BANGLADESH UNIVERSITY OF ENGINEERING AND TECHNOLOGY

"Analyzing the Efficiency of Shell and Tube Heat Exchanger and Desalination of Salt Water for Moving Vessels"

Authors:

Mohammad Khalikujjaman (1312024)

Shajratul Alam Towhid (1312032)

Md. Alauddin Khan (1312048)

Supervisor:

Dr. M. Rafiqul Islam

Professor

Department of Naval Architecture & Marine Engineering

Bangladesh University of Engineering and Technology

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Chapter 01 Introduction

1.1 Desalination

Desalination is a process that takes away mineral components from saline water. More generally, desalination refers to the removal of salts and minerals from a target substance.

1.2 Methods of Desalination

There are several methods, like

- Vacuum distillation
- Multi-stage flash distillation
- Multiple-effect distillation
- Vapor-compression distillation
- Reverse osmosis
- Freeze-thaw
- Solar evaporation
- Electro dialysis reversal
- Membrane distillation
- Wave-powered desalination

1.3 Applicability of Desalination

Desalination processes are adapted to produce drinking water in areas where only seawater or brackish water is available. A number of technologies have been developed for desalination and can be used at different scales from small communities' water supply (e.g. solar distillation) to huge plants (e.g. reverse osmosis) for cities.

Desalination costs, although continuously decreasing, are still higher than conventional drinking water supply processes. Consequently, desalination should be applied if it is the only option.

Pure water produced by desalination processes is corrosive and not appropriate for human consumption. Thus, remineralization (e.g. limestone dissolution) is required before distribution and consumption.

1.4 Experimental techniques

- Waste heat
- Low-temperature thermal
- Thermionic process
- Evaporation and condensation for crops

1.5 Desalination Using Heat Exchanger

In case of the experimental technique of waste heat from engine exhaust is passed through the heat exchanger via a connecting pipe, simultaneously normal sea water is also passed through the smaller tubes of the heat exchanger. Then by taking the high heat from the exhaust gas normal sea water will be converted into steam. After that these steam is passed through a condenser via a condenser pipe where the steam will be condensed as distilled water. Thus desalination is performed using heat exchanger.

Chapter 02 Research Background/ Literature Review

2.1 Ocean thermal energy conversion (OTEC)

Ever since the Industrial Revolution in the 18th century, increasing consumption of fossil fuels has powered the economy and delivered unprecedented affluence to human beings. For many decades, technologies to effectively harness the energy of reliable fossil fuels was well developed but the inexpensiveness of fossil fuels is blamed for being the primary contributor to global warming as they contain high carbon concentration. Fossil fuels are not renewable because of their very long period of renewal, i.e., of an order of millions of years for anaerobic decomposition of organic matter. Due to the rapid depletion of fossil fuels, alternative, renewable energy sources are of vital importance to humans as are water and food.

Thermally-renewable natural energies include solar, geo- and ocean thermal resources. Photovoltaic solar panels are popular due to easy installation and maintenance. Sunlight (as a heat source) is available only during the day, and long-term weather variations hinder the overall performance of solar energy equipment. Due to a couple of performance variables of solar panels, solar energy is competitive. Geothermal power is reliable, sustainable, and environmentally friendly, but has been limited by tectonic plate boundaries. Emission of greenhouse gases from geothermal wells is much lower than that of fossil fuels. Amounts of water injected to geothermal wells are, however, often not fully recovered, which increases the cost of geothermal energy. Solar and geothermal energies are topologically available on land, which occupies approximately 29.2% of the total earth surface, while the rest of the earth surface is covered by water. Renewable energies from ocean include mechanical energy using ocean wave and thermal energy using temperature difference. As the specific heat of water (4.18 kJ/kg ^{o}C) is about three times higher than that of the soil (1.48 kJ/kg ^{o}C), the heat capacity of the ocean is almost one order higher than that of land. Most solar energy is absorbed by a thin layer of the ocean surface and there is barely any sunlight that penetrates below 200 m. Deep seawater temperature is maintained at 4e8 ^{o}C at 600e1000 m depth. When warm surface and cold deep seawater have at least a 20 ^{o}C temperature difference, electric power and/or desalinated water can be produced using the thermal energy gradient. This mechanism, called ocean thermal energy

Conversion (OTEC), has three types: open, closed, and hybrid. By convention, OTEC primarily implies power generation using the ocean temperature difference. Open cycle (OC) OTEC controls vacuum pressure in the flash evaporator to vaporize warm surface seawater, rotates a turbine using the steam that is produced, and condenses the vapor at the same rate as the flash evaporation. A byproduct of OC-OTEC is desalinated water, which is about 0.5e0.6% by volume of the input warm surface seawater. Closed cycle (CC) OTEC uses surface and deep seawater as the heat source and sink, respectively, for continuous cyclic phase transformation of organic working fluids. Hybrid cycle (HC) OTEC attempts to use advantages of CC- and OC-OTEC in an optimized way, i.e., high power generation rate followed by fresh water production. Because the desalination of HC-OTEC uses the leftover thermal energy after power generation of CC-OTEC, the desalination rate is not as high as that of OC-OTEC. Due to low efficiencies of OC- and HC-OTEC, research on OTEC was much more focused on CC-OTEC.

OC-OTEC was invented by French physicist Dr. d'Arsonval in 1881. He proposed the fundamental idea of using the temperature difference between surface and deep ocean water in 1881, but in the same year, the first thermal power plant started in the United States, receiving more public attention. Claudes and Boucherat used Dr. d'Arsonval's idea and did a public experiment at the French Academy of Sciences in Nov. 15, 1926. They used two small flasks that were connected: one filled with ice cubes and the other with 25 L of 28 ^{O}C warm water. Initial exhausting of air decreased the gas pressure in the flasks, and drove the warm water to evaporate and migrate to another flask of ice cubes. This steam flow rotated a turbine of 15 cm diameter at a speed of 5000 rpm, and turned on lights for three electric lamps. This batch experiment ended 8e10 min after the initial exhaustion because the warm water cooled down to 20 ^{O}C after the ice cubes started melting. In the same year, Claude's first experimental sites were selected at Matanzas Bay, Cuba. The first installation of 1.6 m in diameter pipes that were 2 km long failed due to pipe integrity problems in 1929 and the second failure happened in 1930. In the same year of the second failure, Claudes did generated 22 kW of electric power for 10 days using a 14 ^{O}C temperature difference. This is regarded as the first successful demonstration of the OC-OTEC process. In 1933, Claudes converted a 10,000 ton cargo ship, called Tunisie, and sailed to Brazil. Due to rough weather conditions, the cold water pipe disconnected from the deploying floater, and it sank before being re-connected to the ship. In 1940, the Abidjan project was proposed by Claudes. Abidjan is now the capital city of the French West African nation of Cote d'Ivoire (Ivory Coast), where the surface seawater has temperature of 30 ^{O}C. This effort initiated support from the French government, which set up the "Energie des Mers" (Energy from the Sea) project. Unfortunately, in the 1950s large, inexpensive amounts of oil became available, and the Abidjan project was abandoned in 1955. Several key failures of OC-OTEC are primarily ascribed to huge and heavy equipment, which require substantial amounts of initial investment. After Claudes passed away in 1960, Anderson revived Claudes' OC-OTEC in the United States, and he applied for a US patent on a "Sea Water Power Plant" in 1964, revising Claudes' original designs. He proposed to use a working fluid (such as ammonia) of low boiling point in a confined loop and use the surface and deep ocean water as heat source and sink, respectively. Later, this scheme was called a closed cycle OTEC. Japanese researchers were much more interested in Anderson's idea and conducted extensive research on CC-OTEC, requiring in general a smaller-size turbine than OC-OTEC.

In Hawaii, two major long-term experiments were conducted, which are historically important. In 1979, an offshore CC-OTEC barge plant, called Mini-OTEC, was built and successfully operated from August to November. An ammonia turbine was controlled by adjusting the turbine inlet nozzles. In the early 1990s, Natural Energy Laboratory of Hawaii Authority (NELHA) developed a real OC-OTEC plant of 210 kW that continuously operated for six years (1993e1998). This plant is based on a conceptual design proposed in 1990, where 213 kW turbine rotated to generate 84 kW of net power and 0.4 kg/s of desalinated water, using surface and deep seawater of 26 ^{O}C and 6.1 ^{O}C, respectively. Recently, Energy Island Ltd. designed a 1.8 MW OC- OTEC plant, of which a flash evaporator has dimensions of 9.2 m in diameter and 16 m in height. Current research activities of NELHA include efficiency enhancement of condensing heat exchanger as one of the key components of a 5 MW OC-OTEC plant. In addition, a new conceptual design of OC-OTEC, based on the 1990 version, was recently proposed to generate 80 MW (i.e., 414,400 MWh annually) of power and 118,400 m^3/day of fresh water by using five modules of 16 MW OC-OTEC. Economic analysis compared OC-OTEC and CC-OTEC for 50 MW of power generation.

Fundamentals of OC-OTEC in terms of detailed transport mechanisms are lacking in the current OTEC literature. Although OC-OTEC can produce electric power and desalinated water at the same time, their relative production rate is rigidly fixed because the turbine and condenser are connected in series. If a direct condenser is used (as proposed by the original conceptual design), the condensed water is simply mixed with cold deep seawater (taken in) and then discharged onto the ocean surface. If a surface condenser (i.e., heat exchanger) is used, the condensed fresh water is collected into a reservoir as one of the final products. In both cases, a condenser generates fresh water from the steam gas generated by the flash evaporator and then used by the turbine. Therefore, the conventional OC-OTEC design is restricted by the fixed ratios of power generation to desalination and desalination to discharge. We propose a novel design idea to flexibly adjust power and water production rates using multiple condensers. Specifically, this paper aims (1) to design an optimized OC-OTEC plant using multiple condensers at an unprecedented level of theoretical de- tails, (2) to adjust the relative production rate of electric power and desalinated water for required operational guidelines, and (3) to estimate the future potential of using vacuum membrane distillation as an alternative and/or supplementary unit to the flash evaporator.

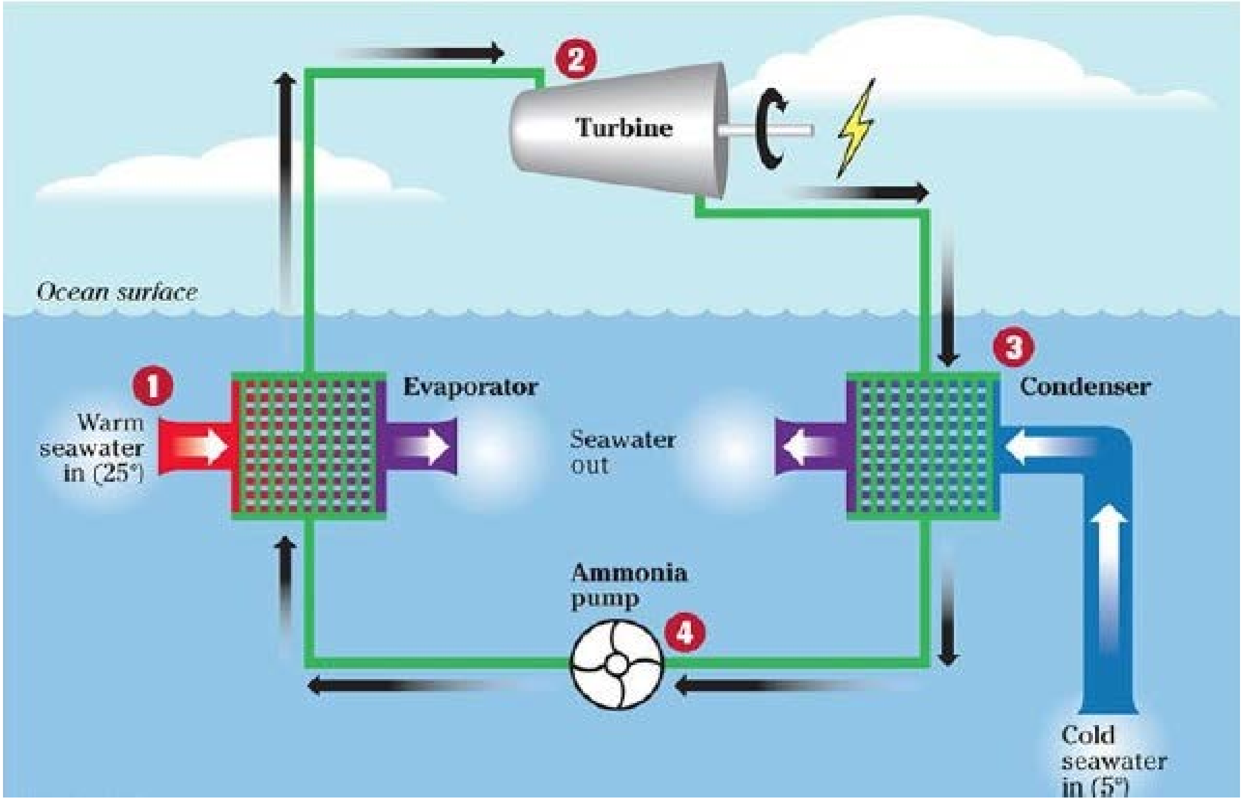


Figure 2.1: Open Cycle OTEC

We first found our idea from OTEC. OTEC is mainly a stationary power plant. But we wanted to develop a similar type of set up in case of a moving vessel. But building a similar kinds of setup on moving vessels is very much difficult. Because arranging a turbine and other equipment's are so much costly. So we tried to develop a system to only desalinate sea water.

2.2 Our topic of Interest

In previous researches in literature the researchers have used flash evaporator to produce steam from sea water. For doing so they had to make vacuum in the flash evaporator first. Water is converted to steam at a pressure of 0.03 bar at atmospheric temperature.

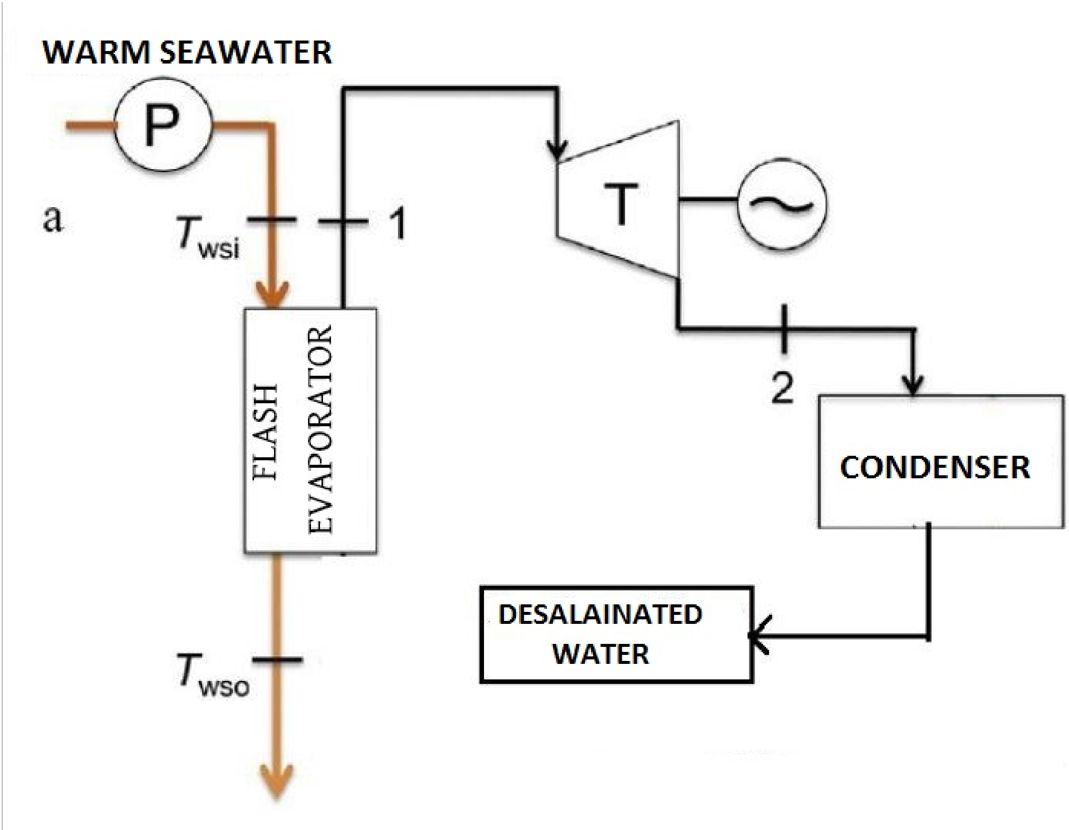


Figure 2.2: Modification of OTEC Cycle

But creating vacuum is so much difficult and costly. So we thought the alternative of flash evaporator. We decided to work with a heat exchanger instead of flash evaporator. In the heat exchanger we thought of using the ship engine exhaust gas as the hot fluid and surface sea water as the cold fluid. Another change in our experiment from previous researches is in the condensation part. In OTEC plants almost one kilometer pipe is used to bring cold water from deep sea. It costs about 1 billion US dollar to produce 100 MW of electricity. So here is our main topic of interest. We thought of a setup in which the use of such long pipes can be avoided. In the heat exchanger the sea water will turn into steam. Then the steam is taken through the surface sea water by a pipe. The ship moves at a certain speed. That ultimately means water with a certain velocity flows over the pipe in which steam is flowing. So sufficient amount of heat transfer will happen there and steam will transform into desalinated water. So here in our experiment we are mainly eager to do three things. They are:

1. Designing of a heat exchanger.
2. Using the ship engine exhaust gas in the heat exchanger as hot fluid.
3. Using auto condensation system without using any power source or using long pipes in deep sea like OTEC.

Chapter 03 Heat Exchanger

3.1 Selection of Heat Exchanger

Transfer of heat from one fluid to another is an important operation for most of the chemical industries. The most common application of heat transfer is in designing of heat transfer equipment for exchanging heat from one fluid to another fluid. Such devices for efficient transfer of heat are generally called Heat Exchanger.

Before setup we had to first select a convenient type of heat exchanger which is much more suitable and efficient than the other types of heat exchangers. There are different types of heat exchangers. Like,

1. Shell and Tube Heat Exchanger.
2. Plate Heat Exchanger.
3. Regenerative Heat Exchanger.
4. Adiabatic Wheel Heat Exchanger.
5. Double pipe heat exchanger.

So we had to select a heat exchanger which has a great heat transfer capacity. Shell and tube heat exchanger is the best option here for selection.

Again shell and tube heat exchangers are of different types. We had to select a type which can be cleaned mechanically because when sea water will evaporate in the heat exchanger it will leave a high amount of salt in the exchanger. So without the option of cleaning it the heat exchanger can be blasted while in operation. The cleaning procedure is described in the later chapter. Here is the advantages and disadvantages of selecting different types of shell and tube heat exchanger:

Shell and Tube Exchangers	*Typical TEMA code*	*Advantages*	*Limitations*
Fixed tube sheet	BEM, AEM, NEN	Provides maximum heat transfer area for a given shell and tube diameter. Provides for single and multiple tube passes to assure proper velocity. Less costly than removable bundle designs.	Shell side / outside of the tubes are inaccessible for mechanical cleaning. No provision to allow for differential thermal expansion developed between the tube and the shell side. This can be taken care by providing expansion joint on the shell side.
Floating-head	AEW, BEW, BEP, AEP, AES, BES	Floating tube sheet allows for differential thermal expansion between the shell and the tube bundle. Both the tube bundle and the shell side can be inspected and cleaned mechanically.	To provide the floating-head cover it is necessary to bolt it to the tube sheet. The bolt circle requires the use of space where it would be possible to place a large number of tubes. Tubes cannot expand independently so that huge thermal shock applications should be avoided. Packing materials produce limits on design pressure and temperature.
U-tube	BEU, AEU	U-tube design allows for differential thermal expansion between the shell and the tube bundle as well as for individual tubes. Both the tube bundle and the shell side can be inspected and cleaned mechanically. Less costly than floating head or packed floating head designs.	Because of U-bend some tubes are omitted at the Centre of the tube bundle. Because of U-bend, tubes can be cleaned only by chemical methods. Due to U-tube nesting, individual tube is difficult to replace. No single tube pass or true countercurrent flow is possible. Tube wall thickness at the U-bend is thinner than at straight portion of the tubes. Draining of tube circuit is difficult when positioned with the vertical position with the Head side upward.

Table 3.1: advantages and disadvantages of selecting different types of shell and tube heat exchanger

From above description it is seen that Floating Head type is best suitable for cleaning purposes. So we have selected this type.

3.2 Thermal Design Consideration

Thermal design of a shell and tube heat exchanger typically includes the determination of heat transfer area, number of tubes, tube length and diameter, tube layout, number of shell and tube passes, type of heat exchanger (fixed tube sheet, removable tube bundle etc.), tube pitch, number of baffles, its type and size, shell and tube side pressure drop etc.

❖ Shell

Shell is the container for the shell fluid and the tube bundle is placed inside the shell. Shell diameter should be selected in such a way to give a close fit of the tube bundle. The clearance between the tube bundle and inner shell wall depends on the type of exchanger. Shells are usually fabricated from standard steel pipe with satisfactory corrosion allowance. The shell thickness of 3/8 inch for the shell ID of 12-24 inch can be satisfactorily used up to 300 psi of operating pressure.

❖ Tube

Tube OD of ¾ and 1‘‘‘‘ are very common to design a compact heat exchanger. The most efficient condition for heat transfer is to have the maximum number of tubes in the shell to increase turbulence. The tube thickness should be enough to withstand the internal pressure along with the adequate corrosion allowance. The tube thickness is expressed in terms of BWG (Birmingham Wire Gauge) and true outside diameter (OD). The tube length of 6, 8, 12, 16, 20 and 24 ft are preferably used. Longer tube reduces shell diameter at the expense of higher shell pressure drop. Finned tubes are also used when fluid with low heat transfer coefficient flows in the shell side. Stainless steel, admiralty brass, copper, bronze and alloys of copper-nickel are the commonly used tube materials:

❖ Tube pitch, tube-layout and tube-count

Tube pitch is the shortest centre to centre distance between the adjacent tubes. The tubes are generally placed in square or triangular patterns (pitch) as shown in the **Figure 3.1**. The widely used tube layouts are illustrated in **Table 3.2.**The number of tubes that can be accommodated in a given shell ID is called tube count. The tube count depends on the factors like shell ID, OD of tube, tube pitch, tube layout, number of tube passes, type of heat exchanger and design pressure.

❖ Tube passes

The number of passes is chosen to get the required tube side fluid velocity to obtain greater heat transfer co-efficient and also to reduce scale formation. The tube passes vary from 1 to 16. The tube passes of 1, 2 and 4 are common in application. The partition built into exchanger head known as partition plate (also called pass partition) is used to direct the tube side flow.

Tube OD, in	Pitch type	Tube pitch, in
$\frac{3}{4}$	Square	1
1		$1\frac{1}{4}$
$\frac{3}{4}$	Triangular	$\frac{15}{16}$
$\frac{3}{4}$		1

Table 3.2: Common Tube Layouts.

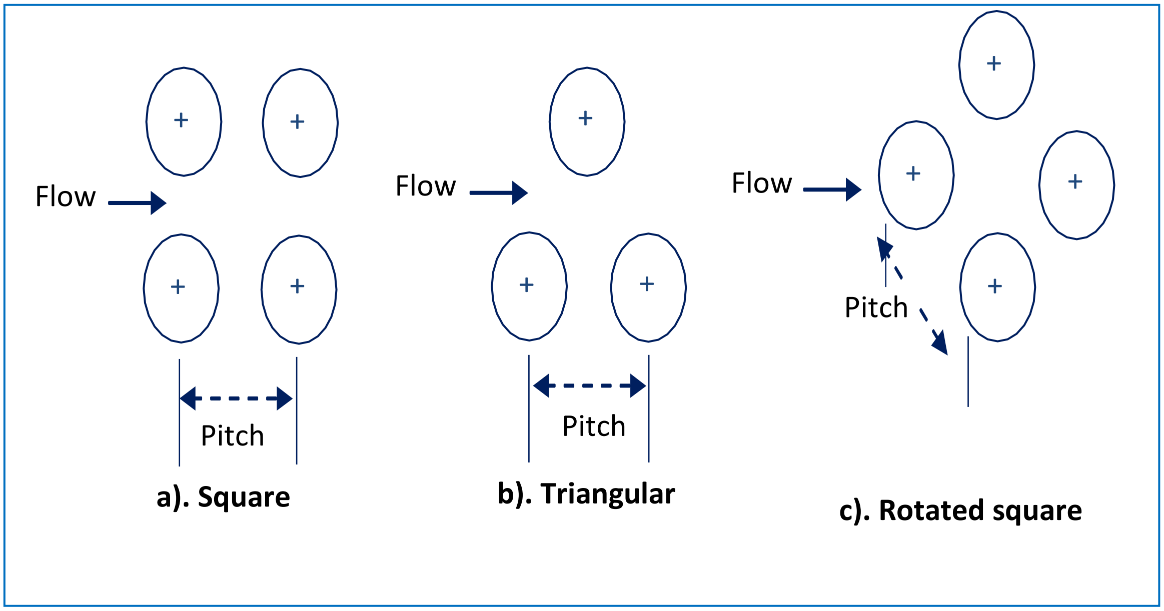


Figure 3.1: Heat Exchanger Tube-layouts

❖ Tube sheet

The tubes are fixed with tube sheet that form the barrier between the tube and shell fluids. The tubes can be fixed with the tube sheet using ferrule and a soft metal packing ring. The tubes are attached to tube sheet with two or more grooves in the tube sheet wall by
„tube rolling‟. The tube metal is forced to move into the grooves forming an excellent tight seal. This is the most common type of fixing arrangement in large industrial exchangers. The tube sheet thickness should be greater than the tube outside diameter to make a good seal. The recommended standards (IS:4503 or TEMA) should be followed to select the minimum tube sheet thickness.

❖ Baffles

Baffles are used to increase the fluid velocity by diverting the flow across the tube bundle to obtain higher transfer co-efficient. The distance between adjacent baffles is called baffle-spacing. The baffle spacing of 0.2 to 1 times of the inside shell diameter is commonly used. Baffles are held in positioned by means of baffle spacers. Closer baffle spacing gives greater transfer co-efficient by inducing higher turbulence. The pressure drop is more with closer baffle spacing. The various types of baffles are shown in **Figure 3.2**. In case of cut-segmental baffle, a segment (called baffle cut) is removed to form the baffle expressed as a percentage of the baffle diameter. Baffle cuts from 15 to 45% are normally used. A baffle cut of 20 to 25% provide a good heat-transfer with the reasonable pressure drop. The % cut for segmental baffle refers to the cut away height from its diameter. **Figure 3.2** also shows two other types of baffles

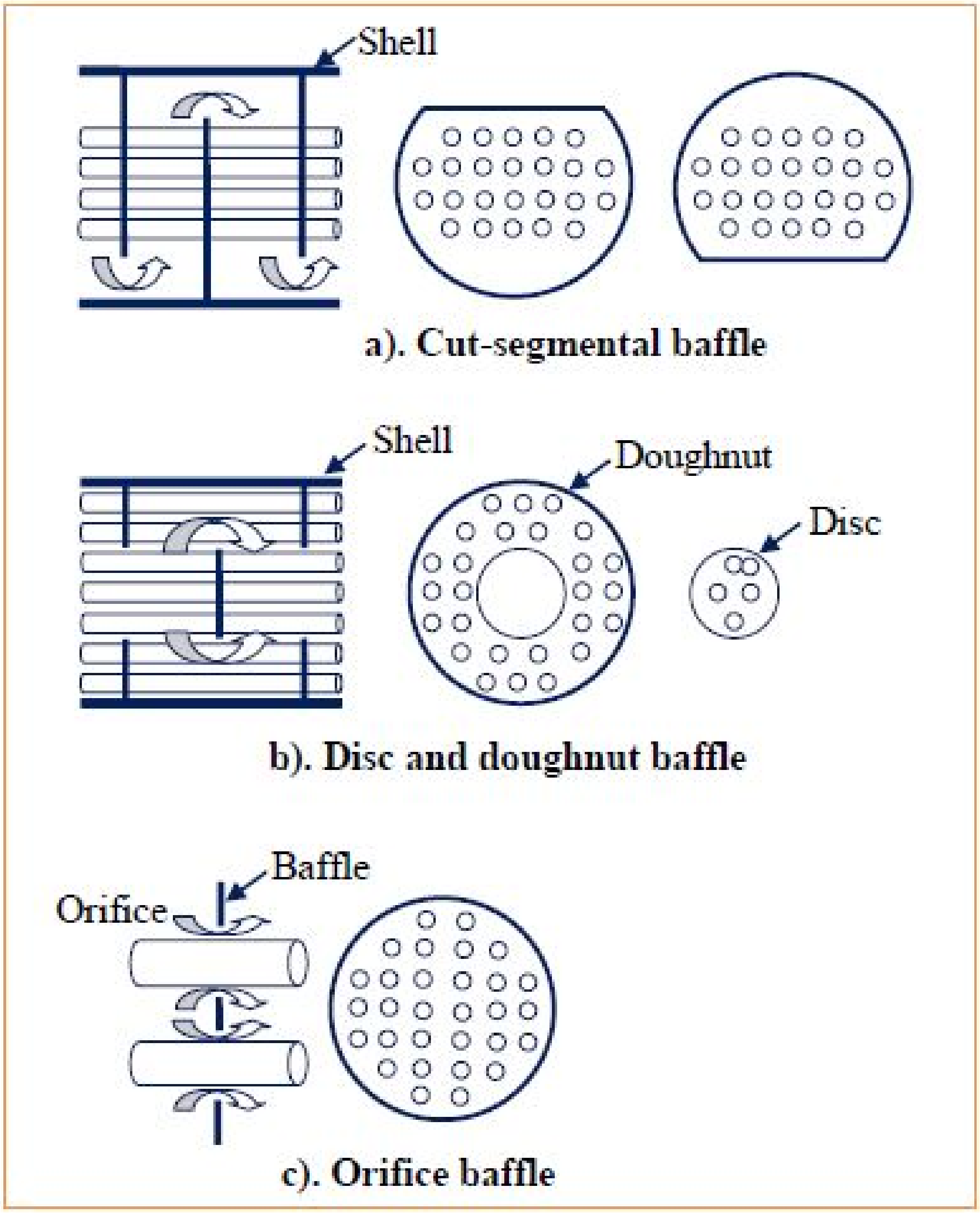


Figure 3.2: Different type of heat exchanger baffles: a). Cut-segmental baffle, b). Disc and doughnut baffle, c). Orifice baffle

❖ Fouling Considerations

The most of the process fluids in the exchanger foul the heat transfer surface. The material deposited reduces the effective heat transfer rate due to relatively low thermal conductivity. Therefore, net heat transfer with clean surface should be higher to compensate the reduction in performance during operation. Fouling of exchanger increases the cost of

(i) construction due to oversizing,

(ii) additional energy due to poor exchanger performance and

(iii) Cleaning to remove deposited materials.

A spare exchanger may be considered in design for uninterrupted services to allow cleaning of exchanger. The effect of fouling is considered in heat exchanger design by including the tube side and shell side fouling resistances. Typical values for the fouling coefficients and resistances are summarized in **Table 3.3**. The fouling resistance (fouling factor) for petroleum fractions are available in the text book **([3]; page 845)**

Fluid	Coefficient ($W.m^{-2}.°C^{-1}$)	Resistance ($m^2.°C.W^{-1}$)
River water	3000-12,000	0.0003-0.0001
Sea water	1000-3000	0.001-0.0003
Cooling water (towers)	3000-6000	0.0003-0.00017
Towns water (soft)	3000-5000	0.0003-0.0002
Towns water (hard)	1000-2000	0.001-0.0005
Steam condensate	1500-5000	0.00067-0.0002
Steam (oil free)	4000- 10,000	0.0025-0.0001
Steam (oil traces)	2000-5000	0.0005-0.0002
Refrigerated brine	3000-5000	0.0003-0.0002
Air and industrial gases	5000-10,000	0.0002-0.000-1
Flue gases	2000-5000	0.0005-0.0002
Organic vapors	5000	0.0002
Organic liquids	5000	0.0002
Light hydrocarbons	5000	0.0002
Heavy hydrocarbons	2000	0.0005
Boiling organics	2500	0.0004
Condensing organics	5000	0.0002
Heat transfer fluids	5000	0.0002
Aqueous salt solutions	3000-5000	0.0003-0.0002

Table 3.3: Typical values of fouling coefficients and resistances

3.3 Design of Heat Exchanger

3.3.1 Calculating the flow rates

1. $(m_{water} \times c_{p,water}) / (m_{gas} \times c_{p,gas}) = 0.5$(Assume)
2. $Q = m_{water} \times c_{p,water} \times 70° + m_{water} \times l_f + m_{water} \times c_{p,steam} \times$ temperature difference.

From this calculated Q,

3. $Q = m_{gas} + c_{p,gas} +$ temperature difference of gas

There are several methods for the design of a heat exchanger, like Log Mean Temperature Difference (LMTD) method, Number of Transfer Unit (NTU) method, the $\Psi - P$ method, The $P_1 - P_2$ method etcetera.

For the application of Log Mean Temperature Difference (LMTD) method both the inlet and outlet temperatures needed to be known along with heat transfer rate. Then for the application of Number of Transfer Unit (NTU) method it is enough only to be known about the inlet temperature, no outlet temperature is needed initially for calculation. And this NTU method is principally based on dimensionless parameter called heat transfer effectiveness.

Now, in our case we only know about the inlet temperature initially, so we have chosen Number of Transfer Unit (NTU) method in order to design an efficient heat exchanger.

3.3.2 Number of Transfer Unit (NTU) Method

The **Number of Transfer Units (NTU) Method** is used to calculate the rate of heat transfer in heat exchangers (especially counter current exchangers) when there is insufficient information to calculate the Log-Mean Temperature Difference (LMTD). In heat exchanger analysis, if the fluid inlet and outlet temperatures are specified or can be determined by simple energy balance, the LMTD method can be used; but when these temperatures are not available **The NTU** or **The Effectiveness** method is used.

To define the effectiveness of a heat exchanger we need to find the maximum possible heat transfer that can be hypothetically achieved in a counter-flow heat exchanger of infinite length.

Therefore *one* fluid will experience the maximum possible temperature difference, which is the difference of $T_{h,i} - T_{c,i}$ (The temperature difference between the inlet temperature of the hot stream and the inlet temperature of the cold stream). The method proceeds by calculating the heat capacity rates (i.e. mass flow rate multiplied by specific heat) C_h and C_c for the hot and cold fluids respectively, and denoting the smaller one as C_{min}..

A quantity:

$$q_{max} = C_{min} (T_{h,i} - T_{c,i})$$

is then found,where q_{max} is the maximum heat that could be transferred between the fluids per unit time. C_{min} must be used as it is the fluid with the lowest heat capacity rate that would, in this hypothetical infinite length exchanger, actually undergo the maximum possible temperature change. The other fluid would change temperature more quickly along the heat exchanger length. The method, at this point, is concerned only with the fluid undergoing the maximum temperature change.

The *effectiveness (€)*, is the ratio between the actual heat transfer rate and the maximum possible heat transfer rate:

$$\epsilon = \frac{q}{q_{max}}$$

Where,

$$q = C_h (T_{h,i} - T_{h,o}) = Cc (T_{c,o} - T_{c,i})$$

Effectiveness is dimensionless quantity between 0 and 1. If we know € for a particular heat exchanger, and we know the inlet conditions of the two flow streams we can calculate the amount of heat being transferred between the fluids by:

$$q = € Cmin (T_{h,i} - T_{c,i}).$$

For any heat exchanger it can be shown that: $\epsilon = f(NTU, \frac{C_{min}}{C_{max}})$

For a given geometry, € can be calculated using correlations in terms of the "heat capacity ratio"

$$C_r = \frac{C_{min}}{C_{max}}$$

and the *number of transfer units*,NTU is , NTU = (UA / C_{min})

Where, U is the overall heat transfer coefficient and A is the heat transfer area.

For example, the effectiveness of a parallel flow heat exchanger is calculated with:

$$\epsilon = \frac{1 - \exp[-NTU(1 + C_r)]}{1 + C_r}$$

Or the effectiveness of a counter-current flow heat exchanger is calculated with:

$$\epsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \exp[-NTU(1 - C_r)]}$$

For $C_r = 1$,

$$€ = NTU/(1 + NTU)$$

Similar effectiveness relationships can be derived for concentric tube heat exchangers and shell and tube heat exchangers. These relationships are differentiated from one another depending on the type of the flow (counter-current, concurrent, or cross flow), the number of passes (in shell and tube exchangers) and whether a flow stream is mixed or unmixed.

Note that the $C_r = 0$ is a special case in which phase change condensation or evaporation is occurring in the heat exchanger. Hence in this special case the heat exchanger behavior is independent of the flow arrangement. Therefore the effectiveness is given by:

$$€ = 1 - \exp[-NTU]$$

3.3.3 Calculation of Tube

1. **Tube area:** $At=(Nt\times3.1416\times IDt^2)/Np\times4$
2. **Velocity inside tube:** $V= m/(density\times At)$
3. **Reynolds no for water**: $(V\times IDt)$/kinematic viscosity
4. **Prandlt no for water:** 3.02
5. **Nusselt no for water:** $0.027\times Re^{0.8}\times Pr^{0.33}$
6. **Convection heat transfer coefficient of** water,h_{water}: $(Nu\times k)/IDt$

3.3.4 Calculation of Shell

1. **Shell Dhs**: $(4\ P_T^2/(3.1416\times ODt))-ODt$
2. **Shell As**: $(Ds\times C\times B)/P_T$
3. **Velocity of gas:** $m_{gas}/(density\times As)$
4. **Reynolds no for gas**: $(Nu\times Dhs)$/kinematic viscosity
5. **Prandlt no for gas:** (dynamic viscosity×C)/k
6. **Nusselt no for gas:**$0.36\times Re^{0.55}\times Pr^{0.33}$
7. **Convection heat transfer coefficient of exhaust** gas,h_{gas}: $(Nu\times k)/Ds$
8. **Overall heat transfer coeffcient,U**:$(h_{water}{}^{-1}+h_{gas}{}^{-1}+R_{water}+R_{gas})^{-1}$
9. **Length of exchanger**: $(NTU\times C_{min})/U\times3.1416\times ODt$

❖ Resulting Value

After making all the calculations and iterations the following main parameters have been taken for design,

- ❖ **Shell ID:** 336.6 mm
- ❖ **Tube OD:**15.875 mm
- ❖ **Tube wall thickness:** 1.651 mm
- ❖ **Baffle spacing :** 200 mm
- ❖ **Pitch :** 31.75 mm
- ❖

3.3.5 Use of Software

We have used HTRI **Xchanger Suite V6.00** software in order to generate the output dimensions of the heat exchanger. And the software gives us number of tubes as 52. Here the length of the exchanger was taken as 2 meters. The software output has been shown below.

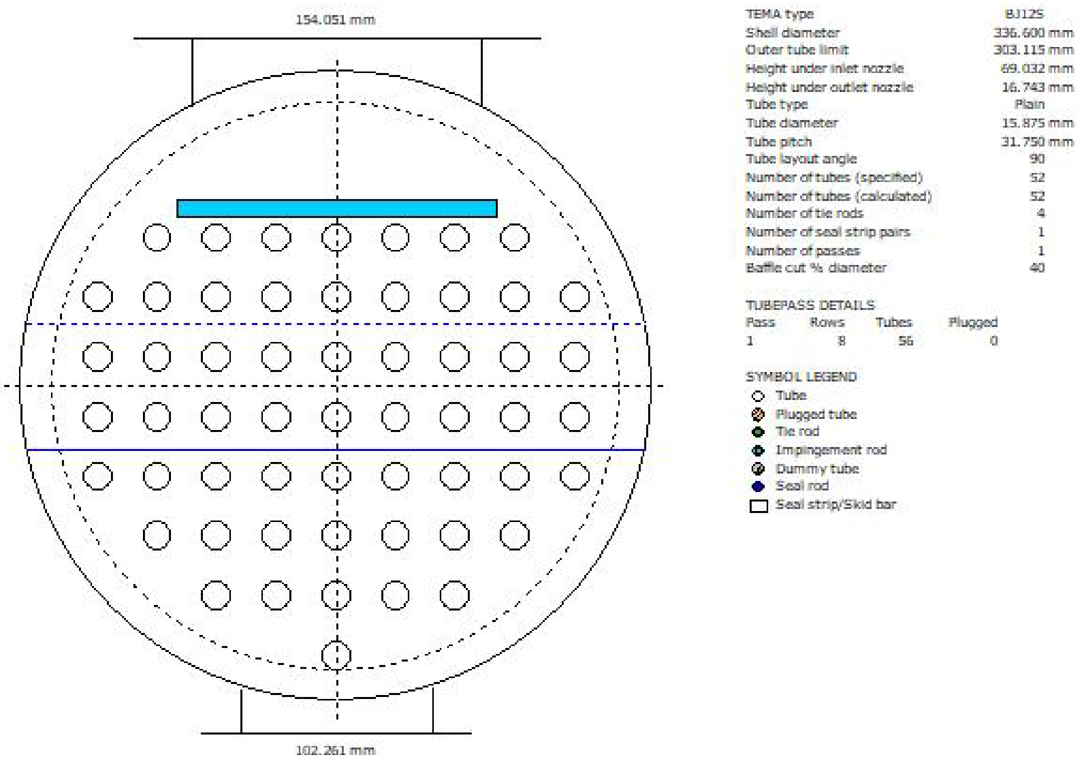


Figure 3.3: Software Output for Heat Exchanger

But it is too much difficult and costly at this stage to build such a heat exchanger having 52 number of tubes. So we further modified the calculation for the heat exchanger to minimize the number of tubes.

And after modification for small length we got final precise output for very small flow rates as follows

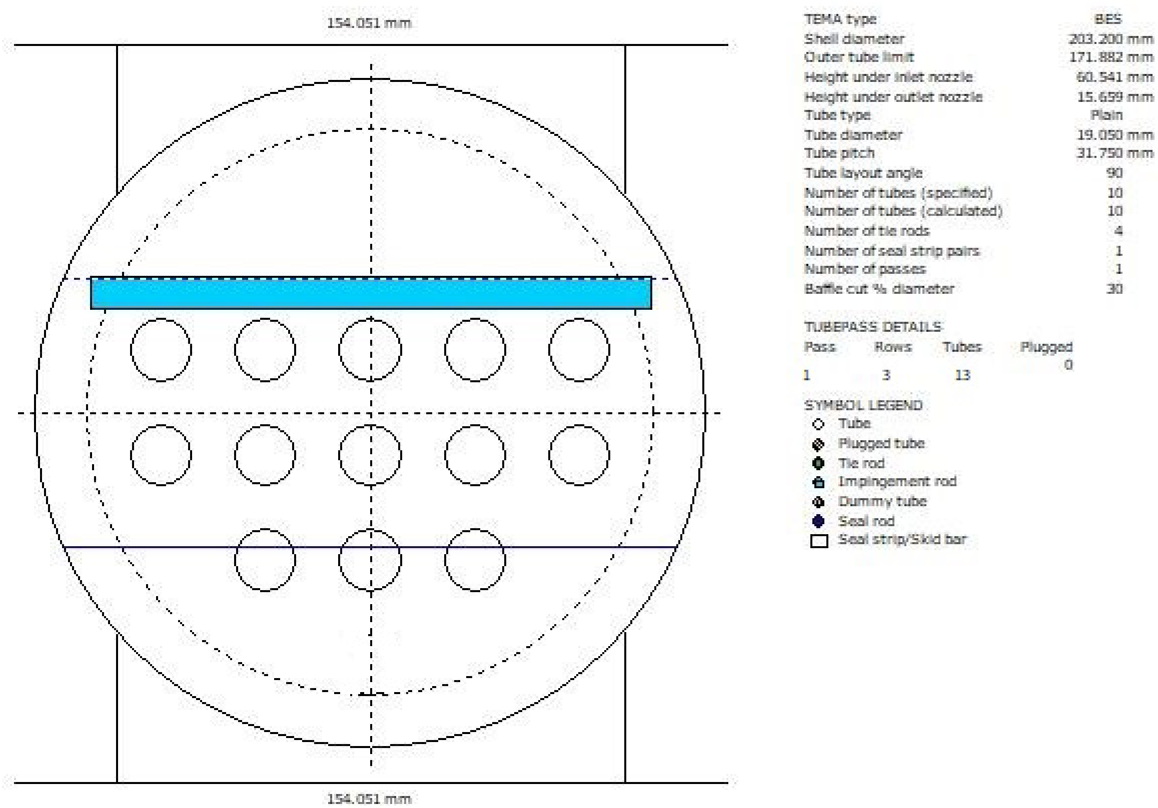


Figure 3.4: Software Output for Heat Exchanger [After precise correction]

We can see from the output that in this time the number of tube is 13 instead of 52, which is quite satisfactory for our experiment.

❖ Final Principal Dimensions

- **Length:** 2m
- **Shell ID**: 200mm
- **Tube OD**: 19mm
- **Tube wall thickness**: 1.5mm
- **Baffle spacing:** 200 mm
- **Pitch:** 31.75 mm

Chapter 04 Experimental Setup

4.1 Engine Specifications

In developing the set up for our experiment primarily we were in need of a running engine with exhaust gas flow at high temperature. We have been privileged by Bangladesh University of Engineering and Technology (BUET) Power Plant Caterpillar Engine which has a capacity of 1MW electric power supply. And the engine exhaust outlet here is almost 569°C temperature at full load. We have used the exhaust gas from this engine. Main specifications of this engine are

- 1 MW Caterpillar engine,1500 RPM
- Exhaust gas flow rate : approximately 6000 FPM
- Exhaust gas temperature: approximately 569°C.

4.2 Principal Equipment’s

1. Running engine with exhaust gas flow at high temperature
2. 20 Feet Iron Pipe [Mild Steel]
3. 6.9 Feet Heat Exchanger [Stainless Steel]
4. 50 Feet Circular Copper Pipe [0.5 inch diameter]
5. Condenser Bucket
6. IR Temperature Measuring Device
7. Water Flowmeter
8. Insulation [Glass Cloth]
9. Water Inlet Pipe & other various engineering tools.

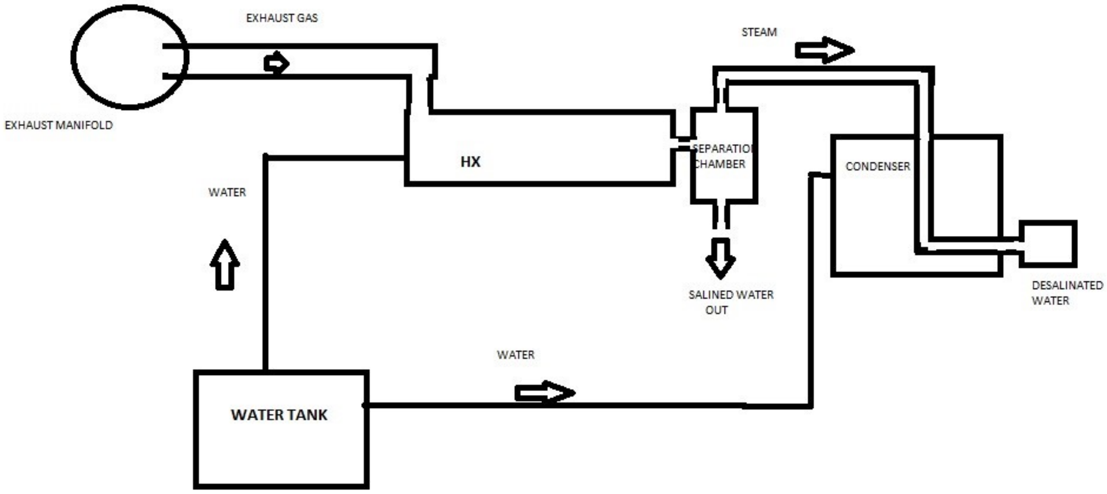


Figure 4.1: Experimental Set-up [Theoretical]

4.3 Preparations of the Accessories and Setup

In the later portion below here a sequential pictorial representation of our experiment set up is shown. It includes all from preparation of the iron pipe for the passage of engine exhaust gas till the last portion of our experiment that is the condensation portion.

Figure 4.2: Twenty Feet Iron Pipe [Mild Steel] Preparation.

Figure 4.3: Taking Exhaust Gas from the Engine Chimney.

Figure 4.4: Triple layer Insulation over Iron Pipe.

Figure 4.5: Preparation of Condenser.

Figure 4.6: Preparation of Separation Chamber

Figure 4.7: Heat Exchanger along with the Separation Chamber.

Figure 4.8: Water and Exhaust gas Inlet Set up.

Figure 4.9: Practical Full Experimental Setup.

4.4 Model Setup Cost Analysis

In setting up the model the following costs were involved

Material Cost:	Tk. 35000
Welding Cost:	Tk. 3000
Wages:	Tk. 17000
Labour Cost:	Tk. 2000
Transportation Cost:	Tk. 6000
Miscellaneous Cost:	Tk. 5000
Total Cost:	**Tk. 68000**

So, it is seen that the total cost for our set-up and model accessories is about **Tk.68,000** approximately, among which we were funded Tk. 40,000 from the University as well as from the Department and rest of the portion was managed by self-finance.

Chapter 05 Calculation and Result

5.1 Calculation

It is very difficult to make water into steam within a small length of the heat exchanger which is 2 m. Because latent heat is related here. The specific heat of vaporization of water is so high and that is 2268000 j/kg. So, we made some tactical application while making the heat exchanger. We have made a space between the water inlet and the tube opening side which is 3 inch. And the first tube row is placed almost at a distance of 0.04 m. We first give some water inside the exchanger until it fulfills the free space inside the exchanger. Then we stopped the flow of water. So water is trapped at that free space. When it starts to vaporize by taking heat from the hot exhaust gas, i.e. it's temperature is 100 degree Celsius then we again started to give the small flow rate of water. So ultimately this inlet water is mixed with the 100 degree Celsius water. Then the temperate of the mixture is just below 100 degree. So ultimately this temperature becomes water inlet temperature. So by this it becomes far easier to convert the water into steam because its temperature is already high.

Here comes the topic, which fluid has the minimum C. The maximum heat transfer possible inside the heat exchanger depends on C_{min}. Because,

$$Q_{max} = C_{min} \, (T_{hot,in} - T_{cold,in})$$

- **Calculation of water trapped inside the exchanger**:

(Assuming trapped area as a rectangle)

Length, L = 3 inch = 0.0762 m

Height, H = 0.04 m

Breadth, B = inner diameter of the exchanger = 200 mm

= 0.2 m

Volume of trapped water= L × B × H

$= 0.0762 \times 0.2 \times 0.04\ m^3$

$= 6.1 \times 10^{-4}$

So, mass of the trapped water, m = density × volume

$= 1000 \times 6.1 \times 10^{-4}$ kg

= 0.61 kg

➢ Calculation of C_{min} :

Trapped water continues to take heat from exhaust gas. When water of mass flow rate of 4.1 kg/sec is supplied inside the exchanger the total mass becomes, m = (0.61 + 0.0041) kg

= 0.6141 kg

So, $C_{water} = m_{water} \times c_{p\ water}$

$= 0.6141 \times 4184$

= 2569.4 J/K

C_p at 371°Celsius = 1.012×10^3 J/kg-k

m of exhaust gas at 470 KW load = 0.175 kg/sec

$C_{exhaust\ gas} = m_{exhaust\ gas} \times Cp_{exhaust\ gas}$

$= 0.175 \times 1.012 \times 10^3$

= 177.163 J/K

So. $C_{min} = C_{exhaust\ gas}$ = 177.163 J/k

➢ Effectiveness Calculation

- Effectiveness = Q/Q(max)
- Q =Actual heat transfer happened = ms × dt + ml_f
- Q(max)= C_{min} ($T_{hot,in}$ - $T_{cold,in}$)
- C = m × s (s at average temperature)

➢ Calculation of exhaust gas flow rate

❖ For 32% load:

- Original flow rate=4.2 kg/sec
- At 32% load flow rate= 4.2 × 32%=1.344
- exhaust entered our pipe= 1.344 × (4/18) × 0.8 =0.2 kg/sec

(Assuming 80% entry of exhaust due to inclination of pipe)

[Approximated 20% loss due to pipe inclination]

❖ For 23.5% load:

- Original flow rate=4.2 kg/sec
- At 32% load flow rate= 4.2 × 23.5%=0.987
- exhaust entered our pipe= 0.987 × (4/18) × 0.8 =0.175 kg/sec

[Approximated 20% loss due to pipe inclination]

		LOAD 640KW (32%)				
Obs. No	m(water),gm/s	T (avg)	Heat Transferred (Q)	m(exhaust gas),kg/s	Q (Max)	Effectiveness
1	4.1	99.49	9210	0.2	60763	15%
2	6.2	99.25	10621	0.2	60811.65	17.40%
3	7.3	99.11	16472	0.2	60839.9	27%
4	9.2	98.89	18423	0.2	60884	30.30%
5	10.4	98.74	19764	0.2	60914	32.40%
6	14.3	98.29	35446	0.2	61005	58%
7	22.7	97.56	35571	0.2	61153	58.16%

Table 5.1: Summary of Heat Transferred, Q_{max} and Effectiveness Calculation.

m(Water),gm/s	Effectiveness
4.1	15%
6.2	17.40%
7.3	27%
9.2	30.30%
10.4	32.40%
14.3	58%
22.7	58.16%

Table 5.2: Chart of Mass Flow Rate of Water and Effectiveness.

❖ Sample Calculation

For Mass flow rate of water 4.1gm/sec:

$m_{H_2O,in}$ = 4.19 gm/sec

$m_{H_2O,Hot}$ = 0.6 gm/sec

m_{steam} = 3.52 gm/sec

✓ T_{avg} Calculation:

$m_{H_2O,in} \times Cp,_{H_2O} \times (T- 30°) = m_{H_2O,exist} \times Cp,_{H_2O} \times (100 - T)$

or, $0.0041 \times 4200 \times (T - 30°) = 0.6 \times 4200 \times (100 - T)$

or, $T_{avg} = 99.49°C$

✓ Calculation of Q:

$Q = m_{H_2O,out} \times Cp,_{H_2O} \times (100-T_{avg}) + m_{H_2O,steam} \times Cp,_{H_2O} \times (100-T_{avg}) + m_{steam} \times m l_f + m_{steam} \times Cp_{steam} \times (101-100)$

$= 0.0006 \times 4200 \times (100-99.49)+0.00352\times4200\times(100-99.49)+0.00352\times1.996\times10^3$

= 9210 Joule

		LOAD 470 KW (23.5%)				
Obs. No	m(water),gm/s	T (avg)	Heat Transferred (Q)	m(exhaust gas),kg/s	Q (Max)	Effectiveness
1	2.5	99.71	3350	0.175	47984	7%
2	3.8	99.56	7022	0.175	47847	14.60%
3	4.9	99.39	7617	0.175	47877	16%
4	5.75	99.3	9017	0.175	47893	18.80%

Table 5.3: Summary of Heat Transferred, Q_{max} and Effectiveness Calculation.

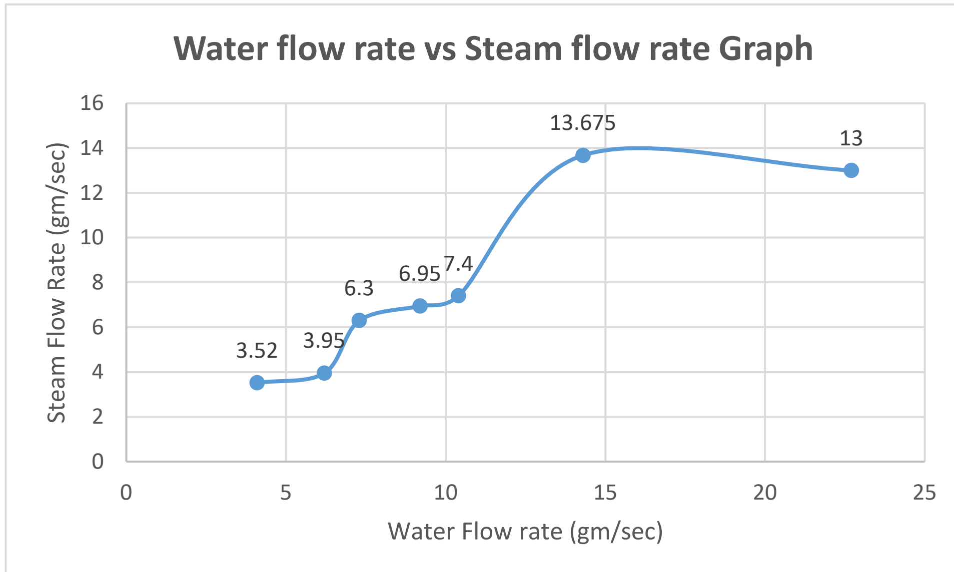
m(water),gm/s	Effectiveness
2.5	7%
3.8	14.60%
4.9	16%
5.75	18.80%

Table 5.4: Chart of Mass Flow Rate of Water and Effectiveness.

5.2 Graphs`

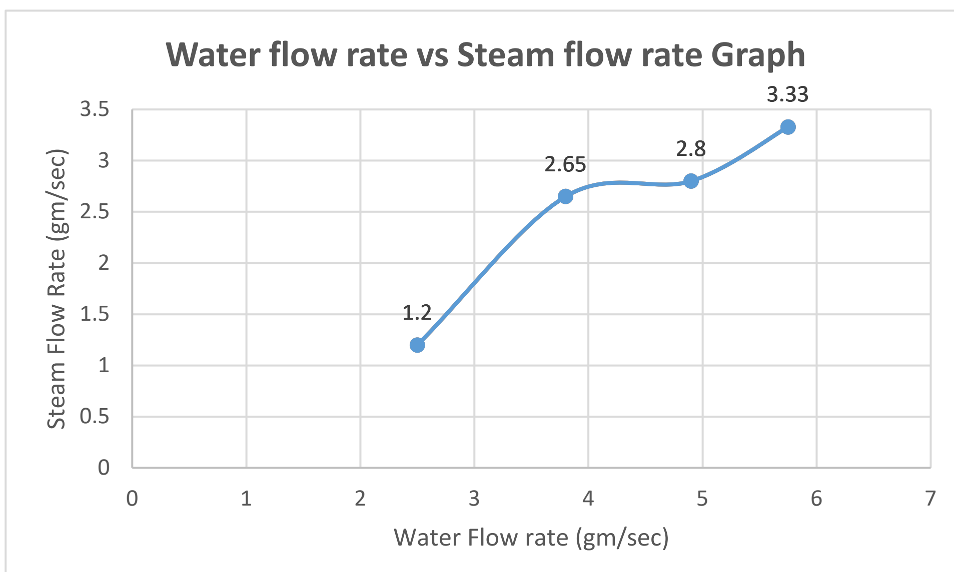
From the obtained result of our calculation we have manipulated those values in order to analyze our resulting values.

Load 645 KW	
Water	**Steam**
4.1	3.52
6.2	3.95
7.3	6.3
9.2	6.95
10.4	7.4
14.3	13.675
22.7	13



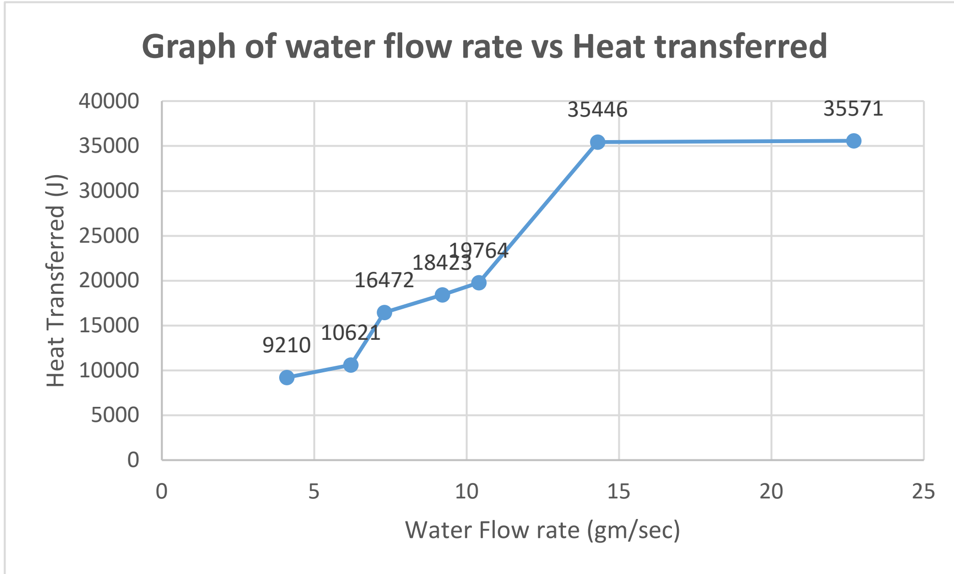
Graph 5.1: Steam flow rate vs Water flow rate.

Load 470 KW	
Water	**Steam**
2.5	1.2
3.8	2.65
4.9	2.8
5.75	3.33



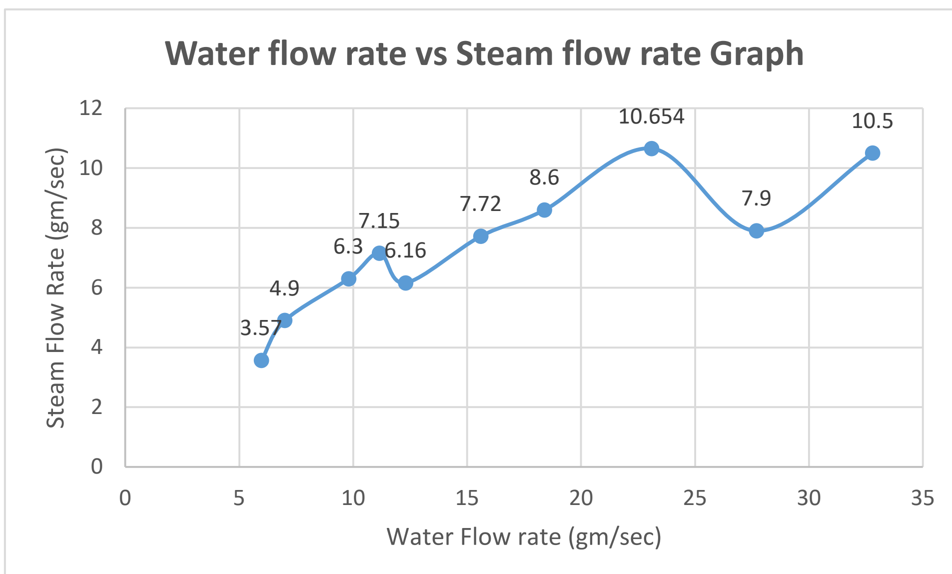
Graph 5.2: Steam flow rate vs Water flow rate.

Load 520KW	
Water	Heat
4.1	9210
6.2	10621
7.3	16472
9.2	18423
10.4	19764
14.3	35446
22.7	35571



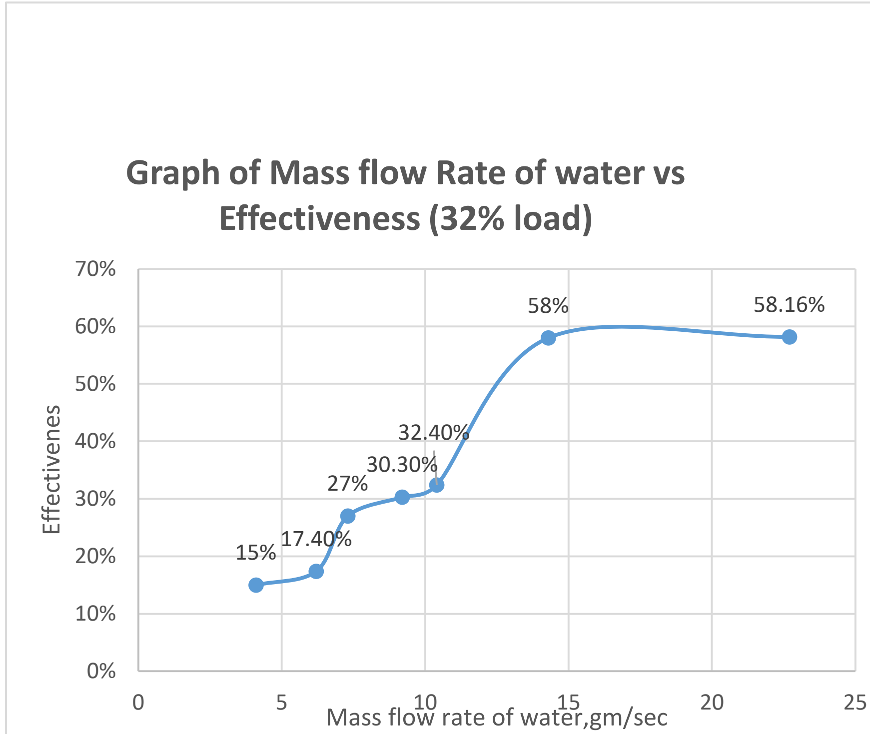
Graph 5.3: Heat transfer vs Water flow rate.

1000KW	
Water	Steam
5.97	3.57
7	4.9
9.8	6.3
11.15	7.15
12.3	6.16
15.6	7.72
18.4	8.6
23.1	10.654
27.7	7.9
32.8	10.5



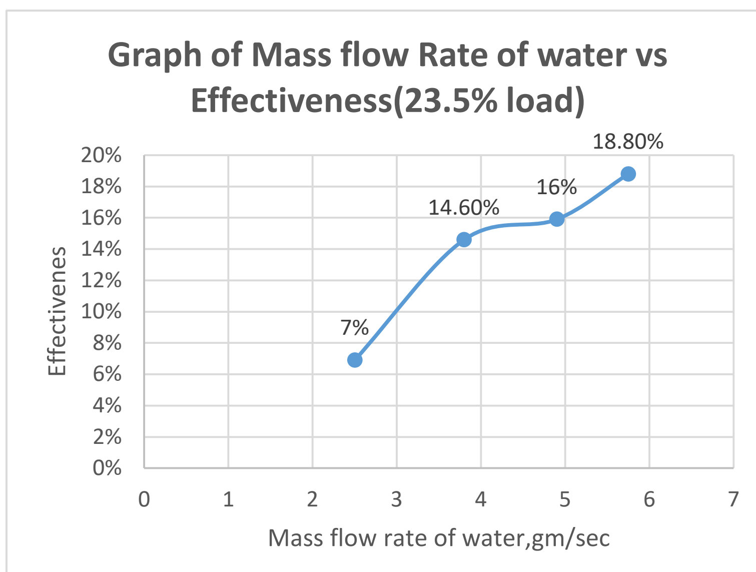
Graph 5.4: Steam flow rate vs Water flow rate.

m(Water),gm/s	Effectiveness
4.1	15%
6.2	17.40%
7.3	27%
9.2	30.30%
10.4	32.40%
14.3	58%
22.7	58.16%



Graph 5.5: Mass flow rate of water vs Effectiveness.

m(water),gm/s	Effectiveness
2.5	7%
3.8	14.60%
4.9	16%
5.75	18.80%



Graph 5.6: Mass flow rate of water vs Effectiveness.

5.3 Result

It is possible to get desalinated water at a flow rate of more than 10gm/sec by the technique used in our experiment (no loss of steam in the separation chamber) when the engine load is more than 1000 kW. We have got 0.7 gm/sec of desalinated water at water inlet flow rate of 4gm/sec when the engine load is 450 KW. Which means approximately 60 kg of desalinated water per day if the ship runs for 24 hours. But using large heat exchanger and proper separation chamber this amount can be far more than our obtained value. Some steam were lost in the separation chamber due to insufficient perfect equipment.

The effectiveness of heat exchanger increased with the increase of inlet water flow rates and with the increase of engine load. Because water passed through more tubes with the increase of water flow rates resulting a greater heat transfer inside the heat exchanger.

❖ Position of the desalinated water tank

The desalinated water tank should be below the sea surface inside the ship . The compartment should be located under the load water line. This types of positioning will eliminate the use of a pump to drive the desalinated water into the tank. The use of a pump means extra power consumption which ultimately reduces the ship's efficiency. The pipe inclination should be in a way which allows the continuous flow of the desalinated water under gravity force. This will also eliminate the risk of the damage of the pump because if somehow some portion of the steam is not condensed then it will enter the pump which will cause cavitation inside the pump and damage will occur. So, this kinds of positioning is safe and cost effective.

Chapter 06 Discussion

6.1 Discussion

In this modern market of shipbuilding industry it is desirable to increase capacity of ship as much as it is possible for one. So when we are using this technique in order to produce fresh water we can easily save the amount of ballast water needed for ship and thus can increase the ship’s capacity little bit. But in performing such operation in a moving vessel we have some limitations or drawback

❖ Major Drawbacks

- Fixing of position for the setup inside the vessel.
- Very small flow rate of water inlet
- Inadequate insulation
- Absence of a thermocouple for measuring exact temperatures of steam and exhaust gas.
- Using of stainless steel pipe instead of copper pipe

❖ Cleaning of salt from heat exchanger

➢ **Daily cleaning the salt by the sea water itself (closing the valve of exhaust gas pipe to stop the entering of exhaust gas into HX):**

If the direct sea water is pumped into the heat exchanger at a high velocity then it will take away the salt with it which is stored inside the heat exchanger during operation. For doing so, at that time of cleaning the exhaust gas should not enter the heat exchanger. If it happens then the sea water will continue to convert into steam and cleaning will not be done. So a valve system should be applied in the chimney of exhaust gas which allows the exhaust gas to go away through alternative pipe passage.

This salt along with sea water will then fall into the sea through the opening of separation chamber.

- **Mechanical cleaning at handsome time interval:**
  Mechanical cleaning of the heat exchanger should be done economically at a handsome interval.

6.2 Future Work

Using the condensation pipe which goes through the surface sea water will increase the frictional resistance of the vessel when it is under operation. Increase of extra resistance will result in extra requirement of power to run the ship at same speed. How much resistance is added by using this technique is a scope of future research.

The condensation pipe should be strong enough to sustain inside the sea water when the ship will move. Fouling and corrosion will also occur on the desalination pipe. So another topic of research is analyzing the strength of the pipe with fouling and corrosion consideration.

We couldn’t use the total exhaust gas from the chimney due to the risk of backflow to the engine. We only used some percentage of the exhaust by relatively small diameter pipe than the outlet diameter of the chimney. So how much maximum exhaust gas is possible to be taken considering reasonable back flow to the engine is another scope of research. Because the more the amount of exhaust gas enters the heat exchanger the more efficiently steam will be formed from the exchanger.

Nomenclature

Please use standard notations. Should you choose to skip this section, all symbols must be clearly defined everywhere relevant in the text. Leave only one blank line between the title of this section and the first parameter description. Please do not forget to delete this paragraph or the entire section accordingly

HX	Heat Exchanger
P	Pressure, kPa, bar
T	Temperature, ˚C, K
T_{avg}	Average Temperature,˚C, K
T_c	Cold fluid Temperature,˚C, K
T_h	Hot fluid Temperature,˚C, K
C_p	Specific heat at constant pressure, kJ/kg.K
C_v	Specific heat at constant volume, kJ/kg.K
T_i	Inlet temperature, ˚C
T_o	Outlet temperature, ˚C
P_i	Inlet pressure, kPa
P_o	Outlet pressure, kPa
T_{Win}	Water inlet temperature, ˚C
T_{Wout}	Water outlet temperature, ˚C
s	Specific entropy, kJ/kg.K
h	Specific enthalpy, kJ/kg
v	Specific volume, m^3/kg
W_{total}	Total plant power, W
$W_{pump, theo}$	Theoretical pump work, W
$W_{pump, act}$	Actual pump work, W
W_{net}	Net plant power, W
m_{water}	Mass flow rate of water, gm/sec
m_{steam}	Mass flow rate of steam, gm/sec
m_{vapour}	Mass flow rate of vapour, gm/sec
m_{gas}	Mass flow rate of gas, gm/sec
Warm water	Mass flow rate of warm ocean water, kg/s
cold water	Mass flow rate of deep-cold ocean water, kg/s
q_{in}	Heat input , kj/kg
q_{out}	Heat output , kj/kg

A_h	Heat transfer area, m^2
A_t	Tube Area, m^2
OD	Outer Diameter
ID	Inner Diameter
t_h	Thickness of heat transfer material, mm
U	Overall heat-transfer coefficient, W/m.˚C
$\dot{V}_{warm\ water}$	Volume flow rate of warm water, m^3/s
$\dot{V}_{cold\ water}$	Volume flow rate of cold ocean water, m^3/s
Q_{loss}	Heat loss, W

Non Dimensional Numbers

R	Reynolds Number
Pr	Prandlt Number
Nu	Nusselt Number

Abbreviations

OTEC	Ocean Thermal Energy Conversion
NTU	Number of Transfer Unit
CC	Closed Cycle
LMTD	Log Mean Temperature Difference
TEMA	Tubular Exchanger Manufacturers Association
HTRI	Heat Transfer Research, Inc.
SS	Stainless Steel
MS	Mild Steel
O&M	Operations and Maintenance

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