ERJ
Engineering Research Journal
Faculty of Engineering
Minoufiya University

PARAMETRIC STUDY OF MARINE STEAM TURBINE PLANTS

M.Mosleh 1, Adel Banawan 2, Wael M. El-Maghlany 3, Ahmed G. Mohamed 4
1,2 Faculty of Engineering , Alexandria university
3 Faculty of Engineering in Ismailia, Suez Canal University
4 Arab academy for since, technology and maritime transport
Emails: momosleh@hotmail.com; abanawan@yahoo.com;ans_shehab@yahoo.com & engahmedgomaa.12 @Gmail.com

ABSTRACT

The steam turbine power plant has until recently been the first choice for very large power main propulsion units. Its advantages of little or no vibration, low weight, minimal space requirements and low maintenance costs are considerable. Furthermore a turbine can be provided for any power rating likely to be required for marine propulsion [1].

This research deals with the parametric study of marine steam power plants. It studies the effect of the ambient temperature which is considered as the cooling water inlet to the condenser (from 24oC to 35 oC) also the boiler pressure (from 30 bar to 60 bar) on the performance of the steam power plants in both fully loaded ship and ballast conditions. The performance of the power plant concerned with the thermal efficiency, the fuel consumption, the steam mass flow rate, and the mass flow rate of cooling water inlet to condenser. The procedures have been carried out for METHANE KAREELIN as LNG carrier with steam power plant rated for 29 MW.

The results showed that, the sailing in cold weather is more favorable for high thermal efficiency, less fuel and steam consumption, also less mass flow of the cooling water inlet to the condenser. Also, it is better to work under high steam pressure for high thermal efficiency, less fuel and steam consumption, and less mass flow of the cooling water inlet to the condenser. But, this pressure is limited to the metallurgy of metals that are bear this pressure as well as not based on the designer. Finally, carrying more cargo increases both the fuel consumption and steam consumption, to keep the ship speed constant as well as possible.

في هذا البحث تمت دراسة بارامترية لمحطات التربينات البخارية البحرية، وقد تم دراسة كل من تأثير درجة الحرارة المحيطة التي تعتبر درجة حرارة مياه تبريد المكثف، وكذلك تأثير ضغط الغلاية على أداء محطات التوليد البخارية مثل الكفاءة الحرارية، استهلاك الوقود، معدل الأستهلاك النوعى للبخار، ومعدل تدفق مياه تبريد المكثف، وذلك في كل من حالتي السفينة بكامل حمولتها وفى حالة الصابورة (بدون حمولة)، وقد تمت هذه الدراسة على احدى ناقلات الغاز الطبيعي المسال التى تعمل بنظام دفع بخاري بقدرة ٢٩ ميغاوات. تم تغير درجة حرارة مياه تبريد المكثيف من ٢٤ درجة إلى ٣٥ درجة وضغط الغلاية من ٣٠ بارإلى ٦٠ بار وكانت النتائج كالتالى عندما تنخفض درجة حرارة مياه تبريد المكثف ينخفض ضغط المكثف، والعكس صحيح. فهناك علاقة عكسية بين الكفاءة الحرارية ودرجة حرارة مياه تبريد المكثف،كما أن هناك علاقة طردية بين الكفاءة الحرارية وضغط الغلاية. وكذلك هناك علاقة طردية بين معدل الأستهلاك النوعى للوقود و درجة حرارة مياه تبريد المكثف،و له علاقة عكسية مع ضغط الغلاية أيضا. وهناك علاقة طردية بين استهلاك الوقود و حالة التحميل أثناء الإبحار. كما أن معدل الأستهلاك النوعى للبخار له علاقة طردية مع درجة حرارة مياه تبريد المكثف، و له علاقة عكسية أيضا مع ضغط الغلاية. وهناك علاقة طردية بين استهلاك البخار و حالة التحميل أثناء الإبحار.وأخيرا، فإن معدل التدفق لمياه التبريد إلى المكثف لديه علاقة طردية مع درجة حرارة مياه تبريد المكثف، و له علاقة عكسية مع ضغط الغلاية.

Keywords: Steam turbine; LNG carrier; propulsion system; Boiler pressure; Condenser pressure; Specific fuel consumption

1. INTRODUCTION

History of marine shipments of LNG began in the late 1950s. The first commercial trades started in the early 1960s and by the 1970s international trades had been established with the subsequent requirement for LNG carriers. The LNG trade has been fairly stable in this period, characterized by long term supply contracts. [2] Reports the world fleet of LNG ships as 146, and about half of these are 20 years old. Steam turbine plant has so many records as marine propulsion system and it is sufficiently attractive even at present because of maintenance - free and low NOx Emission for the environment. [3]

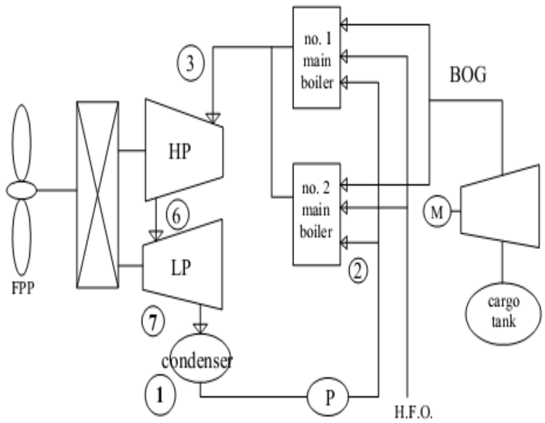
New demands in LNG shipping industry and increasing concern about environmental protection are supposed to lead the trend toward an alternative propulsion application for LNG carriers other than steam turbine, which had dominated propulsion power generators onboard the vessels for decades. The steam turbine has surely performed high reliability for a long time since it was installed on LNG carriers, except for just a few problems with high-speed reduction gearing part. It has been considered as well-proven, reliable and infrequent low-cost maintenance machinery. However the drawback is mainly in its efficiency [4]. The maximum total plant efficiency of the steam propulsion system is approximately 30% at full load and the efficiency becomes lower as the turbine load goes down. Until a few years ago, steam turbines were used for almost all LNG carriers for the propulsion plant because of their operation profitability, reliability, operability, maintainability and initial investment. Another reason is that the steam turbine plant could safely treat boil-off gas (BOG) generated from the cargo tank and could also use heavy oil as fuel. In order to enhance plant efficiency of steam turbine propulsion system, the newly developed concept has been introduced in the market, so called as Ultra Steam Turbine (UST) it used instead of Conventional Steam Turbine (CST). In (CST) the steam flow from boiler to high pressure steam turbine, low pressure turbine final to condenser and the steam flow in (UST) is from boiler to high pressure turbine to reheater, intermediate pressure turbine, and low pressure turbine final to condenser [5].

As the capacity of LNG carriers are getting bigger and they require more power, it seems that the application of steam turbine shall be carefully considered in case of very large LNG carriers due to the limitation in its output power and engine room space [6].

The purpose of heat balance is to set forth data and practices for marine steam power plant heat balance calculations and diagrams. This research is specifically applicable to oil fired but with suitable modifications it can be used for other service rating, other type of fuels, prime movers and auxiliary equipment. A heat balance is a necessary tool on the design of a marine power plant. Once the shaft horsepower, steam conditions and basic cycle have been established, a heat balance calculation can be made. From it, the various steam, exhaust, feed and condensate flows can be determined and used in selecting suitable machinery and equipment. Heat balance may also be prepared to determine optimum steam conditions and cycle design or to analyze the performance of a power plant on trials or in service [7].

2. PROPULSION SYSTEM CONFIGURATION

The propulsion system of METHANE KARE-ELIN



is steam plant shown in Figure (1), and figure (2) show the T-S diagram of steam propulsion system.

Figure (1) Propulsion System Configuration

The propulsion system consists of:

- boiler to generate steam to the cycle the boilers which used is KAWASAKI UME 67/51 with steam condition at superheater outlet of 62 Kg/cm2 and 515 ^{o}C, feed water temperature at inlet economizer 138 ^{o}C, maximum evaporation 67000 Kg/h and normal evaporation 51000 Kg/h [8].
- Steam turbine to give the useful power to the propeller, the steam turbine which used is Kawasaki UA-400 its properties are output range 26500 -29400 kW with total weight 305 ton [9].
- Condenser vacuum 722mmHG at 24 ^{o}C sea water gearbox to reduce the turbine speed to 90 rpm with fixed pitch propeller (FPP) [10].

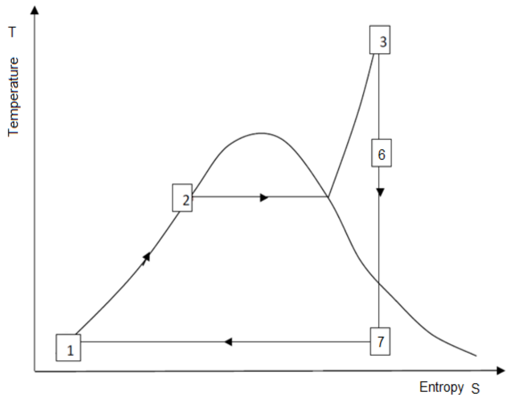


Figure (2) T-S Diagram for Steam Propulsion System

3. Case Study Data Reduction

Before to study the effect of changing parameters (the boiler pressure and the cooling water temperature inlet to condenser) on the performance of steam propulsion system (studies parameters). It is required to calculate the ship powering at every case of loading during sailing, first calculate the ship resistance then the powering calculation, from equations (1) to (9) [11, 12 1ND 13].

$R_T = R_V + R_W + R_A + R_{APP} + R_{SW}$ (1)

$Rv = \frac{1}{2} \rho \, S \, V^2 \, C_v$ (2)

$\frac{Rw}{W} = C_1 C_2 C_3 \, e^{(m1 \, Fn^d + m2 \cos(\lambda Fn^{-2}))}$ (3)

$R_A = \frac{1}{2} \rho_{air} \, A_s \, V^2 \, C_A$ (4)

$EHP = R_T \, V$ (5)

$\eta_H = \frac{EHP}{THP}$ (6)

$\eta_P = \frac{THP}{DHP}$ (7)

$\eta_{shaft} = \frac{DHP}{SHP}$ (8)

$\eta_{gear} = \frac{SHP}{BHP}$ (9)

By determined the break horsepower (BHP), it is the designed power.

The condensate temperature (Ts) and hence the condenser pressure is a function of condenser heat load (Q) and circulating water inlet temperature (Tcwi). The condensate temperature has a linear relationship with the heat load and also cooling water inlet temperature. In practice, the condenser pressure is frequently used instead of condenser temperature. In operation, the condenser temperature (or pressure) cannot be such that the condenser terminal difference is less than 3^{o}C. If this should occur in calculations, the Heat Exchange Institute recommends that the initial temperature difference equation Ts = Tcwo + 3 be used for an estimate of condenser temperature, as shown in figure (3)

$$T_s = \frac{\frac{Q}{m_{cw} Cp_w} + T_{cwi}\left(1 - e^{-\left(\frac{UA}{m_{cw} Cp_w}\right)}\right)}{\left(1 - e^{-\left(\frac{UA}{m_{cw} Cp_w}\right)}\right)} \quad (10)$$

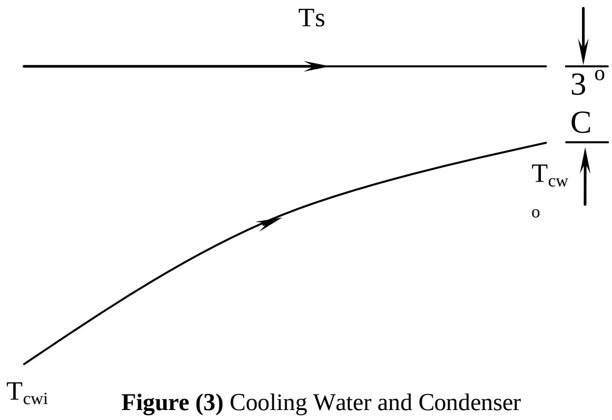


Figure (3) Cooling Water and Condenser Saturation Temperatures

After calculation of ship powering at every case of loading during sailing and determination of the condenser pressure at every cooling water temperature inlet into condenser, now it could be to study the affected parameters combined with changing of cooling water temperture inlet to condenser and boiler pressure, from equations (11) to (15) [14].

$\eta_{th} = 1 - (h_1 - h_7)/(h_3 - h_2)$ (11)

$\dot{m}_s = WT/(h_3 - h_7)$ (12)

$S.S.C = 3600/(h_3 - h_7)$ (13)

$\dot{m}_f = \frac{\dot{m}s(h3 - h2)}{\eta_b \, CV}$ (14)

$S.F.C = \dot{m}f/power$ (15)

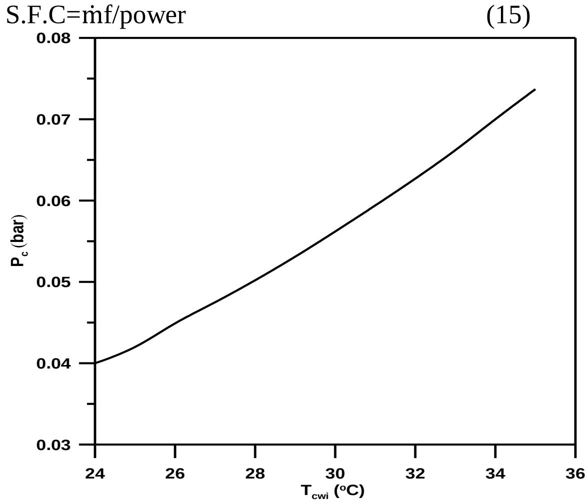


Figure (4) Relationship between Cooling Water Inlet Temperature and Condenser Pressure.

3.1 Result and Discussion

3.1.1 Ship Resistance and Power Calculations

Parameter	Equation	Values	
		Full load condition	Ballast condition
R_v	2	1326 kN	1207 kN
R_w	3	218.4 kN	164.3kN
R_A	4	0.1325 kN	0.22 kN
R_T	1	1698 kN	1372 kN
EHP	5	17548 kW	14174 kW
THP	6	17906 kW	14463 kW
DHP	7	28198 kW	25599 kW
SHP	8	28773 kW	26121 kW
BHP	9	***29361 kW***	***26654 kW***

3.1.2 Condenser and Boiler Pressures Effect

As mentioned above, there is a relationship between the condenser pressure as main study parameter and cooling water temperature inlet to condenser as show in Figure 4. As the inlet temperature to condenser increase from 24°C to 35°C the condenser pressure increase from 0.04 bar to 0.737 bar.

3.1.2 .1 Thermal Efficiency

The overall thermal efficiency of steam cycle is affected by many parameters as boiler pressure, condenser pressure and steam temperature. In fully load condition as shown in Figure 5 and in ballast condition in Figure 11 it was found that the thermal efficiency decreased when the condenser pressure increased at constant boiler pressure and steam temperature. For example at boiler pressure 30 bar and steam temperature 515℃ the thermal efficiency decreased from 38.5% to 36.4% when the condenser pressure increased from 0.04 bar to 0.075 bar and also at all boiler pressure the thermal efficiency decreased when the condenser pressure increased. This condenser pressure based on change the cooling water temperature inlet to condenser. On the other side it was found that the high efficiency was corresponding to the high boiler pressure and lower condenser pressure as the steam temperature was fixed at 515℃ . In Figure 5 or in Figure 11 the best efficiency at high boiler pressure of 60 bar and low condenser pressure of 0.04 bar is 38.5%. By comparison between the two conditions of sailing found that the thermal efficiency was not change in the two conditions because the thermal efficiency depends on the boiler pressure and condenser pressure did not depend on the ship cargo carrying (displacement).

3.1.2 .2 Fuel Consumption

If the fuel consumption at any plant is decreased (the main object to economy consideration), it led to save the money paid. It is the best economical solution for any owner to decrease the expenses. In fully load condition in Figure 6 it was found that the fuel consumption increased when the condenser pressure increased at constant boiler pressure and steam temperature of 515℃ the fuel consumption increased from 2.05 Kg/s to 2.15 Kg/s when the condenser pressure increased from 0.04 bar to 0.075 bar at all boiler pressures the fuel consumption increased when the condenser pressure increased led to increase the expenses from 26819 $/day to 28127 $/day. It was also found that the lower fuel consumption at high boiler pressure and lower condenser pressure and the higher fuel consumption at low boiler pressure and high condenser pressure with steam temperature fixed at 515 ℃ . In Figure 6 the lower fuel consumption at boiler pressure 60 bar and condenser pressure at 0.04 bar. In Figure 6 at condenser pressure 0.04 if the boiler pressure increased from 30 bar to 60 bar, the fuel consumption decreased by 0.11 Kg/s it save in day approximately 1439 $. In ballast condition from Figure 12 it was found that the fuel consumption increased when the condenser pressure increased at constant boiler pressure and steam temperature, for example, at boiler pressure 30 bar and steam temperature 515 ℃ the fuel consumption increased from 1.88 Kg/s to 1.98 Kg/s when the condenser pressure increased from 0.04 bar to 0.075 bar and the same at all boiler pressures. This condenser pressure based on change the cooling water temperature inlet to condenser, this led to increasing the expenses from 24595 $/day to 25903 $/day. It also found also that the lower fuel consumption at high boiler pressure and lower condenser pressure and the higher fuel consumption at low boiler pressure and high condenser pressure with constant steam temperature at 515℃. In Figure 12 the low fuel consumption at boiler pressure 60 bar and condenser pressure at 0.04 bar, it was also observed that from Figure 12 at condenser pressure 0.04 as the boiler pressure increased from 30 bar to 60 bar, the fuel consumption decreased by 0.1 Kg/s led to saving in day approximately 1308 $.. By comparison between the two conditions of sailing found that the fuel consumption decreased from the fully condition to ballast condition from 1.94 kg/s to1.87 kg/s respectively this save approximately 915.8 $/day to kept the ship speed constant due to that the power is decreased when the cargo weight decreased from 29361 KW to 26654 KW.

3.1.2.3 Specific Fuel Consumption

It is a measuring of fuel consumption with respect to power generation by notification of the ratio of fuel consumed used by an engine at a certain force with speed such as the amount of power that the engine produced. Specific fuel consumption compared the engines performance of different sizes to determine which the most efficient fuel consumption is. It allows manufacturers to see which engine will be used for lowest fuel consumption while still producing a high amount of power. From Figures 7 and 13 it was found that the specific fuel consumption increased when the condenser pressure increased at constant boiler pressure and steam temperature. for example at boiler pressure 30 bar and steam temperature 515 ℃ the specific fuel consumption is increased from 0.252 Kg/KW hr to 0.266 Kg/KW hr when the condenser pressure increased from 0.04 bar to 0.075 bar, also at all boiler pressure. It was also found that the low specific fuel consumption at high boiler pressure and low condenser pressure because the steam temperature was fixed at 515℃. In Figures 7 and 13 at boiler pressure 60 bar and condenser pressure 0.04 bar the specific fuel consumption was 0.238 Kg/kW hr. In two loading conditions during sailing the specific fuel consumption was approximately the same because the lower fuel consumption related to the lower power required for full load and ballast conditions.

3.1.2.4 Steam Consumption

The steam consumption is directly related to the fuel consumption discussed in the previous section, but its importance is related directly to the size of steam plant components which affect the weight of the ships. Its value reflected the amount of steam required in cycle to produce the require work done or useful power. It must be minimum as possible. In fully load condition from Figure 8 it was found that the steam consumption increased when the condenser pressure increased at constant boiler pressure and steam temperature, for example at boiler pressure 30 bar and steam temperature 515 ℃ , the steam consumption increased from 22.8 Kg/s to 24.3 Kg/s when the condenser pressure increased from 0.04 bar to 0.075 bar. It was also found that the lower steam consumption at high boiler pressure and lower condenser pressure while the steam temperature was fixed at 515℃. In Figure 8, at boiler pressure of 60 bar and condenser pressure of 0.04 bar the steam consumption decreased to 21.7 Kg/s. In ballast condition from Figure 14 it was found that steam consumption increased when the condenser pressure increased at constant boiler pressure and steam temperature, for example at boiler pressure 30 bar and steam temperature 515℃ the steam consumption increased from 21 Kg/s to 22.3 Kg/s when the condenser pressure increased from 0.04 bar to 0.075 bar. It was found also that the lower steam consumption at high boiler pressure and lower condenser pressure because the steam temperature was fixed at 515℃. In Figure 14 at boiler pressure 60 bar and condenser pressure 0.04 bar the steam consumption is 20 Kg/s. By comparison between the two conditions of sailing found that the steam consumption decreased from fully load condition to ballast load at boiler pressure 60 bar and condenser pressure 0.04 bar from 21.7 kg/s to 20 kg/s respectively to kept the ship speed constant due to that the power decreased when the cargo weight decreased from 29361 KW to 26654 KW.

3.1.2.5 Specific Steam Consumption

It is a criterion of steam plant performance, specific steam consumption relate the power output of a steam plant to the steam flow necessary to produce it. Low values of specific steam consumption mean that the size of a plant and its component parts are smaller than a plant of the same power output, but with a higher specific steam consumption value. In fully loaded in Figure 9 and in ballast condition in Figure 15 it was found that the specific steam consumption increased as the condenser pressure increased at constant boiler pressure and steam temperature, for example at boiler pressure 30 bar and steam temperature 515℃ the specific steam consumption increased from 2.79 Kg/kW hr to 2.99 Kg/kW hr and hence if the condenser pressure increased from 0.04 bar to 0.075 bar at all boiler pressure, the specific steam consumption increased when the condenser pressure increased. It was also found that the low specific steam at high boiler pressure and low condenser pressure with steam temperature fixed at 515℃. In Figures 9 and 15 at boiler pressure 60 bar and condenser pressure 0.04 bar the specific steam consumption is 2.68 Kg/kW h.

3.1.2.6 Cooling Water Mass Flow Rate

The condenser-cooling-water temperature rise and mass-flow rate are related to the rejected heat load. The cooling water inlet to condenser from the sea inside the tubes to absorb heat from the steam is very important in calculations because it directly related to the pressure drop and condenser sizing, the mass flow rate and pressure drop are the main parameters for pumping system performance. In Figure 10 found that the cooling water mass flow rate increased when the condenser pressure increased at constant boiler pressure and steam temperature. It could be observed for example at boiler pressure 30 bar and steam temperature 515℃ that the cooling water mass flow rate is increase from 1120 kg/s to 1220 kg/s lead to high pumping power when the condenser pressure increase from 0.04 bar to 0.075 bar for all boiler pressure are the same as the cooling water mass flow rate increased when the condenser pressure increased. It was also found that the low cooling water mass flow rate at high boiler pressure and low condenser pressure as the steam temperature was fixed at 515℃. In Figure 10 at boiler pressure of 60 bar and condenser pressure of 0.04 bar the cooling water mass flow rate is 1018 kg/s.

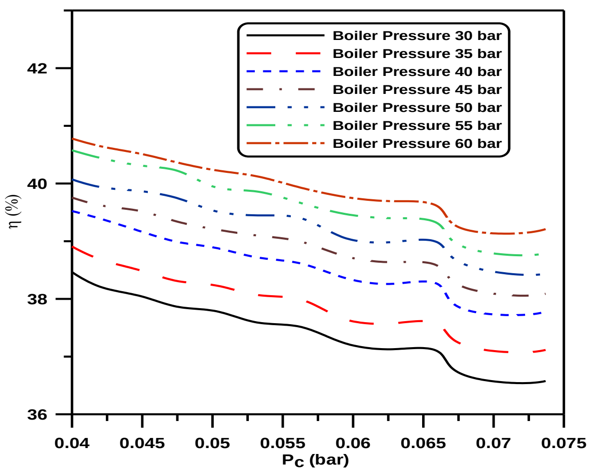


Figure (5): Condenser and Boiler Pressures Effect on Thermal Efficiency (Full Load Conditions)

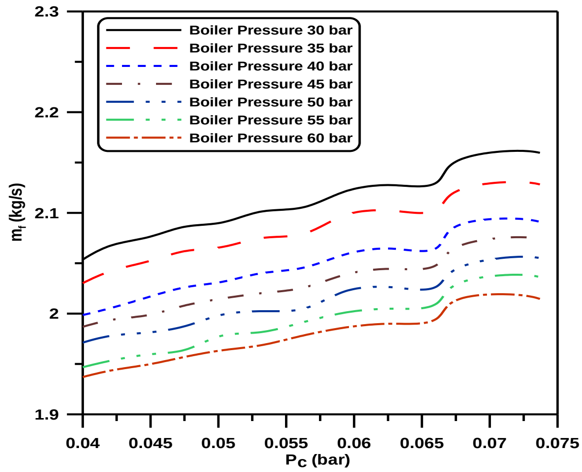


Figure (6) Condenser and Boiler Pressures Effect on Fuel Consumption (Full Load Conditions)

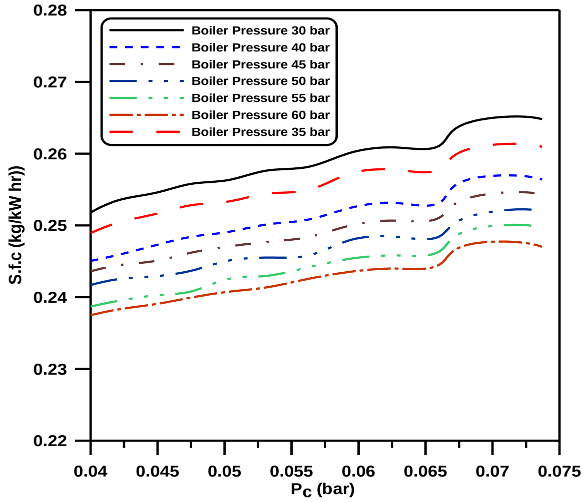


Figure (7): Condenser and Boiler Pressures Effect on Specific Fuel Consumption (Full Load Conditions)

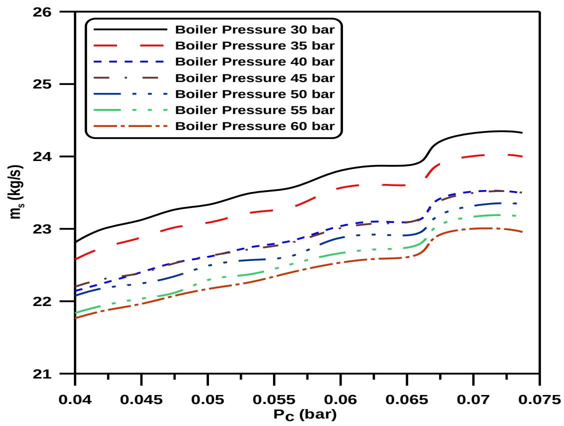


Figure (8) Condenser and Boiler Pressures Effect on Steam Consumption (Full Load Conditions)

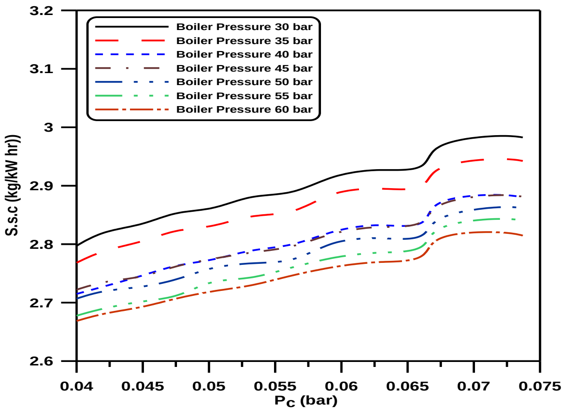


Figure (9) Condenser and Boiler Pressures Effect on Specific Steam Consumption (Full Load Conditions)

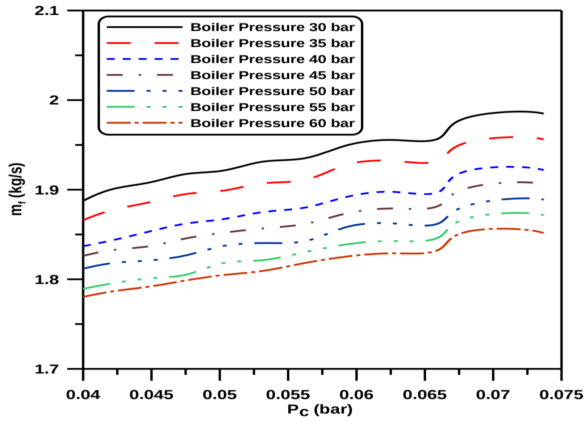
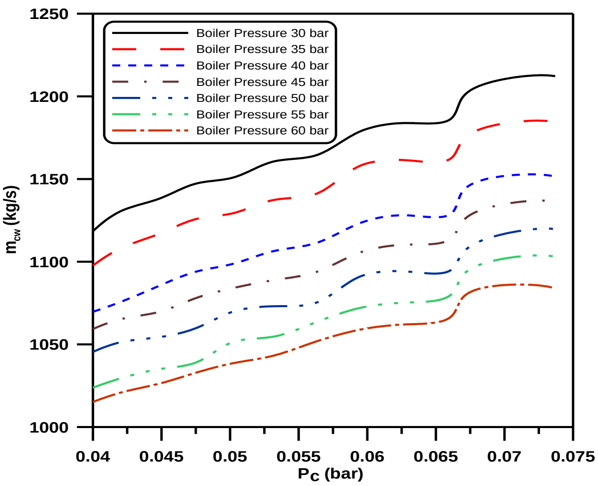


Figure (12) Condenser and Boiler Pressures Effect on Fuel Consumption (Full Load Conditions)



Figure(10) Condenser and Boiler Pressures Effect on Cooling Water Mass Flow Rate (Full Load Conditions)

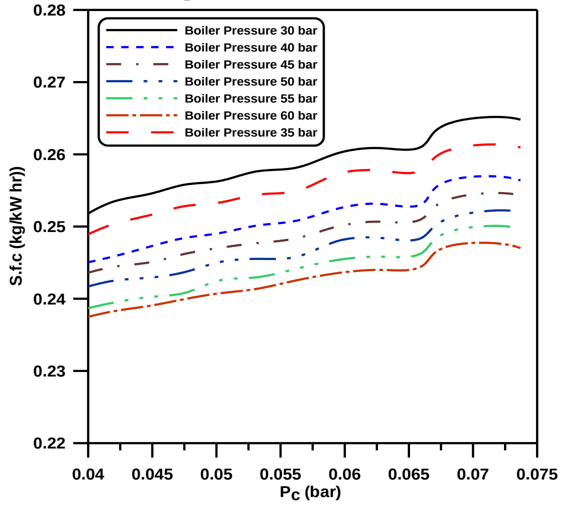


Figure (13) Condenser, Boiler Pressures Effect on Specific Fuel Consumption (Ballast Condition)

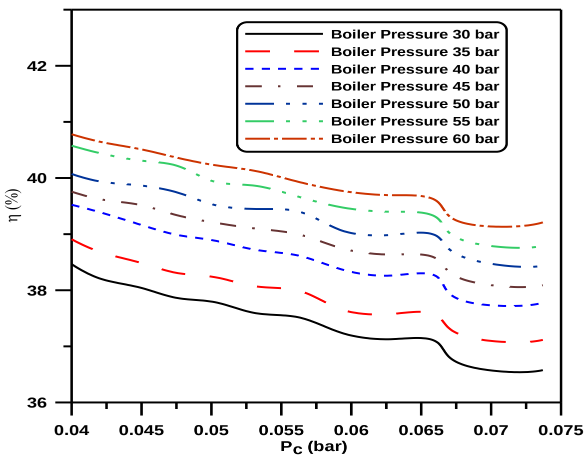


Figure (11) Condenser and Boiler Pressures Effect on Thermal Efficiency (Ballast Conditions)

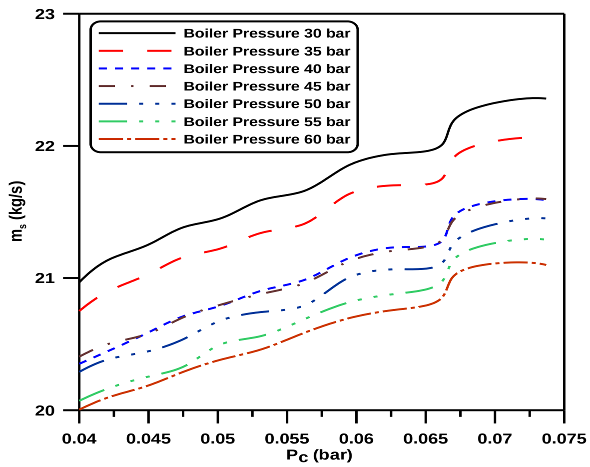


Figure (14) Condenser and Boiler Pressures Effect on Steam Consumption (Ballast Condition)

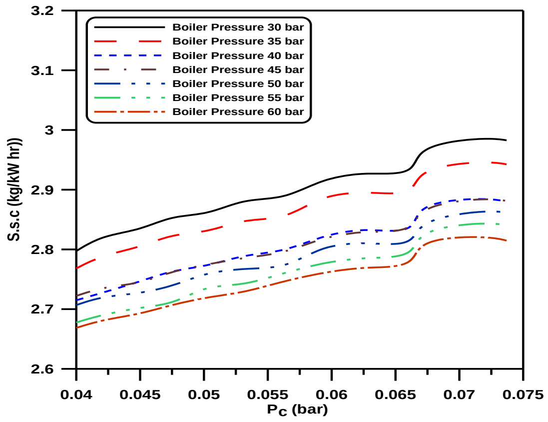


Figure (15) Condenser and Boiler Pressures Effect on Specific Steam Consumption (Ballast Condition)

4. CONCLUSIONS

From the study it was found that

- As the cooling water temperature inlet to condenser decreased, the pressure is decreased, and vice versa.
- There is an inverse relationship between the thermal efficiency and the cooling water temperature inlet to condenser.
- There is a proportionalrelationship between the thermal efficiency and theboiler pressure.
- The specific fuel consumption has proportional relationship with cooling water temperature inlet to condenser, where it has an inverse relationship with the boiler pressure.
- Also, there is a proportional relationship between the fuel consumption and sailing loading condition.
- The specific steam consumption has proportional relationship with cooling water temperature inlet to condenser, where it has an inverse relationship with the boiler pressure.
- There is a proportional relationship between the steam consumption and sailing loading condition.
- Finally, the mass flow rate of cooling water inlet to condenser has a proportional relationship with its temperature, where it has an inverse relationship with the boiler pressure.

5. REFERENCES

[1] D. A. Taylor, "Introduction to marine engineering", Elsevier Butterworth Hbnemann, 1996.
[2] Dongil Yeo, Byungsam, Jinmo Kim, Ilyong Kim and Samsung Heavy Industries Co., Ltd, " Propulsion alternatives for modern LNG carriers ", 2005.
[3] James Gaughan, "Economics, environment driving new generation of LNG carrier", 2007.
[4] R M Pitblado, J Baik, G J Hughes, C Ferro and S J Shaw, "Consequences of LNG Marine Incidents", CCPS Conference Orlando, 2004.
[5] Makoto Ito, Kazuyoshi Hiraoka, Shoichi Matsumoto and Kenji Tsumura, "Development of high Efficiency Marine Propulsion Plant (Ultra Steam Turbine) ", 2007.
[6] J. T. Holm, "Marine steam power plant heat balance practices ", 1980.
[7] George G. Dimopoulos and Christos A. Frangopoulos, "Thermo economic Simulation of Marine Energy Systems for a Liquefied Natural Gas Carrier", 2008.
[8] Kawasaki URA turbine catalogue, 2001.
[9] Kawasaki UM boiler catalogue, 2001.
[10] Kawasaki UA turbine catalogue, 2001.
[11] Volker Bertram, "Practical ship hydrodynamic", Butterworth-Heinemann, 2000.
[12] K.J.Rawson and E.C.Tupper, "Basic ship theory", Butterworth-Heinemann, 2001
[13] E.C. Tupper, "Introduction of naval architecture", Butterworth-Heinemann, 2004.
[14] Yunus A. Cengel and Michael A. Boles, "Thermodynamic", McGraw-Hill Professional, 2001.

6. NOMENCLATURE

Symbols

A_s projected area to wind, m^2
BHP Break power, kW
C_1 Coefficient depend on hull form
C_2 Coefficient depend on shape of bow
C_3 Coefficient depend on shape of stern
C_A Air resistance coefficient
C_f Frictional resistance coefficient
C_{pw} Specific heat of water, kJ/kg ^oC
C_v Viscous resistance coefficient
CV Colorific value of fuel, kJ/kg
DHP Delivered power,kW
EHP Effective power, kW
F_n Froude number
h_1 Enthalpy at exit of condenser, kJ/kg
h_2 Enthalpy at inlet to boiler, kJ/kg
h_3 Enthalpy at inlet to HP turbine, kJ/kg
h_6 Enthalpy at exit of HP turbine, kJ/kg
h_7 Enthalpy at inlet of LP turbine, kJ/kg
m_1 Coefficient depend on hull form
m_2 Coefficient depend on hull form
$\dot{m}_{cw}$ Cooling water mass flow rate kg/s
$\dot{m}_f$ Fuel consumption, kg/s
$\dot{m}_s$ Steam mass flow rate, Kg/s
P_c condenser pressure, bar
Power steam station power (BHP), kW
Q_{add} Heat added to the system, kW
R_A Air resistance, kN
R_{app} Appendage resistance, kN
R_{sw} Sea wave resistance, kN
R_T Total ship resistance, kN
R_V Viscous resistance, kN
R_w Wave making resistance, kN
S Wetted surface area, m^2
SHP Shaft power, kW
T_{cwi} Cooling water inlet temperature, ^oC
T_{cwo} Cooling water outlet temperature, ^oC
T_s Saturated temperature, ^oC
THP Thrust power, kW
U Heat transfer coefficient, W/ m^2 C
V Ship speed, m/s
W_T Turbine work done, kW

Greek symbols

η_b Boiler efficiency
η_H Hull efficiency
η_{gear} Reduction gear efficiency
η_p Propeller efficiency
η_{shaft} Shaft efficiency
η_{th} Thermal efficiency
λ Coefficient depend on hull form
ρ Density of sea water, ton/m^3
ρ_{air} Air density, ton/m^3

List of abbreviations

BOG Boil of Gas
CST Conventional Steam Turbine
FPP Fixed pitch propeller
LNG Liquefied Natural Gas
SFC Specific Fuel Consumption
SSC Specific Steam Consumption
UST Ultra Steam Turbine
VLCC Very Large Crude Carrier