ANALYSIS OF STEAM RECUPERATIVE SYSTEM TO COGAS PLANT

E. A. Ogbonnaya and H. U. Ugwu
Department of Mechanical Engineering, Michael Okpara University of Agriculture, Umudike-Umuahia, Nigeria
E-Mail: ezenwaogbonnaya@yahoo.com

ABSTRACT

In a bid to recover lost energy in the gas turbine and improve efficiency, many modifications to power plants have been made. The COGAS plant is one of such modifications to existing power plants. Due to the high efficiency obtained from such plants theoretically, it is seen that the COGAS plant is of immense benefit for practical applications. However, there exists no chart or table that predicts the efficiency of the COGAS plant at any operating condition since two COGAS plants of the same specification can provide different efficiencies when operated in different regions or places e.g. (Canada in winter and Nigeria in Harmattan). Thus basic parameters of the COGAS plant were examined as to further understand their effect and enhance efficiency. This analysis was achieved using VB.Net computer programming language and data collected from gas and steam turbines on industrial duty for electricity generation. The result is a table similar to the presently existing thermodynamic steam table which will assist designers and engineers to pick optimum parameters with a targeted efficiency and place of operation in mind.

Keywords: COGAS, steam recuperative system, gas turbine, steam turbine, efficiency.

1. INTRODUCTION

The continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants. The difficulties involved in the Carnot cycle has made designers and engineers seek better practicable and more efficient power plants.

The binary vapour cycle is one of such modifications of power plants developed. With the exception of a few specialized applications the working fluid predominantly used in a vapour power cycle is water. Water is the best working fluid presently available, but it is far from being the ideal one. The binary cycle is an attempt to overcome some of the short comings of water and to approach the ideal working fluid by using two fluids (Yunus, *et al*., 1998).

As much as we cannot change the behavior of water during the high temperature part of the cycle, we can certainly replace it with a more suitable fluid. The result is a power cycle that is actually a combination of two cycles, one in the high-temperature region and the other in the low temperature region. The high-temperature cycle is also referred to as toping cycle while the low-temperature cycle is called the bottoming cycle. The COGAS plant is a combination of a gas turbine (Brayton cycle) topping a steam turbine (Rankine cycle).

The combined-cycle unit combines the gas turbine (Brayton cycle) and the steam turbine (Rankine cycle) by using heat recovery boilers to capture the energy in the gas turbine exhaust gas for steam production to supply a steam turbine. This is shown in Appendix 2.

In a gas turbine, the sole aim of the plant of the system is to convert a portion of the heat transferred to the working fluid to work, which is the most valuable form of energy. The remaining portion of the heat is rejected to rivers, lakes, oceans, or to the atmosphere as waste heat. Wasting a large amount of heat is a price to pay to produce work, because electrical or mechanical work is the only form of energy on which many engineering devices can operate (Yunus, A. C. *et al*., 1998).

Gas turbine cycles typically operate at considerably higher temperatures than steam cycles. As a result, the waste heat can be recovered for use by a steam turbine plant. It is noteworthy that by so-doing, both plants are therefore doing work from only one source of heat supply (in the gas turbine combustion chamber) to increase the work output as well as the thermal efficiency.

The maximum fluid temperature at the turbine inlet is about 620^0C for modern steam power plants, but over 1150^0C for gas-turbine power plants. The use of higher temperatures in gas turbines is made possible by recent developments. Cooling the turbine blades and coating the blades with high temperature-resistant materials such as ceramics is one of such developments. Due to the higher average temperature at which heat is supplied, gas-turbine cycles have a greater potential for higher thermal efficiencies. However the simple gas turbine suffers from relatively low cycle efficiency since the exhaust gases leave the gas turbine at a very high temperature (usually above 500^0C) which wipes out any potential gains in thermal efficiency. Consequently, the thermal efficiency of gas turbine plants, in general, is relatively low.

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. A combination that is achieved by the use of steam recuperative system, and offers higher cycle efficiency than both gas and steam turbines run independently.

There are many variables in the design of the COGAS system. The design process involves finding the required parameters that maximizes the cycle efficiency of interest while keeping the place of operation in mind. Since the conditions of operation of the COGAS plant plays a significant role to the overall efficiency of the system, this paper thus provides designers with a more easy and reliable means of picking optimal design parameters for a targeted efficiency thus making the design process easy and prompt.

1.1 COGAS history

COGAS plants find its operation both in marine propulsion and electricity generation. Though the use of COGAS plants for ship propulsion is just finding its way into the maritime sector, this power plant option has been used ashore since the late 1970s.

In 1985, a 1090-MW Tohoku COGAS plant that was put in commercial operation in Niigata, Japan, is reported to operate at a thermal efficiency of 44%. The plant has two 191 MW steam turbines and six 118 -MW gas turbine. Hot combustion gases enter the gas turbine at 1154^{0}C, and steam enters the steam turbines at 500^{0}C. Steam is cooled in the condenser by cooling water at an average temperature of 15^{0}C. The compressor, have a pressure ratio of 14, and the mass flow rate of air is 443 Kg/s. The Nigashi Niigata combined cycle Gas turbine (CCGT) power plant Japan is a modification of the Tohoku COGAS plant. An anti-fouling system has been added to the 1610MW CCGT plant at Higashi Niigata, Japan. This plant, which increased the stations total capacity to 6000MW, was Japan's first large CCGT project and is operated by Tohoku Electric Power Company. The power station was the first to be equipped with a new generation of Mitsubishi gas turbines. Before the turbine began commercial operation in 1999, the plant had a capacity of 2990MW. The existing installation comprised two 350MW steam turbines (Minato Unit one and two), two 600MW steam turbines (units 1 and 2) and a 1090 MW CCGT (unit 3). Unit 3 was completed in 1984; it has six MW701D gas turbine (Higashi Niigata Gas Turbines, 2004).

The Higashi Niigata unit 4 plant comprises two power trains, each with two 270MW Mitsubishi 701GI turbines. The turbines are operated with a reduced inlet temperature of 1400^{0}C (design rating is 308MW at 1500^{0}C). Each gas turbine has a heat recovery steam generator (HRSG) with a reheat temperature of 566^{o}C. The steam from each pair of Heat Recovery Steam Generator (HRSGs) drives a 265MW two-cylinder steam turbine generator.

The overall efficiency is about 50%. This can be compared with the performance of the older unit 3 plant, which operates with a combined -cycle efficiency of 44%. In single-cycle operations the 70IG has a thermal efficiency of 38% compared with 37% for its predecessor. Mitsubishi claims that the 701G Gas Turbine as shown in Figure-1 is capable of offering better than 58% efficiency in combined cycle plant. Tohoku expects that the new plant will deliver savings of around 180,000 tons of Liquefied Natural Gas (LNG) per year compared to the old plant. (Higashi Niigata Gas Turbines, 2004).

In another case, a 1350-MW combined cycle plant built in Ambarli, Turkey; in 1988 by siemens of Germany is the first commercially operating thermal plant in the world to attain an efficiency level as high as 52.5% at design operating conditions. This plant has six 150-MW gas turbines and three 173-MW steam turbines. Some recent COGAS power plant has achieved efficiencies above 60%. (Yunus, A. C. *et al*., 1998)

Recent developments in gas-turbine technology have made the combine gas-steam cycle (COGAS) economically very attractive (New Marine Digest, 2006). According to Nkoi (2007), the combined cycle increases the efficiency without appreciable increasing the initial cost. As a result, many new power plants operate on combined cycles, and many more existing steam or gas turbine plants are being converted to combined-cycle power plants. These conversions and the COGAS plant show the usefulness of the application of steam recuperative systems as the COGAS plant cannot exist without a meaningful means of heat recovery.

2. COGAS SYSTEM DESCRIPTION

The schematic diagram of a combined-cycle unit which combines the gas turbine (Brayton cycle) and the steam turbine (Rankine cycle) by using available means of steam recuperation to capture the energy in the gas turbine exhaust gas for steam production to supply a steam turbine is shown in Appendix 2.

From the outline, it can be seen that a COGAS plant comprises the operation of the gas turbine, steam turbine and the steam recuperative system. These will now be analyzed separately.

2.1 Gas turbine operation

The simple gas turbine plant operates on a Brayton cycle. The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today, it is used for gas turbines only where both the compression and expansion processes take place in rotating machinery.

Gas turbines usually operate on an open cycle, with four basic processes as shown in Figures 1, 2 and 3 below. (Ogbannya, E. A., 2004).

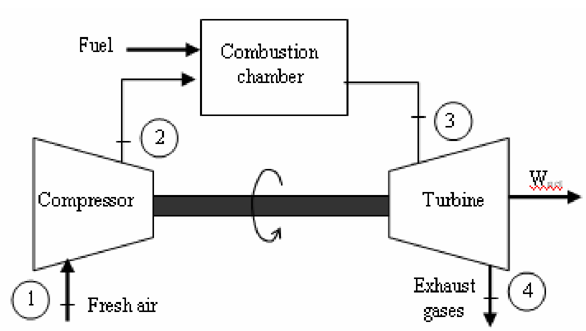


Figure-1. Open cycle gas turbine engine.

Process 1-2: Isentropic compression and takes place in the compressor. Air at ambient conditions is draw into the compressor, where its temperature and pressures is raised.

ARPN Journal of Engineering and Applied Sciences

Process 2-3: Constant pressure heat addition, takes place in the combustion chamber. Here the high-pressure air proceeds to the combustion chamber, where the fuel is burned at constant pressure.

Process 3-4: Isentropic expansion, takes place in the turbine. Here the resulting high-temperature gases enter the turbine, where they expand to the atmospheric pressure, thus producing power.

Process 4-1: Constant pressure heat rejection. The exhaust gases leaving the turbines are thrown out and not reticulated, causing the cycle to be classified as an open cycle.

On a unit mass basis, the energy balance will be analyzed as steady-flow processes, since all four processes of the Brayton cycle are executed in steady flow devices. (Oteh, 2006).

Now for the processes

Compressor work = h2 - h1 = cp (T2-T1) (1)

Turbine work = h3 - h4 = cp (T3-T4) (2)

Heat supplied = h3 - h2 = cp (T3-T2) (3)

Heat rejected = h4 - h1 = cp (T4-T2) (4)

$$\text{Thermal efficiency} = \frac{\text{Net work energy transfered}}{\text{Heat energy supplied}} \tag{5}$$

$$\frac{\text{work of Turbine} + \text{work of compressor}}{\text{Heat energy supplied}} \tag{6}$$

$$= \frac{c_p\left(T_3 - T_4\right) + c_p\left(T_2 - T_1\right)}{c_p\left(T_3 - T_2\right)} \tag{7}$$

Re-arranging, this becomes

$$= \frac{c_p\left(T_3 - T_2\right) - c_p\left(T_4 - T_1\right)}{c_p\left(T_3 - T_2\right)} \tag{8}$$

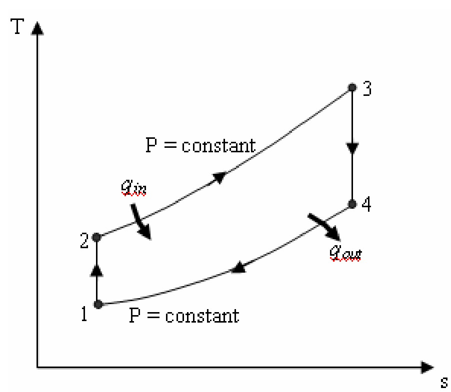


Figure-2. T-S diagram of Brayton cycle.

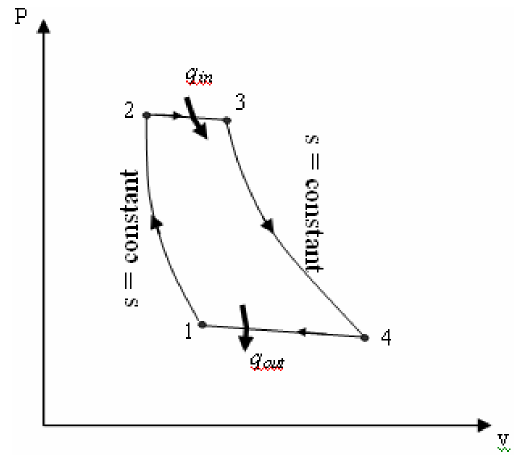


Figure-3. P-V diagram of Brayton cycle.

Further analysis of the gas turbine calculation is shown in Appendix 1.

2.2 Steam turbine operation

The steam turbine plant operates on the Rankine cycle. In this cycle, steam enters the turbine and is expanded until it leaves the turbine as wet steam, which is then condensed to saturated water. The water is displaced from the condenser into the boiler by a pump, which raises its pressure from condenser value to that of the boiler. This is shown in Figures 4 and 5 below:

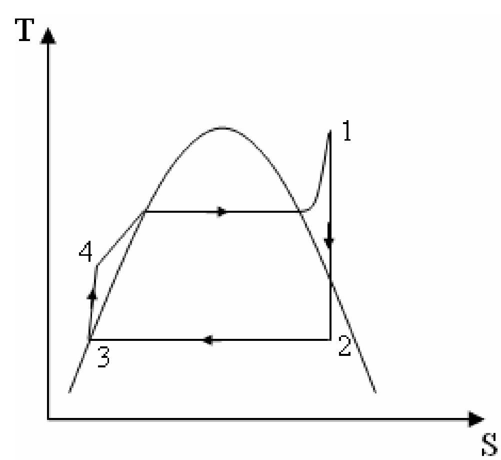


Figure-4. T - S diagram of Rankine cycle.

Heat transfer in boiler = $h_1 - h_4$ (9)

Heat transfer in condenser = $h_2 - h_3$ (10)

Work output of turbine = $h_1 - h_2$ (11)

Work input to pump = $V_f\,(P_4\text{-}P_3)$ x 10^2

= h_4 - h_3 (12)

Network of cycle

= Turbine work - Pump work (13)

$$\text{Cycle efficiency} = \frac{\text{Network}}{\text{Heat supplied}} \qquad (14)$$

$$= \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} \qquad (15)$$

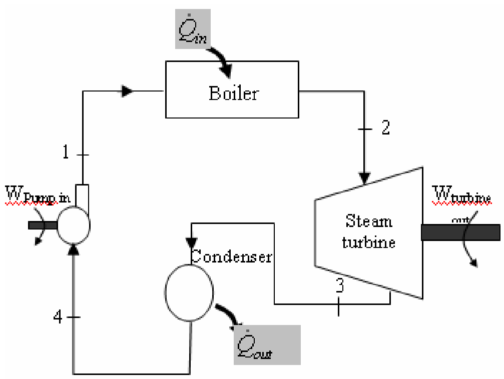


Figure-5. Rankine cycle plant.

2.3 COGAS plant operation

COGAS plant uses heat from the exhaust gas of the gas turbine to heat water through a series of tubes I a heat exchanger under high pressure to become superheated steam (Figure-1). 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.

2.4 The recuperative system

There are various ways by which steam can be recuperated for use in a combined cycle operation. Of the various ways, only the heat exchanger was used in this analysis.

Heat exchangers are devices that provide the flow of thermal energy between two or more fluids at different temperatures (In this case between the exhaust gases of the gas turbine and the water from the feed of the steam turbine). The heat transfer occurs through a separating wall or through the interface between the streams.

3. THE COMPUTER MODEL

VB.net computer programming language was used for the analysis of the COGAS plant using data collected from gas and steam turbines on industrial duty for electricity production. The data and results which corresponded with theoretical estimations were used as a reference point from which the values of other variables were altered at regular intervals and the resulting results were tabulated in the form of the presently available thermodynamic steam table.

The computer flowchart developed for the program used in this analysis is shown in Figure-6.

3.1 Analysis of results

From the analysis obtained from the computer model, the following graphs were plotted to demonstrate the behavior of COGAS efficiency. Figure-7 shows the graph of COGAS efficiency versus compressor inlet temperature. It was observed that as the compressor inlet temperature increased, the thermal efficiency of the COGAS plant reduced. This shows that the relationship between the compressor inlet temperature and the COGAS efficiency is that of inverse proportionality.

In Figure-8, a graph of COGAS efficiency was plotted against the turbine inlet temperature. In this case, as the turbine inlet temperature increased, there was a corresponding increase in the efficiency. This shows that the relationship between them is that of direct proportionality.

In Figure-9, the COGAS efficiency was plotted against the enthalpy at the superheated temperature. It was observed that there was a corresponding increase in the efficiency as the enthalpy at the superheated temperature increased.

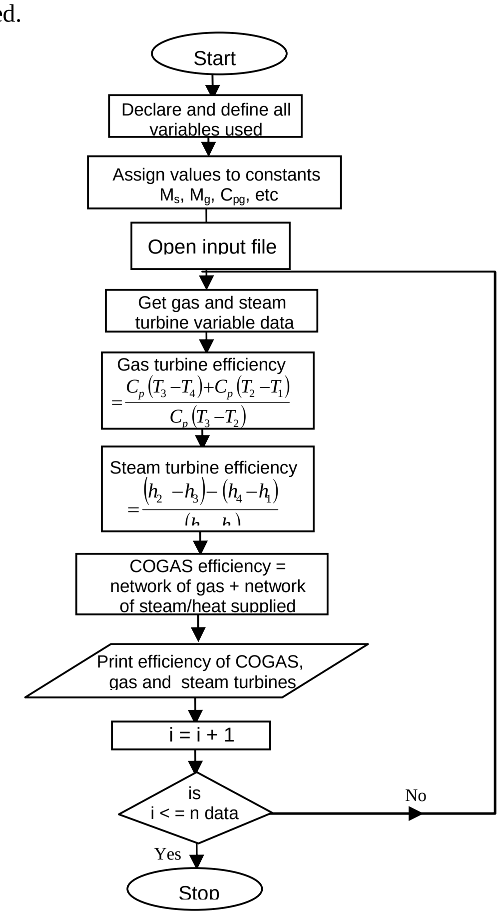


Figure-6. Flowchart to calculate the efficiency of the COGAS plant.

VOL. 7, NO. 5, MAY 2012 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

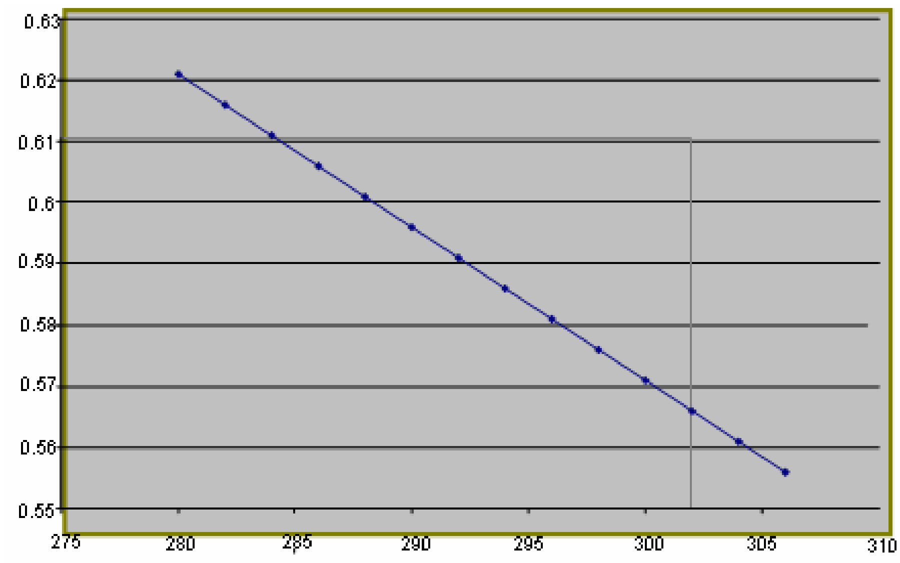


Figure-7. Efficiency of COGAS plant versus compressor inlet temperature.

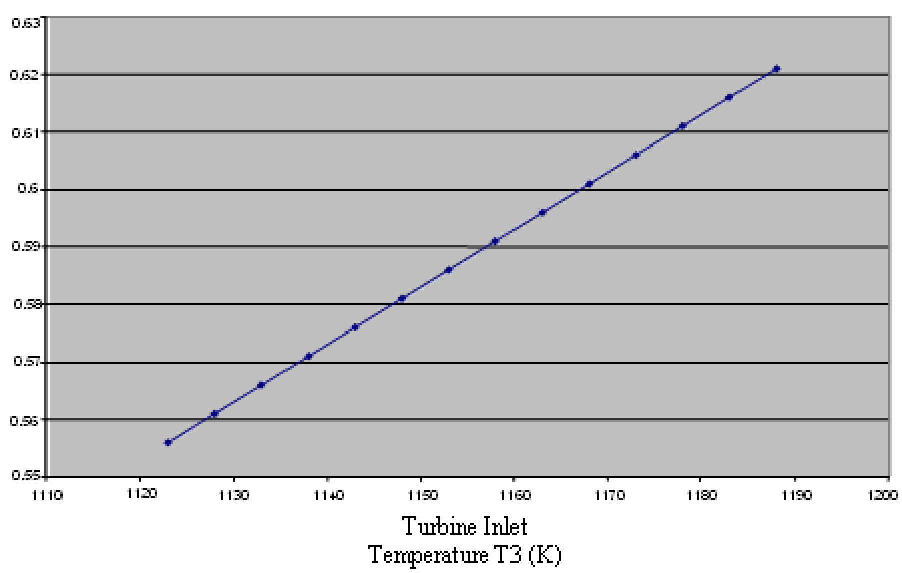


Figure-8. Efficiency of COGAS plant versus turbine inlet temperature, T3.

VOL. 7, NO. 5, MAY 2012 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

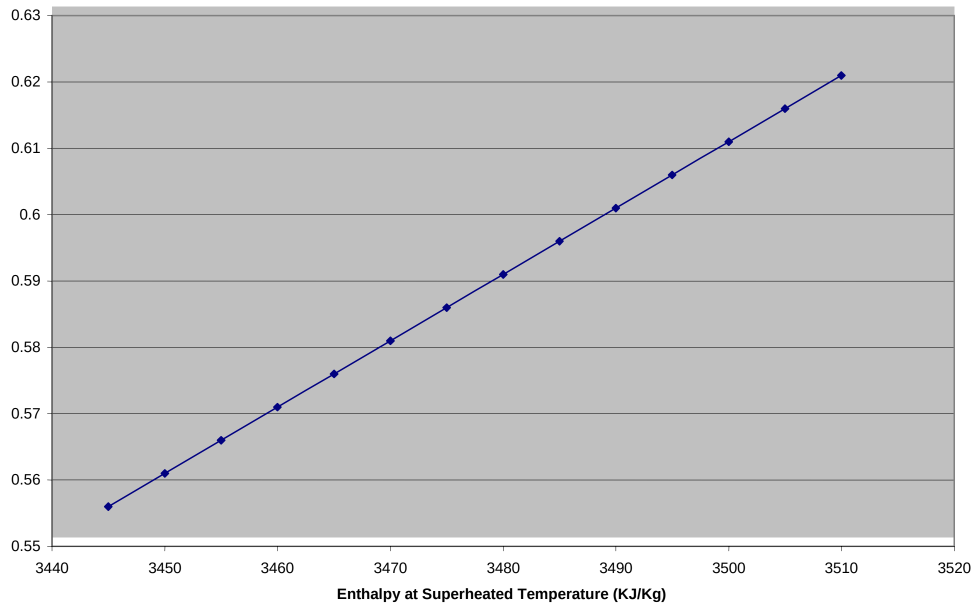


Figure-9. Efficiency of COGAS plant versus enthalpy at superheated temperature (kJ/kg).

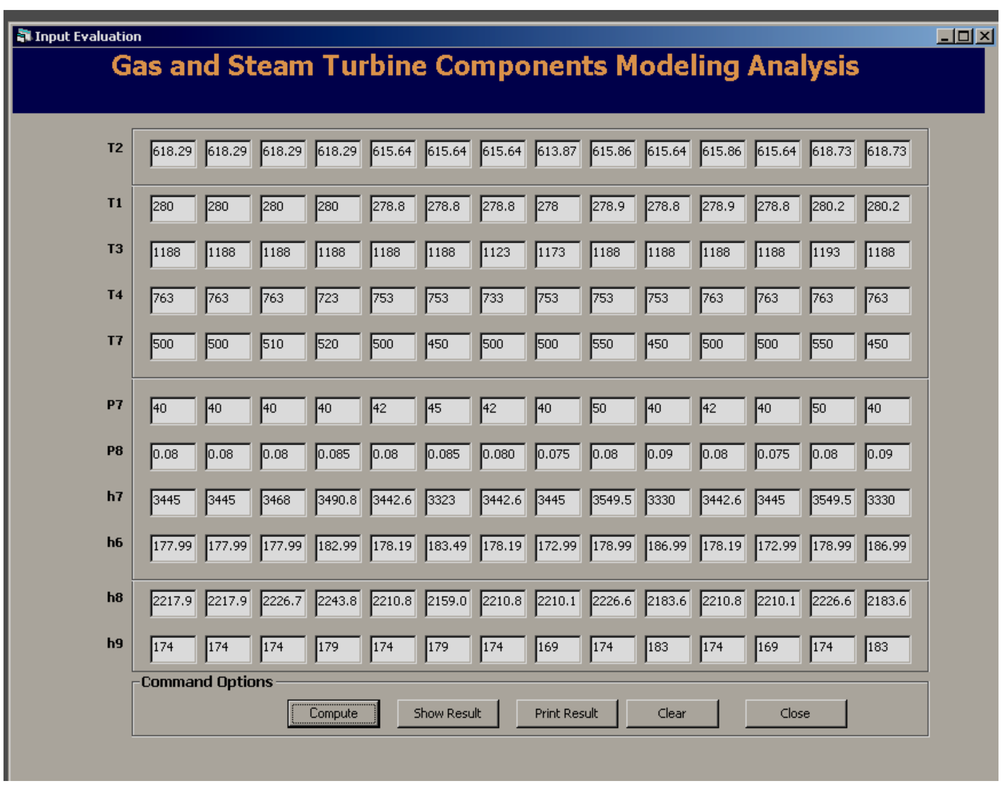


Figure-10. Screen capture of a typical computer print out during COGAS operations.

4. CONCLUSIONS

This analysis is important as satisfactory performance of the COGAS plants depend greatly on the thermodynamic properties and efficiency of the systems. Operators and maintainers must therefore understand the thermodynamic behavior of the plant, what properties and parameters affect the performance of the associated systems and how changes in these parameters affect the thermal efficiency of the system (Dumpe, 2008). To produce the chart that looked like the recent thermodynamic table, the present state of the data was varied using an interval of ten for both temperature and pressure, keeping the mass flow rate constant. The corresponding dryness fraction of the steam, work output and efficiency was recorded against that value. For intermediary pressures and temperatures, linear interpolation will suffice to get the direct value.

REFERENCES

Anderson C. 2008. Planet in Peril. CNN Special Edition. (2008, May 14, last update) [ONLINE] Available at: www.cnn.com/planetinperil.html.

Brady E. F. 1981. Energy conservation for Propulsion of Naval Vessels. Naval Engineers Journal. April. p. 133.

Dumpe B. 2008. Study of the Application of Steam Recuperative System to Combined Cycles. B. Tech. Marine Engineering Thesis, Faculty of Engineering, Rivers State University of Science and Technology, Port Harcourt.

Combs R. M. 1979. Waste Heat Recovery Unit Design for Gas Turbine Propulsion System. U. S Naval Postgraduate School, also available at Defense Technical Information Center Technical Report. 80-20599.

Engineering, Procurement and Construction Services (EPC): Combined Cycle Power Plants (2007, May 14, Last update). [Online] Available at: www.combinedcyclespowerplants.com.

Futtsu - TEPCO's Power City. Diesel and Gas Turbine Worldwide. (May, 2007). p. 20.

Giblon R. P. and Rolih H. 1979. COGAS: Marine power plants for energy savings. Marine Technology. p. 223.

Howard J. L. and Kvamsdal R. S. 1982. Energy Efficient LNG Carriers. Transaction of the 1982 Ship Cost and Energy Symposium.

Halkola J. T., Campbell A. H. and Jung D. 1983. Energy of the Transactions of the ASME. Journal of Engineering for Power. 105: 21.

Harbach J. A. 1988. Optimization of Exhaust Gas Turbo-Generator Systems using TK Solver". Third Chesapeake Marine Engineering Symposium.

Higashi Niigata Gas Turbines (2004, June 28, last update) [ONLINE] Available at: www.power-technology.com/project/higashi [2008, March 20]

Marine Engineering and Offshore. 2006. Diesel Engines versus Gas Turbines. "New Marine Digest". Journal of the Institute of Marine Engineering, Science and Technology, Nigeria Branch. pp. 25-27.

Mills R. G. 1977. Greater Ship Capability with Combined -Cycle Machinery. Naval Engineers Journal. October 1977. p. 17.

Nkoi B. 2007. Design for Combined-Cycle Operation of AFAM V Power Station. M. Tech. Mechanical Engineering Thesis, Faculty of Engineering, Rivers State University of Science and Technology, Port Harcourt, Nigeria.

Ogbonnaya E. A. 2004. Thermodynamics of Steam and Gas Turbines. 1st Ed. Oru's Press Ltd. Port Harcourt. pp. 71-73.

Ogbonnaya E. A. and Koumako K.E.E. 2006. Basic Automatic Control. 1st Ed. King Jovic Int'l Press Port Harcourt. p. 51.

Pinch Technology (2002, March 15, last update) [ONLINE] Available at: www.cheresources.com/pinchtech4.shtml.

Oteh U. 2006. Engineering Thermodynamics. 1st Ed. Author House™. pp. 34-45.

Cengel Y. A. and Boles A. M. 1998. Thermodynamics: An Engineering Approach. 3rd Ed. McGraw-Hill. p. 589.

Sadik K. and Horigtan L. 2002. Heat Exchangers: Selection, Rating and Thermal design. 2nd Ed. CRC Press. [ONLINE] Available at: http://books.google.com.ng/book?id=qiwufokpbpic.

Wiggins E. G. 2008. COGAS Propulsion for LNG Ships. Transactions of the Society of Naval Architects and Marine Engineers.

VOL. 7, NO. 5, MAY 2012 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

Appendix-1

Gas Turbine Calculation
Compressor work =

h_2 - h_1 = cp (T_2-T_1) (1)

Turbine work =

h_3 - h_4 = cp (T_3-T_4) (2)

Heat supplied =

h_3 - h_2 = cp (T_3-T_2) (3)

Heat rejected =

h_4 - h_1 = cp (T_4-T_2) (4)

Thermal efficiency =

$$\frac{\text{Net work energy transfered}}{\text{Heat energy supplied}} \quad (5)$$

$$\frac{\text{workof Turbine+ workof compressor}}{\text{Heatenergysupplied}} \quad (6)$$

$$= \frac{c_p\left(T_3 - T_4\right) + c_p\left(T_2 - T_1\right)}{c_p\left(T_3 - T_2\right)} \quad (7)$$

Re-arranging, this becomes

$$= \frac{c_p\left(T_3 - T_2\right) - c_p\left(T_4 - T_1\right)}{c_p\left(T_3 - T_2\right)} \quad (8)$$

Simplifying, gives;

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2} \quad (9)$$

Processes 1-2 and 2- 4 are Isentropic, and P2 = P3 and P4 = P1
thus;

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{(r-1)}{\gamma}} \quad and \quad \frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{(r-1)}{\gamma}} \quad (10)$$

but $P_2 / P_1 = r_P$

thus, $(r-1) / \gamma$

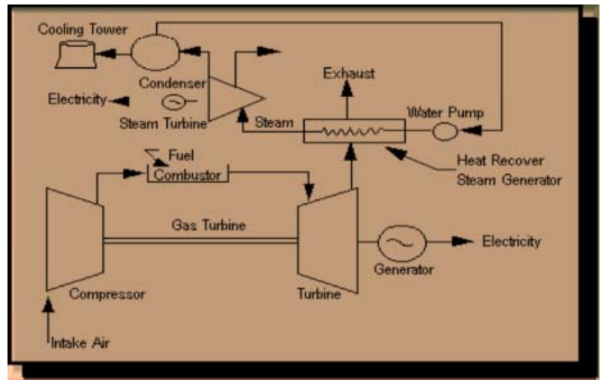
$T_2 = T_1 \, r_P^{(r-1)/r}$ and $T_3 = T_4 r_P^{(r-1)/r}$

Substituting into equation 9 and simplifying gives,

$$\eta = 1 - \frac{1}{r_p^{\left(\frac{r-1}{r}\right)}} \quad (11)$$

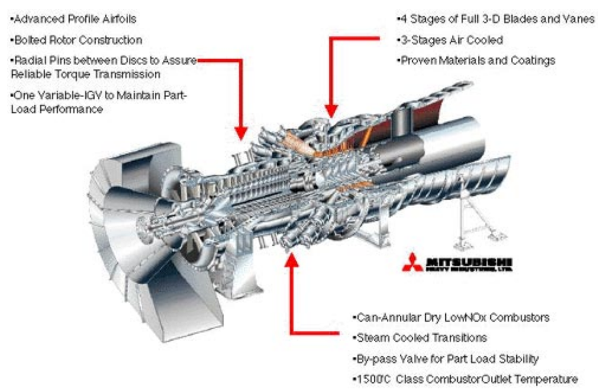
where rp is the pressure ratio and r is the specific heat ratio.

Appendix-2



Combined steam and gas turbine plant.

Appendix-3



MHI'S 270MW -rated model 701GI Gas turbines.