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DESIGN CALCULATIONS FOR THE COOLING WATER SYSTEM OF A TUG BOAT

Samson Nitonye *

Department of Marine Engineering, Rivers State University, Port Harcourt, Nigeria.

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***Corresponding Author**
Samson Nitonye
Department of Marine Engineering, Rivers State University, Port Harcourt, Nigeria.

ABSTRACT

The cooling water system is primarily used to ensure that the temperature of the main engine is kept within the acceptable limit for maximum performance during operations. The system also is used to remove or conduct heat out of hot surfaces or materials either directly or indirectly. This system comprises of different components, sub-systems, piping and fittings, valves, sea water and fresh water pumps, filters, tanks and condition monitoring device amongst other. The sea water cooling system was calculated mathematically to obtain the volumetric flow rates of the sea water for the cooling of the shaft-line bearing, reduction gear lube-oil, and the hot well. Similarly the fresh water flow rate and the transfer surface areas within the main engine were also obtained for the jacket water cooler, piston water cooler, crankcase lube-oil cooler and charger air cooler. All designs were carried out obeying classification rules relating to this capacity of the Sea-going Tug Boat bearing in mind the capacity of the load to be transported.

KEY WORDS: Tug Boat, Pumps, Coolers, Valves, Piping, Fittings and Condition Monitoring device, Vessel, Discharge.

INTRODUCTION

The primary purpose for the cooling water system is to remove or conduct heat out of hot surfaces or materials from the engine either directly or indirectly. Cooling enables the engine metal to retain the mechanical properties and keep the temperature of the main engine within the acceptable limit for maximum performance during operations. There are three basic types of cooling system commonly used in the marine diesel engines on board the vessel. These include the direct cooling system, the keel cooling system and the heat exchanger cooling system.[1]

The direct cooling system involves drawing sea water from the sea in which the ship is floating and then circulate round the engine by the use of pumps which is later expelled overboard. The associated problem to this system is ensuring an optimum coolant temperature and secondly, contamination of the water supply with consequent deposition inside the engine. Furthermore, the system is subject to high rate of corrosion and erosion due to the nature of sea water. Cause this effects sea water is no longer used for direct coolant, although it is a subsidiary coolant in the heat exchangers.[2]

The keel cooling system involves freshwater circulating in a close circuit of the engine, part of which passes along pipes exposed to cooling influence of the surrounding sea water along the keel areas. This method eliminates some problems of the direct cooling. It is also unaffected by the cleanness of the water in which the ship is floating. A thermostat to maintain optimum operating temperature may be higher than that of the direct cooling system.

The heat exchanger cooling system operates on the same principle as the keel cooler, except a special radiator is incorporated in the freshwater circuit instead of cooling pipes along the keel. The system uses two water pumps, one of which is of sea water drawn in through the bottom of the ship via a strainer combined with a cock. The later circulates the freshwater which can be closed when the engine is not in operation. The advantage of the keel cooling and that of the heat exchanger is that the sea water corrosion is reduced. Generally, cooling system for this type consist of two installation namely freshwater and sea water installations.[3]

The Fresh Water Cooling System

The freshwater installation is built in close circuit. It led to the provision of header tanks in the cooling system to ensure the availability of freshwater for the cooling device at all times. The fresh water used for cooling engine components directly must meet certain requirements. The prevention and formation of scales, the hardness of freshwater must be controlled and the P^H value should be slightly alkaline.[8 to 9]

Figure 1 shows a fresh water cooling system for a slow-speed diesel engine. This can be divided into two separate systems: one for cooling the cylinder jackets, cylinder heads and the turbo-blower; and the for piston cooling

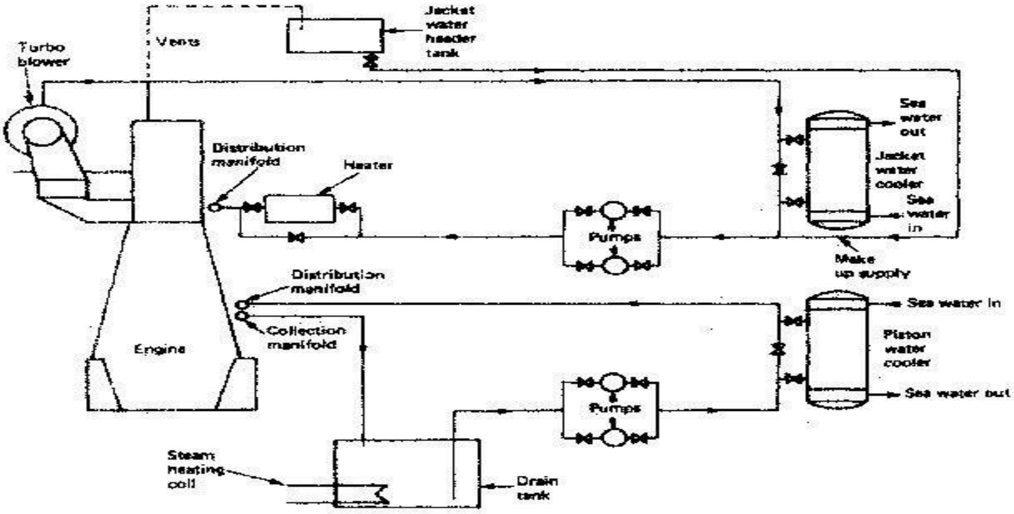

Figure 1: Fresh Water Cooling System for a Slow-Speed Diesel Engine (source – Machinery Spaces.com).

The cylinder jacket cooling water after leaving the engine passes to the sea-water-circulated cooler and the into the jacket-water circulating pumps. It is then pumped around the cylinder jacket, cylinder heads and turbo-blowers. A header tank allows for expansion and water make-up in the system. Vents are led from the engine to the header tank for the release of air from the cooling water. A heater in the circuit facilitates warming of the engine prior to starting by circulating hot water.

The piston cooling system employs similar components, except that a drain tank is used instead of a header tank and the vents are then led to high points in the machinery space. A separate piston cooling system is used to limit any contamination from piston cooling glands to the piston cooling system only.[4]

The Sea Water Cooling System

The sea water cooling system is an open circuit in which the salt water is forced to circulate round the system to extract heat from the freshwater mainly in the heat exchanger. It is later allowed to be discharge overboard. Sea water is not directly used to cool engine component because it is highly corrosive, it is aggressive at elevated temperature, it contains constantly amount of impurities (which after deposition deteriorate engine working performance) and it contains calcium carbonate which causes difficulties to heat transfer process.

The various cooling liquids which circulate the engine are themselves cooled by sea water. The usual arrangement uses individual coolers for lubricating oil, jacket water, and the piston cooling system, each cooler being circulated by sea water. Some modern ships use what is known as 'central cooling system' with only one large sea-water-circulated cooler. This cools a supply of fresh water, which then circulates to the other individual coolers. With less equipment in contact with sea water the corrosion problem are much reduced in this system.

A sea water cooling system is shown in figure 2, from the sea suction one of a pair of sea water circulating pumps provides sea water which circulates the lubricating oil cooler, the jacket water cooler and the piston water cooler before discharging overboard. Another branch of the sea water main provides sea water to directly cool the charge air (for a direct-drive two stroke diesel)

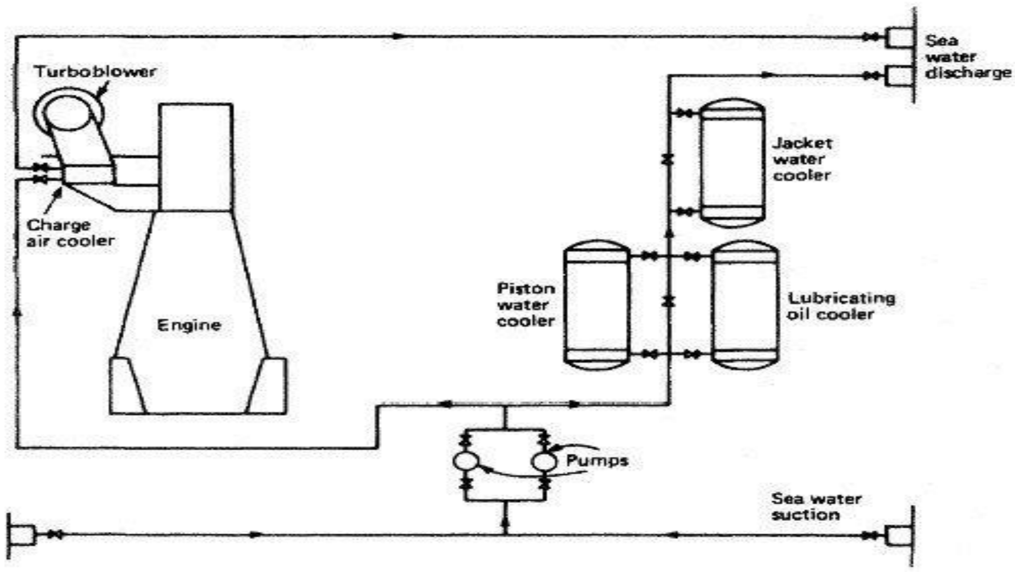

Figure 2: Sea Water Cooling System for a Slow-Speed Diesel Engine (source – Machinery Spaces.com)

The components of the cooling water system include; the sea water pump, the fresh water pump, the coolers valves, piping and fittings and condition monitoring device. All of which have different purpose to ensure that the cooling water system performs the required duty.

Relevant Classification Regulation and Requirement for Engine Cooling Water System

In the main supply, provision is to be made for an adequate supply of cooling water to the main propeller machinery and essential auxiliary engines, also to the lubricating oil and fresh water coolers and air coolers for electric propelling machinery where their coolers are fitted. The cooling water pump(s) may be worked from the engines or driven independently.[5]

The sea inlet for the cooling water system should not be less than two which will be provided with pumps to supply salt water for the cooling system. One of the pumps will serve as the main pump and the other as standby. Those inlets are to be low inlets and one of them may serve as the ballast pump or general service pump.

The cooling water supply to auxiliaries and main engines should be fitted with strainers from the suction pipes which can clean without interruption to the cooling water supply. Cooling water pumps worked from the main engines are provided with relief valves on the pumps discharge. When non-ferrous pipes are proposed for fresh and sea water piping systems, details of the materials and the duty for which they are intended may be submitted to the relevant society for approval. In the selection components for sea water piping system, care should be taken to avoid metal combinations which may lead to galvanic corrosion in service.[6]

Special considerations in the design of the cooling water system of the tug boat

To enhance the working condition of the tug boat at full load and other sea conditions we must take a critical account of the stability of the vessel during any operation.[7] and [8]

2. MATERIALS AND METHODS

The cooling water system design calculations for component, piping and fittings for the Tug boat engines and generators.

The capacity of the engines for the design of the cooling water system included

The Starboard side Main Engine	- 955KW and 1800 rpm
The Port side Main Engine	- 955KW and 1800 rpm
The Starboard side Aux-Engine	- 82KW and 1500 rpm
The Port side Aux-Engine	- 82KW and 1500 rpm
FI-FI pump set	- 133KW and 1800 rpm[9]

General Data for Diesel Fuel Oil (D.F.O)

- Net Calorific value of D.F.O. = 42700 KJ/kg
- Mean overall heat transfer coefficient of fuel = 760KJ/m^2 deg hr
- Specific gravity of .D.F.0.= 900 kg/m^3
- Specific heat of D.F.O. = 1.82KJ/kg°C
- Viscosity of diesel fuel oil -5-15 cst at 20°C

Other parameters of the tug boat

Length of Boat	-	28m
Breadth of Boat	-	9m
Moulded depth of Boat	-	4.5m
Gross Tonnage	-	180
Net Tonnage	-	55

Cooling of shaft-line Bearings

The volumetric rate of flow of sea water can be established from the relation

$$V = \frac{Q_t}{C_{sw}\ x\ \Delta T\ x\ \gamma}\ \frac{m^3}{hr} \qquad (1)$$

Where:-

Q_t - Total energy loss due to friction in all shaft-line bearing KJ/hr

C_{sw} - Specific heat of the sea water = 3.925 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 4^{0}C

γ – Density of sea water = 1020 Kg/m^3

The total energy loss is assessed with the fact that the energy loss to stern tube bearing is approximately equal to 1.5% of the transmitted power

The transmitted power = 955KW

Therefore energy loss to stern tube bearing = 14.33KW

Similarly the energy loss to shaft-line bearing is approximately equal to 0.4% of the transmitted power

The transmitted power = 955KW

Therefore energy loss to shaft-line bearing = 3.82KW

∴ The total energy loss due to friction in all shaft-line bearing

$= 14.33 + 3.82 = 18.15\ KW$

By Simple substitution into equation 1

$$V = \frac{18.15}{3.925\ x\ 4\ x\ 1020} = 4.08\ {}^{m^3}/_{hr}$$

Cooling of reduction gear lubrication oil

The volumetric rate of flow of sea water can be established from the relation

$$V = \frac{Q_t}{C_{sw}\ x\ \Delta T\ x\ \gamma}\ \frac{m^3}{hr} \qquad (2)$$

Where

Q_t - Total energy loss due to friction in the reduction gear KJ/hr

C_{sw} - Specific heat of the sea water = 3.925 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 5.5^{0}C

γ – Density of sea water = 1020 Kg/m^3

The total energy loss is assessed with the fact that the energy loss to reduction gear is approximately equal to 0.5% of the transmitted power

The transmitted power = 955KW

Therefore energy loss to reduction gear = 4.775KW

Similarly the energy loss to the clutches and couplings is approximately equal to 4% of the transmitted power

The transmitted power = 955KW

Therefore energy loss to the clutches and couplings = 38.2KW

∴ The total energy loss due to friction in the reduction gear

$= 4.775 + 38.2 = 42.975\ KW$

By Simple substitution into equation 2

$$V = \frac{42.975}{3.925\ x\ 5.5\ x\ 1020} = 7.026\ {m^3}/{hr}$$

Cooling of Hot well

The volumetric rate of flow of sea water can be established from the relation

$$V = \frac{D\ x\ C_c\ x\ \Delta T_c}{C_{sw}\ x\ \Delta T\ x\ \gamma}\ \frac{m^3}{hr} \qquad (3)$$

Where:-

D – Mass flow rate of condensate drain

C_c = Specific heat of Condensate = 4.200KJ/Kg^{0}C

C_{sw} - Specific heat of the sea water = 3.925 KJ/Kg^{0}C

ΔT_c – Condensate Temperature drop = 35^{0}C

ΔT - Temperature rise of sea water = 12^{0}C

γ – Density of sea water = 1020 Kg/m^3

The rate of flow of cooling media is given as

$$Q = \frac{H_G}{\Delta T \; x \; C_p} \tag{4}$$

$$\therefore H = Q \; x \; \Delta T \; x \; C_p \tag{5}$$

Where

Q – Mass flow rate of coolant Kg/hr

H_G = Heat absorbed by coolant KJ/hr

C_p - Specific heat of coolant = 1.942 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 10^{0}C

The heat absorbed by the coolant is approximately equal to 0.42% of the transmitted power

The transmitted power = 955KW

Therefore heat absorbed by the coolant H_G = 4.011KW

Substituting into equation 4

Q = 0.207 Kg/hr

Hence to obtain the volumetric rate of sea flow, we substitute into equation 3

$$V = \frac{0.207 \; x \; 4.200 \; x \; 35}{3.925 \; x \; 12 \; x \; 1.020} = 0.632 \, {m^3}/{hr}$$

Fresh water cooling system of Main Engine

Total heat rejected to the fresh water cooling media of the diesel engine can be determined with the relation

$$HCM = (LHV \; x \; sfc - 3600)[1 - ff] KJ/KW \;\; hr \tag{6}$$

Where

HCM – Total heat rejected by coolant

LHV – Lower heating value = 42700 KJ/Kg

sfc - Specific fuel consumption 0.200KG/KW hr

Note

For Marine diesel oil ff = 0.55

For I.F.O oil ff = 0.56

For Heavy Diesel oil ff = 0.57

By simple substitution

$$HCM = (42700 \; x \; 0.2 \; - 3600)[1 - 0.55] KJ/KW \;\; hr$$

$$HCM = 2223 \; KJ/KW \;\; hr$$

In the cooling system, the total heat rejected in the engine is distributed to the cylinder jacket, piston cooling, crankcase cooling and charge air cooler at the ratio of 54:15:3:28 percent respectively

Cylinder Jacket Heads and Turbo Charger cooling

The heat dissipated to the cylinder jacket water cooler is 54% of the total heat rejected (2223KJ/KW hr)

∴ H_d = 1200.42 KJ/KW hr

And the rate of flow of jacket water cooler is

$$Q = \frac{H_d}{\Delta T \ x \ C_p} \qquad (7)$$

Q – Mass flow rate of coolant Kg/hr

H_d = Heat absorbed by coolant = 1200.42KJ/KW hr

C_p - Specific heat of coolant = 4.203 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 16^{0}C

The rate of flow of jacket (fresh) water is obtained by substituting into equation 7

$$Q = \frac{1200.42}{16 \ x \ 4.203} = 17.851 \ {m^3}/{hr}$$

The rate of flow of jacket (sea) water is obtained by substituting into equation 7

$$Q = \frac{600.21}{6.6 \ x \ 3.925} = 23.17 \ {m^3}/{hr}$$

And related known data for the jacket water cooler includes

- Fresh water inlet temperature (T_1) = 85^{0}C
- Fresh water outlet temperature (T_2) = 69^{0}C
- Sea water inlet temperature (t_1) = 41.4^{0}C
- Sea water inlet temperature (t_2) = 48^{0}C

To obtain the log mean temperature difference (LMTD =ΔT_m)

$$\Delta T_m = LMTD = \frac{\Delta T_2 - \Delta T_1}{In\frac{\Delta T_2}{\Delta T_1}} \qquad (8)$$

Where

$\Delta T_2 = T_2 - t_1$

$\Delta T_1 = T_1 - t_2$

By simple substitution into equation 8

$\Delta T_m = LMTD = 32.1°C$

Figure 3 shows the temperature profile for counter flow of jacket water cooler and the sea water inlet and outlet temperatures. It also shows the fresh water inlet and outlet temperatures.

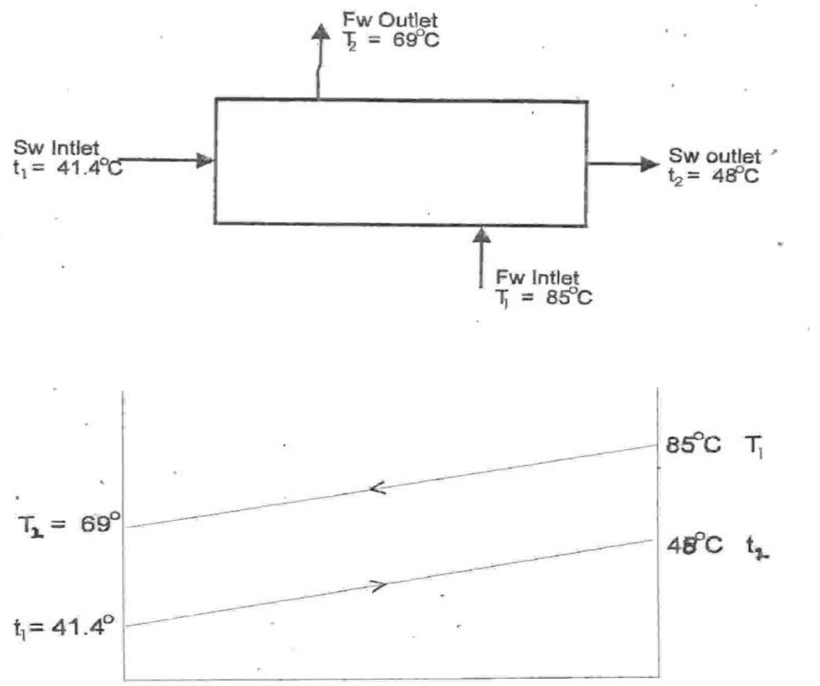

Figure 3: Temperature profile for counter flow of jacket water cooler.

Total heat transfer surface area (A_T) of jacket water cooler is

$$A_T = \frac{Q_T}{K \ x \ F \ x \ \Delta T_m} \quad (9)$$

Where

A_T – Heat transfer surface area

Q_T – Total heat transfer in cooler = 1200.42 KJ/KW hr

K – Mean overall heat transfer coefficient = 4800KJ/KW hr

ΔT_m - Log mean temperature difference = 32^0C

F - Temperature correction factor = 0.95 (0.95 for two-pass cooler and 1.0 for single pass cooler)

By simple substitution into equation 9

$A_T = 29.62 \ m^2$

Piston Water cooler: The heat dissipated to the piston water cooler is 15% of the total heat rejected (2223KJ/KW hr)

∴ H_a = 1200.42 KJ/KW hr

And the rate of flow of piston water cooler is

$$Q = \frac{H_a}{\Delta T \; x \; C_p} \tag{10}$$

Q – Mass flow rate of coolant Kg/hr

H_a = Heat absorbed by coolant = 333.45KJ/KW hr

C_p - Specific heat of coolant = 4.191 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 15.2^{0}C

The rate of flow of piston (fresh) water is obtained by substituting into equation 10

$$Q = \frac{333.45}{15.2 \; x \; 4.192} = 5.2332 \; {}^{m^3}\!/_{hr}$$

The rate of flow of piston (sea) water is obtained by substituting into equation 10

$$Q = \frac{166.725}{6.6 \; x \; 3.925} = 6.436 \; {}^{m^3}\!/_{hr}$$

And related known data for the piston water cooler includes

- Fresh water inlet temperature (T_1) = 70.2^{0}C
- Fresh water outlet temperature (T_2) = 55^{0}C
- Sea water inlet temperature (t_1) = 41.4^{0}C
- Sea water inlet temperature (t_2) = 48^{0}C

To obtain the log mean temperature difference (LMTD =ΔT_m)

$$\Delta T_m = LMTD = \frac{\Delta T_2 - \Delta T_1}{In\frac{\Delta T_2}{\Delta T_1}} \tag{11}$$

Where

$\Delta T_2 = T_2 - t_1$

$\Delta T_1 = T_1 - t_2$

By simple substitution into equation 11

$\Delta T_m = LMTD = 17.5°C$

Figure 4 shows the temperature profile for counter flow of Piston water cooler and the sea water inlet and outlet temperatures. It also shows the fresh water inlet and outlet temperatures

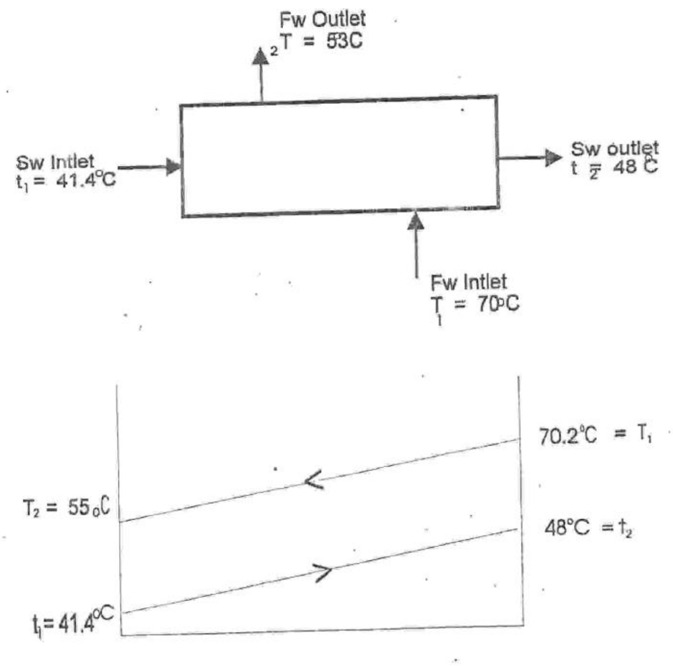

Figure 4 Temperature profile for counter flow of Piston water cooler

Total heat transfer surface area (A_T) of Piston water cooler is

$$A_T = \frac{Q_T}{K \ x \ F \ x \ \Delta T_m} \quad (12)$$

Where

A_T – Heat transfer surface area

Q_T – Total heat dissipated in cooler = 333.45KJ/KW hr

K – Mean overall heat transfer coefficient = 4800KJ/KW hr

ΔT_m - Log mean temperature difference = 17.5^{0}C

F - Temperature correction factor = 0.95 (0.95 for two-pass cooler and 1.0 for single pass cooler)

By simple substitution into equation 12

A_T = 15.1 m^2

Crankcase lubricating oil cooler

The heat dissipated to the Crankcase lubricating oil cooler is 3% of the total heat rejected (2223KJ/KW hr)

$\therefore \ H_l$ = 66.69 KJ/KW hr

And the rate of flow of Crankcase lubricating oil cooler is

$$Q = \frac{H_l}{\Delta T \ x \ C_p} \quad (13)$$

Q – Mass flow rate of coolant Kg/hr

H_a = Heat absorbed by coolant = 333.45KJ/KW hr

C_p - Specific heat of coolant = 1.960 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 5.8^{0}C

The rate of flow of Crankcase lubricating oil is obtained by substituting into equation 13

$$Q = \frac{66.69}{5.8 \ x \ 1.960} = 5.87 \ {}^{m^3}/_{hr}$$

And related known data for the Crankcase lubricating oil cooler includes

-Lube-oil inlet temperature (T_1) = 50.6^{0}C

-Lube-oil outlet temperature (T_2) = 44.8^{0}C

-Sea water inlet temperature (t_1) = 32.2^{0}C

-Sea water inlet temperature (t_2) = 34.5^{0}C

To obtain the log mean temperature difference (LMTD =ΔT_m)

$$\Delta T_m = LMTD = \frac{\Delta T_2 - \Delta T_1}{In\frac{\Delta T_2}{\Delta T_1}} \quad (14)$$

Where

$$\Delta T_2 = T_2 - t_1$$

$$\Delta T_1 = T_1 - t_2$$

By simple substitution into equation 14

$$\Delta T_m = LMTD = 15.52°C$$

Figure 5 shows the temperature profile for counter flow of crankcase lube-oil cooler and the sea water inlet and outlet temperatures. It also shows the lubricating oil inlet and outlet temperatures

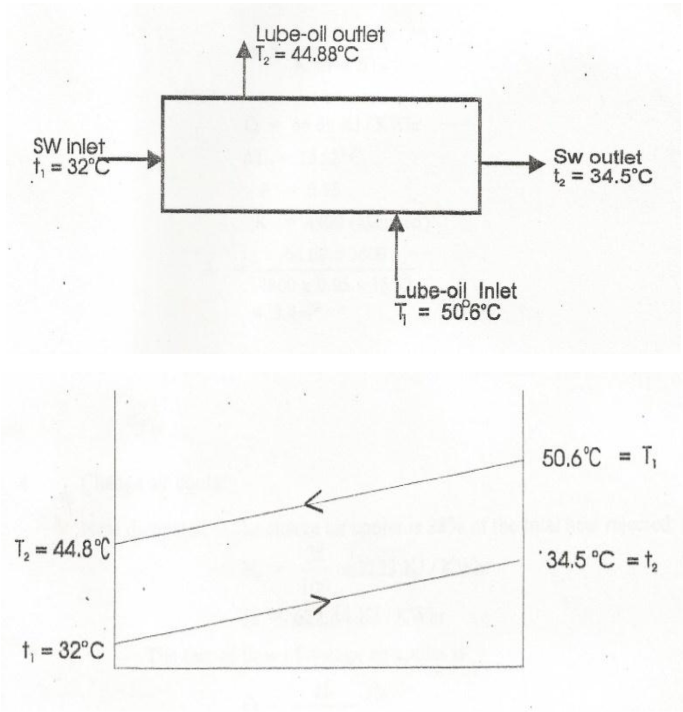

Figure 5: Temperature profile for counter flow of crankcase lube-oil cooler.

Total heat transfer surface area (A_T) of crankcase lube-oil water cooler is determined by

$$A_T = \frac{Q_T}{K \; x \; F \; x \; \Delta T_m} \qquad (15)$$

Where

A_T – Heat transfer surface area

Q_T – Total heat dissipated in cooler = 66.69 KJ/KW hr

K – Mean overall heat transfer coefficient = 4800KJ/KW hr

ΔT_m - Log mean temperature difference = 15.52^{0}C

F - Temperature correction factor = 0.95 (0.95 for two-pass cooler and 1.0 for single pass cooler)

By simple substitution into equation 15

A_T = 3.4 m^2

Charger air cooler

The heat dissipated to the Charger air cooler is 28% of the total heat rejected (2223KJ/KW hr)

$\therefore$ H_c = 622.44 KJ/KW hr

And the rate of flow of Charger air cooler is

$$Q = \frac{H_c}{\Delta T \ x \ C_p} \tag{16}$$

Q – Mass flow rate of coolant Kg/hr

H_a = Heat absorbed by coolant = 622.44KJ/KW hr

C_p - Specific heat of coolant = 1.107 KJ/Kg^{0}C

ΔT - Temperature rise of sea water = 111.86^{0}C

The rate of flow of Charger air is obtained by substituting into equation 16

$$Q = \frac{622.44}{111.86 \ x \ 1.107} = 5.03 \ {m^3}/{hr}$$

And related known data for the Charger air cooler includes

- Exhaust gas temp at turbine inlet (T_1) = 298^{0}C
- Exhaust gas temp at turbine outlet (T_2) = 186.14^{0}C
- Sea water inlet temperature (t_1) = 33.48^{0}C
- Sea water inlet temperature (t_2) = 41.8^{0}C

To obtain the log mean temperature difference (LMTD =ΔT_m)

$$\Delta T_m = LMTD = \frac{\Delta T_2 - \Delta T_1}{In\frac{\Delta T_2}{\Delta T_1}} \tag{17}$$

Where

$\Delta T_2 = T_2 - t_1$

$\Delta T_1 = T_1 - t_2$

By simple substitution into equation 17

$\Delta T_m = LMTD = 200°C$

Figure 6 shows the temperature profile for counter flow of Charger air cooler and the sea water inlet and outlet temperatures. It also shows the Exhaust gas inlet and outlet temperatures

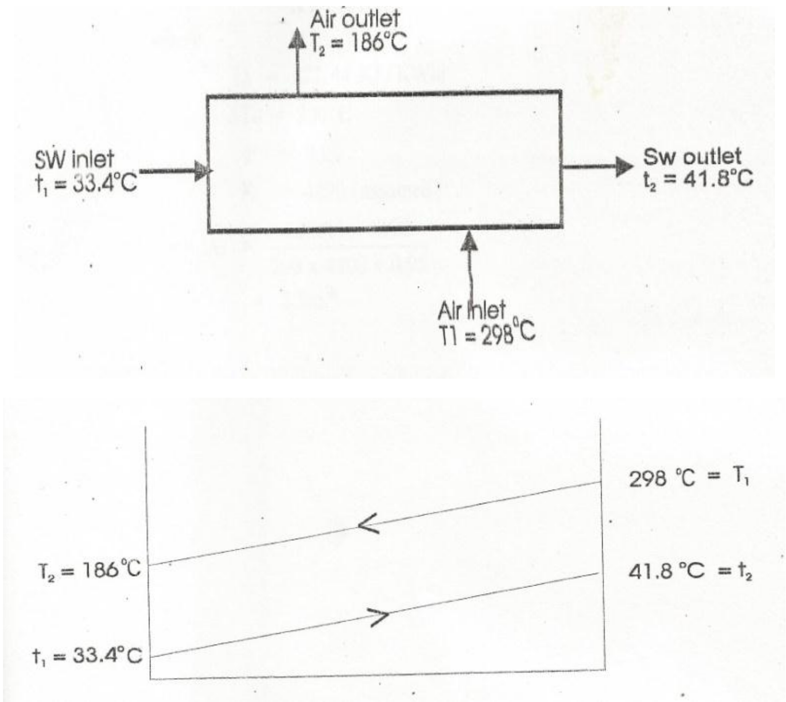

Figure 6 Temperature profile for counter flow of charger air cooler

Total heat transfer surface area (A_T) of Charger air cooler is

$$A_T = \frac{Q_T}{K \; x \; F \; x \; \Delta T_m} \qquad (18)$$

Where

A_T – Heat transfer surface area

Q_T – Total heat dissipated in cooler = 622.4 KJ/KW hr

$K -$ Mean overall heat transfer coefficient = 4800KJ/KW hr

ΔT_m - Log mean temperature difference = 200^0C

F - Temperature correction factor = 0.95 (0.95 for two-pass cooler and 1.0 for single pass cooler)

By simple substitution into equation 18

$A_T = 2.5 \text{ m}^2$

3. RESULTS DISCUSSIONS

The design of the cooling water system for a sea going tug boat has shown that at a transmitting power of 955KW the total energy loss due to friction in the shaft line bearing is about 18KW while the volumetric flow rate of the sea water for cooling is estimated at $4m^3$/hr. It was also discovered that if the power increase there will be also an increase in the energy loss and the volumetric flow rate of the sea water. The volumetric flow rate of the sea water for cooling is estimated at $7m^3$/hr while the total energy loss in transmission in the reduction gear is 43KW. The Jacket water flow rate for the fresh water is obtained as $17.851m^3/hr$ and the area of heat transfer surface is estimated at 30 m^2. The results for piston water cooler and the charger air cooler are $5.23m^3/hr$; 15.1 m^2 and $5.03m^3/hr$; 2.5 m^2 respectively. It is also observed that in the exchanger, the volumetric flow rate of the sea water is always higher than that of the fresh water; this is to ensure that the rate of taking away heat should be higher than the rate at which the heat is formed. This will enables the engine metal to retain its mechanical properties and keep the temperature of the main engine within the acceptable limit for maximum performance during operations. All designs were done in accordance to the Lloyd's specification rules and regulations for a sea going tug boat.

4. CONCLUSION

The cooling water system is used to ensure that the temperature of the main engine is kept within the acceptable limit. The system removes or conducts heat out of hot surfaces or materials either directly or indirectly. This system comprises of different components, sub-systems, piping and fittings, valves, sea water and fresh water pumps, filters, tanks and condition monitoring device amongst other. The design of the cooling water system for a sea going tug boat has shown that at a transmitting power of 955KW, the total energy loss due to friction in the shaft line bearing about 18KW while the volumetric flow rate of the sea water for cooling is estimated at $4m^3/hr$. It was also discovered that if the power increase there will be also an increase in the energy loss and the volumetric flow rate of the sea water. The volumetric flow rate of the sea water for cooling is estimated at $7m^3/hr$ while the total energy loss in transmission in the reduction gear is 43KW. The Jacket water flow rate for the fresh water is obtained as $17.851m^3/hr$ and the area of heat transfer surface is estimated at 30 m^2. The results for piston water cooler and the charger air cooler are $5.23m^3/hr$; 15.1 m^2 and $5.03m^3/hr$; 2.5 m^2 respectively. All designs were done in accordance to the Lloyd's specification rules and regulations for a sea going tug boat.

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