Design Of Heat Exchanger For Effective Cooling Of Marine Gas Turbine Engines

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***Abstract*—The growing demand for an effective cooling system in marine vessel gas turbines (GTs) with an environmentally friendly conditions are the focus target of research scholars. This trend of study is subjected to detail and extensive evaluation of design incorporation of heat exchanger (HE) in a simple turboshaft GT. Thus, the presentation of this paper is one of such using mathematical modelling approach. Analysis of results from study attests high and improved gas path performance of cycle efficiencies between (30.4 – 30.81)%, and (57.08 – 59.09)% for thermal and ideal cycle efficiencies respectively as the compressor pressure ratio (PR) is reduced from its design point (DP) of 14.5 – 13.78 accounting for 5% drop. Subsequently, an improved specific work output (SW) result of 0.5002MJ/Kg at DP was increased to 0.5173MJ/Kg as PR changes in incremental percentage variation. All components parameters such as temperatures and pressures tends to decrease from their DP performance. This gives a clear signal for low production of exhaust gas emission to the environment. However, the maximum/intake temperature ratio ($^{T_3}/_{T_1}$), t yields a better result which determines the metallurgical limit of the stressed GT part. Therefore, obtained results from study can easily predict the life span of component parts for possible routine maintenance.**

Keywords—Deviation, Effectiveness, Gas turbine, Gas path, Heat exchanger, Performance, Pressure ratio.

NOMENCLATURE

CP_c	Specific Heat for Cold Air flow (J/KgK)
CP_h	Specific Heat for Hot Air flow (J/KgK)
p	Pressure (KPa)
P_{amb}	Ambient Pressure (KPa)
Q_{cc}	Heat Supplied at Combustion Chamber (KJ/KWh)
T	Temperature (°K)
T_{amb}	Ambient Temperature (°K)
W_c	Cold Mass Flow of Air (Kg/s)
W_h	Hot Mass Flow of Air (Kg/s)
η_∞	Polytropic Efficiency (%)
η_{ISC}	Compressor Isentropic Efficiency (%)
η_{IST}	Turbine Isentropic Efficiency (%)
η_{Th}	Thermal Efficiency (%)
η_{id}	Ideal Cycle Efficiency (%)
t	Temperature Ratio ($^{T_3}/_{T_1}$)

Greek symbols

γ	Ratio of Specific Heats (gamma)

Abbreviations

CC	Combustion Chamber
CW	Compressor Work (MW)
DP	Design Point
EGT	Exhaust Gas Temperature
FF	Fuel Flow
FCV	Fuel Calorific Value
GT	Gas Turbine
HE	Heat Exchanger
SW	Specific Work Output (MJ/Kg)
TET	Turbine Entry Temperature (°k)
TW	Turbine Work (Mw)

I. INTRODUCTION

Reliability and performance of modern engines are directly dependent on the effectiveness of lubricating cooling systems. To be effective, an engine lubricating cooling system must successfully perform the functions of minimizing excessive high temperature of the engine. Thus, this makes the lubrication oil more efficient for the bearing surfaces of moving parts, dissipating heat, and keeping the engine parts clean by removing carbon and other foreign materials. The essence of the provision of this component is to manager the metal components that will be damaged during the fuel combustion process of the engine. Hence, the continuous cooling of the engine lubricating oil with a HE mostly for the marine propulsion engines is of vital importance in the maritime vessels. HE as defined in a reviewed literature is a device used to transfer thermal energy between two or more fluids at different temperatures in thermal contact. Thus, this equipment effects the transfer of heat energy from a hot fluid to a cold fluid with maximum rate and minimum investment and running costs. Thence, the temperature of each fluid changes as it passes via the exchangers; thus causing variations along the length of the exchanger [1]–[3]. HE has become an essential component in most industrial processes and traces of benefits can be derived from it usage.

However, the application of a HE in GTs primarily is to improve its thermal efficiency. It increases the efficiency of the cycle substantially and markedly reduces the optimum PR for maximum efficiency [4]. It is reported that for a given size of an industrial GT plant, the power output could be reduced by as much as 10% owing to frictional pressure losses in the HE [5]–[7]. Also finding from a scholarly research established that a HE is essential for high efficiency when the cycle PR is low, but becomes less advantageous as the PR is increased [7]. However, currently in design and aerodynamic development of the compressor component, the use of high PRs has achieved over 40% increase in efficiency with simple cycle for power levels of 40MW [7]. According to an unveiled literature, thermal efficiency and the power output reduction of a GT system is as a result of increase in ambient condition such as humidity and ambient temperature [8], [9]. However, in real setting the operation of GT in an ambient condition that is constantly changing affects the performance and efficiency of the plant. Perhaps, this might be a lead way to relatively low production of efficiency of the GT cycle at DP. The indication of a worsen situation may occur further at part load operations and at off-design point (ODP) analysis with fairly high ambient temperature loosing up to 7% output power [10], [11].

In order to reduce and curb these losses is the introduction of a HE in the GT cycle which will maintain a constant air inlet temperature into the GT. Thus, one key element of GT systems is energy optimization, though it is possessed with economic and environmental challenges [12], [13]. It is recorded that the use of HEs in GTs for exhaust gas waste heat recovery can strongly contribute and control emission to the environment. It also leads to significant improvement of GT performance and decreases the consumption of fuel [14]. This has instigated the research for proper design implementation of a HE in GTs for the maximization of its benefits. Conversely, the introduction of recuperators and intercoolers in land based and ship GTs have continuously cooling the cool air to allow a higher turbine entry temperature (TET) to reduce the compressed bleed air. This has been the source of increasing turbine efficiency as well as creating additional air for combustion process [15]. Therefore, in order to enhance an environmentally friendly marine GT with low emissions production and improved SW with less specific fuel consumption (SFC) is to incorporate a HE in its engine with provision of offline monitoring technics.

II. PERFORMANCE SPECIFICATION OF ENGINE MODEL

LM500 marine GT is a product of General Electric (GE) Company PLC. It is derived from GE's CF34 turbofan engine that is currently in use in many military and commercial aircraft applications. The LM500 is basically a CF34 engine without a fan and is very similar in materials and design to GE's industry-leading LM2500. The LM500 is a simple-cycle, one-shaft GT with an aerodynamically coupled power turbine. It incorporates a variable stator compressor driven by an air-cooled, two-stage turbine. The LM500 incorporates the latest in proven design technology and corrosion resistant materials to provide a mature design with maximum reliability and component life. Ideally suited for marine applications requiring light weight and fuel economy, the LM500 offers the highest efficiency of any GT in its output class. Thus, it is considered as the engine of study due to its numerous characteristics. Table 1 and figures 1 – 3 presents LM500 engine specification, model, proposed engine schematic and temperature/entropy diagrams respectively.

Table 1: GE LM500 Engine Specification

S/No	Specifications	LM500 GT
1	Compressor Pressure Ratio	14.5
2	Power Output	6130hp (4571.14 kW)
3	Heat Rate	10916 Btu/kWs-hr (11517.04KJ/Kg)
4	Specific Fuel Consumption	443 Ib/hph (269.5kg/kWh)
5	Exhaust Gas Temperature	1049ºF (565°C)
6	exhaust Gas Flow	36 Ib/sec (16.4 Kg/s)
7	Shaft Numbers	1
8	Power Turbine Speed	7000rpm
9	Relative Humidity	60%

Source: [16]

Table 2: Fixed parameters at DP.

S/No	Parameters	Unit	GE LM500 GT
1	Ambient temperature	^oK	288.15
2	Ambient pressure	Kpa	101.33
3	FCV	KJ/Kg	43,100
4	Compr. Polytropic Eff	%	85
5	Turbine. Polytropic Eff	%	85
6	Combustor Pressure loss	%	5
7	CP_c for cold air	J/Kg/K	1005
8	CP_h for hot air	J/Kg/K	1150
9	γ for cold air		1.4
10	γ for hot air		4/3
11	Exhaust Pressure loss	%	0.75

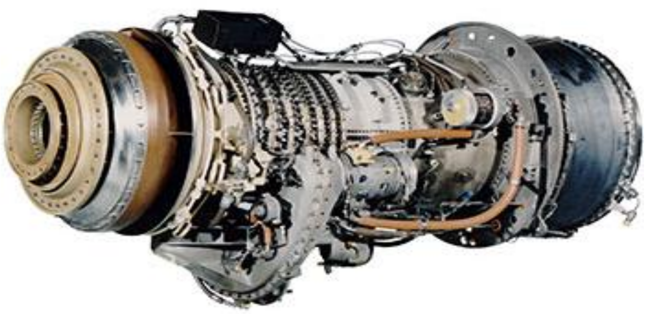

Figure 1: GE LM500 GT
Source: [16]

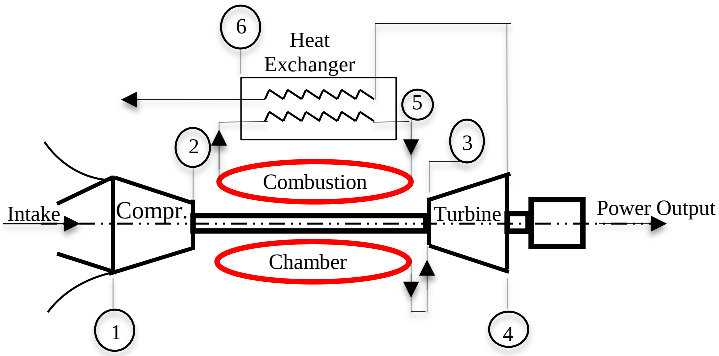

Figure 2: Single Shaft Open Cycle GT with HE

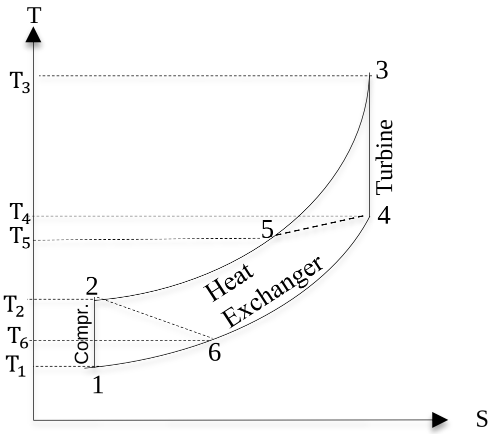

Figure 3: Temperature (T) – Entropy (S) Diagram

III. MATHEMATICAL ANALYSIS

The performance specification parameters also known as the engine DP parameters are the main focus of GT engine from the manufacturer. These configurations are selected to march the performance parameters of the proposed schematic model of the engine of study. Thus, it is essential to define some fixed designed parameters never included in the manufacturer's website due to design ethnics. These parameters are presented in table 2. However, this process is carried out with the following governing equations as shown in equations 1–18 and subsequently, results are presented in the succeeding section.

$$T_1 = T_{amb} \tag{1}$$

$$P_1 = P_{amb} \tag{2}$$

$$\eta_{Isc} = \frac{(PR)^{\gamma-1/\gamma} - 1}{(PR)^{1/\frac{\gamma}{\gamma-1} \times \eta_{Polytropic}} - 1} \tag{3}$$

$$T_2 - T_1 = \frac{T_1}{\eta_{ISC}} \times (PR)^{\gamma-1/\gamma} - 1 \tag{4}$$

$$\frac{P_2}{P_1} = PR \tag{5}$$

$$FF = \frac{Q_{cc}}{FCV} \tag{6}$$

$$CW = W_c \times C_{p_c}(T_2 - T_1) \tag{7}$$

$$Power\ Output = TW - CW \tag{8}$$

$$TW = W_h \times C_{p_h}(T_3 - T_4) \tag{9}$$

$$P_3 = (5\%\ Combustor\ pressure\ loss\ of\ P_2 \tag{10}$$

$$P_4 = (0.75\%\ pressure\ loss\ of\ P_1) + P_1 \tag{11}$$

$$\eta_{IsT} = \frac{1 - (\frac{P_4}{P_3})^{\eta_\infty(\gamma - 1/\gamma)}}{1 - \frac{P_4}{P_3}} \tag{12}$$

$$Q_{CC} = W_h\, x\, C_{p_h}(T_3 - T_5) \tag{13}$$

$$\left(\frac{T_{out}}{T_{in}}\right)_{\text{Turbine}} = \left(\frac{T_{out}}{T_{in}}\right)_{\text{Heat Exchanger}} \tag{14}$$

$$Effectiveness = \frac{T_5 - T_2}{T_4 - T_6} \tag{15}$$

$$SW = C_{p_h}(T_3 - T_2)(1 - \frac{T_1}{T_2}) \tag{16}$$

$$\eta_{Ideal\ cycle} = 1 - \frac{(PR)^{\gamma-1/\gamma}}{t} \tag{17}$$

$$Temperature\ ratio, t = \frac{T_3}{T_1} \tag{18}$$

The fixed parameters defined in table 2 creates an easy path-way in calculating the general performance of any industrial GT. However, the component pressure losses may differ as the case may be in different GTs with a varying polytropic efficiencies ranging from 85 – 88% for both compressor and turbine components. Meanwhile, the numerical combination of these parameters with the governing equations enables the assessment of the effect on the gas path of the GT. Conversely, this result to the determination of the DP calculations as presented in table 3.

Table 3: DP Analysis for GE LM500 GT

S/No	Parameters	Unit	Values
1	Ambient Temperature, T_{amb}	^oK	288.15
2	Ambient Pressure, P_{amb}	Kpa	101.33
3	Compressor Inlet Temperature , T_1	^oK	288.15
4	Compressor Inlet Pressure , P_1	Kpa	101.33
5	Compressor Exit Temperature , T_2	^oK	707.94
6	Compressor Exit Pressure , P_2	Kpa	1469.29
7	Turbine Entry Temperature (TET) , T_3	^oK	1441.4
8	Turbine Entry Pressure , P_3	Kpa	1395.82
9	Turbine Exit Temperature, T_4	^oK	838.15
10	Turbine Exit Pressure , P_4	Kpa	102.09
11	Heat Exchanger Inlet Temp., T_5	^oK	830.74
12	Heat Exchanger Outlet Temp., T_6	^oK	483.06
13	Cycle Thermal Efficiency	%	30.04
14	Cycle Ideal Efficiency	%	57.08
15	Specific Work output (SW)	MJ/Kg	0.5
16	Effectiveness		0.3458
17	Maximum Temperation Ratio (t)		5

Table 4: ODP Analysis for PR Reduction

Parameters	Percentage Reduction				
	1%	2%	3%	4%	5%
T_1 (^oK)	285.15	282.15	279.15	276.15	273.15
P_1 (Kpa)	100.32	99.32	98.33	97.35	94.77
T_2 (^oK)	698.2	688.5	678.84	669.21	659.61
P_2 (Kpa)	1440.09	1411.34	1383.01	1355.11	1305.46
T_3(^oK)	1435.61	1429.85	1424.12	1418.42	1412.75
P_3 (Kpa)	1368.09	1340.77	1313.86	1287.36	1240.18
T_4 (^oK)	838.15	838.15	838.15	838.15	838.15
P_4 (Kpa)	101.07	100.06	99.07	98.08	95.48
T_5 (^oK)	824.95	819.19	813.46	807.76	802.09
T_6 (^oK)	481.63	480.19	478.75	477.31	475.86
Ther. Eff. (%)	30.2	30.35	30.51	30.66	30.81
Ideal Eff. (%)	57.48	57.88	58.28	58.69	59.09
SW (MJ/Kg)	0.5017	0.5032	0.5046	0.5061	0.5074
Effectiveness	0.3555	0.3651	0.3746	0.384	0.3933
Max Temp. Ratio (t)	5.03	5.07	5.1	5.14	5.17

IV. GAS PATH OFF DESIGN ANALYSIS

The GT model under study is based on maritime vessel at offshore where ambient temperature and pressure tends to reduce. This is as a result of the extreme cold experienced on the high sea for a long period of cruise by the vessel. Thus, this sometimes caused freezing of the water supplied to the HE for intercooling. Conversely, this is an advantageous fact because the intercooler incorporated to the system will thereby use this means to provide cooling for the GT and thus rejecting heat to the seawater. However, the airside of the intercooler imposes total pressure loss.

Hence, in respect to the functionality and effectiveness of the HE in a marine GT; some basic analyses are carried out to prognosis and monitor the performance characteristics of the turbine for suitable maritime application. Therefore, an extensive off-design study based on the mathematical model of the engine was carried out to ascertain the reliability and viability of the simple GT model with a HE in the following variation performance test: -

- Reduction of compressor PR by 1%drop from the DP which decreases from 14.5 – 13.78; and subsequent reverse of increase by 1% from DP of 14.5 – 15.23 all at five sequential order.
- Ambient temperature and pressure deviation of 3℃ drop from DP of 15℃ - 3℃; and thus 1% from ambient pressure conditions reducing from (101.33 – 94.77) KPa respectively.

Hence, this leads to the presentation of tables 4 and 5 for the performance variation test of the off-design analysis in the gas path parameters with respect to reduction and increase of compressor PR.

Table 5: ODP Calculation for the Increase of PR

Parameters	Percentage Reduction				
	1%	2%	3%	4%	5%
T_1 (^oK)	285.15	282.15	279.15	276.15	273.15
P_1 (Kpa)	100.32	99.32	98.33	97.35	94.77
T_2 (^oK)	702.99	697.98	692.91	687.76	682.56
P_2 (Kpa)	1469.69	1469.94	1470.03	1469.94	1445.29
T_3(^oK)	1439.73	1438	1436.22	1434.37	1432.49
P_3 (Kpa)	1396.2	1396.44	1396.53	1396.44	1373.03
T_4 (^oK)	838.15	838.15	838.15	838.15	838.15
P_4 (Kpa)	101.07	100.06	99.07	98.08	95.48
T_5 (^oK)	829.07	827.34	825.56	823.71	821.83
T_6 (^oK)	482.65	482.22	481.78	481.32	480.85
Ther. Eff. (%)	30.09	30.13	30.18	30.23	30.28
Ideal Eff. (%)	57.35	57.63	57.91	58.19	58.47
SW (MJ/Kg)	0.5036	0.507	0.5104	0.5139	0.5173
Effectiveness	0.3547	0.3634	0.3722	0.381	0.3898
Max Temp. Ratio (t)	5.05	5.1	5.14	5.19	5.24

V. PRESENTATION OF RESULTS

Based on the mathematical modelling and off-design analysis carried out for the research study with respect to the GT model, the following results in figures 4 – 9 are presented for discussion.

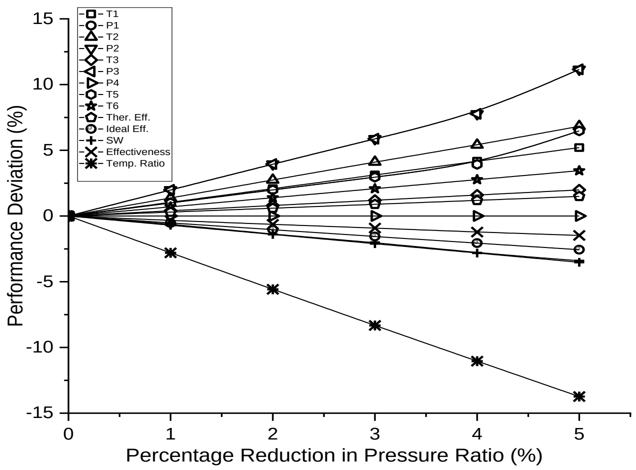

Figure 4: Reduction Variation of PR

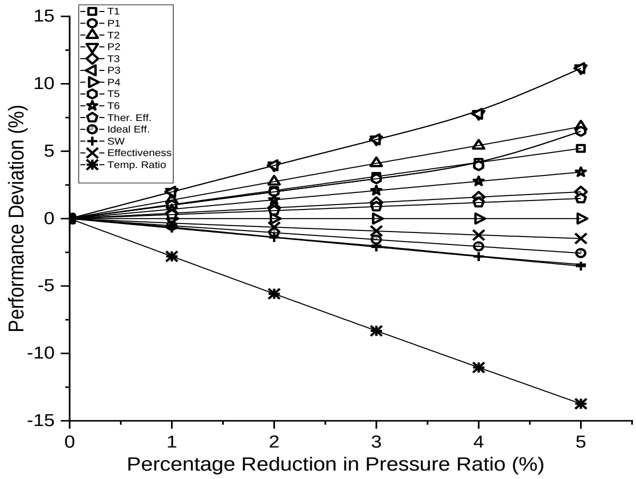

Figure 5: Increase Variation of PR

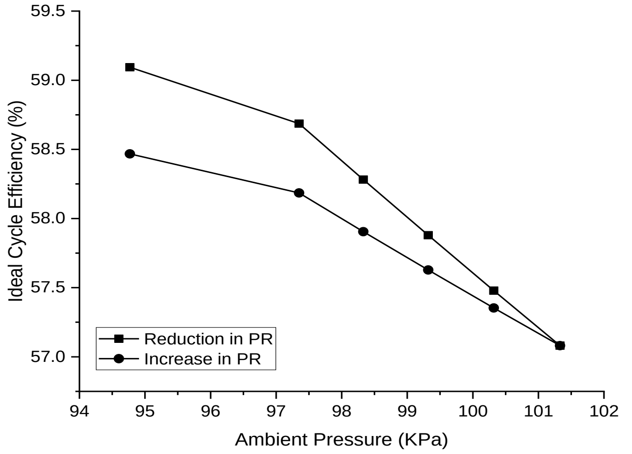

Figure 8: Ideal Cycle Efficiency - Ambient Pressure with HE in Simple Cycle

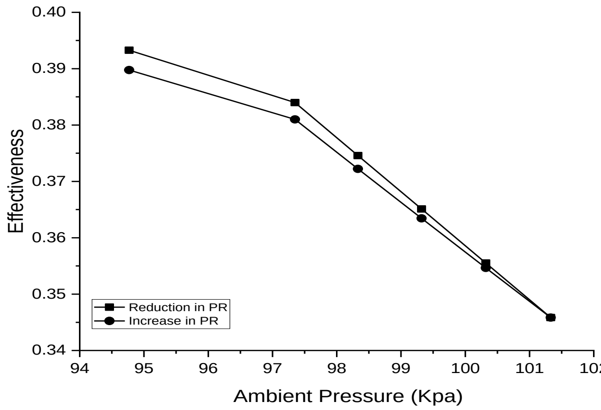

Figure 6: Effectiveness - Ambient Pressure with HE in Simple Cycle

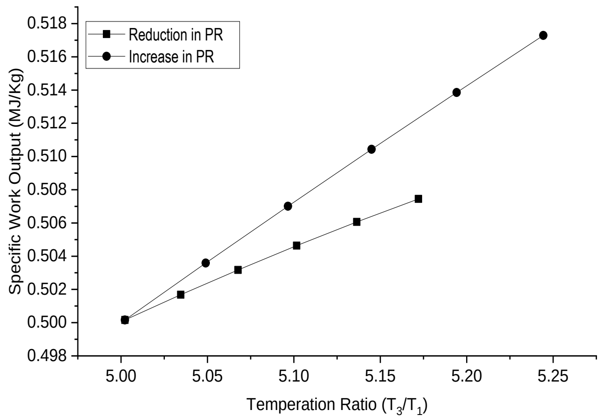

Figure 9: Specific Work Output and Temperature Ratio in a Simple Cycle with HE

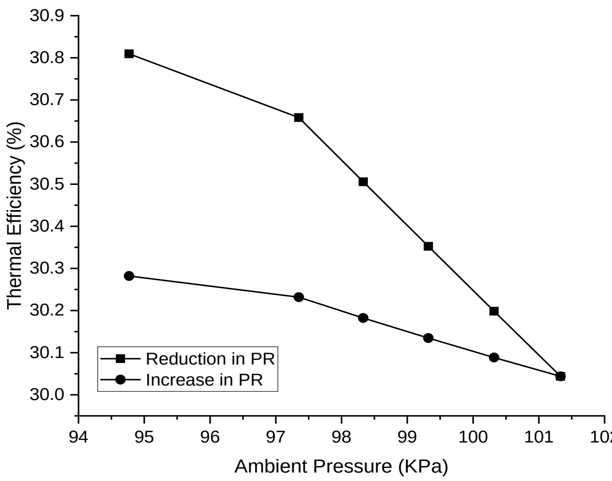

Figure 7: Thermal Efficiency - Ambient Pressure with HE in Simple Cycle

VI. DISCUSSION OF RESULTS

The graphical result plots shown above is a through representation of equation 19, where deviation performance lies on the real or DP parameter and slight variation (off-design point) of compressor PR.

$$Deviation = \frac{Real \ - Variation}{Real} \ x \ 100\% \qquad 19$$

Thus, reduction and increase in PR affects the gas path parameters as shown in figures 4 and 5. It is observed that all components pressures and temperatures decreases from their DP as PR is either reduced or increased. However, this phenomenon gives high performance as registered in HE effectiveness, cycle efficiencies and SW in figures 6 – 9 respectively. The effectiveness of the intercooler rises from 0.3458 of its DP to 0.3933 and 0.3898 for reduction and increase in PR; whereas the thermal and ideal cycle efficiencies improved from 30.04% and 57.08% of their DP to 30.81% and 59.09% for drop in PR while 30.28% and 58.47% for increase in

PR. Meanwhile, a significant increase in SW was also observed with a corresponding value of 0.5002MJ/Kg of its DP to 0.5074MJ/Kg and 0.5173MJ/Kg for both cases of PR drop and rise. A similar scenario across the graphical result plot is the sharp rise of all performance parameters in terms of drop in PR except a reverse situation for the SW.

The results obtained is a clear confirmation from relevant literature according to [7] which attests that the intercooler – HE in a simple GT leads to pressure losses but increases the SW and a worthwhile improvement in cycle efficiencies. Thus, the SW upon which the size of plant for a given power depends on the function of PR and maximum cycle temperature. Hence, the plot of the SW as a function of temperature ratio in figure 9 depends upon this condition which highly stressed parts of the turbine can stand for a required working life of the engine.

Conversely, the life span of a stressed GT part is a function of its metallurgical limit as confirmed in reviewed literature which depends on the temperature ratio, ƫ. Meanwhile, also in record an estimated value of ƫ ranges from 5.0 to 6.0 is established to moderate an air-cooled turbine with the aid of intercooler in simple GT [7]. Thus, result validation from study attest value of ƫ from its DP of 5.0 to 5.17 and 5.24 for PR drop and increase respectively.

VII. Conclusion

By and large, simple justification from the curves in the graphical plots yields appreciable improvement for cycle efficiencies from their DP of view. Thus, this is manifested by the introduction of HE in simple GT with the following conclusions: -

a) The reduction of PR value appreciably less than the optimum for maximum SW or DP should be adhered.
b) It is not necessary to use or increase the PR, because it will lead to increase in maximum cycle temperature otherwise also known as the (TET).
c) An increased PR is liable to increase GT exhaust temperature giving-up more pollutant gases to the environment.

Similarly, the increase of the HE effectiveness as PR is reduced is against its design phenomenon of yielding higher effectiveness with improved larger volume of the HE. Thus, the concession of the cost of the HE design for higher effectiveness is overtaken by the provision of the research study. Therefore, in summary the study is capable of reducing turbine exhaust gases militating environmental pollution. This boycotts the cost involved in larger volume-area production of HE for improved effectiveness. Hence, study is realistic and viable.

REFERENCES

[1] Shah, R.K. and Dusan, P.S. (2003). *Fundamentals of Heat Exchanger Design*. John Wiley & Sons, Inc., Hoboken, New Jersey.
[2] Arora, C.P. (2009). *Refrigeration and Air Conditioning, (3rd Ed)*. Tata Mcgraw Hill Education Private LTD, New Delhi, India. P.49
[3] Rajput, R.K. (2009). *Thermal Engineering, 7th Edition*. Laxmi Publications (P) LTD, Daryagani – New Delhi, India. P.1555
[4] June, K.M., Ji, H.J., Man, Y.H. and Kui, S.K. (2009). High temperature heat exchanger studies for applications to gas turbines. *Heat Mass Transfer*, 46, pp.175–186.
[5] Takeshi Y., Takafumi, N., Takafumi, M. and Yasuyuki, I. (2018). Basic Heat Exchanger Performance Evaluation Method on OTEC. J. Mar. Sci. Eng. 6, 32.
[6] Sudheer, S. (2018). Applied Thermodynamics II. Course Notes, Indian Institute of Technology Patna. Available on http:// http://www.iitp.ac.in, Accessed 25th November, 2018.
[7] Saravanamutto, H.I.H., Rogers, G.F.C., Cohen, H. and Straznicky, P.V. (2009). "*Gas Turbine Theory (6th Ed)*". Upper Saddle River, N. J: Pearson Prentice Hall.
[8] Gonz_alez-Díaz, A., Alcar_az-Calder, A.M., Gonz_alez-Díaz, M.O., Mendez-Aranda, A., Lucquiaud, M. and Gonz_alez-Santal, J.M. (2017). Effect of the ambient conditions on gas turbine combined cycle power plants with post-combustion CO2 capture. *Energy* 134, pp. 221-233.
[9] Liu, Y., Avisekh B., Amar K., Alka, S. and Nita, G. (2017). Effect of Ambient Temperature on Performance of Gas Turbine Engine. *Annual Conference of the Prognostics & Health Management Society,* pp. 1-7.
[10] Abdul Karim, Z.A., Mohd Azmi, M.N.H. and Abdullah, A.S. (2012). Design of a Heat Exchanger for Gas Turbine Inlet Air using Chilled Water System. *Energy Procedia: 2nd. International Conference on Advances in Energy Engineering,* 14, pp.1689 – 1694.
[11] Bengt, S. (2005). High Temperature Heat Exchangers. *Proceeding of the 5th International Conference on Science, Engineering and Technology*. Hoboken, USA
[12] Aziza, B. and Alfred, R. (2016). Gas Turbine: Optimization of Energy Production and High Efficiency by Using Power Electronics. *Procedia Engineering,* 138, pp. 337-346.
[13] Aminov, R.Z. and Kozhevnikov, A.I. (2016). Optimization of the Operating Conditions of Gas-Turbine Power Stations Considering the Effect of Equipment Deterioration. *Thermal Engineering*, 64(10), pp.715-722.
[14] Christina, S., Dimitrios, M., Zinon, V., Michael, F., Stefan, D. and Kyros, Y. (2016). Numerical Modeling of Heat Exchangers in Gas Turbines Using CFD Computations and Thermodynamic Cycle Analysis Tools. *Chemical Engineering Transactions,* 52, pp.517-522.
[15] June, K. M., Ji, H. J., Man, Y. H. and Kui, S. K. (2009). High temperature heat exchanger studies for applications to gas turbines. *Heat Mass Transfer*, 46, pp.175–186.
[16] General Electric (GE), (2017). LM500 Marine Gas Turbine Specifications. Available at *https://www.gemarinesolutions.com/.../ge-lm500-engines-power-republic-korea-navy*. Accessed 18th November, 2018.