IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 13, Issue 3 Ver. IV (May- Jun. 2016), PP 36-44
www.iosrjournals.org

Gas Turbines Waste Heat/Power Recovery in Tropical Climate Zones: Analysis to Inform Decision Making.

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Abstract: *Gas turbines (GTs) are commonly used in the power generation field. In some applications, it is the preferred option due to its compactness and low weight. Site conditions especially ambient temperature has a strong influence on the gas turbine performance. The gas turbines power plant that is currently under operation in northern kingdom of Saudi Arabia consists of 7 gas turbine of 25MW each. In this area, during summer season, the ambient temperature may reach 50 °C when peak demand occurred, resulting in lower power output and consequently, the problem of electricity black-out occurs unless power demand is decreased. The present paper is mainly prepared to support GTs plant decision makers (technically and economically). A mathematical model was prepared firstly to study the effect of site conditions on the performance of the above mentioned GTs plant. Secondly, the investment cost to add an absorption chiller is to be justified. The results showed that, about 20% of the power output is lost due to increasing the ambient air temperature, from 15 oC (design conditions) up to 50 ^{o}C (actual site conditions). Moreover for hot climate countries, adding an absorption chiller for existing GT plants is strongly recommended which is financially justified (payback period=1.14 year or less). Finally, for future GT power plants, gas turbines with inlet air cooling are advised for hot climate zones.*

Keywords: *Gas turbines power plants, power loss, hot climates, and analysis.*

I. Introduction

The operation of a GT is based on the Brayton cycle (Figure 1), where fuel is mixed with compressed air (fuel oil or natural gas) and burned under constant pressure conditions. The resultant hot gas expands through a gas turbine to produce a mechanical work. More than 50 percent of the turbine output power is consumed by the compressor. The simple cycle is the most basic operating cycle of gas turbines with plant efficiencies ranging from about 30% to 40 % (8968-11606 kJ/kWh)[1-8].In some cases of designs, to assure easy starting of GT, two bleeds are fitted at the 6th and 11th stage of the compressor. They remain open during the start up to 90% of the design speed [1-8].Production capacities of GT_S are rated by the International Standards Organization (ISO):Air inlet conditions: air temperature 15C, 60% RH, absolute Pressure 101.325 KPa. Such conditions are not applicable in hot climate zones, especially in summer. The difference between the ISO standard conditions of 15 C and the hot summer peak periods of approximately 40C, may result in a 20% drop in GT output[1-16].Some researches [17-53] were focused on cooling the inlet air for power augmentation. Environmental issues like NOx reduction using bypass air and catalyst systems were focused by other researches [54-77].Using new catalytic combustors, the future target of NOx emission is2 ppm. Also for future plants, CO2 capture and storage system isincluded[59,69].Finally, integration between solar energy and GTs were studied for future applications [53-69].

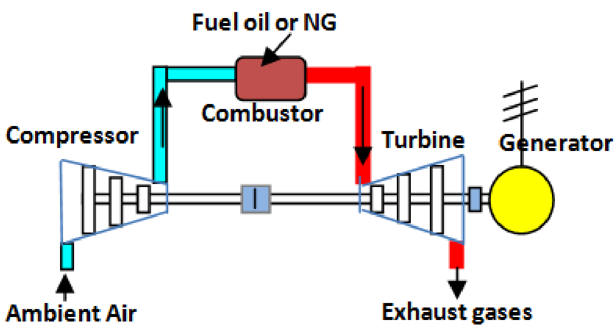

Figure (1): Gas Turbine Simple Cycle [3]

Generally, to increase the gas turbine specific power output, the researchers and designers has been focused on[9]:

- Increasing firing temperature
- Increasing pressure ratio
- Improving component design, cooling and combustion technologies, and advanced materials
- Technology transfer from aircraft gas turbines to stationary gas turbines for power generation and conversion of aircraft gas turbines to power generation applications
- System integration (e.g. combined cycles, inter-cooling, recuperation, reheat, chemical recuperation).

This paper presents a theoretical study to recover the power loss due to high ambient temperature by using a flue gas heat recovery system. The study is mainly prepared to support decision makers either technically or economically. A mathematical model was prepared firstly to study the effect of ambient temperature on the performance of the above mentioned GTs plant. Secondly, the investment cost to add an absorption chiller is to be justifiedtechnically and economically.

II. Mathematical Model[1-16]

In this study, the selected actual gas turbine (TG 20) from FIAT-AVIO[2], with a nominal rating of 25 MW, is an open cycle and a single shaft gas turbine shown in fig.(2).

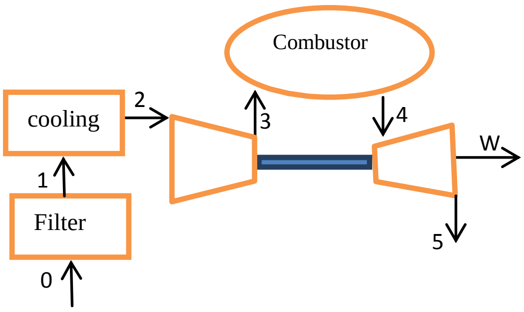

Fig.(2):Schematic of simple GT with inlet cooling

According to the flow direction through GT components, with the aid of fig.(2), the mathematical model is arranged as given below:

2.1 Inlet air cooling

Knowing the pressure drop through the inlet duct ΔP_{inlet} , the compressor inlet pressure can be calculated as follow:

$$P2= P_{atm} - \Delta P_{intake} \quad \text{-----1}$$

Where P_{amb} is the local atmospheric pressure

a-Inlet air cooling using evaporative Cooling system

The water cooler is modeled as an adiabatic air washer that delivers air to the compressor at saturation (about100% relative humidity). The air exit temperature (Tdb2) is related to both the ambient air temperature (Tdb1) and relative humidity and can be calculated as follow:

$$T_{db2}=T_{db1} - \varepsilon\ (T_{db1} - T_{wb1}) \quad \text{----2}$$

Where
ε is the effectiveness of evaporative cooling process,
T_{db1} is the ambient air dry bulb temperature,
T_{wb1} is the ambient air wet bulb temperature.

b- Using absorption chiller

The cooling load (Q) removed from the air flowing at ambient conditions into the power plant can be estimated as follows:

$$\dot{Q}_a = \dot{m}_a\ C_{Pa}\ (T_1 - T_a) \quad \text{-------3}$$

Where
-$\dot{m}_a$ air mass flow rate , Kg/s
- C_{Pa} is the specific heat of the dry air at constant pressure,
-T_a is the ambient air temperature, C

-T_1 is the temperature at the compressor inlet which should not be lower than 5°C in order to avoid condensation and freezing conditions.

The power required to operate the chiller is estimated from the following relation:

$$W_{ac} = \frac{Q_{coil}}{COP} \quad \text{--------4}$$

Where :
COP is the coefficient of performance of the employed chiller. Then, the output net power of gas turbine will be reduced as follows:

$$W_{net} = W_t - W_c - W_{ac} \quad \text{-----5}$$

2.2 Compressor

For a certain pressure ratio r_p, the fluid pressure at compressor outlet can be calculated using the following equation:

$$P_2 = r_p \, . P_1 \quad \text{--------6}$$

Where inlet pressure entering the compressor is:

$$P_1 = P_{atm} - \Delta P_{\text{int}\,ake}$$

The compressor isentropic efficiency can be evaluated using the following empirical equation:

$$\eta_{compressible} = 1 - \frac{[0.09 + (r_p - 1)]}{300} \quad \text{-------7}$$

The isentropic outlet temperature leaving the
Compressor is determined from the following equation:

$$T_{2s} = T_1 (r_p)^{\frac{K_a - 1}{K_a}} \quad \text{----8}$$

The isentropic temperature rise is determined from the following equation:

$$T_{Sr} = T_{2S} - T_1 \quad \text{----}9$$

The actual temperature rise in the compressor is calculated from the definition of isentropic efficiency:

$$T_{ar} = \frac{T_{SR}}{\eta_S} \text{-----}10$$

Then, the actual outlet temperature leaving the compressor is:

$$T_{2a} = T_1 + T_{ar} \text{ ----11}$$

The actual work consumed by the compressor is given by:

$$W_C = \dot{m}_a \; C_{Pa} T_{ar} \text{ ---12}$$

2.3 Combustor

The outlet pressure from the combustor is determined using the following equation:

$$P_3 = P_2 - \Delta P_{combustor} \quad \text{----13}$$

Fuel mass flow rate is determined from the following equation:

$$\dot{m}_f = \frac{Q_{in}/_{HHV}}{\eta_{Combustion}} \quad \text{----14}$$

fuel/air ratio (*F/A)*) is determined from the following equation:

$$F/A = \dot{m}_f \big/ \dot{m}_a \quad \text{----15}$$

heat input to the combustor can be estimated from energy balance across the combustor:

$$\dot{Q}_c = \dot{m}_a C_{Pg} (T_3 - T_{2a}) \text{ ----16}$$

2.4 Turbine

At invariable turbine inlet temperature T_3 and pressure P_3, The turbine isentropic efficiency can be estimated using following empirical equation:

$$\eta_{turbine} = 1 - \frac{[(T_3/P_4) - 1)]}{250} \quad \text{----17}$$

Where Outlet pressure at the turbine's exit is:

$$P_4 = P_{atm} + DBP \quad \text{---18}$$

The expansion ratio is determined from the following equation:

$$r_{p1} = \frac{P_4}{P_3} \quad \text{------19}$$

The isentropic outlet temperature leaving the turbine is determined from the following equation:

$$T_{4s} = T_3 (r_{p1})^{\frac{K_g - 1}{K_g}} \text{ ----20}$$

The isentropic temperature drop is determined from the following equation:

$$\Delta T_{4S}=T_3 - T_{4s} \quad ----21$$

The actual temperature drop is obtained from the definition of turbine's isentropic efficiency:

$$\Delta T_{4a}=\Delta T_{4s}/\eta_{\ turbine} \quad ------22$$

The actual outlet temperature leaving the turbine is determined from the following equation:

$$T_{4a}=T_3 \ - \Delta T_{4a} \quad ---23$$

The total mass flow rate is given by:

$$\dot{m}_{tot} \ = \dot{m}_f + \dot{m}_a \quad -----24$$

The work produced from the turbine is determined by the following equation:

$$W_T \ = \dot{m}_{tot} \ \ C_{Pg} T_{4a} \quad ---25$$

The power output from the gas turbine power plant is:

$$W_{net} \ = W_T - W_c \quad ----26$$

The thermal efficiency without cooling intake air before being introduced to the gas turbine unit is:

$$\eta_{th} \ = \frac{W_{net}}{Q_{in}} \quad -----27$$

2.5 Economic analysis (assuming the cooling system to work only 2 months /year)

The refrigerated cooling systems can lower the air inlet temperature by about 25-30C.The economic study for adding such system to the existing plant is estimated below with the aid of table(1):

-yearly saving of electricity

Based on an amount of power output increase of 20% for two months per year (July and august)=0.2 x7units x25000 KW/turbine x60days/year x24hr/day= $50\text{x}10^6$ Kwh/year

-yearly saving in $ =$50\text{x}10^6$ Kwh/year x 0.12 $/KWh= 6 x 10^6 $/year

- Initial Cost of adding absorption chiller=195.74$/KW increase in power x 0.2x7unitsx25000 KW each=$685.09\text{x}10^4$ $

-Assuming lifetime is 10years, and then the cash flow diagram can be represented below in Fig.(3):

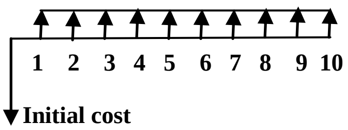

Fig.(3):Simplified Cash Flow diagram.

The financial key indicators are:

-Payback period=investment cost/ saving annually=$685.09\text{x}10^4$ /6 x 10^6 =1.14 years which is highly attractive.

- The other financial indicators can be calculated based on the CFD mentioned above (such as Net present value and internal rate of return).

Cooling system type	Percent inctrease in Power (%)	Cost,$/KW increased
Evaporative cooling	3.32% x110 MW=3.69MW	135.67
Refrigeration inlet cooling	11.51%x110MW =12.77MW	$2.5\text{x}10^6$ $/$12.77\text{x}10^3$= 195.74

Cooling system type	Fuel saving per year,$	Increase in sales revenue per year,$	Total saving per year,$
Evaporative cooling	515,264	396,755	912,019
Refrigeration inlet cooling	605,075	1,379,901	1,984,977

Table(1): cost and saving data for adding cooling system at inlet to compressor[Based on : GT operating at power=110MW, inlet temperature=32C, efficiency=32.92%, HR=10935KJ/KWhr[1].

III. Assumptions

1-The working fluid passing through the compressor is considered as an ideal mixture of air and water vapor, while that passing through the turbine is assumed to be an ideal mixture of combustion gases .
2-The ISO standard conditions are: 59 ^{o}F(15C), 60% relative humidity and 14.7 psia.
3-intake pressure drop (ΔP_{intake}) is taken to be 1 kPa, while the intake temperature is the same as the ambient temperature.
4-Specific heat ratio for air K_a=1.4 and specific heat for air $C_{Pa} = 1.005$ KJ/Kg.K
5-Specific heat of flue gases, $C_{Pg} = 1.15$ KJ/Kg.K, specific heat ratio of exhaust gas K_g=1.332
6-combustion efficiency ,
$\eta_{Combustion} = 0.99$, pressure drop in the combustor $\Delta P_{combustor} = 48\ KPa$
7-Diesel fuel higher heating value is assumed to be around 42 MJ/kg
8-Turbine design back pressure (DBP)=1 KPa

IV. Solution Technique

Gas turbine performance can be estimated using performance curves and site data at both full load and part load (i.e., elevation, ambient temperature, inlet and exhaust pressure drops, and the type of fuel). The set of equations (1-27) are used in calculating the GT performance using Engineering equations solver (EES) software.

V. Results And Discussion

The GT power plant performance is estimated based on actual values of site conditions and gas turbine engine characteristics. The power plant performance is characterized by net power output and heat rate.

5.1 Site Conditions [16-21]:

The max daily temperature for a complete year for Skaka city located in northern Saudi Arabia are given in table(2) [8]. Other parameters related to saturation condition can be obtained using Psychometric chart.

	T_{amb} ^{o}C , daily maximum
Jan	22.7
Feb	27.6
March	33.8
April	41.8
May	48.6
June	52.2
July	55.1
August	55.8
Sept.	51.7
Octob.	42.5
Nov.	30.5
Dec.	23.7

Table(2)Maximum Of The Daily Earth Temperature (°C)(SKAKA City, Lat 29.97 , Lon 40.21)[5].

5.2 Effect of Inlet Air Cooling Systems

Cooling the air at compressor intake helps in increasing the density of air flowing into the GT plant and consequently increasing the generated power from the turbine. Fig.(4) illustrates the effect of compressor inlet temperature (T_2) on power output factor. Let the power output at ISO conditions ($T_{ambient}$=15 C) is 100% (factor =1). Then, from fig.(5)It is clear that the increase of ambient temperature from 15 C to 50 C leads to decrease in power output down to 77.5% .

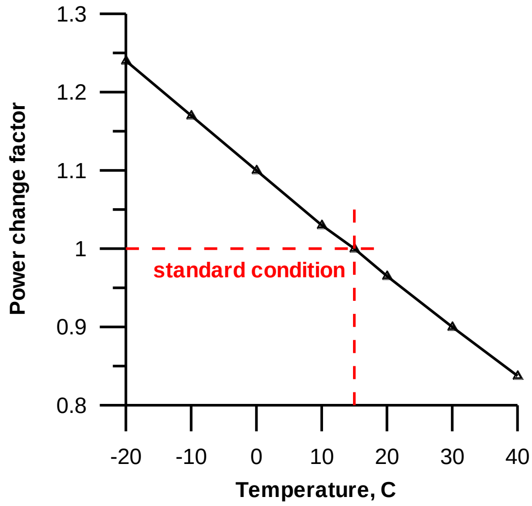

Fig.(4): Effect of compressor inlet temperature (T_2) on power output factor (w_n/ $W_{n\ @\ ISO\ condition}$). The red arrowline points to the ISO condition(i.e T2=15 C).

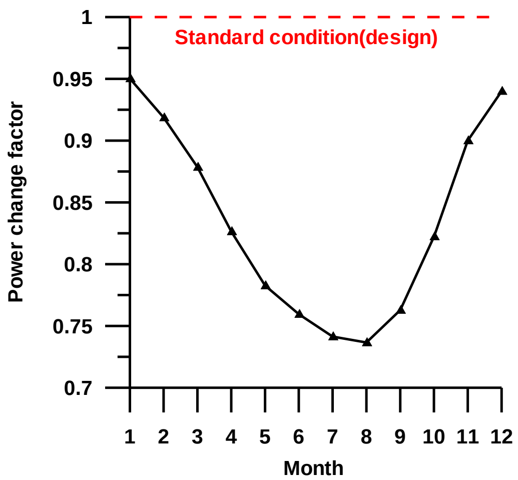

Fig.(5): variation of power change factor with site maximum daily temperature (Estimated monthly at max. daily temperature for SKAKA City[5].

As illustrated in fig.(5), During summer season in SKAKA area, the ambient temperature reaches it max in july and august, when peak demand occurred, resulting in deviation of the plant power from its design conditions (decreased to 78% of design conditions). Consequently, the problem of electricity black-out occurs. So, using the air cooling at compressor inlet is highly recommended

5.3 Verification of the obtained results

As discussed above, the increase in air temperature at inlet of GT compressor, causes a significant reduction in output power, especially in hot climate countries during summer. To verify the obtained results, a comparison with similar results obtained from GT manufacturers (table-3) is made. As shown in fig.(6), a good agreement is noticed between the results obtained from the present study and the GT manufacturers Data.

T_1, C	**Power change factor**
-20	1.24
-10	1.17
0	1.1
10	1.03
15	1
20	0.965
30	0.9
40	0.8375
50	0.775

Table.(3):influence of compressor inlet temperature on power output(base rating) (constant ambient pressure, turbine inlet temperature, and rotating speed)[76].

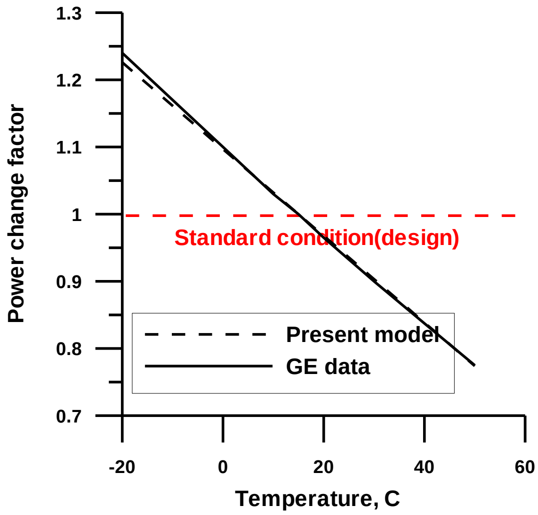

Fig.(6): comparison with similar results obtained from GT manufacturers.

From table(3), as the ambient temperature decreased from 50 C up to 15 C , the net output power increases from 78% of design value up to 100%. Consequently, say 20% increase in power (7units) means adding of 35 MW of power output to overcome the blackout problem during peak load.

5.4 Financial analysis results:

-As estimated in section II.5, the Initial investment Cost of adding absorption chiller=195.74$/KW increase in power x 0.2x7unitsx25000 KW each=685.09x10^4 $

-yearly saving in $ =50x10^6 Kwh/year x 0.12 $/KWh= 6 x 10^6 $/year

-Payback period=investment cost/ saving annually=685.09x10^4 /6 x 10^6 =1.14 years which is highly attractive.

VI. Conclusion

Thepresent study is prepared, regarding power loss recovery for the existing units of GTs due to high ambientconditions. This is achieved by cooling the inlet air using a flue gases heat recovery system. It is concluded that:

-The ambient temperature has a strong influence on the gas turbine output power. As a Summary retrieving of power up to 20% is possiblein hot climate zones.

-For tropical climate zones, adding absorption water chiller to the existing gas turbine for hear/ power recovery is technically and economically justified. Also, for the new GT plant , it should be considered.

Nomenclature:

GE General Electric
GT Gas Turbine
ISO International Standard Organizations
C_{Pa} Specific heat for air (kJ/kg. °K)
C_{Pg} Specific heat of the flue gas (kJ/kg. °K)
COP Coefficient of performance of the mechanical chiller
DBP Design back pressure (kPa)
HHV Diesel fuel High Heating Value (MJ/kg)
F/A Fuel to air mass ratio
*h*Specific enthalpy(KJ/Kg)
*k*Specific heat ratio
$\dot{m}_a$ Air mass flow rate (kg/s)
$\dot{m}_f$ Fuel mass flow rate (kg/s)
$\dot{m}_{tot}$ totalmass flow rate (kg/s)
P_{atm}Atmospheric pressure (kPa)
TIT turbine inlet temperature (^{0}C)
ηefficiency

References

[1]. Meherwan,P,B, 2002"Gas turbine engineering handbook" *the boyce consultancy, Fellow, American society of mechanical engineer and Institute of Diesel and gas turbine engineers, UK*.

[2]. FIAT, TTG, TG20 Gas turbine training manual.

[3]. Frank J. Brooks "GE Gas Turbine Performance Characteristics" *GE Power Systems, Schenectady, NY, GER-3567H*.

[4]. General electric training manual , GEK 111323, 2005

[5]. NASA Surface Meteorology and Solar Energy. Available from: /http://eosweb.larc.nasa.gov/sse.

[6]. J. H. HorlockF.R.Eng., F.R.S., 2003" ADVANCED GAS TURBINE CYCLES" *Whittle Laboratory Cambridge, U.K*.

[7]. M. M. Rahman, Thamir K. Ibrahim and Ahmed N. Abdalla, 2011"Thermodynamic performance analysis of gas-turbine power-plant"*International Journal of the Physical Sciences Vol. 6(14)*, pp. 3539-3550, 18 July, 2011

[8]. P. Centeno, I. Egido, C. Domingo, F. Fernández, L. Rouco , M. González, "Review of Gas Turbine Models for Power System Stability Studies" *Universidad PontificiaComillas*,

[9]. KAWASAKI, 2003 "Gas turbines"Texas Technology Showcase, 17 - 19 March 2003 Houston, TX

[10]. www.kawasaki.com/gtd

[11]. CEGCO, Central Electricity Generating Co., 2000-2006, Annual reports 1999-2005, Amman, Jordan.

[12]. J.O. Jaber, M.S. Mohsen, S.D. Probert, M. Alees, "Future Electricity Demands and Greenhouse Gas Emissions in Jordan".*Applied Energy, Vol. 69*, 2001, 1-18.

[13]. J.O. Jaber, "Greenhouse Gas Emissions and Barriers to Implementation in the Jordanian Energy Sector".*Energy Policy, Vol. 30*, 2002, 385 395.

[14]. J.O. Jaber, S.D. Odeh, S.D. Probert, "Integrated PV and gas turbine system for satisfying peak-demands". *Applied Energy, Vol. 76*, 2003, 305-319.

[15]. J.O. Jaber, O.O. Badran, N. Abu-Shikhah, "Sustainable energy and environmental impact: role of renewables as clean and secure source of energy for the 21st century in Jordan". *Clean Technologies and Environment Policy, Vol. 6*, 2004, 174-186.

[16]. J.O. Jaber, A. Al-Sarkhi, B. Akash, M. Mohsen, "Medium-range planning economics of future electrical power generation". *Energy Policy, Vol. 32*, 2004, 357- 366.

[17]. Y.S.H. Najjar, "Enhancement of Performance of the Gas Turbine Engines by Inlet Air Cooling and Cogeneration System". *Applied Thermal Engineering, Vol. 16*, 1996, 163-173.

[18]. M.M. Alhazmy, Y.S.H. Najjar, "Augmentation of Gas Turbine Performance Using Air Coolers". *Applied Thermal Engineering, Vol. 24*, 2004, 415-429.

[19]. Amell, F.J. Cadavid, "Influence of the Relative Humidity on the Air Cooling Thermal Load in Gas Turbine Power Plant". *Applied Thermal Engineering, Vol. 22*, 2002, 1529-1533.

[20]. Mohanty, J. Paloso, "Enhancing Gas Turbine Performance by Intake Air Cooling Using an Absorption Chiller". *Heat Recovery Systems, Vol. 15*, 1995, 41-50.

[21]. J.S. Andreppont, "Combustion Turbine Inlet Air Cooling, Benefits, Technology Options and Applications for District Energy". *International District Energy Association (IDEA) 91st Annual Conference, Montreal*, 2000.

[22]. M. Ameri, K. Montaser, "The Study of Capacity Enhancement of a 37.5 MW Gas Turbine Using Chilled Water Thermal Energy Storage System". *Proceedings of the 15th International Conference of Efficiency Costs, Optimization, Simulation and Environmental Impact of Energy Systems* (ECOS, 2002), Berlin, Germany, 2002.

[23]. M. Ameri, H. Nabati, A. Keshtgar, "Gas Turbine Power Augmentation Using Fog Inlet Air-cooling System". *Proceedings of ESDA04, 7th Biennial Conference on Engineering Systems Design and Analysis, Manchester, United Kingdom*, 2004.

[24]. R.S. Johanson, "The Theory and Operation of Evaporative Coolers for Industrial Gas Turbine Installations". *Gas Turbine and Aeroengine Congress and Exposition*, June 5-9, Amsterdam, The Netherlands, Paper No. 88-GT-41, 1988.

[25]. I.S. Ondryas, D.A. Wilson, M. Kawamoto, G.L. Haub, Options in Gas Turbine Power Augmentation Using Inlet Air Chilling". *Engineering for Gas Turbine and Power, Transaction of the ASME, Vol. 113*, 1991, 203- 211.

[26]. M. Mercer, "One-Stop Shop for Inlet Cooling Systems". *Diesel and Gas Turbine Worldwide, June Issue*, 2002, 10-13.

[27]. A.M. Bassily, "Effects of Evaporative Inlet and Aftercooling on the Recuperated Gas Turbine Cycle". *Applied Thermal Engineering, Vol. 21*, 2001, 1875- 1890.

[28]. D.A. Kolp, W.M. Flye, H.A. Guidotti, "Advantages of Air Conditioning and Supercharging an LM6000 Gas Turbine Inlet". *Engineering for Gas Turbine and Power, Transactions of the ASME, Vol. 117*, 1995, 513-527.

[29]. M. De Paepe, E. Dick, "Technological and Economical Analysis of Water Recovery in Steam Injected Gas Turbines". *Applied Thermal Engineering, Vol. 21*, 2001, 135-156.

[30]. M.Jonsson, J. Yan,2005" Humidified gas turbines—a review of proposedand implemented cycles"*Energy 30* (2005) 1013–1078.

[31]. Hedman, B.A., 2008, 'Waste Energy Recovery Opportunities for Interstate Natural Gas Pipelines'*INGAA*.

[32]. GAS TURBINES IN SIMPLE CYCLE & COMBINED CYCLE APPLICATIONS" *Claire M. Soares P.E.; Fellow ASME; MBA*.

[33]. Q.M.Jaber, J.O.Jaber, M.A. Khawaldah, 2007"Assesment of power augmentation from GT power plants using different inlet air cooling systems"*jordan journal of mechanical and industrial engineering, vol.1*, no.1, pp.7-15.

[34]. Mark S. Knodle and Mark A. Novaresi, 2009 "INDUSTRIAL APPLICATION OF GAS TURBINES COMMITTEE, Shared Technologies in the Development of the Titan 250 Gas Turbine System" *Presented at the 18th Symposium on Industrial Application of Gas Turbines (IAGT) Banff, Alberta, Canada, Paper No: 09*-IAGT-102

[35]. Oleg Zastavniuk and Erick van den Akker, 2009" MODERN TEST FACILITIES FOR GAS TURBINE ENGINES: REQUIREMENTS AND CHALLENGES" *18TH SYMPOSIUM OF THE INDUSTRIAL APPLICATION OF GAS TURBINES COMMITTEE BANFF, ALBERTA, CANADA*, 2009-IAGT-104

[36]. Edward Wacekt, Warren Ferguson, General Electric, 2009 "APPLICATION OF THE LATEST AERODERIVATIVE GAS TURBINE TECHNOLOGY, 09-IAGT 203

[37]. Randy Bohl, P.Eng, 2009" Waste Heat Recovery from Existing Simple Cycle Gas Turbine Plants A Case Study" *Paper No: 09*-IAGT-205

[38]. Anders Hellberg, 2009"Performance Upgrade of Siemens SGT 700 SGT-Gas Turbine for More Power and Higher Efficiency"*09 IAGT301* .

[39]. Aaron Bolhous, P.Eng, 2010" CORNELL 30 MW COMBINED HEAT AND POWER PROJECT" *IAGT Fall 2010 Course Hamilton Ontario*.

[40]. T. K. Ibrahim, M. M. Rahman, A. N. Abdalla,2011 "Improvement of gas turbine performance based on inlet air cooling systems: A technical review" *International Journal of Physical Sciences Vol. 6(4)*, pp. 620-627, 18 February, 2011.

[41]. Christian Kaufmann ,2010"INDUSTRIAL APPLICATIONS OF GAS TURBINES, Session #7 "Emission Reduction Case Study" Fall 2010 Course.

[42]. W. I. Rowen, "Simplified Mathematical Representations of Single Shaft Gas Turbines in Mechanical Drive Service", *Proc. The International Gas Turbine and Aeroengine Congress andExposition*, Cologne, Germany, 1992.
[43]. A. Bagnasco, B. Delfino, G. B. Denegri, and S. Massucco, "Management and dynamic performances of combined cycle power plants during parallel and islanding operation", *IEEETransactions on Energy Conversion, vol. 13*, pp. 194-201, 1998.
[44]. J. Undrill and A. Garmendia, "Modeling of combined cycle plants in grid simulation studies", *Proc. 2001 Winter Meeting of the IEEE PowerEngineering Society, Columbus, OH, USA*, 28 Jan.- 1 Feb, 2001, pp. 657-63, vol.2.
[45]. W. I. Rowen, "Simplified mathematical representations of heavy duty gas turbines", *Transactions of the ASME. Journal of Engineering for Power, vol. 105*, pp. 865-70, 1983.
[46]. W. I. Rowen, "Operating Characteristics of Heavy Duty Gas Turbines in Utility Service", *Proc. TheGas Turbine and Aeroengine Congress*, Amsterdam, The Netherlands, 1988, pp.
[47]. GER 4250 "The LM2500+G4 Aeroderivative Gas Turbine for Marine and Industrial Applications" by Gilbert H. Badeer, GE Energy September 2005.
[48]. Bammert,K. and Deuster, G., 1974"Layout and present status of the closed cycle Helium turbine plant"*ASME paper*, 74-GT-132.
[49]. Michael T. McCarrick, 2011 "LM2500 to LM2500+DLE Gas Turbine Combined Cycle Plant Repowering" www.ge-energy.com
[50]. Fredrik Haglind,2008" A review on the use of gas and steam turbine combined cycles as prime movers for large ships. Part I: Background and design" *Energy Conversion and Management 49* (2008) 3458–3467
[51]. Development of Key Technology for Ultra-high-temperature Gas Turbines" *Mitsubishi Heavy Industries Technical Review Vol. 47 No. 1 (Mar. 2010).*
[52]. SITAKANT PATRA, 2010 "DESIGN AND MODELING OF AXIAL MICRO GAS TURBINE" national institute of technology, Rourkela-769008 Roll No. – 10603030.
[53]. Peter Schwarzbozl, Reiner Buck, ChemiSugarmen, Arik Ring, Jesu´s Marcos Crespo, Peter Altwegg, Juan Enrile, 2006" Solar gas turbine systems: Design, cost and perspectives"*Solar Energy 80* (2006) 1231–1240
[54]. ERNEST ORLANDO LAWRENCE BERKELEY NATIONAL LABORATORY,1999 "Carbon Emissions Reduction Potential in the US Chemicals and Pulp and Paper Industries by Applying CHP Technologies." LBNL-43739
[55]. Brun, K., Kurz, R., 2008, "The Carbon Footprint of Electric Drives" *Turbomachinery International*, March 2008.
[56]. Ferenc LEZSOVITS, Sándor KÖNCZÖL, and KrisztiánSZTANKÓ" CO EMISSION REDUCTION OF A HRSG DUCT BURNER" *THERMAL SCIENCE*: Year 2010, Vol. 14, No. 3, pp. 845-854
[57]. Roger E. Anderson, Scott MacAdam, Fermin Viteri,2008" ADAPTING GAS TURBINES TO ZERO EMISSION OXY-FUEL POWER PLANTS" *Proceedings of ASME Turbo Expo* 2008: Power for Land, Sea and Air GT 2008 June 9-13, Berlin, Germany
[58]. C.Y. Liu, G. Chen, M. Assadi, X.S. Bai, 2011"characteristics of Oxy-fuel combustion in gas turbines" *El-sevier*, 2011
[59]. Hanne M. Kvamsdal1, Ola Maurstad1, Kristin Jordan, and Olav Bolland" BENCHMARKING OF GAS-TURBINE CYCLES WITH CO2 CAPTURE" *1SINTEF Energy Research, N-7465 Trondheim, Norway*
[60]. Hanne M. Kvamsdal1, Olav Bolland, Ola Maurstad, and Kristin Jorda" A qualitative comparison of gas turbine cycles with CO2 capture" *The Norwegian University of Science and Technology (NTNU),* N-7491 Trondheim, Norway
[61]. Shinji, K. and Nobuhide, K., " performance evaluation of gas turbine –fuel cell hybrid micro generation system" *proceedings of ASME TURBO EXPO 2002*, Amsterdam , The Netherlands.
[62]. A. BoudgheneStambouli and E. Traversa, 2002" Solid oxide fuel cells (SOFCs): a review of an environmentally clean and efficient source of energy" *Renewable and Sustainable Energy Reviews 6* (2002) 433–455
[63]. M J Moore,1997" Noxemission control in gas turbines for combined cycle gas turbine plant" *ProcInstnMechEngrsVol 211* Part A
[64]. C. Fielenbach, Th. Holfeld, C. von Petery, U. Renz" *NOx-Reduction in a Pressurized Pulverized Coal Flame by Flue Gas Recirculation" WÜK, Aachen University, Germany*
[65]. D. Straub, D. Ferguson, K. Casleton, and *G. Richards"* Effects of Propane/Natural Gas Blended Fuels on Gas Turbine Pollutant Emissions" Paper # E06, 5th US Combustion Meeting Organized by the Western States Section of the Combustion Institute and Hosted by the University of California at San Diego March 25-28, 2007.
[66]. A. BoudgheneStambouli and E. Traversa, 2002" Solid oxide fuel cells (SOFCs): a review of an environmentally clean and efficient source of energy" *Renewable and Sustainable Energy Reviews 6* (2002) 433–455
[67]. L. N. Hannett and A. H. Khan, "Combustion turbine dynamic model validation from tests", *IEEETransactions on Power Systems*, vol. 8, pp. 152-8, 1993.
[68]. L. M. Hajagos and G. R. Berube, "Utility experience with gas turbine testing and modeling", *Proc. Winter Meeting of the IEEE PowerEngineering Society*, Columbus, OH, USA, 2001, pp. 671-7.
[69]. John C. Gerretsen, Rainer Kurz, "POWERING PROGRESS WITH CLEAN GAS TURBINE SYSTEMS" *Solar Turbines Incorporated,* San Diego, CA, USA
[70]. A.B. Stambouli, E. Traversa, 2002 "Technical characteristics of fuel cells " *Renewable and Sustainable Energy Reviews 6* (2002) 433–455).
[71]. Catalano, L. A., *et al*, "Thermal Design and Emissions of Duct-Burners for Combined-Cycle and Cogenerative Plants" *International Journal of Environmental Technology and Management, 7* (2007), 1-2, pp. 165-177
[72]. Brian G. Swanson, Paul Lawrence,2009" An Alternative Approach to Continuous Compliance Monitoring and Turbine Plant Optimization using a PEMS (Predictive Emission Monitoring System)" 18TH SYMPOSIUM OF THE INDUSTRIAL APPLICATION OF GAS TURBINES COMMITTEE BANFF, ALBERTA, CANADA.
[73]. Carbon Emissions Reduction (LBNL-43739 , June 1999)
[74]. Catalano, L. A., *et al*, "Thermal Design and Emissions of Duct-Burners for Combined-Cycle and Cogenerative Plants" *International Journal of Environmental Technology and Management, 7* (2007), 1-2, pp. 165-177
[75]. *A.B. Stambouli, E. Traversa, 2002 "*Technical characteristics of fuel cells "*Renewable and Sustainable Energy Reviews* vol.6 ,pp 433–455).
[76]. PSS/E-30, *PSS/E-30 Online Documentation*, 2004.
[77]. (http://inventors.about.com)
[78]. Elson,A., Tidball,R., Hampson, A, 2015 "Waste Heat to Power Market Assessment" *Oak Ridge National Laboratory*.