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Seawater Desalination Using Waste Heat Recovery on Passenger Ship

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ABSTRACT:

This paper presents a study for sea water desalination on board of passenger ships using waste heat from the engine. Three thermal methods for water desalination were explored, namely Forward Feed Multiple Effect Evaporation, Once Through Multi-Stage Flash, and Brine Circulation Multi-Stage Flash. Computer simulation has been developed to calculate the parameters of the desalination plant; the required amount of fresh water and the corresponding total heat transfer area. The optimum plant selection is the one which achieves the required distillate flow rate with minimum heat transfer area. The effect of different variables on the plant selection has been studied, i.e. steam temperature, exhaust temperature, and intake water flow rate. An existing passenger ship has been selected to examine the proposed method where the optimum desalination method has been selected for her using the developed software. To assess the effectiveness of the proposed method, the economical, environmental, and technical gains are numerically analyzed. Using the waste heat recovery leads to reducing the unit product cost of freshwater by circa 30%. The plant reduced the emissions by about five thousand tons of CO_2, 100 tons of NO_x and 35 tons of SO_2 per year. Applying the optimum design of the proposed salt water desalination on the case study saved 2.7 $/ m^3 as a minimum comparing to the average cost of fresh water in ports. These savings can cover the plant capital cost in six years at most.

Keywords: Desalination, Water generation, Waste heat recovery, Passenger ship

1. INTRODUCTION

In 2003, B. R. Smalley, (1996 Noble Laureate in Chemistry) defined the humanity's top ten problems for the next 50 years [1]. Energy and water exist on the top of his list. The challenge of permanently providing clean, safe, and fresh water is the key of life. While the dilemma of fuel price and availability is an inevitable fact. For passenger ships, fresh water usually represents a high percentage of the dead weight [2]. The average water consumption per person in new ships rises to more than 300 liters per day [3]. The above facts lead directly to the concept of sea water desalination on board passenger ships. This will lead in a direct way to the reduction of dead weight and the cost of purchasing water.

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Water desalination is usually done by one of two methods; membrane methods and thermal methods. The thermal process is one of the oldest desalination methods which are based on water evaporation. Fresh water is formed when the condensate vapor is formed while the brine remains with high concentration and usually thrown back to the sea. Since the thermal methods require source of energy, the waste heat from the engine can be used as this source. This improves the efficiency of the engine, reduces the emissions and creates an environmental friendly system for water generation without using extra energy source.

Fresh water generation on shipboard is indeed not novel, but it has been applied on many shipboards especially on passenger ships. Morsy et al. [4] proposed fresh water generator onboard passenger ship which uses the waste heat recovered from scavenging air to provide the heat required to evaporate sea water. The proposed system minimizes the fresh water supply by 8 tons/day; an amount sufficient for 20 persons per day. Based on the fact that the brine which passes to over board and the steam will stay condensate in the steam condenser converting into fresh water. It was recommended to use brine in heating purpose or use this losses energy in next study.

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Mehner and Penney [5] presented a full-scale shipboard system that incorporates multimedia filtration and ultra-filtration, yet requires minimal space and optimal power usage. This study received the first place award given by the US Navy in a competition made to improve the pretreatment for shipboard Reverse Osmosis potable water system. The annual savings were $17,000, which resulted in a payback period of 14 years.

In 2015, Guler el al. [6] studied the Reverse Osmosis, RO system capability under different conditions together with cost analysis examination on a relatively small cruise ship. The system used had a daily water treatment capacity of 30 m^3. The study revealed that the used system is capable of meeting the needs of the drinking and potable water. Cost-wise, the resulting water was more costly compared to water supply costs on land, as the above authors expected in the beginning of the research. Therefore, the authors recommended that one should evaluate alternative energy sources, such as solar, wind or wave energy for seawater desalination.

Grand Princess Cruise ship was the largest and most expensive passenger ship ever built in 1998. The ship, which uses more than 260,000 gallons of fresh water per day, has implemented evaporator management programs aimed at optimizing the operation of fresh water evaporators by utilizing the waste heat generated by the ships engines. The cooling water passed through these engines is re-used to heat the salt water in the evaporators [7].

The aim behind this paper is to study the sea water desalination onboard passenger ship using thermal methods. This goal is done using the waste heat from engine as an energy source for the desalination plant. Desalination is made by three methods, namely; the Forward Feed Multiple Effect Evaporation (MEE-FF), Once Through Multi Stage Flash (MSF-OT) and Brine Circulation Multi Stage Flash (MSF-BC).

A computer simulation has been developed to calculate the heat transfer area required for fresh water desalination unit on ship board based on the required water demand. The program is calibrated versus another research results to confirm its validity. *MS Freedom of the sees* ship has been selected to examine the proposed method. The program is used to find the optimum specification of water desalination plant fitted onboard the ship. Besides, it is also used to study the effect of different variables on the distillate flow rate and the total area of the plant. Eventually, the optimum desalination method is selected for the candidate ship; which gives the required water demand for minimum heat transfer area.

2. WATER DESALINATION METHODS

The provision of freshwater around the world highly depends on seawater desalination. Saline seawater is separated into two streams in a typical desalination process; fresh water stream with low concentration of salts and concentrated brine stream. It requires energy to desalinate, while several different technologies for separation can be used. This section reviews the different technologies that have been developed over the years. Figure 1 shows the desalination methods that consume energy.

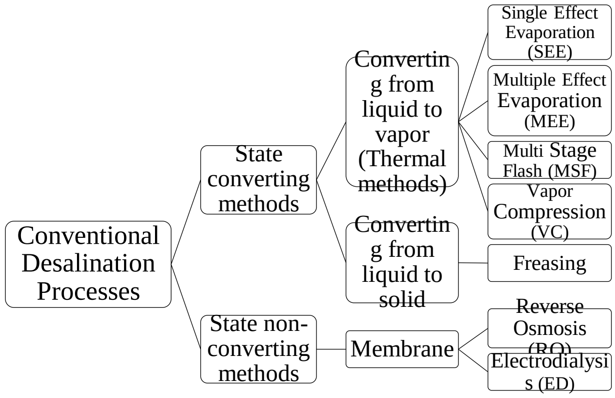

Figure 1: Conventional desalination process

For the special environment on ships, the most accessible access is to reclaim the waste heat of the main engine for freshwater production. Therefore, among types of technology, thermal distillation that applies the waste heat from the exhaust gas and jacket water cooling has been proposed. Both MSF and MEE belong to heat-operated type units [7].

2.1. Foreword Feed Multiple Effect Evaporation

The forward feed multiple effect evaporation (MEE-FF) process occurs in a series of evaporators, called *effects*. It makes the feed water (M_f) boils without additional heat after first effect via reducing the pressure inside the effects. The flow direction of the brine and vapor is in the lowering pressure which is gradually reduced from effect one to n. The process consists of n number of evaporators, seawater pre-heaters, equal to $(n - 2)$, a train of collecting boxes, equal to $(n - 1)$, down condenser and a vacuum system.

The intake seawater stream (M_{cw}+M_f) passes through the down condenser where it exchanges heat with vapor produced from the last effect and increases its temperature from a cooling water temperature T_{cw} to feed temperature T_f, Figure 2. A part of this stream M_{cw} is rejected to the sea to remove the excess heat added to the system.

The remaining feed water M_f is chemically treated and pumped through a series of pre-heaters to increase its temperature from T_f to t_2. Heating of the feed seawater in the pre-heaters is made by condensing the flashed off vapor from the effects d_j and the collecting boxes $d_j^{\backslash}$. The feed water is sprayed over the outside surface of the first evaporator tubes. The brine temperature rises to the boiling temperature T_1 which corresponds to the pressure of the vapor space. The vapor formed in the first effect flows as heating vapor to the second effect. The non-evaporated seawater from the first effect B_1 enters the second effect as feed water. From the second effect onward, freshwater is produced inside the effect by two different mechanisms; boiling D_j and flashing d_j [8,9]. Motive steam M_s, extracted from an external source drives vapor formation in the first effect [10].

2.2. Brine Circulation Multi Stage Flash

The multistage flashing desalination units (MSF) are typically constructed with large capacity that may vary from 50,000 to 75,000 m^3/d. This capacity is almost 2–3 times the conventional units installed in 1980's [11]. The process is based on the principle of flash evaporation, as seawater is evaporated by reducing the pressure due to raising the temperature.

In MSF distillation process vapor formation takes place within the liquid bulk instead of the surface of hot tubes. The hot brine is allowed to freely flow and flash in a series of chambers; this feature keeps the hot and concentrated brine from the inside or outside surfaces of heating tubes. This is a major advantage over the original concept of thermal evaporation, where submerged tubes of heating steam are used to perform fresh water evaporation.

The process elements are illustrated in Figure 3 where the flashing stages are divided among the heat recovery and heat rejection sections. The main aim of adding heat rejection section is to control the temperature of the intake seawater and to reject the excess heat added in the brine heater. This process starts with the brine heater and finishes with the rejection section and between them there is recovery section that consists of a set of evaporation stages.

The intake seawater stream $M_f + M_{cw}$ is introduced into the condenser tubes of the heat reject section where its temperature is increased by absorbing the latent heat of the condensing fresh water vapor. The warm stream of intake seawater is divided into two parts; cooling seawater M_{cw} and feed seawater M_f .

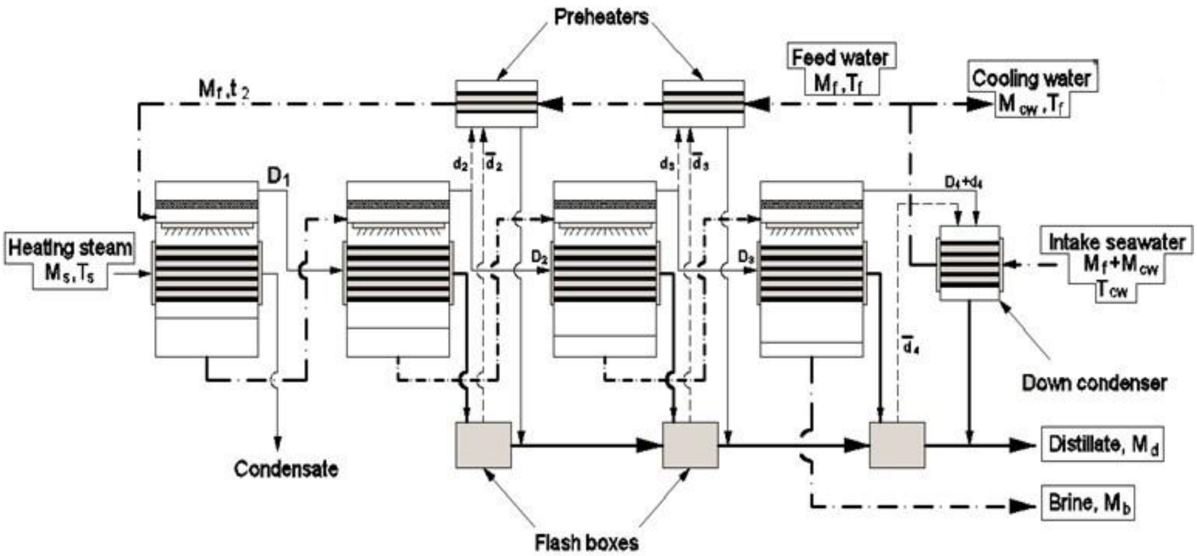

Figure 2: Conventional of a MEE process (forward feed configuration) [8]

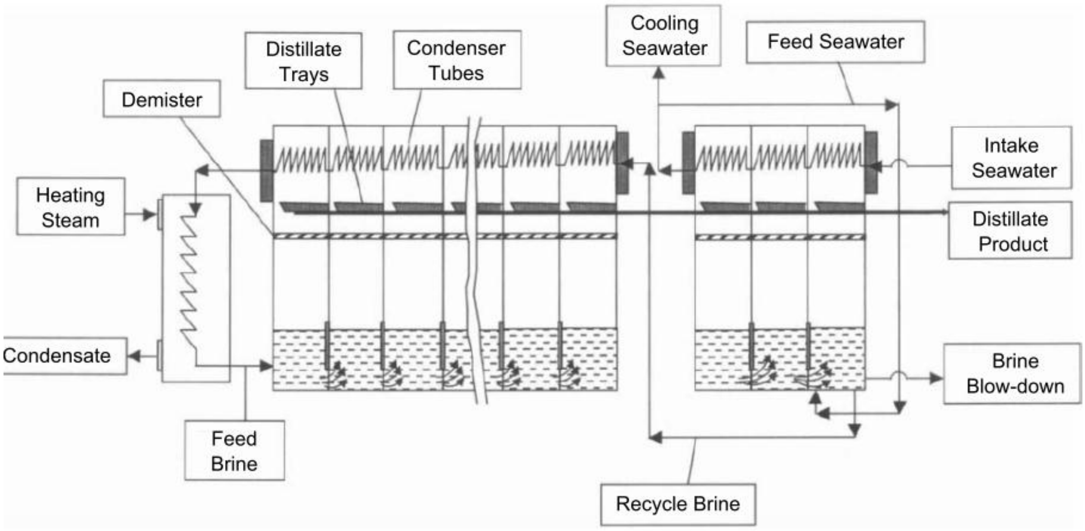

Figure 3: Brine circulation multi stage flash desalination process (MSF-BC) [9]

The cooling sea water is rejected back to the sea and it represents 70% of the intake water. While, feed seawater M_f is chemically treated and then mixed with the brine pool of the last flashing stage in the heat rejection section.

The rejection section consists of two–four stages where the intake water flows from the last stage which has the lowest pressure and the lowest temperature. The cooling seawater temperature rises by seven or eight degrees in the heat rejection section. The brine recycle stream M_r is extracted from the brine pool of the last stage in the heat rejection section and is introduced into the condenser tubes of the last stage in the heat recovery section. As the stream flows in the condenser tubes across the stages, it absorbs the latent heat of condensation from the flashing vapor in each stage.

The brine recycle stream M_r enters the brine heater tubes where the heating steam M_s is condensed on the outside surface of the tubes. The brine stream absorbs the latent heat of condensing steam and its temperature is increased to its maximum design value, i.e. known as the top brine temperature T_0. The hot brine enters the flashing stages in the heat recovery section and then in the heat rejection section where a small amount of fresh water vapor is formed by brine flashing in each stage. The flashing process takes place due to decreasing the stage saturation temperature which in turn causes a reduction in the stage pressure.

Salt water flows in two levels and two opposite directions in the recovery section. In the upper level, the water flows inside the tube bundle in the direction of brine heater where is used as cooling water condensate the flashed off vapor. Whereas in the lower level, the salty water flows in the direction of heat rejection section after leaving the heat exchanger in the previous stage.

As the brine recycle stream M_r flows in the lower level in the brine pools of the stages and under suitable conditions of pressures and temperature, the water remains in turmoil and the flashing evaporation occurs. Moreover, the flashed off vapor is produced and the brine water moves from a stage to the next where the pressure and temperature are decreased by the control halls between the stages. Whenever the steam is produced, the salt concentration of the brine water is increased. The brine reaches the most salt concentration in the last stage in the heat rejection section. There are disposed of a small portion of a highly concentrated brine, then the feed water is added to restore the concentration degree to the required level. Afterward, this water is pumped to the upper level in heat recovery section. The flashed off brine flows through the demister where most of the entrained brine droplets are captured by the demister wires. Continuous removal of the droplets increases the captured droplet size. This may result in detachment of the large droplets and settling back to the brine pool.

The flashed off vapors condense on the outside surface of the condenser tubes which are collected in the distillate tray in each stage. These vapors flow through the stages where its quantity is increased until forming the distillate product stream M_d which moves from the last stage of heat rejection section to outside of the unit. The flashing process and vapor formation are limited by the rise of the specific vapor volume at lower temperatures and difficulties encountered for operation at low pressures. The limit of temperature of the last stage ranges from 30 to 40 ˚C for winter and summer operation, respectively like the MSF-OT method. The stage width and length should be large enough to maintain the vapor velocity below 6 m/s [9]. This is necessary to limit entrainment of brine droplets in the vapor stream and to allow for settling of the brine droplets.

3. ENERGY BALANCE ANALYSIS OF SHIP MACHINARY

In general, diesel engines represent the majority of prime movers and auxiliaries for seagoing ships. This pervasiveness of diesel engines is attributed to the inexpensive heavy oil and the highest efficiency compared with all other heat engines. Nevertheless, its efficiency typically does not exceed 51% while the rest energy is discharged into the atmosphere in the form of exhaust gas, jacket water cooling, heating of lubricating oil and small part of this energy goes out as radiation [7,12].

There are two main strategies for increasing the diesel engine efficiency. The first strategy is a direct method by the improvement of the combustion process. Whether Homogeneous Charge Compression Ignition (HCCI), Lean Combustion or Stratified Combustion, all of them flow toward fuel consumption and emissions reduction. Unfortunately, achieving these improvements and meeting the demands of practical combustion systems confronts some unresolved challenges that have kept these technologies from being applied in commercial engines [13,14].

The second strategy is an indirect method to improve the efficiency by recovering of the available waste heat from exhaust gas and cooling water. The Waste Heat Recovery (WHR) for internal combustion engines is the optimal method to increase their efficiency especially for the engines of large ships that run in constant speed for a long time.

Several technologies have been investigated for waste heat recovery (WHR) including thermoelectric generators (TEG), turbochargers, six-stroke cycle internal combustion engines, the Rankine Cycle (RC), etc.

Rankine Cycle (RC) is a thermodynamic cycle that converts thermal energy into mechanical work, which is commonly found in thermal power generation plants.

The conventional RC system consists of four components: pump, evaporator, expander and condenser. The pump drives the working fluid to circulate through the loop, and the evaporator utilizes a waste heat source to vaporize the working fluid. The fluid vapor expands in the expander and converts thermal energy into mechanical power output. Then, the expanded vapor flows through a condenser to turn back into liquid phase, thus completing the cycle. The most common and simple Rankine Cycle system structure is shown in Figure 4, which utilizes the exhaust gas as the only heat source to evaporate the working fluid. The heat from engine coolant is dissipated to the environment through the radiator and is not recovered by the RC system [15,16].

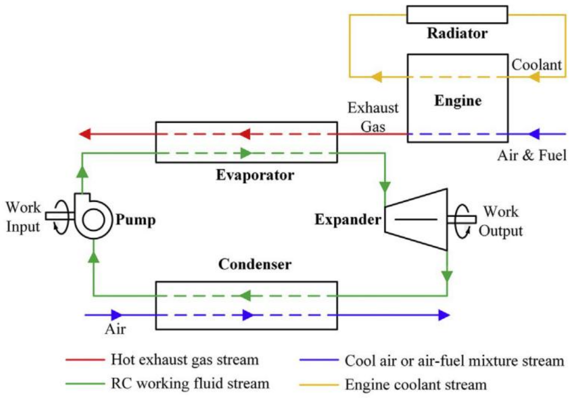

Figure 4: The conventional RC system [17]

In the turbocharger, the energy of the exhaust heat is converted to kinetic energy of power compressor. The turbocharger consists of a turbine and compressor on the same shaft. The exhaust heat energy transfers to the turbine which drives the compressor to compress ambient air. Normally, the air heated by the compression passes through a cooler which reduces its temperature and increases its density, and then is delivered to the air intake manifold of the engine at higher pressure. Thus, the amount of air entering the engine cylinders is greater, allowing more fuel to be burnt. As a consequence, the engine produces more power without increasing the engine size. Figure 5 shows typical arrangement for a 4-sroke engine with turbo charging [12].

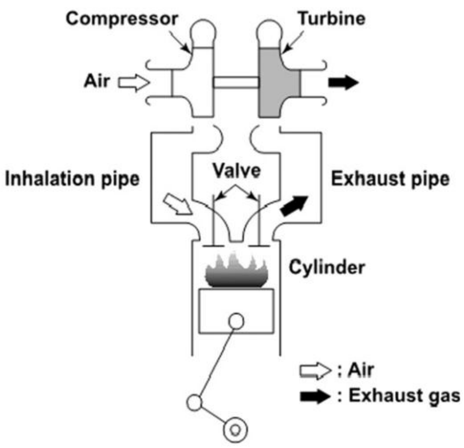

Figure 5: Typical arrangement for a 4-sroke engine with turbo charging [12]

Before selecting the proper waste heat recovery method, one should consider the engine size, the ship type, route, the loading condition, the surrounding environment, the energy balance, the machinery arrangement, and the operating scenarios. It should be noted that the energy balance is the most important factor considered for WHR.

The heat balance calculation process is just an application for the first law of thermodynamics on internal combustion engines. The first law of thermodynamics is applied to identify the amount of the recoverable heat from the internal combustion IC engines. The law states that: energy can be converted from one form to another with the interaction of heat, work and internal energy, but it cannot be created nor destroyed, under any circumstances. For a thermodynamic cycle of a closed system, which returns to its original state, the heat Q_{in} supplied to a closed system in one stage of the cycle, minus that Q_{out} removed from it in another stage of the cycle, equals the net work W done by the system. The steady flow first law of thermodynamics for an IC engine is expressed by Equation 1.

$$Q = Q_{in} - Q_{out} = Work\ done \quad (1)$$

$$Q_s = P_b + Q_w + Q_{lub} + Q_{exh} + Q_{rad} \quad (2)$$

where

P_b is the brake power, Q_{ex}, Q_w, Q_{lub}, Q_{rad} are the heat loss by exhaust gas, the cooling water, the lubricating oil and the radiation, respectively. The supplied fuel energy Q_s in kW is given by equation 2 and 3.

$$Q_s = FC \cdot LCV \quad (3)$$

where

FC is the fuel consumptions in kg/s,
LCV is the lower calorific value of the fuel in kJ/kg.

According to the heat transfer formula from one point to another, the cooling water loss can be calculated as given by Equation 4.

$$Q_w = m_w \cdot c_w \cdot \Delta T_w \quad (4)$$

where

m_w is the mass flow rate of cooling water in kg/s,
c_w is the specific heat of water in kJ/(kg.K), and
ΔTw is the temperature difference between outlet and inlet water in kelven.

The amount of heat carried away by the lubricating oil Q_{lub} is calculating using Equation5.

$$Q_{lub} = m_{lub} \cdot c_{lub} \cdot \Delta T_{lub} \quad (5)$$

where

m_{lub} is the mass flow rate of lubricating oil in kg/s,
c_{lub} is the specific heat of oil in kJ/(kg.K), and
ΔT_{lub} is the temperature difference between outlet and inlet oil.

The Exhaust heat losses given by Equation 6.

$$Q_{exh} = m_{exh} \cdot c_{air} \cdot T_{exh} - m_{air} \cdot c_{air} \cdot T_{air} \quad (6)$$

Where
m_{exh} and m_{air} are the mass flow rate of the exhaust gas and the intake air in kg/s,
c_{air} is the specific heat of air in kJ/(kg.K), and
T_{exh} and T_{air} are the temperatures of exhaust gas and air respectively
The radiation heat loss is given by Equation 7.

$$Q_{rad} = Q_s - (P_b + Q_w + Q_{lub} + Q_{exh}) \quad (7)$$

The above mentioned mathematical equations can be used to identify the amount of waste heat which can be exploited in water desalination.

4. PROPOSED FRAMEWORK OF FRESH WATER GENERATION IN SHIPS

In this section, the exploitation of waste heat energy to produce fresh water is presented. To this end, three thermal desalination methods, which use the waste heat from the exhaust gas and jacket water, are examined, namely; the Multiple Effect Evaporation (MEE), the Once Through Multi-Stage Flash (MSF-OT) and the Brine Circulation Multi-Stage Flash (MSF-BC) where all methods belong to the heat-operated type units.

Figure 6 shows the proposed coupling of a desalination plant with the heat recovery cycle. The first contribution of the paper is reusing the salt water going out of the cooler as an intake water to the desalination unit. The fresh water-cooling system has two parts; high temperature (HT) and low temperature (LT) cooling system. The former (HT) cools cylinders, cylinder heads and the first stage of the charge air cooler. While the later (LT) cools the second stage of the charge air cooler, and the lubricating oil in an external

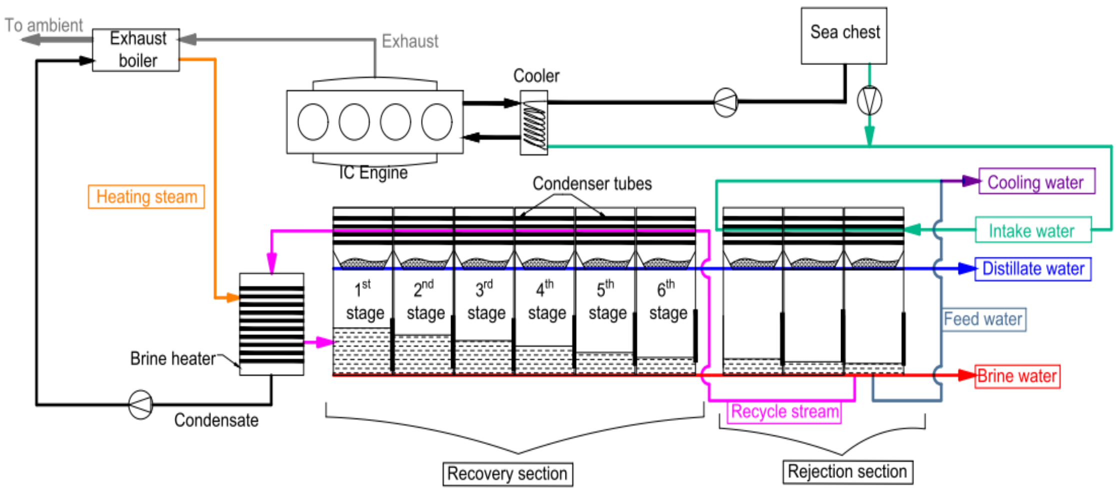

Figure 6: Proposed desalination plant with heat recovery cycle

cooler. The low temperature cooling system has been used for heating the intake seawater. This is because most of ships already have a waste heat recovery system that uses a high temperature cooling system for propulsion or for electric power generation. Consequently, the low temperature cooling system has to be used instead of the high temperature system for generalizing the proposed fresh water generation system on passenger ships. The fresh water flows out of the cooler to the engine with a temperature $T_{out(FW)}$. The flow rate for this stream is defined by the primary machine characteristics. The fresh water enters the cooler with higher temperature $T_{in(FW)}$ due to engine heat, whereas, the salt water enters the cooler with a temperature $T_{in(SW)}$ and leaving with a higher temperature $T_{out(SW)}$ which is typically acquired from the fresh water. The out flow from the cooler is used as intake flow rate for the desalination plant. The temperature of the fresh water entering the cooler from the machine depends on the thermal load of the cooler Q_{cooler} as well as the temperature of the salt water going out of the cooler.

The capacity of the sea-water pumps is determined according to the type of the coolers used and the dissipated heat. It usually flows with a rate in the range between 1.2 and 1.5 relative to the fresh water flow according to Equation 8:

$$Q_{cooler} = M_{FW} \cdot C_p \cdot \left(T_{in(FW)} - T_{out(FW)}\right) = M_{SW} \cdot C_p \cdot \left(T_{in(SW)} - T_{out(SW)}\right) \quad (8)$$

where
Q_{cooler} is the thermal load of the cooler (kW),
M_{FW} is the a flow rate of fresh water in (kg/s),
C_p is specific heat capacity in (kJ/kg.K) and
M_{SW} is the salt water flow rate in (kg/s).

The intake seawater stream M_{int} is introduced into the condenser tubes of the heat rejection section where its temperature is increased. The warm stream of intake seawater is divided into two parts: the cooling seawater M_{cw} and the feed seawater M_f . The cooling sea water is rejected back to the sea. While the feed seawater M_f is mixed with the brine pool of the last flashing stage in the heat rejection section. The brine recycle stream M_r is extracted from the brine pool of the last stage in the heat rejection section and then is introduced into the condenser tubes of the last stage in the heat recovery section. As the stream flows in the condenser tubes across the stages, it absorbs the latent heat of condensation from the flashing vapor in each stage.

The second contribution of the paper is using the exhaust to generate the heating steam. After the exhaust leaves the turbocharger and before going out to the air, it passes through an exhaust boiler to evaporate fresh water which is used as heating steam M_s in the thermal desalination plant. This heating steam enters to the brine heater to increase the feed water temperature. The heating steam M_s is condensed on the outside surface of the brine heater's tubes and then it returns back to the exhaust boiler. The feed seawater M_f enters the first stage of the desalination plant where the pressure is intentionally reduced. The pressure reduction results in evaporating a part of the feed water that condensates on the condenser tubes. The rest of the feed water (which has not been evaporated) is forwarded to the next stage with reducing the pressure. This process is repeated till the last stage.

The heating steam temperature and the flow rate of water flowing out of the heat exchanger are determined from equation 9.

$$M_w \cdot (H_s - H_w) = M_{exh} \cdot C_{p(air)} \cdot (T_{exh(inlet)} - T_{exh(outlet)}) \quad (9)$$

where M_w is water flow rate in (kg/s), H_s is specific enthalpy of the steam in (kj/kg),

H_w is specific enthalpy of the water in (kj/kg), M_{exh} is exhaust flow rate in (kg/s), $T_{exh(inlet)}$ is the exhaust temperature before boiler after turbocharger, and $T_{exh(outlet)}$ is exhaust temperature after boiler going to the air. The steam flow rate is controlled by changing the temperature of heating steam and the exhaust gases after the boiler.

5. NUMERICAL ANALYSIS

Three desalination methods have been applied on shipboard; MEE-FF, MSF-OT and MSF-BC. A visual basic (VB) program has been developed to estimate the required heat transfer area and the distillate flow rate for a desalination plant. The energy source of desalination plant is taken from the heating steam produced from the boiler and a part of intake seawater coming from the cooler. Figure 7 shows the flow chart of the visual basic program.

The program starts with calculating the value of pump capacity of salt cooling water and the salt water temperature after the cooler which will be considered as the intake flow rate M_{int} and the cooling water temperature T_{cw} for a desalination unit. The steam temperature and the steam flow rate after the boiler are calculated. This steam is used as a heating steam in the desalination plant. The first part of the program shown in Figure 7 ends with selecting a desalination method. After calculating the distillate flow rate M_d and the total heat transfer area A_T for the three methods, one method will be selected as the optimum method to be applied onboard ships.

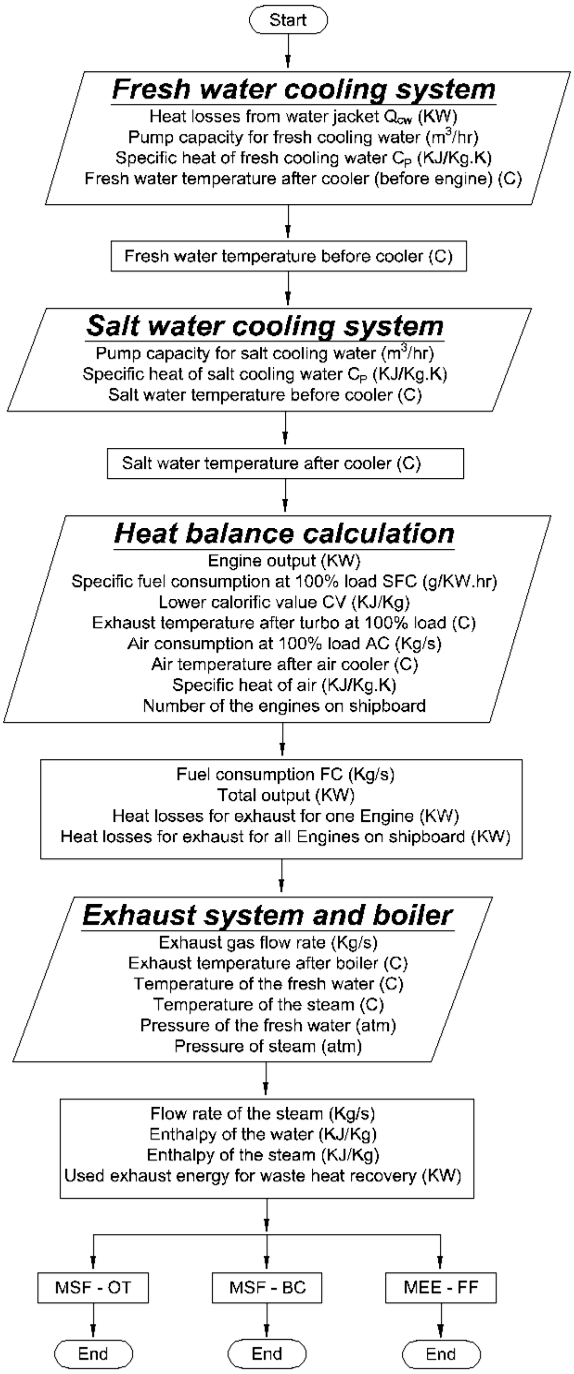

Figure 7: Flow chart of the water desalination program

As an example, Figure 8 shows the flow chart of the MSF-BC component. Then, the program starts with calculating the flow rates; M_r, M_d, M_f, M_b *and* M_{cw}. Then, the heat transfer areas A_b, A_r *and* A_j are computed. The output of this stage is: the distillate and the brine flow rates, the salinity, the temperature, the feed temperature, the brine density and the pressure of each stage. The next stage in the VB program is written to calculate the flow rates *D*, *B*, the temperatures *T*, T_f, and the salinity *X* . This process is made by the "do-loop" structure which operates in *N* iterations where *N* represents the total number of stages. Afterward, the stage length is determined in the last iteration.

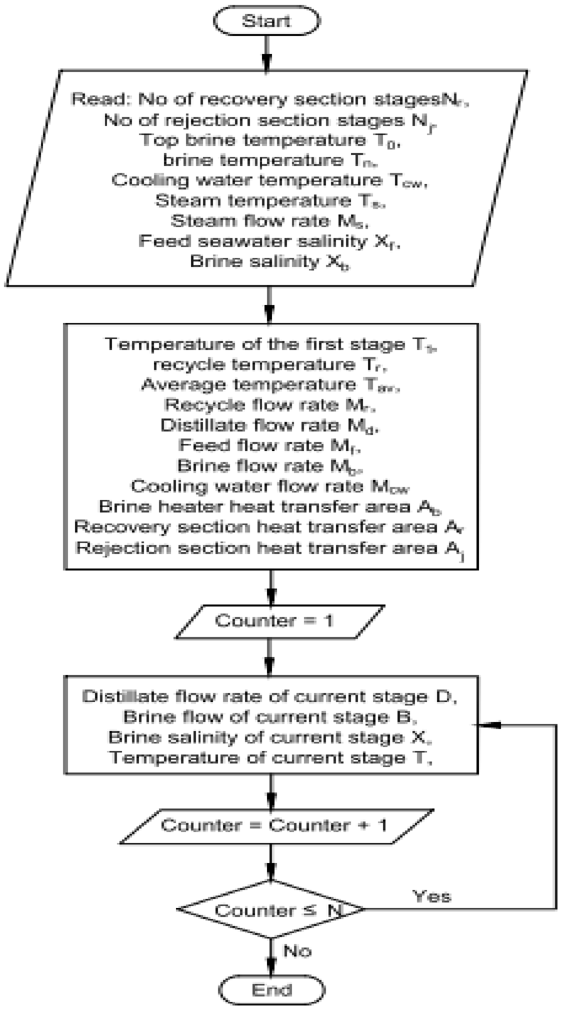

Figure 8: Flow chart of the MSF-BC program

6. PROGRAM VALIDATION

The results obtained from the VB program are checked versus the results presented by Dessouky and Ettouney [9], and Darwish et.al, 2006 [18]. Comparing with Darwish et.al, 2006 [18], the results of MEE-FF program indicates that the cumulative distillate flow rate *ΣD*, and the brine flow rate are smaller by 2.8% and 1.6%, respectively, as shown in Figure 9 and Figure 10. These differences occur due to computing the values of the latent heat of the heating steam, and the produced vapor in the program. While, the values of these parameters is set to a fixed number in Darwish et.al, 2006 [18]. This change in the latent heat strongly affects the heat transfer area of the effects and the feed pre-heaters where they decrease by 16.9% and 10.9%, respectively. These relationships are shown from Figure 9 to Figure 11.

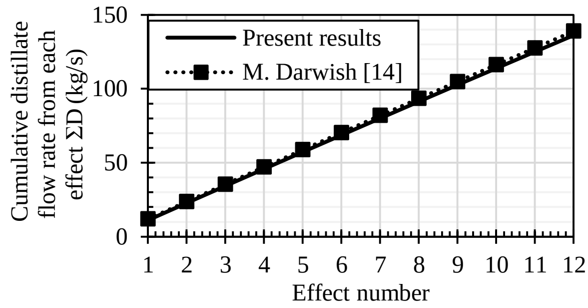

Figure 9: Comparison in terms of the cumulative distillate flow rate for the MEE-FF method

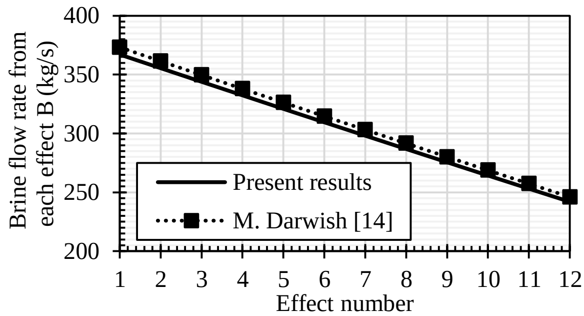

Figure 10: comparison in terms of the brine flow rate for the MEE-FF method

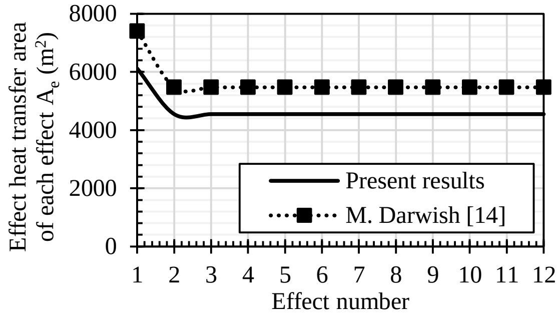

Figure 11: Comparison in terms of the effect heat transfer area for the MEE-FF method

For the MSF-OT method, strong uniformity was found in the values of the top brine and feed temperatures. Additionally, there exists an unremarkable difference in the values of the summation distillate and brine flow rates and brine salinity. The differences do not exceed 0.4% where the same results used in Dessouky and Ettouney [9] were adopted. However, one can explain this small difference by pointing out that Dessouky and Ettouney [9] used charts for solving the problem. Figure 12 and Figure 13 show that the results obtained by the VB programs and Dessouky and Ettouney [9] are identical. After this validation, the developed VB program is now used to find the optimum desalination method for an existing ship.

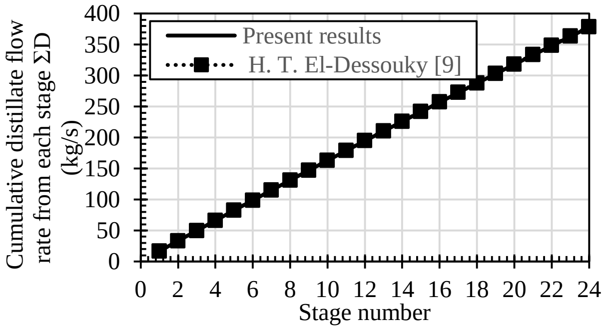

Figure 12: Comparison in terms of the cumulative distillate flow rate for the MSF-OT method

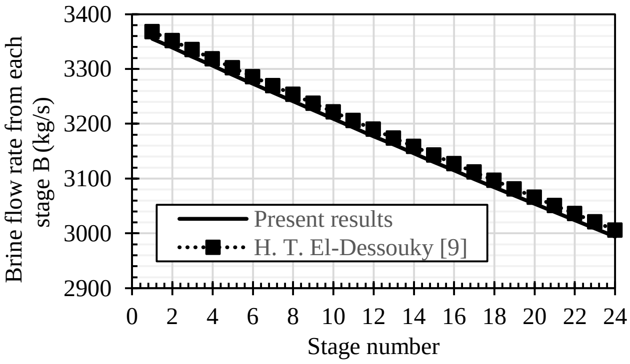

Figure 13: Comparison in terms of the brine flow rate for the MSF-OT method

7. PROPOSAL EXAMINATION

The developed VB program is now used to find the specification of the desalination plant which fulfills the ship requirements from fresh water. The calculations are done for the three methods; MEE-FF, MSF-OT and MSF-BC. Eventually, the optimum specification is recommended, which has the minimum area together with giving the required amount of fresh water.

The proposed method is examined on The *MS Freedom of the Seas* ship. Table 1 shows the characteristics of the ship [19]. The ship sails at eastern and western Caribbean routes and she is powered by six Wärtsilä 46 V12 diesels each rated at 12.6 MW driving electric generators at 514 rpm [20]. The ship is fitted with three ABB Azipod podded electric propulsion units, two of them azimuthing and one central fixed unit. There are also four bow thrusters for maneuvering.

According to the technical report released by the Royal Caribbean Cruises, in 2012, the fresh water consumption is an average of 0.208 m3 (54 gallons) per person per day [21]. This implies that it is required to select a desalination unit with 14 kg/s capacity.

Table 1: Specifications of the MS Freedom of the Seas [19]

Item	Value
Overall Length	338.77 m
Perpendicular Length	303.21 m
Breadth	38.60 m
Draft	8.5 m
Height	63.7 m
Gross Tonnage	154,407 t
Cruising Speed	21.6 kt
Passengers	4,375
Crew	1,360
Cabins	1,817
Power	6 × Wa¨rtsil¨a 12V46 (6 × 12,600 kW)

The program starts with heat balance for the main engine. Then, the waste heat energy from the exhaust and cooling water are used to operate the desalination plant. The effect of the following variables on the total area required for the desalination plant selection and the distillate flow rate is to be studied.

The three desalination methods are now analyzed to select the optimum desalination method which meets the required distillate flow rate M_d with minimum heat transfer area A_T to reduce the plant size and cost.

7.1. MSF-BC Parameters Optimization

Table 2 shows the input data of MSF-BC program. The effect of controllable variable is now studied. The column *input value* corresponds to the initial values used while the column *Range of change* corresponds to the range by which the variables are controlled to study its effect.

Table 2: The input data of MSF-BC program

	Variable	Input value	Range of change
Controllable variables	No of effects	NA	3-8
	Steam temperature T_s	NA	110-200˚C
	Exhaust temperature after boiler T_{exh}	250 ˚C	250 – 300 ˚C
	Intake flow rate M_{int}	90 kg/s	90 – 115 kg/s
	Top brine temperature T_0	65 ˚C	90 - 120 ˚C
	Brine Salinity X_b	45,000 ppm	45,000 - 80,000 ppm
	No of rejection section stages N_j	3	3-7
	Feed water Salinity X_f	36,000 ppm	NA
	Exhaust temperature before boiler	355 ˚C	NA
	Sea water temperature before cooler	24 ˚C	NA

7.1.1. Effect of steam temperature T_s

Both flow rate M_d and heat transfer area A_T take the same reaction with increasing the temperature T_s from 110˚C to 200˚C. The flow rate M_d is decreased by about 19% and the area A_T is reduced by 38% as shown in Figure 14 and Figure 15.

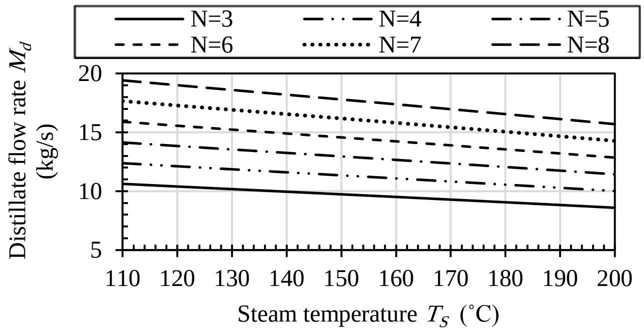

Figure 14: The effect of steam temperature on the distillate flow rate (MSF-BC)

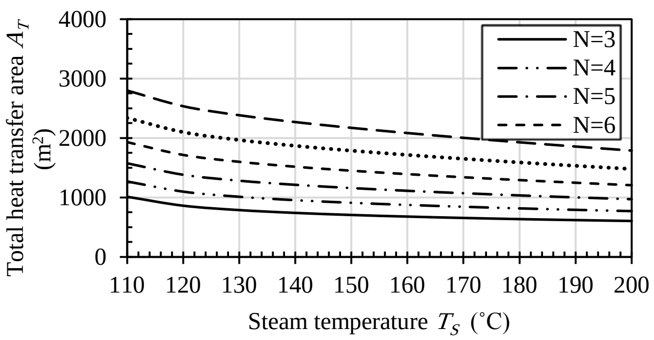

Figure 15: The effect of steam temperature on the total heat transfer area (MSF-BC)

7.1.2. Effect of exhaust temperature after boiler

The diesel engine exhaust gases vary with the speed and load. Specifically, the high loads and high speeds result in the highest temperatures. Generally, temperatures of 500-700°C are produced in the exhaust gases from diesel-cycle engines at 100% load to 200-300°C with no load [22]. Consequently, increasing the temperature T_{exh} from 250˚C to 300˚C, the distillate flow rate M_d is decreased to half value and the total heat transfer area A_T is decreased by 46%, as shown in Figure 16 and Figure 17.

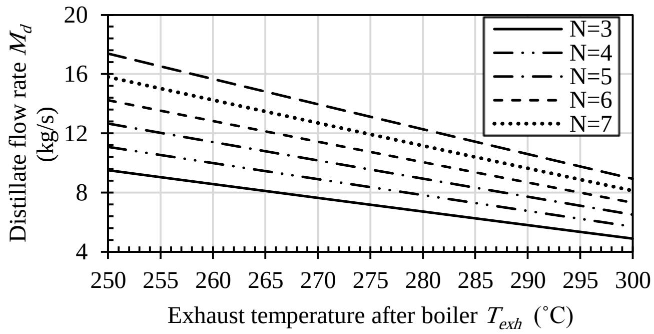

Figure 16: Effect of Exhaust temperature after boiler on the distillate flow rate (MSF-BC)

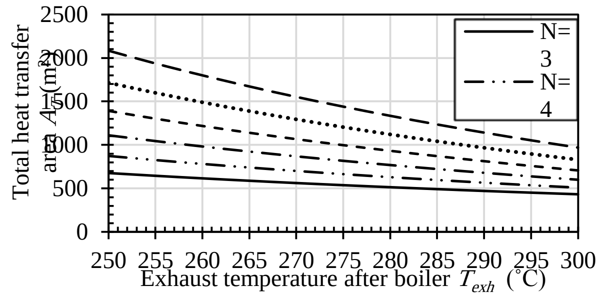

Figure 17: Effect of Exhaust temperature after boiler on the total heat transfer area (MSF-BC)

7.1.3. Effect of intake water flow rate

As mentioned before, the capacity of the sea-water pumps usually flows with a rate in the range between 1.2 and 1.5 relative to the fresh water flow [20]. Consequently, the intake flow rate M_{int} is studied between the range 90 kg/s to 115 kg/s. The intake flow rate M_{int} has minor effect on the distillate flow rate M_d; it reduced with less than 1%. The total heat transfer area A_T decreased by 4.7% when increasing the intake flow rate. Consequently, the intake flow rate for the next study will be set to 115 kg/s.

7.1.4. Effect of top brine temperature

The MSF plants usually operate at the top brine temperatures of 90-120˚C [23]. Therefore, the escalation of top brine temperature T_0 will be at this range. The distillate flow rate M_d is almost constant. Hence, the selection of the temperature T_0 value depends on the trend of heat transfer areas. The total heat transfer area A_T is inversely proportion with the temperature T_0, as shown in Figure 18. This is a result of increasing the brine heater heat transfer area A_b by 49% and decreasing in condensers heat transfer area A_c by 33% with the variation of top brine temperature. Therefore, the value of 120˚C of the temperature T_0 is suitable for the desalination plant selection.

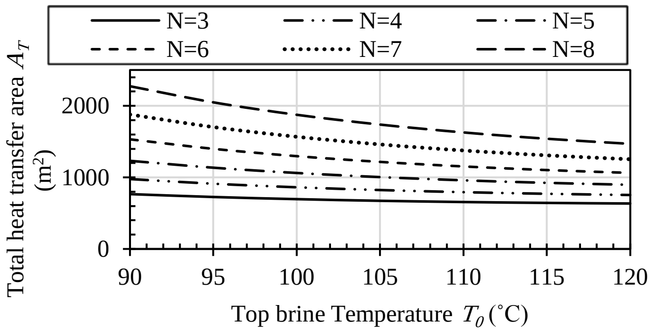

Figure 18: Effect of top brine temperature on the total heat transfer area (MSF-BC)

7.1.5. Effect of brine salinity

The brine salinity does not have any effect on the distillate flow rate M_d and it increases the total heat transfer area A_T by only 4%. Therefore, the brine salinity of 45,000 ppm is the optimal value for the area A_T and the distillate flow rate M_d.

7.1.6. Effect of Number of Rejection Section Condensers

In this section, the number of rejection section condensers N_j is changed from three to seven condensers. This analysis is performed to study the effect of number N_j on the flow rate M_d and the different heat transfer areas. Figure 19 and Figure 20 show that the number N_j has a negative impact on the distillate flow rate M_d and the total heat transfer area A_T. The distillate flow rate M_d is decreased by 36% and 33% for the total heat transfer area A_T. Hence, we adopt seven condensers in the rejection section. According to these results, it is found that three condensers in the rejection section and six condensers in recovery section are enough for producing the fresh water demand for the *"MS Freedom of the Seas"* ship.

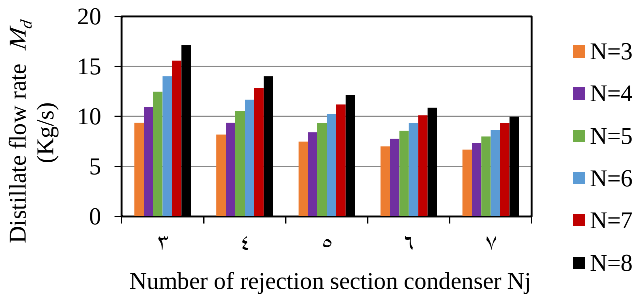

Figure 19: The effect of rejection section condensers number on the distillate flow rate (MSF-BC)

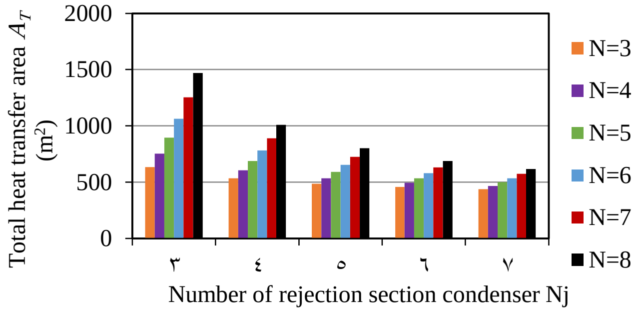

Figure 20: The effect of rejection section condensers number on the total heat transfer area (MSF-BC)

Table 3 summaries the optimum values of the controllable variables which meet the water demand of the ship with the minimum heat transfer area.

Table 3: Optimum desalination plant selection of MSF-BC method

Variable	Optimum value
No of stages	6
No of rejection section stages	3
Steam temperature T_s	160 ˚C
Exhaust temperature after boiler T_{exh}	250 ˚C
Intake flow rate M_{int}	115 kg/s
Top brine temperature T_0	120 ˚C
Brine Salinity X_b	45,000 ppm
Distillate flow rate M_d	14 kg/s
Total heat transfer area A_T	1060 m^2

7.2. MEE-FF Parameters Optimization

The previous steps have been applied to the MEE-FF method. The effect of changing the same variables showed almost similar trends as the MSF-BC method. Increasing the number of effect increases the total area and the distillate flow rate. The results are summarized as follows:

- Increasing the steam temperature T_s lead to 11% reduction in the total area. The steam temperature T_s has no effect on distillate flow rate M_d.
- The effect of increasing exhaust temperature T_{exh} is same as increasing steam temperature. The total heat transfer area A_T decreased with about 23%. The temperature T_{exh} is more effective than the steam temperature on the heat transfer area.
- Increasing intake water flow rate M_{int} from 90 kg/s to 115 kg/s led to increasing the distillate flow rate M_d to 42% and increasing the total heat transfer area A_T with about 28%.
- Increasing top brine temperature T_1 has a positive effect. The distillate flow rate increases about 6%

and in the same time, the total area A_T decreased about 79%.

- The brine salinity X_b does not have any effect on distillate flow rate. But the total heat transfer area decreases with about 8% with increasing the brine salinity from 45,000 ppm to 80,000 ppm.

Table 4: Optimum desalination plant selection of MEE-FF method

Variable	Optimum value
No of effects	5
Steam temperature T_s	200 ˚C
Exhaust temperature after boiler T_{exh}	300 ˚C
Intake flow rate M_{int}	115 kg/s
Top brine temperature T_1	90 ˚C
Last effect brine temperature T_n	44 ˚C
Condenser exit temperature T_{fn}	40 ˚C
Brine Salinity X_b	80,000 ppm
Distillate flow rate M_d	14 kg/s
Total heat transfer area A_T	1590 m^2

7.3. MSF-OT Parameters Optimization

Similar to the previous two methods, this section studies the effect of the controllable variables on the distillate flow rate and the heat transfer area. Table 5 shows the input data of MSF-OT program.

Table 5: The input data of MSF-OT program

	Variable	Input Value	Range of change
Controllable variables	No of effects	NA	3-8
	Steam temperature T_s	NA	110-200˚C
	Exhaust temperature after boiler T_{exh}	250 ˚C	250 – 300 ˚C
	Intake flow rate M_{int}	90 kg/s	90 – 115 kg/s
	Top brine temperature T_0	100 ˚C	90-110 ˚C
Feed water Salinity X_f		36,000 ppm	NA
Exhaust temperature before boiler		355 ˚C	NA
Sea water temperature before cooler		24 ˚C	NA

When increasing the steam temperature, the distillate flow rate M_d does not change with the number of stages and it is increased by 15%, Figure 21. While, A_T had minimum value at steam temperature 150˚C and starts to increase after this point, Figure 22. Consequently, this steam temperature was fixed at this value during this study.

The temperature T_{exh} affects positively on the flow rate M_d. The flow rate M_d is increased in the range between 250˚C and 300˚C by around 33% and reach to 8.7 kg/s, as shown in Figure 23. However, the exhaust temperature of 280˚C is chosen to be the optimum temperature due to the increasing total heat transfer area.

Increasing the feed water flow rate M_f from 90 to 115 kg/s will correspondingly increase the distillate flow rate by 41%. The total heat transfer area is increased by about 108%. Although the total heat transfer area is increased more than two times, it is assumed in this section that the flow rate M_f is set to 115 kg/s. The distillate flow rate at this stage is 11 kg/s which still less than required demand.

The flow rate M_d is increased by 64% with increasing the top brine temperature from 90˚C to 120˚C. The total heat transfer area is increased by 133%, as shown in Figure 24. Consequently, the top brine temperature value of 115˚C is the optimum value of this case in MSF-OT method because the required demand is achieved.

The number of stages does not make any considerable impact on the flow rate M_d and simultaneously has an inverse proportion with the heat transfer area A_T. Therefore, eight stages are suitable for this method. Table 6 summaries the optimum values of the controllable variables from the previous analysis.

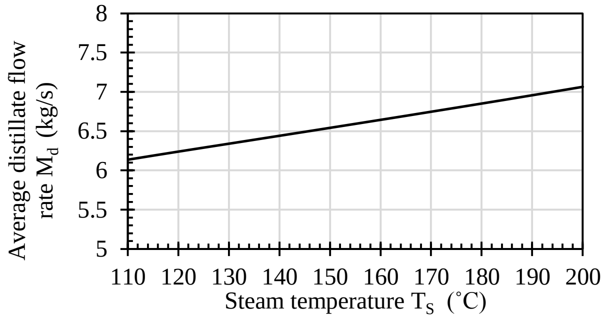

Figure 21: Effect of steam temperature on distillate flow rate MSF-OT

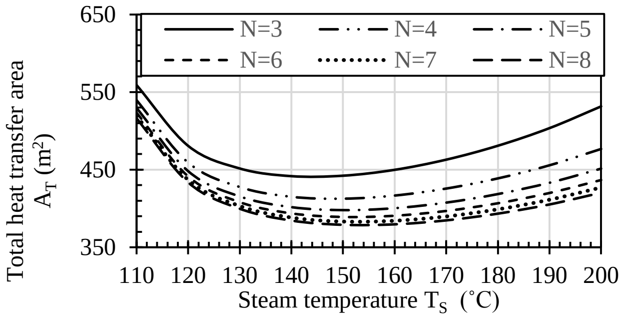

Figure 22: Effect of steam temperature on total heat transfer area MSF-OT

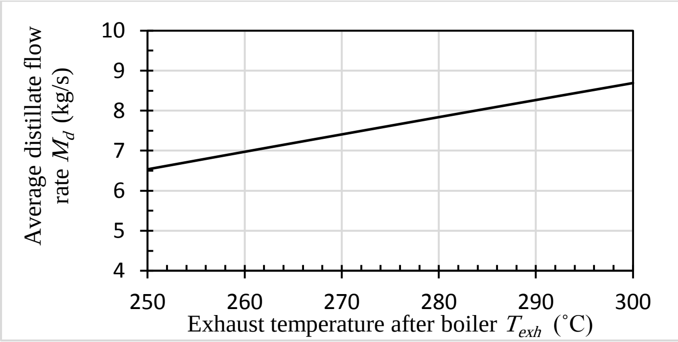

Figure 23: Effect of exhaust temperature on distillate flow rate MSF-OT

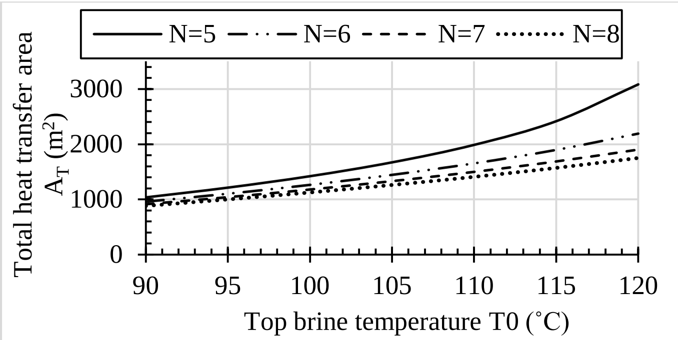

Figure 24: Effect of Top brine temperature on total area MSF-OT

Table 6: Optimum desalination plant selection of MSF-OT method

Variable	Optimum value
No of stages	8
Steam temperature T_s	150 ˚C
Exhaust temperature after boiler T_{exh}	280 ˚C
Intake flow rate M_{int}	115 kg/s
Top brine temperature T_0	115 ˚C
Distillate flow rate M_d	14 kg/s
Total heat transfer area A_T	1570 m^2

7.4. Optimum Selection

From Table 3, Table 4, and Table 6 one can conclude that the MAF-BC desalination method gives the minimum heat transfer area this area is 30% less than of the other two methods. Figure 25 summarizes the results. It is also seen from the results that the MSF-BC method gives the lowest exhaust temperature, which is better for the environment. Table 3 summarizes the obtained results.

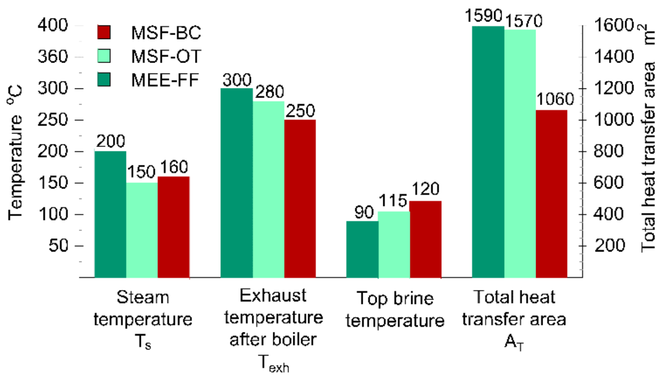

Figure 25: Results summary

8. ECONOMICAL, TECHNICAL, AND ENVIRONMENTAL STUDY

In this section, a study of the gains achieved by seawater desalination using the waste heat recovery on passenger ships is performed. The gains are interpreted in terms of the economical costs, environmental implications, and technical aspects.

8.1. Economical Comparison

In general, the most important cost indicator for desalination plants is the cost of unit volume of water, namely unit cost $/$m^3$. The unit cost reflects the total costs and liabilities of a desalination plant divided by the volume of produced water. In fact, the unit cost depends on the desalination technology and on the production capacity. However, several local factors like the geographic location, the energy costs, the energy availability, or the input water quality have also a considerable impact on the unit cost [24]. Equation 10 is used to calculate an approximate capital cost of a new plant (of a known capacity) based on the known capital cost of an existing plant.

$$\left(\frac{Capital\ cost_{Plant1}}{Capital\ cost_{plant2}}\right) = \left(\frac{Plant\ capacity_{plant1}}{Plant\ capacity_{plant2}}\right)^m \qquad (10)$$

The power law model has been fitted to develop the relationship between capital cost and plant capacity, as shown in Equation 4. Simple linear regression using least squares was performed to determine the power law exponent, m, for various desalination technologies.

$$\ln(Capital\ cpst) = m \cdot \ln(Capacity) + constant \qquad (11)$$

According to [25], the exponent *m* is set to 0.7 and the constant value is set to 4.86 for the MSF desalination method. The capital cost of the optimal desalination unit can be estimated for a capacity of 1211 m^3/day. After computing the capital cost, Equation 12 is used to determine the Unit Product Cost *UPC* which is defined as the sum of the depreciated capital cost and the operating costs. When unspecified, the plant life has been assumed to be 20 years with plant availability of 90%.

$$UPC = \frac{\left(\text{Capital cost}/_{\text{plant life}}\right) + \text{Annual operating cost}}{\text{Plant capacity} \cdot \text{Plant availability}} \quad (12)$$

Equations 10 and 11 produce two extreme values of the unit's capital cost. Specifically, Equation 3 estimates the maximum capital cost by $ 5,435,000. In this case, the unit product cost is equal 1.3 \$/m^3. About 2.7 \$/m^3 can be saved from the average fresh water price in some ports. Consequently, the capital cost can be returned after 6 years. Whereas, Equation 4 estimates the minimum capital cost by $ 1,155,000 where this value can be returned after one year. Accordingly, the capital return can be achieved in a range between one and five years due to 0.26 $/ton for unit product cost. In both cases, the unit's lifetime reaches 20 years. Therefore, a significant profit is gained even with considering the worst case. Using waste heat energy to generate fresh water reduces the *UPC* by 40%. This value is estimated comparing with the capital cost of MSF unit in Table 7 [25].

Table 7: Capital cost of various desalination processes

Process	Plant capacity m3/day	Unit-capital cost $/(m3/day)
MEE	37,850	1860
MSF	37,850	1598

8.2. Environmental Comparison

In case of using dedicated fuels for the desalination unit, Equation 13 can be used to estimate the fuel consumption where m_f is fuel consumption in kg/s, *CV* is the calorific value of the fuel in kj/kg, m_{steam} is the steam flow rate in kg/s, C_p is the specific heat of the water in kj/kg and ΔT is the difference temperature between the output steam and the inlet water. Assuming that the steam boiler is operated with Heavy Diesel Oil HDO, thus the fuel consumption is calculated to be 253.5 kg/hr. This fuel combustion produces an amount of exhaust emission that can be saved when using the waste heat energy. Legislation of exhaust emission levels has focused on carbon monoxide CO, hydrocarbons HC, nitrogen oxides NOx, and particulate matter PM [26]. According to the fuel consumption, the amount emission which was supposed to be emitted is shown in Table 8. Preventing these emissions obviously has a positive effect on the environment.

$$m_{fuel} \cdot CV = m_{steam} \cdot C_P \cdot \Delta T \quad (13)$$

Table 8: Emission saving due to using WHR

Emission factor	Emission saving (kg/hr)
CO_2	800
NOx	16
CO	0.7
HC	0.7
Particulates	0.6
SO_2	5.3

8.3. Technical Comparison

Eventually, the efficiency of the main diesel engine is studied before and after applying the proposed system; η_1 and η_2. Equation 14 states that the efficiency of the engine is increased by adding the desalination gain power. Specifically, the thermal efficiency is increased by 2% and η_2 reaches 50.5%.

$$\eta_2 = \frac{\text{Engine power} + \text{desalination gain power}}{\text{Total input}} \quad (14)$$

9. CONCLUSION

In this paper, the fresh water generation on board of a passenger ship has been studied. Water desalination is performed using the waste heat energy produced from the main engine exhaust and the cooling water was proposed.

Three methods have been used for comparison; Forward Feed Multiple Effect Evaporation MEE-FF, One Through Multi-Stage Flash MSF-OT, and Brine Circulation Multi-Stage Flash MSF-BC. Waste heat from the engine is used for the operation of the desalination plant. A VB program has been developed to calculate the desalination plant characteristics. The program was validated and then it was used to calculate the heat transfer.

The results showed that the MSF-BC gives the minimum heat transfer area for the required distillate flow rate; 30% less than the other two methods. It is found that the steam temperature T_s, the exhaust temperature T_{exh}, and the top brine temperature T_0 are inversely proportion with the total heat transfer area A_T. Consequently, these variables have been assigned their maximum values so that the heat transfer area is minimized. Conversely, the brine salinity X_b has a direct proportion with the area A_T. Therefore, increasing the salinity normally results in increasing the area A_T.

Applying the optimum selection of the proposed salt water desalination on the case study saved 2.7 $/ m^3 as a minimum comparing to the average cost of fresh water in ports. This saving can cover the plant capital cost in six years at most. However, in the best case, the capital cost was $ 1,115,000 and the unit product cost UPC was 0.26 $/m^3.

The plant reduced the emissions by about five thousand tons of CO_2, 100 tons of NO_x and 35 tons of SO_2 per year.

Finally, the thermal efficiency is of the engine is increased by 2% (from 48.5% to 50.5%).

So far, the paper proved the efficiency of exhaust gas and cooling water for deriving the water desalination units. As an outlook, the work in this paper can be extended through investigating the efficiency of other waste heat sources, such as the air cooler and the lubricating oil. Additionally, the paper focused on thermal water desalination methods. As an extension is the application of the reverse osmosis method or hybrid methods for water desalination on ships. This method is broadly superior over other methods in the literature thanks to its simplicity while being adopting.

NOMENCLATURE

A_b	Heat transfer area of the brine heater	m^2
A_j	Heat transfer area of the rejection section stage	m^2
A_r	Heat transfer area of the recovery section stage	m^2
B	Brine flow rate in special effect	kg/s
C_{air}	Specific heat of air	kj/kg.K
C_{lub}	Specific heat of lubricating oil	kj/kg.K
C_p	Specific heat	kj/kg.K
C_w	Specific heat of water	kj/kg.K
D	Distillate flow rate in special effect	kg/s
FC	Fuel consumption	kg/s
H_s	Specific enthalpy of the steam	kj/kg
H_w	Specific enthalpy of the water	kj/kg
LCV	Low calorific value	kj/kg
M_{cw}	Cooling water flow rate	kg/s
M_d	Distillate flow rate	kg/s
m_{exh}	Mass flow rate of exhaust gas	kg/s
M_f	Feed flow rate	kg/s
M_r	Recycle stream flow rate	Kg/s
M_{FW}	Flow rate of fresh water	kg/s
m_{lub}	Mass flow rate of lubricating oil	kg/s
M_s	Steam flow rate	kg/s
M_{FW}	Flow rate of fresh water	kg/s
M_{SW}	Flow rate of salt water	kg/s
m_w	Mass flow rate of cooling water	kg/s
M_w	Water flow rate	kg/s
P_b	Brake power	kW
Q_{cooler}	Thermal load of the cooler	kW
Q_{exh}	Heat loss by exhaust	kW
Q_{lub}	Heat loss by lubricating oil	kW
Q_{rad}	Heat loss by radiation	kW
Q_s	Supplied fuel energy	kW
Q_w	Heat loss by cooling water	kW
T_0	Top brine Temperature	˚C
T_b	Brine water temperature	˚C
T_{cw}	Cooling water temperature	K
$T_{exh(inlet)}$	Exhaust temperature before boiler after turbocharger	K
$T_{exh(outlet)}$	Exhaust temperature after boiler going to the air	˚C
T_f	Feed water temperature	˚C
$T_{in(FW)}$	Inlet fresh water temperature to the cooler	K
$T_{in(SW)}$	Inlet salt water temperature to the cooler	K
$T_{out(FW)}$	Outlet fresh water temperature from the cooler	K
$T_{out(SW)}$	Outlet salt water temperature from the cooler	K
X_b	Brine salinity	ppm
X_f	Feed water salinity	ppm
X_r	Recycle stream salinity	ppm

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