND DISCUSSION

The assumed and derived costs used for calculations are tabulated in Table 1 and the cost summary in Table 2 shows the cost variations (in brackets). All designs were thermodynamically feasible. For treatment of ballast water, a minimum temperature of 55°C was kept as the target.

Case 1 was excluded because the optimum outlet temperature was much below the targeted temperature. Case 5 was used only as a reference for the geometrical values.

Case 2 had the highest objective function value followed by Case 3 and hence both were not considered, though other optimal values were tangible.

Amongst the rest of the cases, Case 8 had the least value of objective function, even lower than the direct costs. The next two cases with low objective function values were Case 7 and Case 4. The objective function value (Lagrangian) varied by 0.03% for Case 4 and 0.07% for Case 7, when compared with direct cost values.

But the obtained value of sea water outlet temperature (optimum value) was highest (>80 °C) for Case 4. This established the scope for higher mass flows and temperatures which are crucial for treating large volumes of ballast water[3]. So, Case 4 with least variation as also the highest optimum sea water temperature was identified as the optimum design and all other geometric values were calculated for this design.

4.0 CONCLUSION

Aviable heat exchanger design employing simple optimisation techniques has been verified. Since cost was the chosen objective, another approach was employed to verify if the objective function values were closer and so the optimisation could be validated. For further validations, a heat exchanger model based on the chosen design has to be erected. The conditions assumed in the design have to be simulated to obtain the calculated temperatures. The species mortalities at these temperatures have to be assessed which will prove the effectiveness of the treatment method.

The scope of these discussions has been limited to cost computations only. The thermodynamic and geometric values used in developing the design and all equations related to heat exchanger design are not projected.

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Table 1 Costs derived for design and verification

Costs (in 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
Cost of energy ashore C_{EL}	0.06/kWh
Reference cost $I_{EXo\ ref}$	250/m^2

Table 2 Summary of cost computation (All costs in US$)

	Area (m^2)	**Annual Cost**	I_{Ppump}, $C_{EL}=0.2$	$I_{Ppump}=0$, $C_{EL}=0.2$	$I_{Ppump}, C_{EL}=0.06$	$I_{Ppump}=0$, $C_{EL}=0.06$	I_{Ppump}, $C_{EL}=0.2$	$I_{Ppump}=0$, $C_{EL}=0.2$	$I_{Ppump}, C_{EL}=0.06$	$I_{Ppump}=0$, $C_{EL}=0.06$
			Interest rate 5% payback 10 years				No Interest & payback			
t_2 Optimum										
Case 1 C_u= 0 C_i = 0.2 C_o= 0.2	363.32	14153.16	22167.63 (-0.57)	20292.63 (-0.43)	18833.99 (-0.33)	16958.99 (-0.20)	32704.25 (-1.31)	17704.25 (-0.25)	29370.61 (-1.08)	14370.61 (-0.02)
Case 2 C_u= 0.04 C_i = 0.2 C_o= 0.2	482.38	22124.65	24994.63 (-0.13)	23119.63 (-0.04)	21660.99 (0.02)	19785.99 (0.11)	35060.08 (-0.58)	20060.08 (0.09)	31726.45 (-0.43)	16726.45 (0.24)
Case 3 C_u= 0.04 C_i = 0.2 C_o= 0	474.65	22077.93	24820.50 (-0.12)	22945.50 (-0.04)	21486.86 (0.03)	19611.86 (0.11)	34914.97 (-0.58)	19914.97 (0.10)	31581.33 (-0.43)	16581.33 (0.25)
Case 4 C_u= 0.04 C_i = 0 C_o = 0.2	494.96	17477.46	25275.60 (-0.45)	23400.60 (-0.34)	21941.96 (-0.26)	20066.96 (-0.15)	35294.22 (-1.02)	20294.22 (-0.16)	31960.58 (-0.83)	16960.58 (0.03)
t_2 fixed										
Case 5 No costs; Software	383.70	n.a	22675.85	20800.85	19342.22	17467.22	33127.77	18127.77	29794.13	14794.13
Case 6 C_u= 0.04 C_i = 0.2 C_o= 0.2	435.61	20302.73	23922.63 (-0.18)	22047.63 (-0.09)	20588.99 (-0.01)	18713.99 (0.08)	34166.75 (-0.68)	19166.75 (0.06)	30833.11 (-0.52)	15833.11 (0.22)
Case 7 C_u= 0 C_i = 0.2 C_o= 0	428.56	16924.19	23757.02 (-0.40)	21882.02 (-0.29)	20423.39 (-0.21)	18548.39 (-0.10)	34028.74 (-1.01)	19028.74 (-0.12)	30695.11 (-0.81)	15695.11 (0.07)
Case 8 C_u= 0 C_i = 0 C_o= 0.2	434.90	12427.60	23906.00 (-0.92)	22031.00 (-0.77)	20572.36 (-0.66)	18697.36 (-0.50)	34152.89 (-1.75)	19152.89 (-0.54)	30819.25 (-1.48)	15819.25 (-0.27)