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International Journal of Current Research
Vol. 6, Issue, 03, pp.5459-5463, March, 2014

INTERNATIONAL JOURNAL OF CURRENT RESEARCH

ISSN: 0975-833X

RESEARCH ARTICLE

THERMAL ANALYSIS OF A GAS TURBINE POWER PLANT AT URAN, INDIA

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ARTICLE INFO

Article History:
Received 19th December, 2013
Received in revised form
30th January, 2014
Accepted 04th February, 2014
Published online 25th March, 2014

Key words:
Gas Turbine, Compressor ratio,
Thermal Efficiency,
Turbine Inlet Temperature.

ABSTRACT

In the present work the thermal performance analysis of a simple open cycle gas turbine power plant which is situated at Uran in district Raigad of Maharashtra, India is carried out at varying ambient temperature and pressure ratio and a comparison is made from the stipulated ISO conditions. The analysis of the cycle has been carried out using software developed in C++. The results show that for every 5^0C rise in ambient temperature the gas turbine loses 0.44 % in terms of thermal efficiency and 0.5 MW of its useful power. Also there is a loss of 3.36 % in thermal efficiency at pressure ratio 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 3.89 % at pressure ratio 21, which reveals that, as pressure ratio increases percentage loss in thermal efficiency increases on increasing the ambient temperature. The thermal performance analysis reveals that, the ambient temperature and compression ratios are strongly influence the performance of gas turbine power plant.

INTRODUCTION

The gas turbine prime movers were first used in 1919 for large central station service. Since then several stations have been built with gas turbine to drive electric generators. This is due to some inherent advantages of the gas turbine, such as, simplicity and flexibility of design and installation, compactness, low first cost, small building space requirement, little cooling water requirement, etc. The delivery and installation time for these plans is much less than for steam plants. These prime movers can be started quickly and can be put on load within a few minutes. Efficiency can be improved considerably of employing heat reclaiming devices. However, fuel costs in these plants are usually higher than those in other plants, though maintenance costs are lower than these costs for diesel plants. Since the fuel costs are relatively higher and initial cost lower, these plants are well suited for meeting peak load demands. The Gas Turbine power plant works on a Joule-Brayton cycle (Zhang *et al.*, 2009). The use of heat sinks in the basic Joule-Brayton cycle in order to exploit its available heat sources leads to a more advanced mixed (auto-combined) cycles (Lazzaretto and Manete 2009). The gas turbine is a complex machine, and its performance and reliability are governed by many standards. The American Society of Mechanical Engineers (ASME) performance test codes (American Society of Mechanical Engineers, 2005) have been written to ensure that test are conducted in a manner that guarantees that all turbines are tested under the same set of rules and conditions to ensure that the test results can be compared in a judicious manner. The reliability of the turbines depends on the mechanical codes that govern the design of gas turbines (American Society of Mechanical Engineers, 2005). The mechanical standards and codes have been written by both ASME and the American Petroleum Institute (API) amongst others.

The performance of the gas turbine is reliant on the efficiency achieved at the compressor of the turbine. Hot air, being less dense, de-rates the gas turbine's performance (Basrawi *et al.*, 2011). In case studies carried out previously, the effect of ambient temperature on electricity production and fuel consumption of a simple cycle plant has been documented at temperatures closer to ISO conditions in Turkey (Erden and Sevelgen 2006). Further, the performance improvement of the gas turbine is dependent on the maximum temperature tolerance of the first stage blades and is also reliant on inter stage cooling at the compression stage (Carniere *et al.*, 2006). Several methods and technologies are available to augment this power loss but this entails additional plant and equipment installation as well as additional operational requirements (Ibrahim *et al.*, 2010). Many of these methods such as use of air cooler (AlHazmy and Najjar 2004), regenerative steam injection (Nishada *et al.*, 2005), effusive blade cooling techniques (Cerri *et al.*, 2007; Rahman *et al.*, 2010), use of desiccant-based evaporative cooling (Zadpoor and Golshan 2006) or absorption chillers (Ameri and Hejazi 2004) are commonplace. The effect of relative humidity (Mathioudakis and Salavoutas 2001) on the gas turbine power plant addresses

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issues of the air cooling (Amell and Cadaies 2002; Sephehr and Majtaba 2010; Jean and Francoise 2009) and enhances compressor efficiency. However, humidity prior to filtration system imposes a penalty on gas turbine performance. Analytical methods have been researched for evaluation of gas turbine performance when subjected to inlet air cooling in combined cycle power plant (Yang *et al.*, 2009). The effects, whether positive or detrimental on the gas turbine compressor and engine performance, as well as their operability to use of water cooling techniques for inlet air can be effectively assessed for their merits (Roumeliotis and Mathioudakis 2010). Also, analytical studies are performed to confirm that increasing the turbine inlet temperature no longer means an increase in cycle efficiency, but increases the work. When applied with intercooled gas turbines, these studies have shown that increasing turbine inlet temperature and pressure ratio can still improve the performance of the intercooled gas turbine (Ibrahim *et al.,* 2010). However, such use of additional plant is seldom encountered in desert conditions, primarily due to the high cost of such application and its maintenance as water for such application needs to be specially generated using desalination technologies, which are high cost applications.

Methodology

Reversible simple (Joule-Brayton) cycle

A simple OCGT power plant has been considered for the present study and analysis.gas turbine power plant consists of four components, compressor, combustion chamber, turbine and generator. T-s diagram for an actual Bryton cycle is shown in fig1. The pressure loss in the combustion chamber is represented by $p_2 - p_3$ and the exhaust hood $p_4 - p_1$. In this cycle,

1-2 is isentropic compression.
1-2' is actual compression.
3-4 is isentropic expansion.
3-4' is actual expansion.

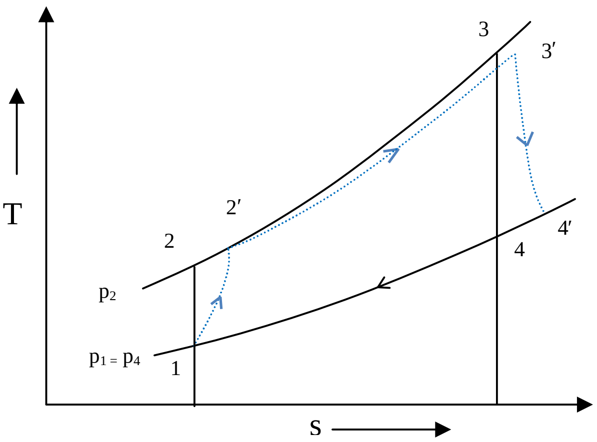

Fig 1 Actual gas turbine cycle representation on T-s chart

For compressor

The compressor efficiency, also known as isentropic compressor efficiency, $\boldsymbol{\eta_c}$ is-

The compressor efficiency $\eta_c = \frac{\text{Isentropic compression vn work}}{\text{Actual compression wowork}} = \frac{w_c}{w_{ca}}$

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1} \tag{1}$$

For compression process 1-2, we have

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}$$

where γ= 1.4 for air

$$T_2 = T_1 (r_p)^{(\gamma-1)/\gamma}$$

where r_p(pressure ratio) $= \frac{p_2}{p_1}$

$$T_2' = T_1 + \frac{T_2 - T_1}{\eta_C} \tag{2}$$

For compression process 1-2', we also have

$$\frac{T_2'}{T_1} = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma.\eta_{pc}} \tag{3}$$

For combustion chamber

For combustion process 2-3, we have

$$T_3 = \frac{\left((\text{C.V.}) \times \eta_{comb} + c_{pa} \times A/F \times T_2'\right)}{(C_{pg}) \times (A/F+1)} \tag{4}$$

For turbine

The turbine efficiency,

$$\eta_t = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{w_{ta}}{w_t}$$

$$\eta_t = \frac{T_3 - T_4'}{T_3 - T_4} \tag{5}$$

For expansion process 3-4, we have

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{(\gamma-1)/\gamma}$$

where γ= 1.34 for gas

$$T_4' = T_3 - \eta_t (T_3 - T_4) \tag{6}$$

For expansion process 3-4', we also have

$$\frac{T_3}{T_4'} = \left(\frac{p_3}{p_4}\right)^{\{(\gamma-1).\eta_{pt}/\gamma\}} \tag{7}$$

For pressure calculations, we have

$$p_2 = p_3 = r_p \times p_1 \quad (8)$$

$$p'_3 = p_2 \times 0.97 \quad (9)$$

$$p_4 = p_1 \times 1.02 \quad (10)$$

The thermal efficiency

The thermal efficiency of the plant η_{tha},

$$\eta_{tha} = \frac{\{C_{pg}(T_3 - T'_4) - C_{pa}(T'_2 - T_1)\}}{(C_{pg}.T_3 - C_{pa}.T'_2)} \quad (11)$$

Net power

Total net power available = $w_{net} \times \dot{m}_f$ MW (12)

Specific fuel consumption

It is one of the most important parameters expressed in kg/kWh. Thus,

$$sfc = \frac{(3600).\dot{m}_f}{W_{net}} \frac{kg}{kWh} \quad (13)$$

The cycle work ratio

The work ratio, WR, is defined as the ratio of net work to the turbine work. Thus,

$$WR = \frac{W_{net}}{W_t} \quad (14)$$

It is clear that for a good gas turbine the work ratio should be high.

The cycle air rate

Air rate, AR, is defined as the air flow required per kWh output.

$$AR = \frac{3600}{W_{net}} \frac{\text{kg of air}}{kWh} \quad (15)$$

The reciprocal of air rate is termed as specific power. Air rate is actually the criterion of the size of the plant i.e. the lower the air rate the smaller the plant. From figure 3.1, process 1, 2, 3, 4 represents the ideal Brayton cycle, the thermal efficiency is given by, The thermal efficiency, $\eta_{th} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}$, where r_p = pressure ratio = $\frac{p_2}{p_1}$

Software

The analysis performance has been performed using software developed in C++ and used for the calculation of thermal efficiency and base load. The results of various parameters have been calculated and graphs have been plotted in Microsoft Excel 2007.

RESULTS AND DISCUSSION

In the following section the results of the present work have been plotted.

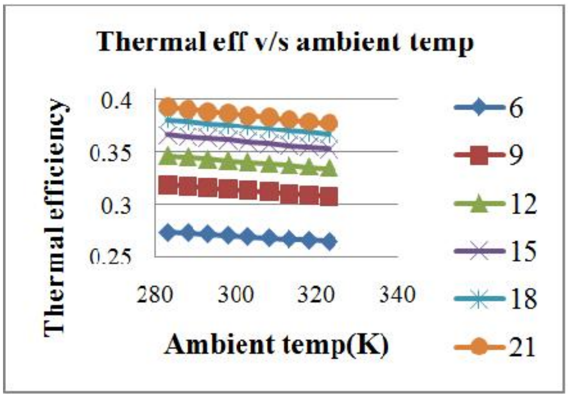

Fig 2 Variation of thermal efficiency and ambient temp at different pressure ratios

Fig 2 shows the variations of thermal efficiency and ambient temperature at different pressure ratios. It is evident from the Fig 2 that, as pressure ratio increases from 6 to 21 the thermal efficiency also increases for a given ambient temperature, but from the graph it also reflect that as ambient temperature increases from 283 K to 313 K, the thermal efficiency decreases for all ranges of pressure ratios under consideration. It is higher at lower temperatures but its value slightly decreases as temperature increases. There is a loss of 3.36 % in thermal efficiency at 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 3.89 % at 21 pressure ratio, which reveals that, as pressure ratio increases percentage loss in thermal efficiency increases on increasing the ambient temperature. The thermal efficiency is affected by ambient temperature due to the change of air density and compressor work. Since a lower ambient temperature leads to a higher air density and a lower compressor work that in turn gives a higher gas turbine output which in turns a higher thermal efficiency. But at higher ambient temperature leads to a lower air density and a higher compressor work. Here compressor work is more as compared to turbine work which in turns lower net work done, which leads to lower thermal efficiency at higher ambient temperatures.

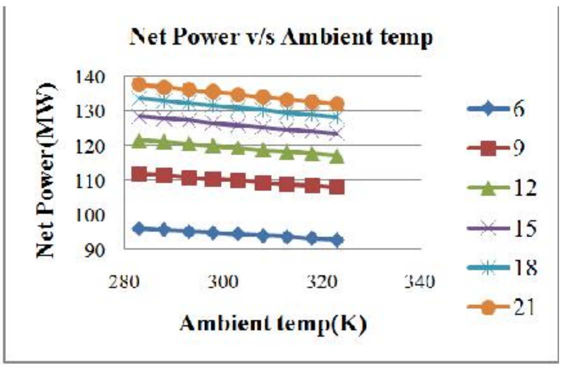

Fig 3 Variation of net power and ambient temp at different pressure ratios

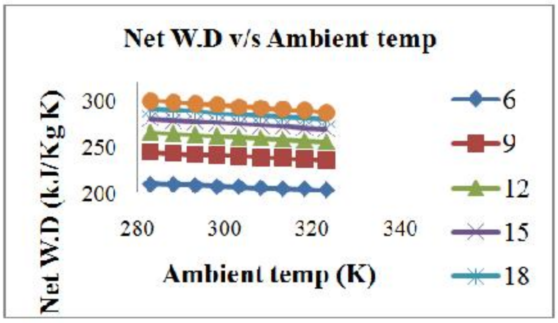

Fig 4 Variation of net work done and ambient temp at different pressure ratios

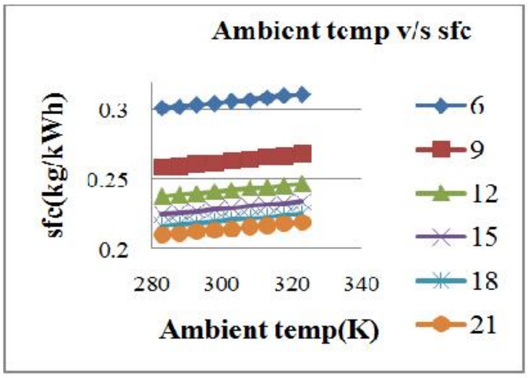

Fig 5 Variation of sfc and ambient temperature at different pressure ratios

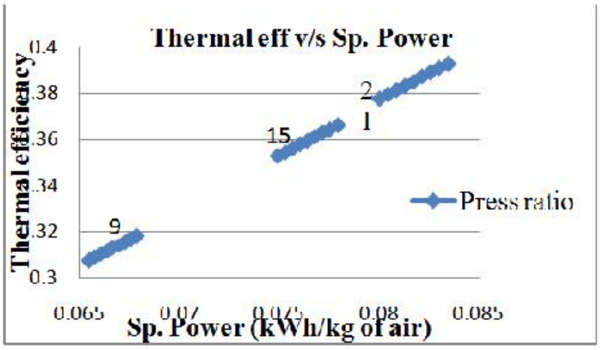

Fig 6 Variation of thermal efficiency and sp. power at different pressure ratios

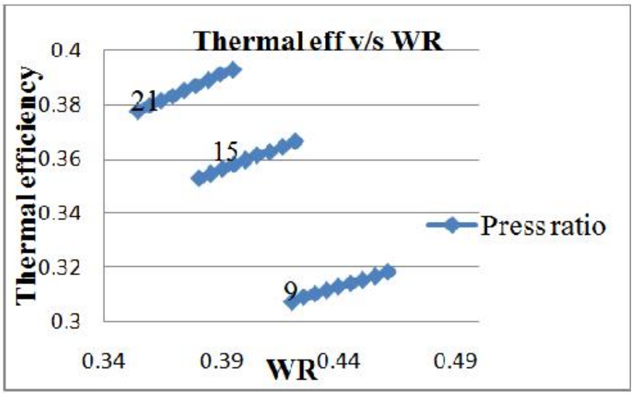

Fig 7 Variation of thermal efficiency and work ratio at different pressure ratios

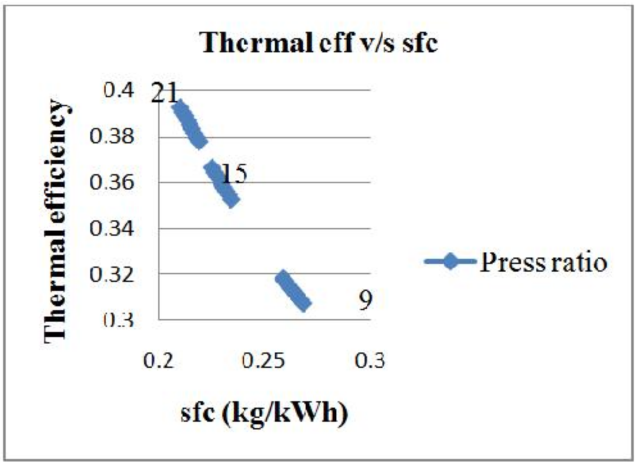

Fig 8 Variation of thermal efficiency and sfc at different pressure ratios

Fig 3 shows the variations of net power and ambient temperature at different pressure ratios. It is evident from the Fig 3 that as pressure ratio increases from 6 to 21 the net power also increases for a given ambient temperature, but from the graph it also reflects that as ambient temperature increases from 283 K to 313 K, the net power decreases for all ranges of pressure ratios under consideration. It is higher at lower temperatures but its value slightly decreases as temperature increases. There is a loss of 3.95 MW or 3.53 % in net power at 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 5.68 MW or 4.12 % at 21 pressure ratio, which reveals that, as pressure ratio increases percentage loss in net power increases as ambient temperature increases. Here we observe that loss of net power is more as compared to thermal efficiency at higher pressure ratios. Fig 4 shows variation of net work done and ambient temperature at different pressure ratios. It is evident from the fig 4 that as pressure ratio increases from 6 to 21 the net work done also increases for a given ambient temperature, but from the graph it also reflect that as ambient temperature increases from 283 K to 313 K, the net work done decreases for all range of pressure ratios under consideration. It is higher at lower temperatures but its value slightly decreases as temperature increases. There is a loss of 3.53 % in net work done at 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 4.13 % at 21 pressure ratio, which reveals that, as pressure ratio increases percentage loss in net work done increases as ambient temperature increases. Here we observe that loss of net work done is more as compared to thermal efficiency at higher pressure ratios.

Fig 5 shows variation of sfc and ambient temperature at different pressure ratios. There is a reverse phenomenon of sfc as compared to thermal efficiency and net power. Fig shows that as pressure ratio increases the value of sfc decreases but for a particular pressure ratio as ambient temperature increases value of sfc increases. This is because, the air mass flow rate inlet to compressor increases with increase of the ambient temperature. So, the fuel mass flow rate will increase, since air to fuel ratio is kept constant. The power increase is less than that of inlet compressor air mass flow rate, therefore the specific fuel consumption increases with the increase of ambient temperature. As the pressure ratio increases, the air exiting the compressor is hotter; therefore less fuel is required to reach the desired TIT and specific fuel consumption decreases as pressure ratio increases. Fig 6 shows variations of thermal efficiency and specific power at different pressure ratios. Fig shows that there is a linear relationship between thermal efficiency and specific power at all pressure ratios, as thermal efficiency increases specific power also increases and as thermal efficiency decreases specific power also decreases. This is due to the reason that thermal efficiency and specific power are directly proportional to net work done. Fig 7 shows variations of thermal efficiency and work ratio (WR) at different pressure ratios. As we know that the WR is the ratio of net work done to the turbine work. It is clear from earlier discussion that thermal efficiency increases with increase of pressure ratios at all ambient temps, but WR decreases as pressure ratio increases. This is due to the reason that turbine work increases more rapidly as compared to net work done as pressure ratio increases, which results in decrease of WR at higher pressure ratios. Fig 8 shows variations of thermal efficiency and specific fuel consumption (sfc) at different pressure ratios. Fig shows that as pressure ratio increases value of sfc decreases and as thermal efficiency increases sfc also decreases for all pressure ratios. So sfc decreases as pressure ratio increases.

Conclusion

The graphical representation of data as seen in figs, demonstrate that the gas turbine thermal efficiency and its useful power output varies with the ambient temperature. At higher ambient temperatures (than ISO conditions) the thermal efficiency and useful power output tend to be lower. The gas turbine inlet temperature being a limiting factor as dictated by the turbine blade metallurgy and mass flow of air being reduced at higher temperatures, hence it is observed that a turbine thermal efficiency de-rating upon rise in ambient temperature. As a direct consequence of this, the power generated by the gas turbine and supplied to the grid has a significant drop and therefore is a matter of concern.

REFERENCES

AlHazmy M.M., Y.S.H. Najjar, 2004. Augmentation of gas turbine performance using air coolers, *Applied Thermal Engineering* 24 (2-3) 415-429.

Amell A.A., F.J. Cadaies, 2002. Influence of the relative humidity on the air cooling thermal load in gas turbine power plant, *Applied Thermal Engineering* 22 (13) 1529-1533.

Ameri M., S.H. Hejazi, 2004. The study of capacity enhancement of the chabahar gas turbine installation using an absorption chiller, *Applied Thermal Engineering* 24 (1) 59-68.

American Society of Mechanical Engineers, 2005. Performance Test Code on Test Uncertainty: *Instruments and Apparatus* PTC 10, 19.1, 22, 46. ASME, New York,

Ashley De Sa, Sarim Al Zubaidim, 2011. “Gas turbine performance at varying ambient temperature”, *Applied thermal engineering* 31 2735-2739.

Basrawi F., T. Yomada, K. Nakanishi, S. Naing, 2011. Effect of ambient temperature on the performance of micro gas turbine with cogeneration system in cold region, *Applied Thermal Engineering* 31 (6-7) 105-1067.

Carniere H., A. Willocx, E. Dick, M De. Paepe, 2006. Raising cycle efficiency by inter cooling in air cooled gas turbine, *Applied Thermal Engineering* 26 (16) 1780-1787.

Cerri G., A. Giovannelli, L. Battisti, R. Fedrizzi, 2007. Advances in effusive cooling techniques of gas turbines, *Applied Thermal Engineering* 27 (4) 1529-1533.

Erden H.H., S.H. Sevelgen, 2006. Case study: effect of ambient temperature on the electricity production and fuel consumption of a simple cycle gas turbine in Turkey, *Applied Thermal Engineering* 26 (2-3) 320-326.

Ibrahim T.K., M.M. Rahman, A.N. Abdalla, 2010. Study on the effective parameter of gas turbine model with intercooled compression processes, *Academic Journals - Scientific Research and Essays*. ISSN: 1992-2248 5 (Suppl. 23) ISSN: 1992-2248 3760-3770.

Ibrahim T.K., M.M. Rahman, A.N. Abdalla, 2010. Study on the effective parameter of gas turbine model with intercooled compression process, Scientific Research and Essays. ISSN: 1992-2248 5 (23) ISSN: 1992-2248 3760-3770.

Jean P.B., S. Francoise, 2009. Gas turbine performance increase using an air cooler with phase change energy storage, *Applied Thermal Engineering* 29 (5-6) 1166-1172.

Lazzaretto A., G. Manete, 2009. Analysis of superimposed elementary thermodynamic cycles: from the Brayton-Joule to advanced mixed (Auto-Combined) cycles, *International Journal of Thermodynamics*. ISSN: 1301-9724 12 (3) ISSN: 1301-9724 123-130.

Mathioudakis K., A.T. Salavoutas, 2001. Uncertainty Reduction of Gas Turbine Performance Diagnostics by Accounting for Humidity Effects, in: ASME Turbo EXPO 2001 - Land, Sea & Air, the 46th ASME *International Gas Turbine & Aero Engine Technical Congress*; Jun 4-7; New Orleans, Louisiana, USA

Nishada K., T. Takagi, S. Kinoshita, 2005. Regenerative steam injection gas turbine system, Applied Energy 81 231-246.

Rahman A., Al-Ibrahim, V. Abdulhadi, 2010. A review of inlet air cooling technologies for enhancing its performance of combustion turbines in Saudi Arabia, *Applied Thermal Engineering* 30 (14-15) 1879-1888

Roumeliotis I., K. Mathioudakis, 2010. Evaluation of water injection effect on compressor and engine performance and operability, *Applied Energy* 87 (4) 1207-1216.

Sephehr S., T. Majtaba, 2010. Analysis of gas turbine operating parameters with inlet fogging and wet compression processes, *Applied Thermal Engineering* 30 (2-3) 257-262.

Xiaojun S., A. Brian, C. Defu, G. Jiamin, 2010. Performance enhancement of conventional combined cycle power plant by inlet air cooling, inter cooling and LNG cold energy utilization, *Applied Thermal Engineering* 30 (14-15) 2003-2010.

Yang C., Z. Yang, R. Cai, 2009. Analytical method for evaluation of gas turbine inlet air cooling in combined cycle power plant, *Applied Energy* 86 (6) 848-856.

Zadpoor A.A., A.H. Golshan, 2006. Performance improvement of a gas turbine cycle by using a desiccant-based evaporative cooling system, Energy 31 2652-2664.

Zhang W., O. Lingen, S. Fengrui, 2009. Power and efficiency optimization for combined Brayton and inverse Brayton cycles, *Applied Thermal Engineering* 29 (14-15) 2885-2894.

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