iJRASET
International Journal For Research in Applied Science and Engineering Technology


INTERNATIONAL JOURNAL FOR RESEARCH
IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 Issue: X Month of publication: October 2019

DOI: http://doi.org/10.22214/ijraset.2019.10047

# Performance Optimization of Combined Gas and Steam Power Plant using Matrix Laboratory (MATLAB)

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***Abstract:** The ever increasing demands for higher thermal efficiency from power plants have resulted to wide range of research in this regard. This work is one of such researches that deal with the performance optimization of a COGAS plant using MATLAB. A mathematical model of the COGAS plant was developed based on thermodynamic analyses and energy balance equations. This was used to develop a program and run in MATLAB environment for analysis and Optimization of the COGAS thermal efficiency.*

***Results show that the program simulation yielded an overall maximum operational thermal efficiency of 69.7% at about 1750K, considering metallurgical limits of the topping cycle turbine components. The resulting model could predict the optimized performance output of the system with high degree of accuracy and reliability. The novelty of this research is essentially in its methodology and will remain invaluable in future works that will focus towards improving the thermal efficiency of power plants.***

***Keywords:** Gas turbine, Steam turbine, Optimization, Efficiency, MATLAB*

## I. INTRODUCTION

A gas turbine (GT) is an internal combustion engine that uses the gaseous energy of air to convert chemical energy of fuel into mechanical energy. The steam turbine is a device that extracts thermal energy from pressurized steam and uses it to produce mechanical work. Bringing together these two systems thermodynamically to function as a single system is termed “cogeneration” which means generation of both work (shaft power) and heat (steam) simultaneously.

Cogeneration systems can work with high capacity and high efficiency nearly all year. Cogeneration systems save 30 % more energy than conventional energy systems [1].

Thermodynamically, when two thermal cycles are combined in a single power plant the efficiency that can be achieved is higher than that of one cycle alone and energy is conserved [2]. The exhaust temperature of GT can be within the range of about 500-600$^{o}$C [3]. Combination of cycles with different working media is quite interesting because their advantages can complement one another. Normally, when two cycles are combined, the cycle operating at the higher temperature level is called the "topping cycle". The waste heat it produces is then used in a second process that operates at a lower temperature level and is therefore called the "bottoming cycle" [4]. It thus makes engineering sense to take advantage of the very desirable characteristics of the gas-turbine cycle at high temperatures and to use the high-temperature exhaust gases as the energy source for the steam power cycle [5]. The combination most widely accepted for commercial power generation and marine propulsion application is that of a gas topping cycle with a steam bottoming cycle [6].

Along with its wide and successful application in land-based power plants, the combined gas and steam turbines (COGAS) concept is being extended to provide an alternative form of power plant for ships [7]. COGAS should not be confused with combined steam and gas power plants, which employ oil-fired boilers for steam turbine propulsion during normal cruising and the gas turbines is supplemented for high speed and faster response/reaction times. COGAS plant uses heat from the exhaust gas of the gas turbine to heat water through a series of tubes in a heat exchanger under high pressure to become superheated steam. In this plant, the network done is a combination of the work done in the gas and steam turbines all from only one source of heat supply [5]. Flexibility provided by these systems satisfies utility power- generations, industrial-cogeneration and ship propulsion applications where the efficiency of these systems can exceed 60% [8]. Modeling and simulation of combined cycles has always been a powerful tool for their performance optimization. However, the need to develop accurate and reliable models of COGAS for different objectives and applications has been a strong motivation for researchers to continue to work in this fascinating area [9]. MATLAB has shown a high and strong potential to be considered as a reliable alternative to the conventional modeling approaches, simulation and optimization methodologies due to their independence and adaptability. This work will deal with novel methodology for performance optimization of a COGAS plant using MATLAB.

## II. METHODOLOGY

The performance optimization of the units that make up the entire system of this COGAS plant with technical parameters shown below in Fig. 1, are implemented utilizing the approach stated below: modeling the COGAS plant, writing a program to implement the modeling in MATLAB and to use the obtained operational data from a COGAS plant for performance optimization of the COGAS plant. The technical details of the COGAS system used for this research are shown in Appendix A.

### *A. Analytical Model of the COGAS System*

For the purpose of this research, fig. 1 shows the schematic diagram of the COGAS plant used for the modeling. The modeling were carried out in segment for mathematical convenience and simplification starting with the gas turbine modeling, then steam turbine modeling and finally the combined cycle following the nomenclature of the schematic diagrams of each power plant.

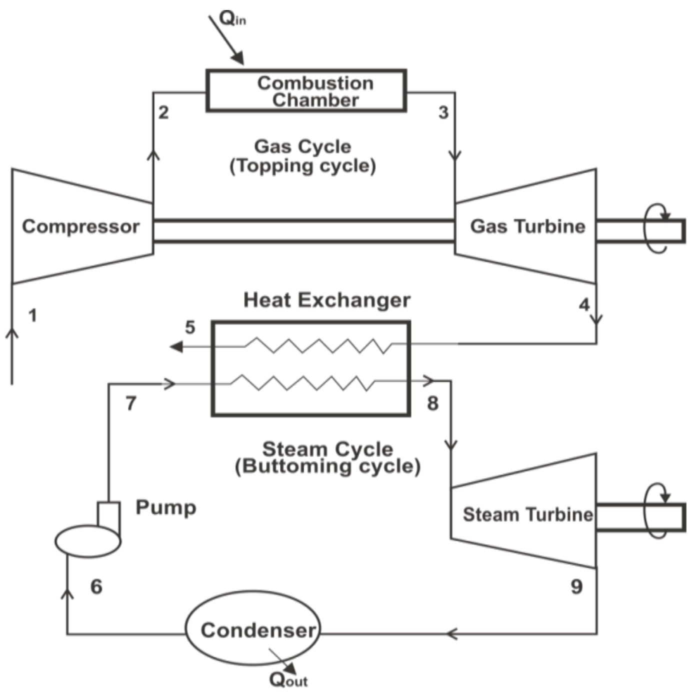


Fig. 1 A schematic diagram of a COGAS plant [10]

### *B. Modeling the Gas Turbine*

In the GT cycle (topping cycle) as shown in fig 1, the air is compressed isentropically in the compressor from state 1 to 2 where its temperature rises from $T_1$ to $T_2$.The compressed air then enters the combustion chamber where the combustion of fuel takes place isobarically. This results in rise of temperature of combustion product from $T_2$ to $T_3$. The high temperature gases enter the turbine where it expands to the final temperature $T_4$. According to Ogbonnaya [5], [11] the work done in the compressor is given by:

$$W_{gC} = m_a C_{p_a}(T_2 - T_1) \quad (1)$$

$$= mC_p T_1\left(\frac{T_2}{T_1} - 1\right) \quad (2)$$

But the pressure ratio is given by the expression below;

$$\frac{T_2}{T_1} = P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \quad (3)$$

Considering the pressure ratio of the turbine, equation (1) becomes

$$W_{gc} = m_a C_p T_1 \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right) \tag{4}$$

The expression for the work done, $W_{gt}$ by the turbine is:

$$W_{gt} = m_a C_p (T_3 - T_4) \tag{5}$$

Similarly;

$$T_3 = T_4 P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \tag{6}$$

From equation (2.6), the turbine outlet temperature is given by;

$$\frac{T_3}{T_4} = \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right) \tag{7}$$

Substituting equation (7) into equation (5), the expression for the work done in the gas turbine is;

$$W_{gt} = m_a C_p T_4 \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right) \tag{8}$$

The heat supplied, $Q_{sg}$ to the system is obtained by the relations below:

$$Q_{sg} = m_a C_p (T_3 - T_2) \tag{9}$$

Substituting equation (3) into equation (9), the heat supply to the system is given by the expression in equation (10);

$$Q_{sg} = m_a C_p \left( T_3 - T_1 P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right) \tag{10}$$

Where:

$T_3 = inlet\ temperature\ to\ the\ turbine$

$Q_{sg} = heat\ supplied$

The network done by the topping cycle is derived from the expression below:

$$W_{net.gas} = W_{gt} - W_{gc} \tag{11}$$

According to Cengel and Boles the above, the efficiency of the gas turbine is [12]:

$$\eta_{gas.tur} = \frac{W_{net.gas}}{Q_s} \tag{12}$$

$$\eta_{gas.tur} = \frac{m_a C_p \left( T_3 - \left( \frac{T_3}{P_r^{\left(\frac{\gamma-1}{\gamma}\right)}} \right) \right) - m_a C_p T_1 \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right)}{m_a C_p \left( T_3 - T_1 P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right)} \tag{13}$$

$$\eta_{gas.tur} = \frac{\left[ \left( T_3 - \left( \frac{T_3}{P_r^{\left(\frac{\gamma-1}{\gamma}\right)}} \right) \right) - T_1 \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right) \right]}{\left( T_3 - T_1 P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right)} \tag{14}$$

### *C. Modeling the Steam cycle (Bottoming Cycle)*

In the steam cycle, feed water flows through the heat exchanger where it absorbs heat from the exhaust gas of the GT cycle and in this way the temperature of the air rises [13]. The high pressure and high temperature steam leaving the heat exchanger enters the turbine where it expands to the final state as shown in fig. 2. Therefore the expression for energy balance in the heat exchanger is given by:

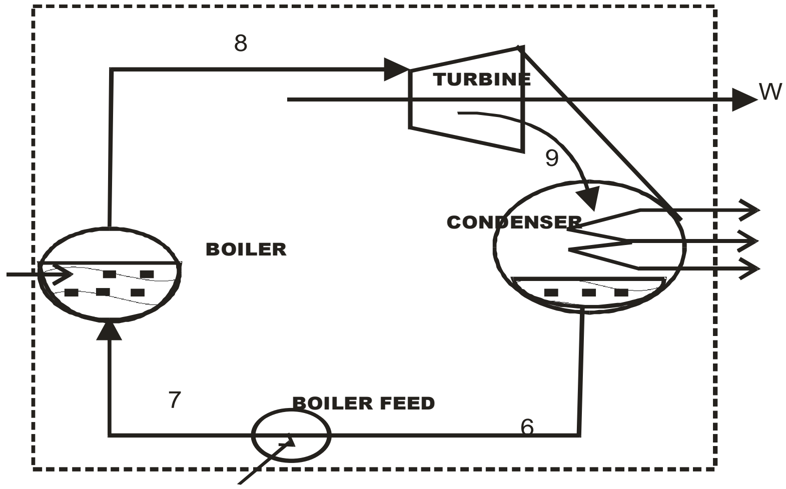


Fig. 2 A schematics diagram of a steam turbine [10]

The energy balance in the heat exchanger is expressed by equation 15

$$m_a C_P(T_4 - T_5) = m_s(h_8 - h_7) \quad (15)$$

According to Ogbonnaya (2004), the work done by the steam turbine is;

$$W_{st} = m_s(h_8 - h_9) \quad (16)$$

The work done by the pump is given by:

$$w_p = v_f(P_7 - P_9)X100 \quad (17)$$

Therefore, the net work done by the steam turbine is given by the expression:

$$W_{net.steam} = W_{st} - w_p \quad (18)$$

$$W_{net.steam} = m_s(h_8 - h_9) - v_f(P_7 - P_9) \quad (19)$$

Equation (2.18) can be written as;

$$W_{net.steam} = m_s(h_8 - h_9) - m_s(h_7 - h_6) \quad (20)$$

The heat supplied to the system can be obtained from the expression below:

$$Q_{ss} = m_s(h_8 - h_7) \quad (21)$$

Therefore, the steam cycle efficiency will be given by;

$$\eta_{st} = \frac{m_s[(h_8 - h_9) - (h_7 - h_6)]}{m_s(h_8 - h_7)} \quad (22)$$

From Cengel and Boles [12] and Shalan et al [9], the net efficiency of the combined cycle can be obtained from the expression:

$$\eta_{combined} = \frac{(W_{net.gas} + W_{net.steam})}{Q_{sg}} \quad (23)$$

$$\eta_{combined} = \left[ \frac{\left[ m_a C_p \left( T_3 - \left( \frac{T_3}{P_r^{\left(\frac{\gamma - 1}{\gamma}\right)}} \right) \right) - T_1 \left( P_r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1 \right) \right] + m_s[(h_8 - h_9) - (h_7 - h_6)]}{m_a C_p \left( T_3 - T_1 P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right)} \right] \quad (24)$$

*D. Computer Model and Programming of the COGAS System*

MATLAB (R2016a) programming language was used to write script files for analyzing the various parameters and for optimizing the COGAS thermal efficiency using technical data collected from COGAS plant for ship propulsion. To obtain a maximally optimized and to ensure good generalization characteristic of the COGAS model, a comprehensive computer code was generated and run in MATLAB. Table 1 and Fig. 3 show the input parameters for the COGAS optimization and flow chart developed for the program to optimize the thermal efficiency of the COGAS system respectively.

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue X, Oct 2019- Available at www.ijraset.com

Table 1
Input Parameters for COGAS Optimization

| Parameters | Symbol | Unit | Operational Range |
|---|---|---|---|
| GT compressor inlet temperature | $T_1$ | K | [273.15; 328.15] |
| GT compressor inlet pressure | $P_1$ | bar | [1.01325; 21.0325 |
| GT pressure ratio | $P_r$ | - | [11.5; 20.8] |
| GT inlet temperature to the turbine | $T_3$ | K | [1650; 1850] |
| GT air mass flow Rate | $m_a$ | Kg/sec | [67.9268; 77.9268 |
| GT fuel mass flow rate | $m_g$ | Kg/sec | [0.00367; 0.2661] |
| ST steam mass flow rate | $m_s$ | Kg/sec | [0.79; 60.75] |
| ST enthalpy before entering the pump | $h_6$ | KJ/kg | [174.0; 194.0] |
| ST enthalpy after the pump | $h_7$ | kJ/kg | [182.06; 202.0] |
| ST enthalpy after the boiler | $h_8$ | kJ/kg | [3398.0; 3599.0] |
| ST enthalpy after the turbine | $h_9$ | kJ/kg | [2102.8; 2302.8] |
| ST inlet temperature | $T_5$ | K | [500.0; 550.0] |
| ST boiler pressure | $P_5$ | Bar | [80.0; 100.0] |
| Specific heat capacity of air | $C_p$ | kJ/kgk | [1.005; 1.010] |
| Ratio of specific heat | $\gamma$ | - | [1.33; 1.44] |
| ST Condenser pressure | $P_6$ | Bar | [0.08;0.10] |

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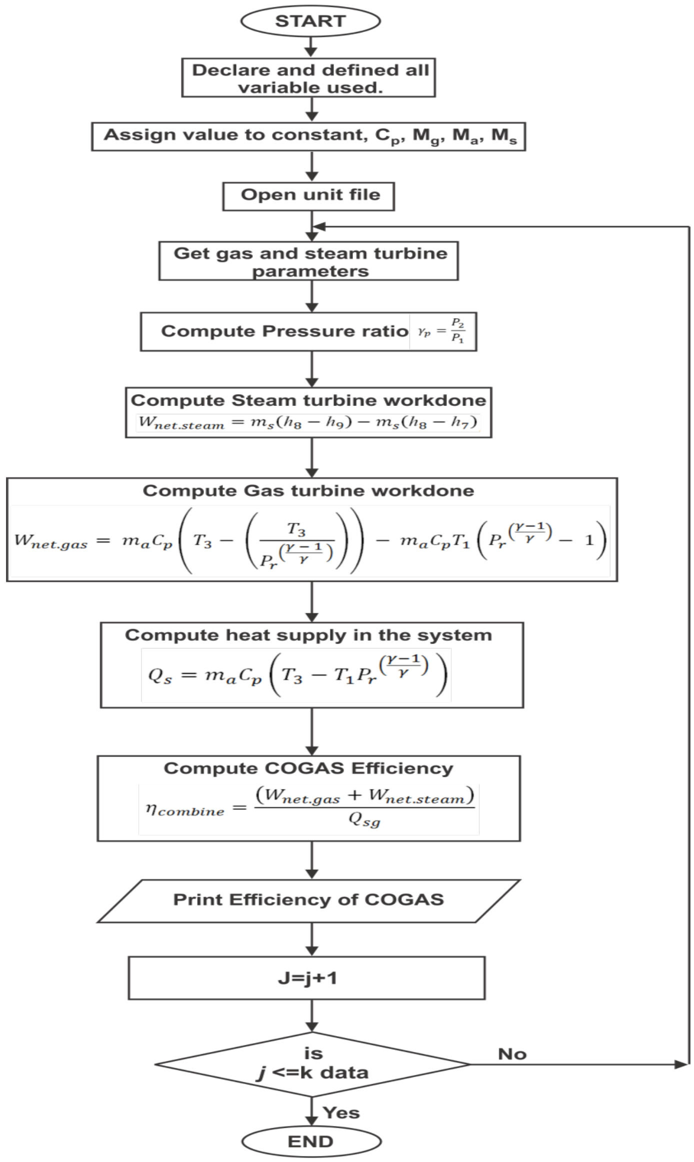


Fig. 3 A Flow Chart to Calculate the Efficiency of COGAS Plant

## III. RESULT AND ANALYSIS

To obtain and ensure a good optimization of the COGAS model, a comprehensive code was developed and run in MATLAB. From the analysis obtained from the computer model, the following graphs were plotted to demonstrate the relationship between COGAS thermal efficiency and some selected parameters from the MATLAB program. Fig. 4 shows a graph of COGAS efficiency against the steam turbine mass flow rate. In this case, as the turbine mass flow rate increases, there is a corresponding increase in the thermal efficiency. Steam mass flow rate influence on the COGAS thermal efficiency is observed to be at a value of 0.195kg/sec, after which both parameter increases proportionately.

Fig. 5 shows a graph of compressor pressure ratio against COGAS efficiency. It was observed that, as the pressure ratio increases, the COGAS efficiency increases correspondingly. This indicates a proportional relationship between these two quantities of COGAS system. Fig. 6 shows the variation of COGAS efficiency against the gas turbine inlet temperature. The COGAS efficiency begins to show an increase at value of 0.6912 as the turbine inlet temperature increases. It shows that the rate of rise in efficiency of COGAS increases as the gas turbine inlet temperature increase. The final overall data generated in MATLAB with which ANN was trained is shown in Appendix B.

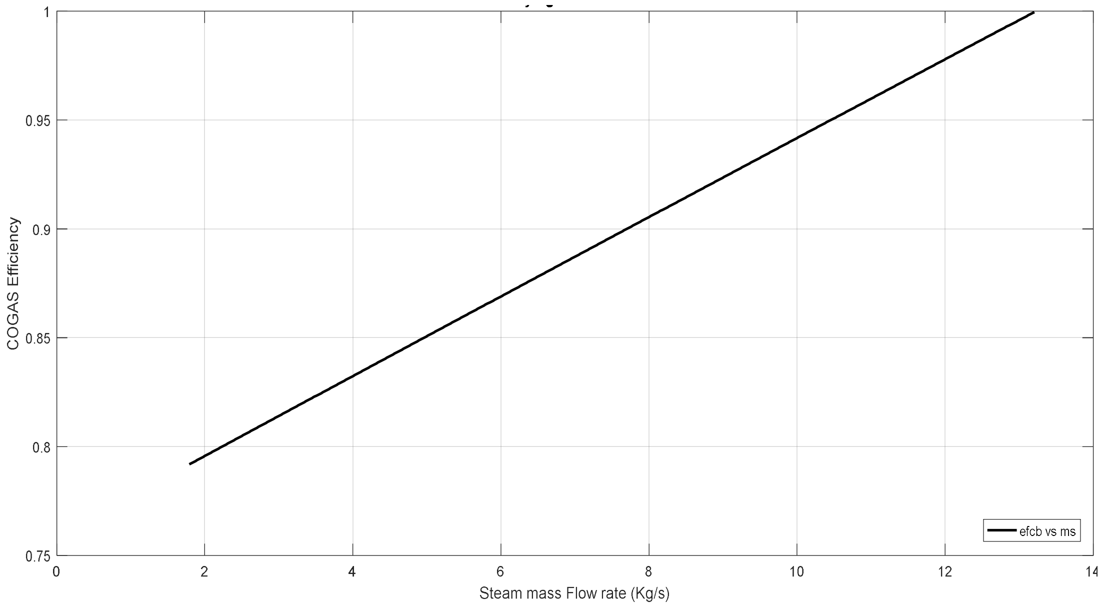


Fig. 4 COGAS efficiency against Steam mass flowrate

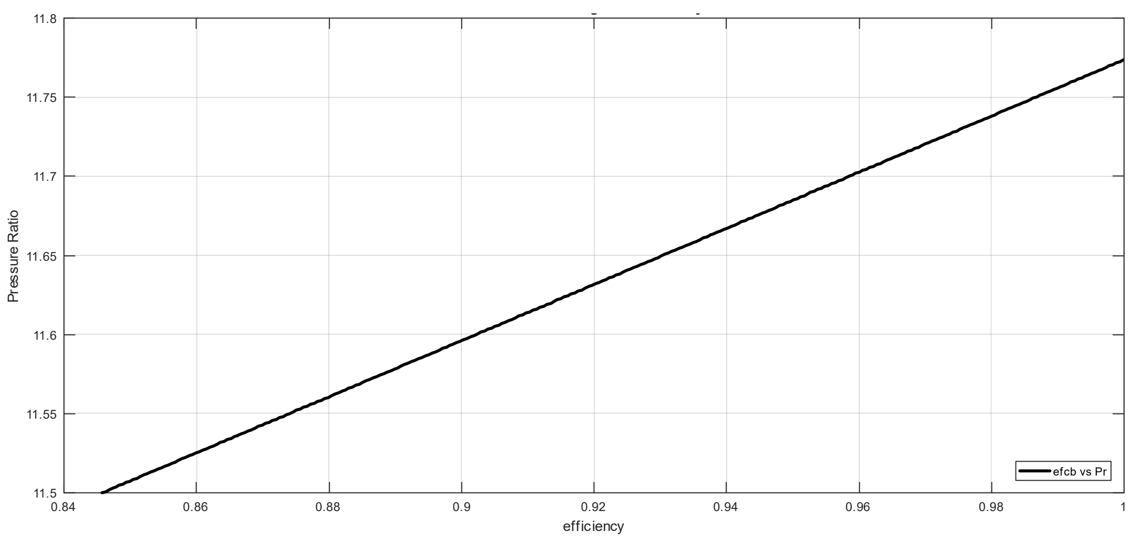


Fig. 5 Graph of Pressure Ratio against Efficiency

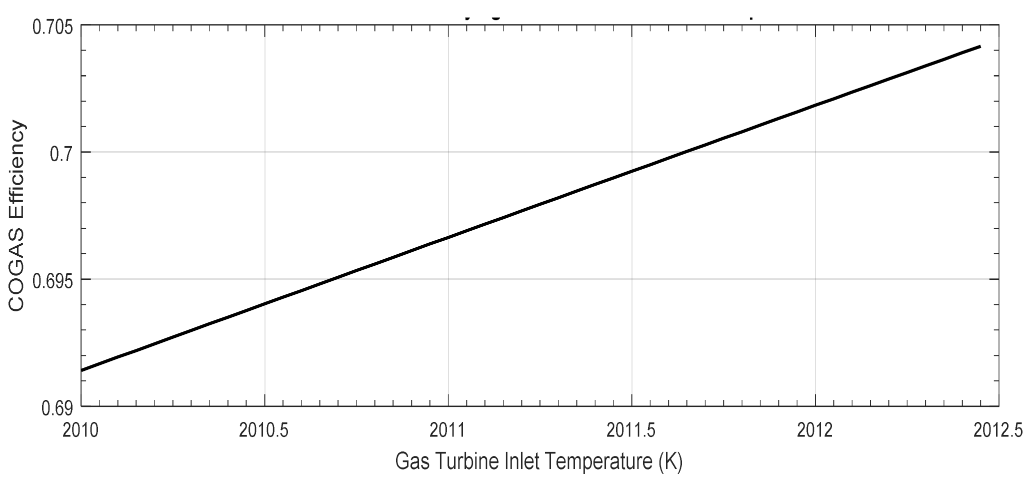


Fig. 6 COGAS efficiency against Gas turbine inlet temperature

## IV. CONCLUSION

In this research work, thermodynamics and energy balance equations were employed to model the COGAS plant. A comprehensive computer program code was generated and run in MATLAB environment using the COGAS data obtained for ship propulsion. The main objective of the research was to explore and optimize the thermal efficiency of the COGAS plant using MATLAB. This analysis is vital as COGAS plant satisfactory performance depends extensively on the thermodynamic properties and efficiency of the systems. The program simulation yielded an overall maximum operational thermal efficiency of 69.7% at about 1750K, considering metallurgical limits of the topping cycle turbine components. Similarly, some COGAS parameters have a proportional relationship with the thermal efficiency of the system. The results are evident to conclude that modeling, simulation and analysis can be handled using MATLAB to produce results with a high degree of accuracy and reliability.

### *A. Recommendations*

Based on the findings from the computer model and analysis results, recommendations are as follows:

*1)* MATLAB should be employed to maximize the optimization characteristics of COGAS systems.
*2)* MATLAB methodology should be used to predict the thermal efficiency and performance of similar COGAS systems.
*3)* Reasonable attention and consideration should be given to the thermodynamic properties of COGAS systems

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ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.177
Volume 7 Issue X, Oct 2019- Available at www.ijraset.com

## APPENDIX A

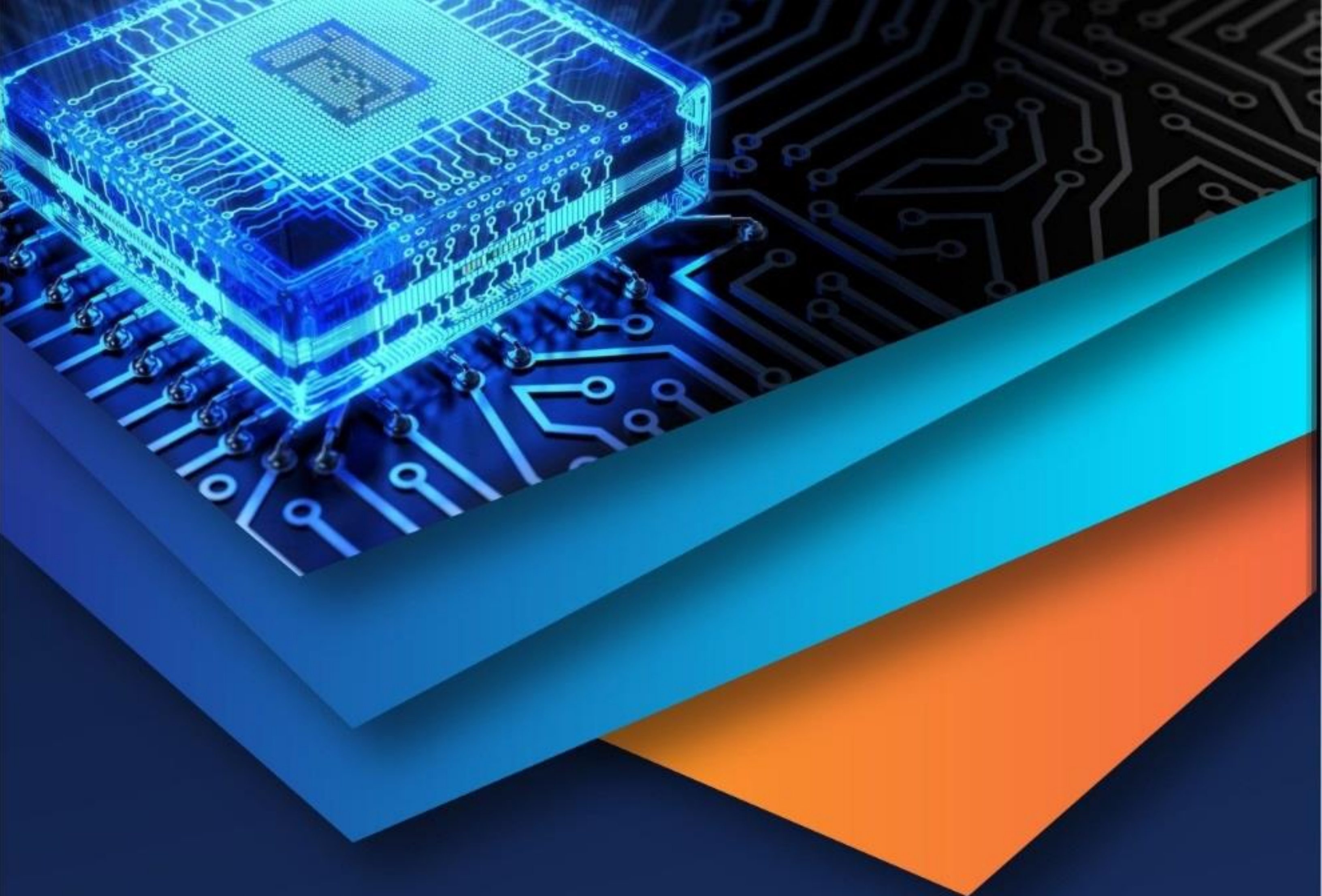
Technical Details of the COGAS Used In This Work

| | |
|---|---|
| Gas turbine: | |
| Manufacturer | General Electric |
| Model | GE9351FA |
| Fuel | Natural Gas |
| Number of shaft | 1 |
| Frequency | $50H_z$ |
| Pressure ratio | 15.8 |
| Compressor inlet temperature | $273.15K$ |
| Turbine inlet temperature | $1950K$ |
| Exhaust temperature | $872K$ |
| Power | $259.5MW$ |
| Thermal Efficiency | 37.3% |
| Heat rate | $9643\,KJ/Kwh$ |
| Air flow rate | $802\,kg/s$ |
| Steam turbine | |
| Manufacturer | Babcock |
| Model | D2248B |
| Steam flow rate | $70.74\,kg/s$ |
| Power | $120\,MW$ |
| Thermal efficiency | 42% |
| Inlet pressure | $80bar$ |
| Condenser pressure | $0.08bar$ |
| Inlet temperature | $500^0$C |
| Calculated parameters | |
| Steam enthalpy at state 6, $h_6$ | $174\,Kj/kg$ |
| Steam enthalpy at state 7, $h_7$ | $182.06\,Kj/kg$ |
| Steam enthalpy at state 8, $h_8$ | $3398\,Kj/kg$ |
| Steam enthalpy at state 9, $h_9$ | $2102.8\,Kj/kg$ |

## APPENDIX B

COGAS generated data in MATLAB for ANN training

| GT Compressor inlet Temperature(T1) | GT compressor inlet Pressure(P1) | GT pressure ratio (Pr) | GT inlet Temperature to the turbine(T3) | GT inlet Temperature to the turbine(ma) | GT fuel mass flow rate(mf) | ST steam mass flow rate(ms) | Enthalpy before entering the pump (h6) | Enthalpy after entering the pump(h7) | Enthalpy before the boiler (h8) | Enthalpy after the boiler(h9) | ST inlet temperature (T5) | ST boiler pressure(P5) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 253.15 | 1.01325 | 10.5 | 2010 | 800.9268 | 0.00367 | 0.75 | 174 | 182.06 | 3398 | 2102.8 | 500 | 80 |
| 253.225 | 1.05326925 | 10.5053 | 2010.05 | 800.9318 | 0.00393243 | 0.90925 | 174.02 | 182.07994 | 3398.201 | 2103 | 500.05 | 80.02 |
| 253.3 | 1.0932885 | 10.5106 | 2010.1 | 800.9368 | 0.00419486 | 1.0685 | 174.04 | 182.09988 | 3398.402 | 2103.2 | 500.1 | 80.04 |
| 253.375 | 1.13330775 | 10.5159 | 2010.15 | 800.9418 | 0.00445729 | 1.22775 | 174.06 | 182.11982 | 3398.603 | 2103.4 | 500.15 | 80.06 |
| 253.45 | 1.173327 | 10.5212 | 2010.2 | 800.9468 | 0.00471972 | 1.387 | 174.08 | 182.13976 | 3398.804 | 2103.6 | 500.2 | 80.08 |
| 253.525 | 1.21334625 | 10.5265 | 2010.25 | 800.9518 | 0.00498215 | 1.54625 | 174.1 | 182.1597 | 3399.005 | 2103.8 | 500.25 | 80.1 |
| 253.6 | 1.2533655 | 10.5318 | 2010.3 | 800.9568 | 0.00524458 | 1.7055 | 174.12 | 182.17964 | 3399.206 | 2104 | 500.3 | 80.12 |
| 253.675 | 1.29338475 | 10.5371 | 2010.35 | 800.9618 | 0.00550701 | 1.86475 | 174.14 | 182.19958 | 3399.407 | 2104.2 | 500.35 | 80.14 |
| 253.75 | 1.333404 | 10.5424 | 2010.4 | 800.9668 | 0.00576944 | 2.024 | 174.16 | 182.21952 | 3399.608 | 2104.4 | 500.4 | 80.16 |
| 253.825 | 1.37342325 | 10.5477 | 2010.45 | 800.9718 | 0.00603187 | 2.18325 | 174.18 | 182.23946 | 3399.809 | 2104.6 | 500.45 | 80.18 |
| 253.9 | 1.4134425 | 10.553 | 2010.5 | 800.9768 | 0.0062943 | 2.3425 | 174.2 | 182.2594 | 3400.01 | 2104.8 | 500.5 | 80.2 |
| 253.975 | 1.45346175 | 10.5583 | 2010.55 | 800.9818 | 0.00655673 | 2.50175 | 174.22 | 182.27934 | 3400.211 | 2105 | 500.55 | 80.22 |
| 254.05 | 1.493481 | 10.5636 | 2010.6 | 800.9868 | 0.00681916 | 2.661 | 174.24 | 182.29928 | 3400.412 | 2105.2 | 500.6 | 80.24 |
| 254.125 | 1.53350025 | 10.5689 | 2010.65 | 800.9918 | 0.00708159 | 2.82025 | 174.26 | 182.31922 | 3400.613 | 2105.4 | 500.65 | 80.26 |
| 254.2 | 1.5735195 | 10.5742 | 2010.7 | 800.9968 | 0.00734402 | 2.9795 | 174.28 | 182.33916 | 3400.814 | 2105.6 | 500.7 | 80.28 |
| 254.275 | 1.61353875 | 10.5795 | 2010.75 | 801.0018 | 0.00760645 | 3.13875 | 174.3 | 182.3591 | 3401.015 | 2105.8 | 500.75 | 80.3 |
| 254.35 | 1.653558 | 10.5848 | 2010.8 | 801.0068 | 0.00786888 | 3.298 | 174.32 | 182.37904 | 3401.216 | 2106 | 500.8 | 80.32 |
| 254.425 | 1.69357725 | 10.5901 | 2010.85 | 801.0118 | 0.00813131 | 3.45725 | 174.34 | 182.39898 | 3401.417 | 2106.2 | 500.85 | 80.34 |
| 254.5 | 1.7335965 | 10.5954 | 2010.9 | 801.0168 | 0.00839374 | 3.6165 | 174.36 | 182.41892 | 3401.618 | 2106.4 | 500.9 | 80.36 |
| 254.575 | 1.77361575 | 10.6007 | 2010.95 | 801.0218 | 0.00865617 | 3.77575 | 174.38 | 182.43886 | 3401.819 | 2106.6 | 500.95 | 80.38 |
| 254.65 | 1.813635 | 10.606 | 2011 | 801.0268 | 0.0089186 | 3.935 | 174.4 | 182.4588 | 3402.02 | 2106.8 | 501 | 80.4 |
| 254.725 | 1.85365425 | 10.6113 | 2011.05 | 801.0318 | 0.00918103 | 4.09425 | 174.42 | 182.47874 | 3402.221 | 2107 | 501.05 | 80.42 |
| 254.8 | 1.8936735 | 10.6166 | 2011.1 | 801.0368 | 0.00944346 | 4.2535 | 174.44 | 182.49868 | 3402.422 | 2107.2 | 501.1 | 80.44 |
| 254.875 | 1.93369275 | 10.6219 | 2011.15 | 801.0418 | 0.00970589 | 4.41275 | 174.46 | 182.51862 | 3402.623 | 2107.4 | 501.15 | 80.46 |
| 254.95 | 1.973712 | 10.6272 | 2011.2 | 801.0468 | 0.00996832 | 4.572 | 174.48 | 182.53856 | 3402.824 | 2107.6 | 501.2 | 80.48 |
| 255.025 | 2.01373125 | 10.6325 | 2011.25 | 801.0518 | 0.01023075 | 4.73125 | 174.5 | 182.5585 | 3403.025 | 2107.8 | 501.25 | 80.5 |

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