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Performance Evaluation of a Shell-and-Tube Heat Exchanger in a Reefer Vessel Voyaging from Deep Sea into Inland Waters

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Abstract-This study analyzed the performance parameters of a shell and tube heat exchanger in a reefer vessel refrigeration plant, as the vessel voyaged from the deep-sea water into inland water. Sea water cooling inlet temperature and pressure recordings for 10 days in both deep-sea water and inland water were collected from the vessel refrigeration plant log book and analyzed using various energy equations implemented in MATLAB. The result of the analysis showed that Logarithmic Mean Temperature Difference, heat duty, overall heat transfer coefficient, effectiveness, fouling factor, and pressure drop are, respectively, 38.99^{o}C, 117706.80W, 766.20W/m^{2o}C, 0.05, 0.0001m^{2o}C/W and 0.016239bar. Within the period the vessel voyaged in the inland water way, logarithm mean temperature difference, fouling factor and pressure drop increased to 40.61^{o}C, 0.0007m^{2o}C/W and 0.016400bar. Heat duty, overall heat transfer coefficient and effectiveness decreased to 82465.82W, 532.65W/m^{2o}C and 0.0352, respectively. The result is a strong indication that the heat exchanger had deviated from the design specifications and the major phenomenon that has such significant adverse impacts on heat exchanger performance and effectiveness is attributed to fouling, which can consequently render the heat exchanger inadequate for the services. It was therefore recommended that maintenance work be carried out periodically as long the vessel is in the inland water ways.

Keywords*-Deep-Sea and Inland Water Temperatures, Fouling Factor, Pressure Drop, Reefer Vessel Refrigeration Plant, Shell-and-Tube Heat Exchanger*

I. INTRODUCTION

The Shell-and-tube heat exchanger is an integral component of the reefer vessel refrigeration plant, which transports frozen foods from one continent to another [1]. Refrigerant flows through the shell with a high temperature while seawater is pumped through the tube bundles in order to reduce the high temperature of the refrigerant. However, performance of the shell-and-tube heat exchanger will be reduced when the tubes come in contact with seawater polluted by oil spills, routine shipping, run-offs and dumping on daily basis.

Reference [2] studied the influence of fouling on heat exchanger effectiveness in a polyethylene plant. Data were obtained through steady state monitoring and direct measurements from the plant, and analyzed using various energy equations and a computer program to determine the performance parameters, including the overall heat transfer coefficient that was 51.60% less than the design value.

Reference [3] evaluated the thermal performance of an industrial heat exchanger for process in the offshore industry. Steady state monitoring and data were collected and evaluated for three performance scenarios of the exchanger. The results showed performance trends and deviations from the design values which were attributed to fouling.

Reference [4] presented a paper on impact of nonuniform fouling on operating temperatures in heat exchanger networks. The heat exchangers in the network were subjected to various fouling mechanisms that were simulated. The simulation results indicated that nonuniform fouling in heat exchangers are significantly higher than those predicted by conventional analyses using uniform fouling.

Reference [5] presented a study on performance analysis of a shell-and-tube heat exchanger. The study investigated the effect of shell diameter, tube length and baffle spacing on heat transfer coefficient and pressure drop for both triangular and square pitches. The results showed that increasing tube length by 0.61m gave 2.2% increase in heat transfer coefficient and 21.9% increase in pressure drop.

Reference [6] carried out a research on the kinetics and mechanisms of fouling in crude oil heat transfer. Kinetic model equations of fouling deposition and removal processes were developed and solved with an iterative method. The results showed the influence of fluid velocity and temperature on the fouling rate, and highlighted possibilities to mitigate fouling by minimization of particles initially present in the crude oil.

Reference [7] presented a paper on basic heat exchanger performance evaluation method on ocean thermal energy conversion. The paper proposed a new simplified overall performance evaluation method for heat exchangers, which takes into consideration the heat transfer, effectiveness and pressure drop of the heat exchanger, and can be applied to evaluate the performance of existing heat exchangers.

Thus, a significant number of researches have been conducted in an attempt to evaluate various performance criteria of shell-and-tube heat exchangers in different heat transfer plants. However, many concluded that more research is needed to evaluate the performance of modern-day heat exchangers in various heat transfer plants, which often deviate significantly from the design performance criteria. This study employed the heat energy equations implemented in MATLAB to evaluate the various performance parameters of shell-and-tube heat exchangers in a reefer vessel refrigeration plant voyaging from deep-sea into inland water that is polluted with oil spills.

II. MATERIALS AND METHODS

The materials used in this study are the records of the sea water cooling outlet temperature and pressure from the refrigerator log book, catalogue of reefer vessel heat exchanger, and thermophysical properties of sea water [8] and thermophysical properties of Refrigerant 22. [9]

The methods used in this study are discussed in the following sections:

A. *Description of the Refrigeration Plant*

The reefer vessel refrigeration plant contains four major components: the compressor, heat exchanger (condenser), expansion device and evaporator. Refrigerant remains piped between these four components and is contained in the refrigerant loop. The refrigerant begins as a cold vapor and heads to the compressor as shown in Figure 1. The working fluids in the heat exchanger are chlorodifluoromethane (R-22) and seawater. The seawater is pumped through the sea chest of the vessel to the heat exchanger to cool the R-22 looping the refrigeration plant.

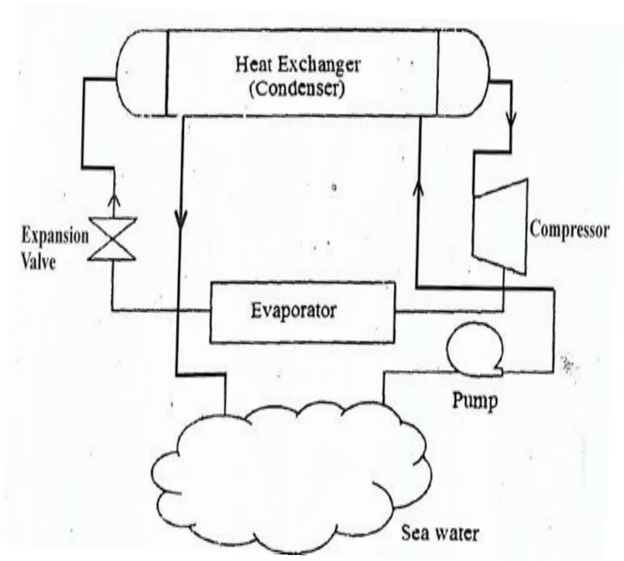

Figure 1. Schematic of a Reefer Vessel Refrigeration Plant [10]

B. *Assessing Performance Data*

The data collected include the temperature of seawater pumped in and out of the heat exchanger when the vessel voyaged in deep-sea and in inland water.

C. *Performance Evaluation*

Energy equations were used to analyze and evaluate the performance of the heat exchanger at intervals of 20 days; 10 days in deep sea and 10 days inland water. The performance parameters in deep sea and inland were analyzed and compared with the design specifications.

D. *Thermodynamic Equations*

Figure 2 shows the schematic diagram that graphically illustrate the temperature distribution in the heat exchanger.

Figure 2. Schematic of a Shell and Tube Heat Exchanger

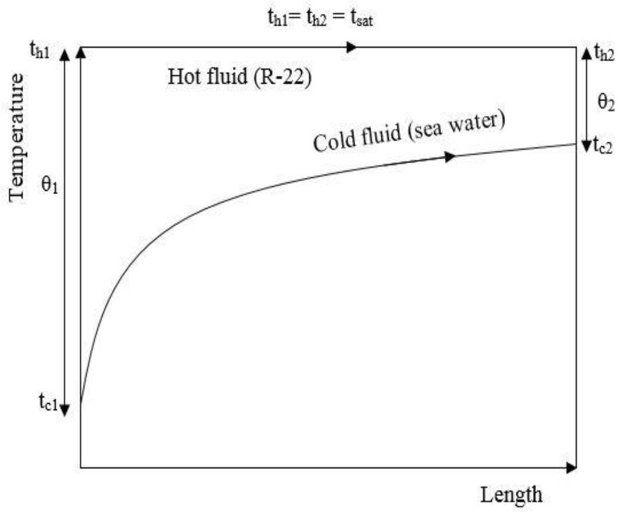

Figure 3. Schematic of Temperature Distribution [11]

1) *Heat Transfer*

The heat transfer in the heat exchanger is both by cold and hot fluid, and are given respectively as: (Holman, 2002)

$$Q_c = \dot{m}_c C_{pc}(t_{c2} - t_{c1}) \tag{1}$$

and

$$Q_h = \dot{m}_h h_{fg} \tag{2}$$

where $\dot{m}_c$ = mass flow of sea water (kg/s), C_{pc} = heat capacity at constant pressure for sea water (kJ/kgK), t_{c2} = outlet temperature of sea water (°C), t_{c1} = Inlet temperature of sea water (°C), $\dot{m}_h$ = mass of hot fluid (kg/s), h_{fg} = latent heat of hot fluid (kJ/kg).

2) Heat Duty

The heat duty is the amount of heat needed to transfer from the hot side to the cold side per unit time. Heat duty (Q) is given as [12]

$$Q = m_c C_{pc}(t_{c2} - t_{c1}) = m_h h_{fg} \tag{3}$$

3) Logarithm Mean Temperature Difference (LMTD)

LMTD is average temperature difference across the tube length and accounts for the exponential decay nature of temperature along the tube. LMTD (Θ_m) is given as [13].

$$\theta_m = \frac{(t_{h1} - t_{c2}) - (t_{h2} - t_{c1})}{\ln \dfrac{(t_{h1} - t_{c2})}{(t_{h2} - t_{c1})}} \tag{4}$$

where t_{h1} = inlet temperature of R-22 (°C), t_{h2} = outlet temperature of R-22.

4) Overall Heat Transfer Coefficient

The overall heat transfer coefficient (U_o) is expressed in Watt per meter square Kelvin(W/m^{2}K) and is given as [11]

$$U_o = \frac{Q}{N_t \pi D_t L_t \theta_m} \tag{5}$$

where N_t = number of tubes, D_t = diameter of tube (m), L_t = length of tube (m)

5) Effectiveness

The effectiveness of a heat exchanger can be defined in terms of number of transfer unit (NTU), which is the ratio of the product of overall heat transfer coefficient and the contact surface area to the minimum heat capacity rate of the transfer fluid [14]. NTU and effectiveness (ε) of a heat exchanger are given respectively as [11]

$$NTU = \frac{U_o N_t \pi D_t}{C_{\min}} \tag{6}$$

$$\varepsilon = 1 - e^{(-NTU)} \tag{7}$$

where C_{min} = minimum of heat capacity of seawater or heat capacity of R-22 (kJ/K).

6) Fouling Factor

The fouling factor is the theoretical resistance to heat transfer resulting from build-up of a layer of dirt on the tube surfaces of the heat exchanger. Fouling factors are determined for different data as [12]

$$R_{fc} = \frac{1}{U_{dirty}} - \frac{1}{U_{clean}} \tag{8}$$

where U_{dirty} = overall heat transfer coefficient in dirty conditions (W/m^{2}K), U_{clean} = overall heat transfer coefficient in clean situations (W/m^{2}K).

7) Pressure Drop

Pressure drop is created when a flow is disturbed, due to difference between the flow pressure at the beginning of a passage (that is always higher) and the flow pressure at the end of the passage [14]. The pressure drop in a tube depends on different variables which are determined as follows:

a) Area of Cross Flow

Area of cross flow (A_c) is expressed in metre squared (m^2) and is given as [14].

$$A_c = \frac{\pi (D_t^2)(N_t)}{4N_p} \tag{9}$$

where N_p= number of passes.

b) Velocity of the Cross Flow

Velocity of the cross flow (V) is expressed in meters per second (m/s) and is given as [14]

$$V = \frac{m_c}{\rho A_c} \tag{10}$$

where ρ = seawater density (kg/m^3).

c) Reynolds Number

The Reynolds number for pipe flow (Re_D) is given as [11]

$$\mathrm{Re}_D = \frac{\dot{m} D_t}{A_c \mu_c} \tag{11}$$

where μ_c = viscosity of seawater (kg/ms)

d) Friction Factor

Friction factor for turbulent flow (f) is given as [11]

$$f = 0.0008 + \frac{0.05525}{(\mathrm{Re}_D)^{0.237}} \tag{12}$$

e) Viscosity Correction Factor

Viscosity correction factor (φ) is given as [12]

$$\varphi = \left(\frac{\mu_h}{\mu_c}\right)^{0.14} \tag{13}$$

where μ_h = viscosity of R-22 (kg/ms).

Pressure drop (ΔP) is given as [12]

$$\Delta P = \frac{f N_p L_t (\rho V)^2}{2\rho s \varphi D_t} \tag{14}$$

where s = specific gravity of seawater

III. RESULTS AND DISCUSSION

The results of the study are presented and discussed as follows:

A. Design Parameters

The heat exchanger in the reefer vessel refrigeration plant was designed to operate with R-22 and sea water as working fluids, R-22 is the hot fluid while sea water is the cold fluid. The design specifications are shown in Table 1

TABLE I. DESIGN SPECIFICATIONS OF THE HEAT EXCHANGER-

Parameters	Hot Fluid in Shell Side	Cold Fluid in Tube Side
Working Fluid	Chlorodifluoromethane(R-22)	Sea Water
Heat Duty		117791.4kW
Mass Flow	0.96kg/s	14.1kg/s
Inlet/Outlet(Temperature)	$70^oC/70^oC$	$29^oC/31^oC$
Design Pressure	19bar	2 bar
Pressure Drop	0.012bar	0.012bar
Tube Length	-	2.850m
Tube Diameter	-	0.020m
Number of Tubes	-	22
Shell Diameter	0.609m	-
Shell Length	3.180m	-
Number of Passes	1	1

Source: [15]

B. Analysis of Performance Parameters

The various energy equations were implemented in MATLAB Program code which was used to determine the performance of the system at different ambient temperatures of the deep-sea water and inland water ways for different days that the vessel voyaged. The results are shown in Tables 2 and 3.

1) Analysis of Performance in Deep Sea

From the refrigerator log book the ambient temperature of 31^oC in deep-sea water remains constant for 5 days the vessel voyaged deep-sea, and the performance of the heat exchanger was analyzed as follows:

a) Analysis of Heat Duty

Applying Equation 1 and 2, the heat transferred by cold and hot fluids are determined, respectively, as: $\dot{m}_c$= 14.1kg/s, t_{c1} = 29^oC, t_{c2} = 31^oC (from Table 1), C_{pc} = 4.177kJ/kgK [8], $\dot{m}_h$= 0.96kg/s, h_{fg} of R-22 at 70^oC = 122.99kJ/kg [9]

$$Q_c = 14.1 \times 4.177 \times (31 - 29) \approx 118kW$$

And

$$Q_h = 0.96 \times 122.99 \approx 118kW$$

Using Equation 3, the heat duty is obtained thus:

$$Q \approx 118kW$$

b) Analysis of LMTD

From Equation 4, the logarithm mean temperature difference is obtained as: $t_{h1} = t_{h2} = 69^oC$, t_{c1}= 29^oC, $t_{c2} = 31^oC$ (Table 1)

$$\theta_m = \frac{(69-29)-(69-31)}{In\left[\frac{69-29}{69-31}\right]}$$

=38.99K

c) Analysis of Overall Heat Transfer Coefficient

Using Equation 5, the overall heat transfer coefficient is calculated as: n = 22, D = 0.02m, L = 2.850m (from Table 1), Q = 117813.96W, Θ_m = 38.99K,

$$U_o = \frac{117813.96}{22 \times \pi \times 0.02 \times 2.85 \times 38.99} = 767W/m^2K$$

d) Analysis of Effectiveness

Applying Equations 6 and 7, respectively, the effectiveness is determined as follows: U_o= 767W/m^{2}K, n = 22, D = 0.02m, L = 2.850m, $\dot{m}_c$= 14.1kg/s, $\dot{m}_h$ = 0.96kg/s (from Table 1), C_{pc} = 4177.80J/kgK [8], C_{ph} of R-22 at 70^oC = 1743J/kgK [9], C_{min} = 0.96 x 1743 = 1673.28W/K

$$NTU = \frac{767 \times 22 \times \pi \times 0.02 \times 2.850}{1673.28} \approx 1.8$$

$$\varepsilon = 1 - e^{(-1.8)} \approx 0.84$$

e) Analysis of Fouling Factor

Equation 8 was used to obtain the fouling factor as: U_{clean} = 767W/m^{2}K, U_{dirty} = 703.72W/m^{2}K

$$R_f = \frac{1}{703.72} - \frac{1}{766.20} = 0.0001m^2K/W$$

f) Analysis of the Pressure Drop

From Equation 9, the heat transfer area was obtained as: D_t = 0.02m, N_t = 22, N_p = 1 (from Table 1)

$$A_c = \frac{\pi \times 0.02^2 \times 22}{4 \times 1} = 0.0069m^2$$

Using Equation 10, the velocity of the fluid is calculated as follows: A_c = 0.0069m^2, $\dot{m}_c$ =14.1kg/s (from Table 1), ρ at 31^oC and 35% salinity of seawater = 988.3kg/m^3 [8]

$$V = \frac{14.1}{988.3 \times 0.0069} = 2.067m/s$$

Applying Equation 11, the Reynolds number is obtained as: A_c = 0.0069m, D_t = 0.02m, $\dot{m}_c$ = 14.1kg/s (from Table 1), μ_c at 31^oC and 35% Salinity of seawater = 555.1 x 10-6 kg/ms [8].

$$Re_D = \frac{14.1 \times 0.02}{0.0069 \times 555.1 \times 10^{-6}} = 73625.59$$

Recall Equation 12, Re_D = 73625.59

$$f = 0.0008 + \frac{0.05525}{973625.59)^{0.235}} = 0.004688$$

Using Equation 13, friction factor is calculated as follows: μ_h = 123.9 x 10^{-6}kg/ms [9], μ_c = 552.10 x 10^{-6}kg/ms [8]

$$\phi = \left(\frac{123.9 \times 10^{-6}}{552.04 \times 10^{-6}}\right)^{0.14} = 0.81062$$

Using Equation 14, the pressure drop is obtained thus: f =0.004683, S = 1, L = 2.85m, ρ = 1022kg/m^3, V = 2m/s, φ = 0.81062, N_p = 1, D_t = 0.02m

$$\Delta P = \frac{0.004688 \times 1 \times 2.85 \times (988.3 \times 2.067)^2}{2 \times 988.3 \times 10.81062 \times 0.02} = 1623.9 \, N/m^2$$

$$= 0.016239 bar$$

TABLE II. SUMMARY OF THE PERFORMANCE PARAMETERS IN DEEP SEA

Outlet Temperature (oC)	Heat Duty (kW)	Logarithmic Mean Temperature Difference (oC)	Overall Heat Transfer Coefficient(W/m2oC)	Effectiveness	Pressure Drop(bar)	Fouling factor(m2oC/W)
31	118	38.99	766.20	0.84	0.016239	0.0001
31	118	38.99	766.20	0.84	0.016239	0.0001
31	118	38.99	766.20	0.84	0.016239	0.0001
31	118	38.99	766.20	0.84	0.016239	0.0001
31	118	38.99	766.20	0.84	0.016239	0.0001

Table 2 shows that the performance parameters of the heat exchanger in the deep-sea water with constant outlet temperature.

2) Analysis of the Performance in Inland Water

The performance parameters in inland water were analyzed using the outlet temperature of seawater recorded in the in the refrigerator log book for day 11 that the vessel started voyaging inland water as follows:

a) Analysis of Heat Duty

Applying Equation 1 and 2, the heat transferred by cold and hot fluids are determined, respectively, as: $\dot{m}_c$= 14.1kg/s, t_{c1} = 29^{o}C, t_{c2} = 30.47^{o}C, (from refrigerator log book), C_{pc} = 4.177kJ/kgK [8], : $\dot{m}_h$= 0.7kg/s, h_{fg} of R-22 at 70^{o}C = 122.99kJ/kg [9]

$$Q_c = 14.1 \times 4.177 \times (30.47 - 29) = 86.58 kW$$

$$U_o = \frac{86580}{22 \times \pi \times 0.020 \times 2.850 \times 40.46} = 539.41 W / m^2 K$$

and

$$Q_h = 0.7 \times 122.99 \approx 86.58 kW$$

Using Equation 3, the heat duty is obtained thus: $Q \approx 86.58 kW$

b) Analysis of LMTD

From Equation 4, the LMTD is calculated as follows: t_{h1} = t_{h2} = 70^{o}C, t$_{c1}$= 29^{o}C, (from Table 1), t_{c2} = 30.47^{o}C (from refrigerator log book)

$$\theta_m = \frac{(70-29)-(70-30.47)}{\ln\left(\frac{70-29}{70-30.47}\right)} = 40.46^o C$$

c) Analysis of Overall Heat Transfer Coefficient

Equation 5 was used to determine overall heat transfer coefficient; n = 22, D_t = 0.02m, L = 2.850m (from Table 1), Q_c = 86.58kW, Θ_m = 40.46^{o}C

$$NTU = \frac{539.41 \times 22 \times \pi \times 0.02 \times 2.850}{1220.1} \approx 1.7$$

$$\varepsilon = 1 - e^{(-1.7)} \approx 0.82$$

d) Analysis of Effectiveness

Applying Equation 6 and Equation 7, respectively, the effectiveness is obtained as: U_o= 539.41W/m^{2}K, n = 22, D = 0.02m, L = 2.850m, $\dot{m}_c$= 14.1kg/s, $\dot{m}_h$ = 0.7kg/s, C_{pc} = 4177J/kgK [8], C_{ph} of R-22 at 70^{o}C = 1743J/kgK [9], C_{min} = 0.7 x 1743 = 1220.1W/K

e) Analysis of Fouling Factor

Using Equation 8, the fouling factor is calculated as follows: U_{clean} = 766.20W/m^{2}K, U_{dirty} = 539.41W/m^{2}K

$$R_f = \frac{1}{703.72} - \frac{1}{766.20} = 0.0001 m^2 K / W$$

f) Analysis of the Pressure Drop

Equation 9 was used to compute the available heart transfer area as follows; D_t = 0.02m, N_t = 22, N_p = 1 (Table 1)

$$A_c = \frac{\pi \times 0.02^2 \times 22}{4 \times 1} = 0.0069 m^2$$

Equation 10 was used to compute the velocity of the cold stream as follows; A_c = 0.0069m^2, $\dot{m}_c$ = 14.1kg/s (Table 4.1), ρ at 30.47^{o}C and 35% Salinity of seawater = 1022kg/m^3 (Nayar *et al.*, 2016)

$$V = \frac{14.1}{1022 \times 0.0069} = 2 m / s$$

Applying Equation 11 the Reynolds number is obtained as follows; A_c= 0.0069m, D_t = 0.02m, $\dot{m}_c$ = 14.1kg/s (Table 1), μ at 30.47°C and 35% Salinity of seawater = 559.06 x 10^{-6} kg/ms [8]

$$\mathrm{Re} = \frac{14.1 \times 0.02}{0.0069 \times 559.06 \times 10^{-6}} = 72464$$

Equation 12 was applied to calculate the friction factor as follows: Re = 72464

$$f = 0.0008 + \frac{0.05525}{(72464)} = 0.004695$$

Applying Equation13, the viscosity correction factor is determined as follows: μ_h = 124.09 x 10^{-6} kg/ms, μ_c = 559.06 x 10^{-6} kg/ms

$$\phi = \left(\frac{124.09 \times 10^{-6}}{559.06 \times 10^{-6}}\right)^{0.14} = 0.80998$$

Using Equation 14, the pressure drop is obtained thus: f =0.004695, S = 1, L = 2.85m, ρ = 1022kg/m^3, V = 2m/s, φ = 0.80998, N_p = 1, D_t = 0.02m

$$\Delta P = \frac{0.004695 \times 1 \times 2.850 \times (2 \times 1022)^2}{2 \times 1022 \times 1 \times 0.80998 \times 0.02} = 1633.23\, N/m^2$$

$$= 0.01633 bar$$

Table 3 shows changes in the performance parameters as the vessel voyaged inland waters with changes in temperature for the remaining 5 days.

TABLE III. PERFORMANCE PARAMETERS IN INLAND WATER IN WET SEASON

Outlet Temperature (°C)	Heat Duty (kW)	Logarithmic Mean Temperature Difference (°C)	Overall Heat Transfer Coefficient (W/m^{2o}C)	Effectiveness	Pressure Drop (bar)	Fouling factor (m^{2o}C/W)
30.85	1082.03	39.03	703.72	0.0463	0.016297	0.0002
30.68	982.60	39.25	635.49	0.0419	0.016306	0.0004
30.55	906.56	39.31	586.46	0.0388	0.016319	0.0005
30.47	859.76	40.46	539.41	0.0357	0.016332	0.0006
30.42	824.66	40.61	532.65	0.0352	0.016400	0.0007

3) Variation of Performance Parameters with Outlet Temperature

In order to facilitate the observation of the variation trend of the performance parameters with the different outlet temperatures of seawater the tabular values were plotted.

Figure 4 shows that the heat duty decreases as the vessel started voyaging inland water. Thus, changing the cooling fluid from deep sea water to inland water decreased the heat duty from 1082.03kW up to 824.66kW for the 5days that the vessel voyaged inland water. This is an indication that the impurities in the inland water cause clogging in the tube of the heat exchanger, thereby reducing its diameter and causing a reduction of heat transfer area.

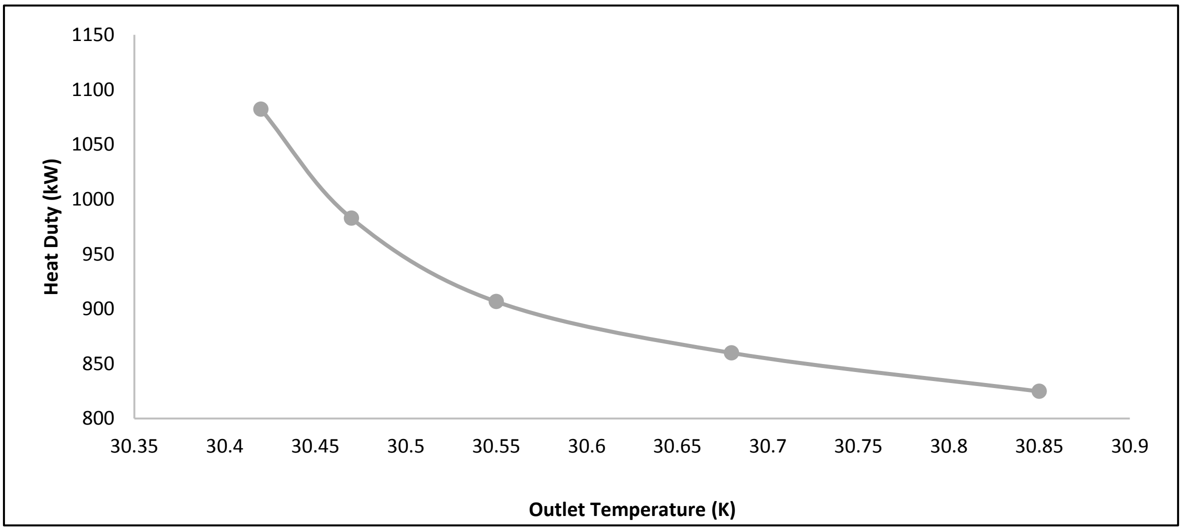

Figure 4. Graph of Heat Duty Against Outlet Temperatures of Seawater

Figure 5 shows that the LMTD increases as the vessel started voyaging inland water. Thus, changing the cooling fluid from deep sea water to inland water increased the LMTD from 38.99^{o}C up to 40.61^{o}C for the 5days that the vessel voyaged inland water which could be due to fouling.

Figure 6 shows that the Overall Heat Transfer Coefficient deceases as the vessel started voyaging inland water. Thus, changing the cooling fluid from deep sea water to inland water decreased the heat transfer coefficient from 766.20W/m^{2o}C up to 532.65W/m^{2o}C for the 5days that the vessel voyaged inland water which could be due to fouling, clogging and pressure drop.

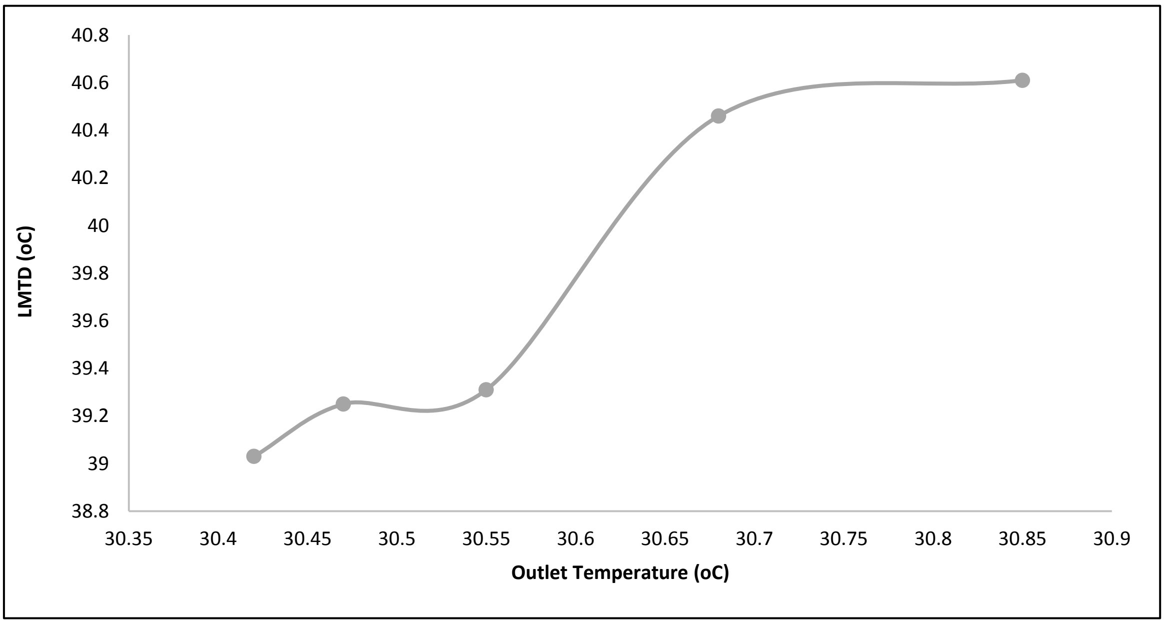

Figure 5. Graph of LMTD Against Outlet Temperatures of Seawater

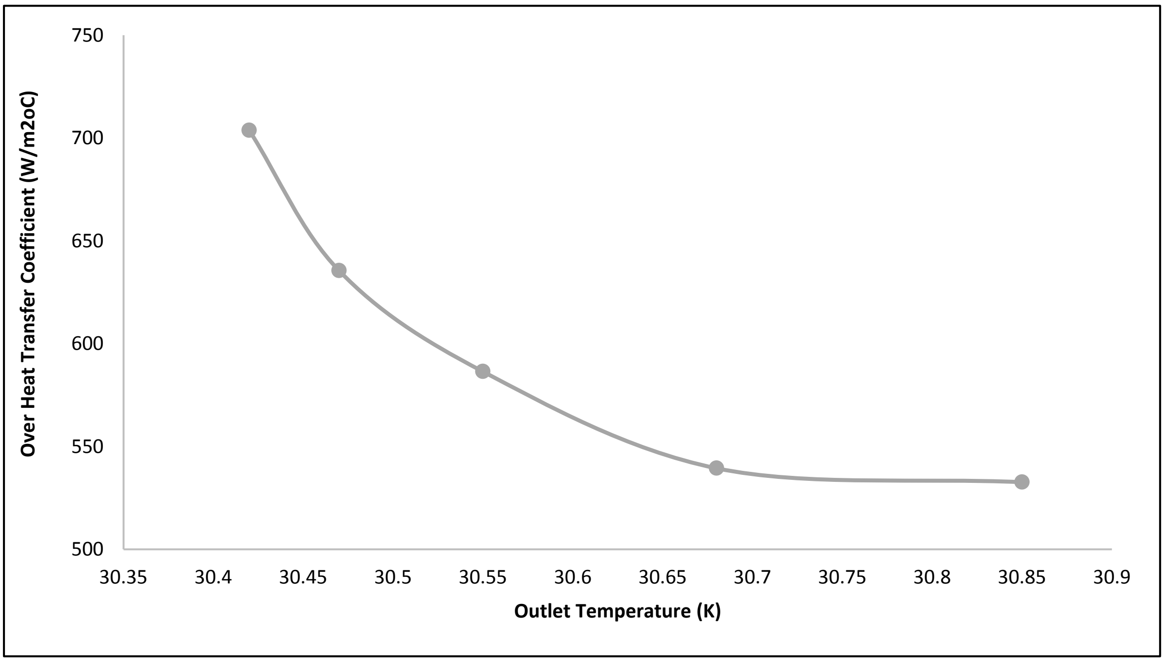

Figure 6. Graph of Overall Heat Transfer Coefficient Against Outlet Temperatures

IV. CONCLUSION

The performance parameters of a shell and tube heat exchanger in a reefer vessel refrigeration plant were analyzed as the vessel voyaged from the deep-sea water into the inland waterway. Sea water cooling outlet temperature and inlet pressure recordings for 10 days were collected from the vessel refrigeration plant log book and

analyzed using various energy equations encoded in MATLAB program code. The findings could be summarized as follows:

1) The analysis carried out on the heat exchanger shows that the performance parameters of the heat exchange remained constant as the vessel voyage the deep-sea water and continually changed as the vessel voyaged in the inland water way to wharf.

2) LMTD, Fouling factor and pressure drop of 38.99^{o}C, 0.0001m^2/^{o}C and 0.016239bar, respectively, were maintained in deep sea and increased to 40.61^{o}C, 0.0007m^2/^{o}C and 0.016400bar respectively as the vessel voyaged in the inland water way for the 5days.

3) Heat duty, overall heat transfer coefficient and effectiveness of 118kW, 766.20W/m^{2o}C and 0.084, respectively, were maintained in deep sea and decreased from 1082.03kW to 824.66kW, 703.72W/m^{2o}C to 532.65W/m^{2o}C and 0.0463 to 0.0352, respectively, as the vessel voyaged in the inland water way.

4) The decrease in the heat transfer of cold fluid is an indication that the impurities in the inland water causes clogging in the tube of the heat exchanger, that reduces it diameter thereby causing insufficient area for heat transfer. The reduction in the tube diameter also resulted in high pressure drops and which could cause mechanical-induced vibrations which might reduce the life span of the heat exchanger

In conclusion, this analysis shows a strong indication that the heat exchanger had deviated from the design specifications and the major phenomenon that could cause such significant adverse impacts on heat exchanger performance is attributed to fouling.

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