Available online at www.sciencedirect.com
ScienceDirect
Energy Procedia 82 (2015) 591 – 598
Energy Procedia

ATI 2015 - 70th Conference of the ATI Engineering Association

Thermoeconomic Analysis of a One-Pressure Level Heat Recovery Steam Generator Considering Real Steam Turbine Cost

Carlo Carcasci[a]*, Beniamino Pacifici[a], Lorenzo Winchler[a], Lorenzo Cosi[b], Riccardo Ferraro[b]

[a]*DIEF: Department of Industrial Engineering – University of Florence – Via Santa Marta, 3 –Florence (I)*
[b]*GE Oil&Gas –Florence (I)*

Abstract

Nowadays, gradual depletion of fossil fuels associated with emissions constraints due to greenhouse gases, leads to reuse wasted heat from power plant in order to increase the global efficiency. One of the implemented technologies for improvements is the application of combined cycles. In this scenario, the steam cycle is frequently combined with a Brayton cycle and the power plant performances and costs are competitive in the global market.
Often, an energetic engineering company defines and studies the performance of the bottom steam cycle, thus it imposes operational conditions of steam turbine and heat recovery boiler and requires these components are built by two different manufacturers. For this reason, the plant cannot be globally optimized.
From a steam turbines manufacturer point of view, the integration between proprietary simulation code and an energy balance code is an opportunity to simulate a complete bottom-cycle in order to define the best plant configuration.
In the present paper, aone-pressure level heat recovery steam generator is studied in term of thermodynamic performance and cost analysis. The thermodynamic analysis is realized using a fixed steam turbine isentropic efficiency (as an energetic engineering company can do) and using anisentropic efficiency determined from steam turbine industrial tool, so a different best performance can be determined. Moreover, a comparison between two academic steam turbine cost correlations and steam turbine cost suggested by industrial cost is carried out.

Peer-review under responsibility of the Scientific Committee of ATI 2015

Keywords: Gas Turbine Combined Cycle; Heat Recovery Steam Generator; Thermoeconomic Analysis; Manufacturing Cost, Steam Turbine

* Corresponding author. Tel.: +39-055-2758783.
E-mail address: carlo.carcasci@unifi.it.

Peer-review under responsibility of the Scientific Committee of ATI 2015
doi:10.1016/j.egypro.2015.11.877

1. Introduction

In an increasingly competitive market, reducing costs for generating electricity becomes very important, in order to provide a rapid return on investment, but without decreasing the power plant reliability or flexibility.

Combined-cycle systems using Brayton Cycle gas turbine and Rankine Cycle steamsystem with air and water/steam as working fluidscan achieve efficient, reliable, and economic powergeneration [1].

Often, an energy engineering company defines and studies the performance of the bottom steam cycle imposing the operational conditions of the steam turbine and the heat recovery boiler. However, the company requires to take these two components from two different manufacturers. Thus, the energy-balance design company can obtain the real steam turbine performance and cost only when the entire bottoming cycle is defined and, for this reason, the plant cannot be globally optimized.From a steam turbines manufacturer point of view, the integration between proprietary simulation code and an energy balance code is an opportunity to simulate a complete bottom-cycle in order to define the best steam turbine model for the proposed bottoming cycle.

Several authors [2–13] carried out a thermoeconomic analysis of this kind of combined cycle. Attala et al. [2]have realized a tool for a thermoeconomic evaluation and optimization of thermal power plants; Roosen et al. [3] treated the optimization of a combined cycle power plant, following a rigid direct cost evaluation. Rao and Francuz [4] identified and assessed advanced improvements to the combinedcycle that will lead to significant performanceimprovements in coal based power systems.Carapellucci and Giordano [5]compared two different methodologies for optimizing CCGTs.Furthermore, Facchini and Carcasci [6-13] have studied GT power plants in design and off-design conditions. Some thermodynamic cycles are studied like a comparison between two heavy duty gas turbines for combined cycle application [7], a Chemically Recuperated Gas Turbine cycle with a detailed HRSG and Mass Steam Reformer analysis [9, 10, 11], thermoeconomic district heating analysis using a gas turbine [8], Joule-Joule combined cycle [13] and others thermodynamic cycle [12].

The aim of this paper is to model a combined cycle with one-pressure level HRSG, focusing on the effect on CCGT,in terms of thermodynamic and economic performances, using two academic steam turbine model correlations or an industrial tool. Their effects on best design pressure that optimize steam turbine output power and COE (Cost of Energy)are compared; the steam turbine industrial tool can supply a reliable cost of the machine and a correct value of isentropic efficiency. ESMS (Energy System Modular Solver)modular code is used for modelling the cycle.

Nomenclature

A	Steam Turbine outlet section	[m^2]
c_x	Axial velocity	[m/s]
C	Cost	[$]
COE	Cost of Energy	[$/MWh]
m	Mass flow rate	[kg/s]
P	Pressure	[bar]
T	Temperature	[K]
v	Specific volume	[m^3/kg]
x	Steam quality	[-]
W	Power	[kW]
η	Efficiency	[-]

2. ESMS Cycle Analysis Code

Power plants based on gas turbine engines are not very complex, but to simulate them, a flexible and detailed tool is necessary: power plant designers use ad-hoc toolsor commercial codes to simulate each component because many details are necessary, but a modular code like ESMS can be very useful for designing phases.The most important feature of this modular simulation code is the ability to simulate a new power plant configuration without creating a new source program. The power plant configuration is defined by connecting a number of elementary components representing different unit operations such as compressors, combustion chambers, mixers and so on. Each component is defined as a black box capable of simulating a given chemical and thermodynamic transformation. All equations are then solved simultaneously using a classic matrix method; thus the procedure is essentially a fully implicit linear approach. The reader is referred to references [6, 7, 8, 9 and 10] for a complete presentation of the code, related theory and some engineering applications.

Correlations for the main equipment costsof plant are implemented in the ESMS code. The reader is referred to references [2, 3 and 5] for a complete presentation of cost correlations.

Table 1. Thermodynamic parameters.

Parameter		Value	Unit
P	Pressure	vary	bar
P_{cond}	condenser Pressure	0.18	bar
ΔT_{pp}	Pinch Point	5.0	°C
ΔT_{app}	Superheater Approach	30.0	°C
ΔT_{sub}	Economizer Subcooling	15.0	°C
x_{lim}	steam quality limit	0.86	-
η_{ST}	steam turbine efficiency	0.86	-

Table 2. LM6000 turbine data sheet [14, 15].

LM6000 data sheet	Value	Unit
ISO Rated Power	43076	kW
Heat Rate	8707	kJ/kWh
Electrical Efficiency	41.3	-
Pressure Ratio	30.0	-
Exhaust Mass Flow	125.2	kg/s
Turbine Speed	3600	rpm
Exhaust Temperature	449.0	°C

3. Gas Turbine Combined Cycle Description

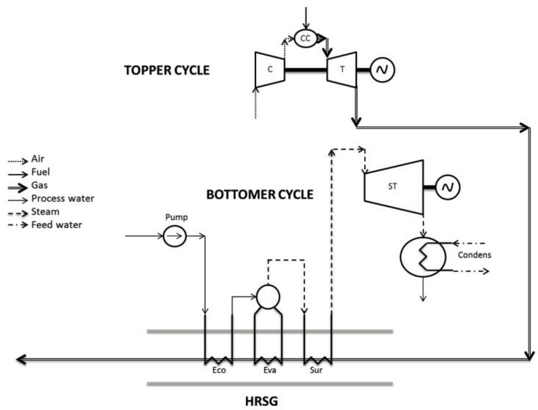

Figure 1.Combined Cycle scheme.

Figure 1 shows a typical model of a combined cycle witha one pressure level Heat Recovery Steam Generator (HRSG).Topper cycle is a classical industrial cooled gas turbine:exhaust gasesfromgas turbine, crossing the boiler, release heat in countercurrent to the section of the superheater (Sur), the evaporator (Eva) and the economizer (Eco).Those sections are crossed by water that ispressurizedby a pump and absorbs heat from exhaust gases through heat exchangers. Water turns into steam and evolves into the steam turbine (ST) generating power. The steam finally reaches a condenser in which is led back to the liquid state.Table 1 shows the main thermodynamic parameters used for the simulation.

3.1. Topper Gas Turbine

The topper cycle, used in this work, is a real aeroderivative gas turbine (LM6000, [14, 15]). Its efficiency and operational flexibility make the LM6000 a cost-effective choice for all applications.The LM6000 is a simple-cycle, two-shaft, high-performance gas turbine that is derived from GE's CF6-80C2 high bypass turbofan aircraft engine. The compressor is an eleven-stage axial flow designed with a 30:1 pressure ratio.Rotational speed is 3600 rpm with a mass flow of 125.2 kg/s. Hot gas parts are cooled by air extracted from the axial compressor.

Gas turbine LM6000, whose main data are listed in Table 2, is used to model the topper: they are referred to ISO conditions. In the present analysis, gas turbine is not simulated, but exhaust data are directly used.

3.2. Steam turbine

GEdesigns and manufactures a lot of steam turbines destined for Oil & Gas business. The production includes machine for mechanical drive and power generation, condensing and backpressure turbines, with or without extraction.

Backpressure turbine makes use of the pressure drop available from two steam systems at different pressures: it is used for operating at low values of pressure and temperature (up to 90 bar and 520°C). When steam is available at higher values, different types of machine are used, enabling pressures of up 140 bar and 540°C to be employed.

Production include impulse and reaction stages. This double design allows to ensure the better solution between increasing enthalpy drop and reducing the size of the machine with high level of operability.

4. Cost Analysis

The economic optimization of a combined cycle can be obtained by minimizing the objective function represented by the cost per unit of energy, called Levelized Cost of Energy (LCOE) or simply COE, usually measured in $/kWh or $/MWh. This index represents the price at which the electricity should be generated from a specific source in order toreach the break-even point. COE is represented by an economic balance of all costs considered over the life of the system: initial investment, operation and maintenance, fuel costs, capital costs. This index is very useful to calculate the final cost of the electricity generation from different sources and for comparing technologies with different operating characteristics [16]. COE can be defined by a formula recommended by the International Energy Agency (IEA) [17]:

$$COE = \frac{TCR \cdot CRF + O\&M + FP}{E}$$

COE is composed of three parts: TCR (Total Capital Requirement), defined as the sum of capital costs,interest during construction and pre-production costs, multiplied for “Capital Recovery Factor” to take account of discount

Table 3. COE Main economic assumptions.

Economic parameters	Value	Unit
%TEC=BOP	12.0%	-
%TEC=Engineering Costs	8.0%	-
%TEC=Contingencies	5.0%	-
Fuel price	6	$/GJ
Fixed O&M	14	$/kW-y
Variable O&M	0.5	mill$/kWh
Yearly operating hours	7500	h/y
Discount rate	10.0%	-
Plant working life	25	Y
Capacity factor	85.62%	-

rate, *O&M* (Operating and Maintenance) costs, and *FP* (Fuel Price).Operating and maintenance costs includea fixed and a variablecontribution, and they are evaluated using the assumptions summarized in Table 3.Fuel price (FP)is imposed of 6.0 \$/GJ.*E* represents the energy.

Correlations for the main equipment costs of plant are implemented in the ESMS code. A complete presentation of the cost correlations is shown in previous studied [2, 3 and 5]. In this paper, a focus on steam turbine cost is carried out and used steam turbine cost correlations are:

by Attala et al. [2]:

$$C_{ST} = 3197280 \cdot A^{0.261} + 823.7 \cdot W^{1.543}$$

by Roosen et al. [3]:

$$C_{ST} = 3880.5 \cdot W^{0.7} \cdot \left(1 + \frac{0.05}{1 - \eta_{is}}\right)^3 \cdot \left(1 + 5 \cdot e^{\frac{T_{in} - 866\,\mathrm{K}}{10.42\,\mathrm{K}}}\right)$$

In both correlations, steam turbine output power is considered, but other parameters are neglected: in the correlation introduced by Roosen et al., an effect of thermodynamic efficiency and inlet steam temperature is present, but there is not any effect of exit turbine section*A*, which is proportional to exit steam mass flow rate and exit steam quality. In fact, considering the mass flow rate equation, the specific volume v can be written based on vapor and liquid value because of saturation condition of steam at turbine exit. Considering that liquid specific volume $v_l$is negligible respect to the vapor one, *A*can be obtained:

$$A \cong \frac{m \cdot v}{c_x} = \frac{m \cdot [v_l + x(v_v - v_l)]}{c_x} \cong \frac{m \cdot x \cdot v_v}{c_x}$$

Thus, the exit steam turbine area is proportional to steam mass flow rate, steam quality and vapor specific volume, which depends on condenser pressure.

The steam turbine industry uses own tools to determine the efficiency of a commercial steam turbine and its cost. So, academic cost correlations (by Attala et al. and Roosen et al.) can be compared with an industrial tool. Steam turbinesmodels are selected usinga tooldeveloped byGE.

Initially, a simulation of the entire bottoming power plant is performed using a simplifiedthermodynamic model, then, using obtainedresults (like inlet steam mass flow rate, pressure and temperature condition) a simulation of theindustrial tool to select steam turbine type can be done, so its real performance and cost can be obtained. At this point, we can perform again the simulation with the new efficiency value until we reach a convergence and so the cost can be determined from the industrial tool. Otherwise, if we use the academic correlations we have to impose a fixed efficiency value and so the cost is directly evaluated. Thus, a comparison between performances and costs of the entire power plant data, obtained from industrial tooland from with literature cost correlation, all linked to ESMS, was carried out.

5. Thermoeconomic analysis

5.1 Thermodynamic analysis

The performance and the cost are studied varying the outlet heat recovery boiler of the water/steam circuit (from 10 to 100 bar) while hot gas conditions from gas turbine are not been modified in the simulations because the topper cycle works in ISO condition. Efficiencies, pressure drops, pinch point temperature difference, approach point and subcooling of the exchangers are imposed(Table 1).

Increasing heat recovery boiler pressure, the steam mass flow rate decreases (Figure 2) and considering that the maximum steam temperature is constant for all simulations (exhaust gas temperature from gas turbine and approach point are fixed), the exit steam quality x grow less (Figure 3). However,

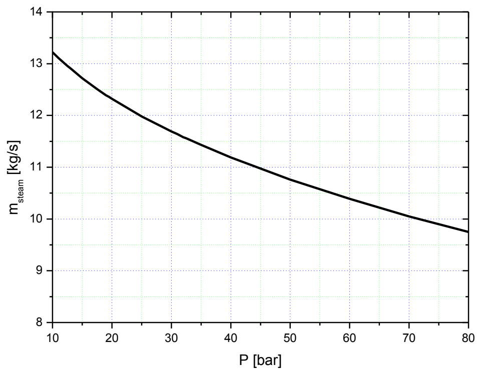

Figure 2.Steam mass flow rate
versus boiler pressure

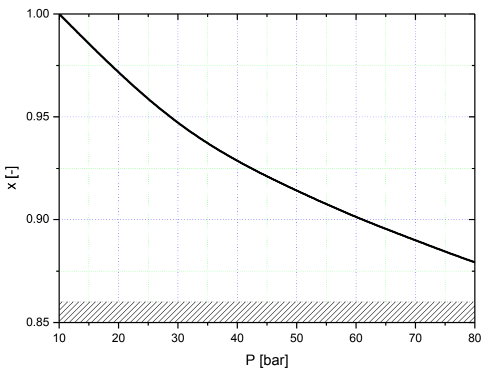

Figure 3.Steam quality at the exit
of steam turbine versus boiler pressure

the steam turbine specific work rises. Figure 4 shows the bottoming power output (net specific work multiplied by mass flow rate) respect to maximum pressure: its value initially grows, then a maximum is present at P=28 bar and finally it goes down.

Using industrial tool to evaluate steam turbine performance, steam turbine isentropic efficiency is not constant anymore (Figure 5) and the trend is not regular because changing the pressure and mass flow rate a different steam turbine model is selected.Using the industrial tool, the optimum pressure changes (about 30-32 bar) and the maximum power increases of 0.3%. This effect is due because the industrial selector tool identifies a best isentropic efficiency in a range from 25.0 to 32.5 bar (Figure 5). A little output power step is present in the range from 30.0 to 32.0 bar (Figure 4) but the optimum pressure using a simple thermodynamic simulation (28bar) is out of this range.

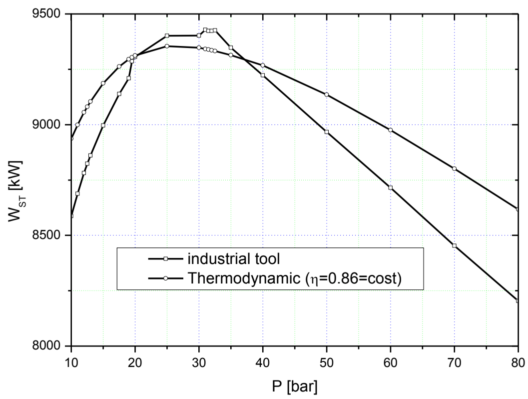

Figure 4.STpowerobtainedwith ESMS and industrial tool

Thus, if steam turbine industrial tool is not integrated to a code like ESMS, there is a strong risk that steam turbine manufactory reply to an offer with an optimized steam turbine referred to the thermodynamic conditions of the offer, but that maybe the best steam turbine integrated with that bottoming cycle is a totally different model with different performances.

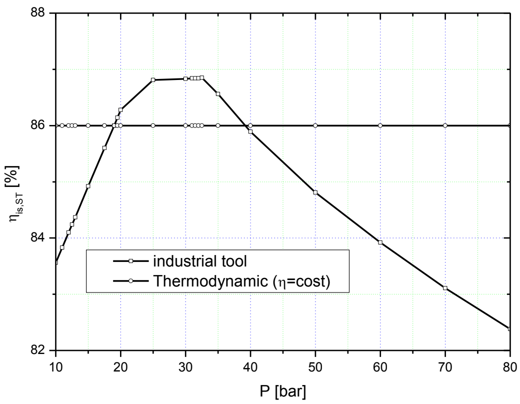

Figure 5.Isoentropic efficiencyversus boiler pressure

5.2 Economic analysis

Changing the type of steam turbine will affect not only thermodynamic performancebut also steam turbine cost.Figure6shows curves of steam turbine costs. Roosen et al. cost correlation trend is equal to the outputpower because it is a function of this parameter only, so maximum value is present at 28 bar. Attala et al. correlation decreases when the pressure increases, because output power variation is not high, but when the pressure rises, steam mass flow rate (Figure 2) and steam quality (Figure 3) decrease and so the outlet steam turbine area decreases (see previous equations). The trends of Attala et al. and Roosen et al. correlations are different, anyway the trend are continuous. The industrial tool determines a steam turbine cost that decreases when inlet boiler pressure rises: this happened because a different model of steam turbine is selected every time the pressure increase too much, and changing the model leads to a smaller outlet area, so the cost curve presents a gap.

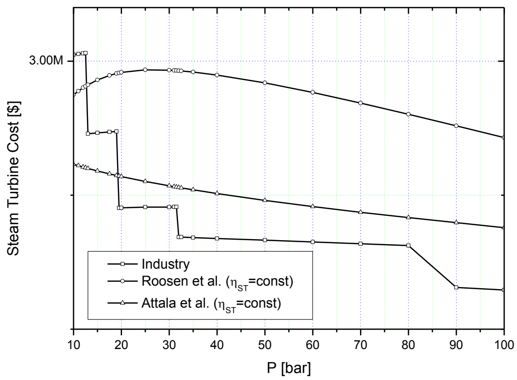

Figure6. Trend of ST cost with correlations and industrial tool.

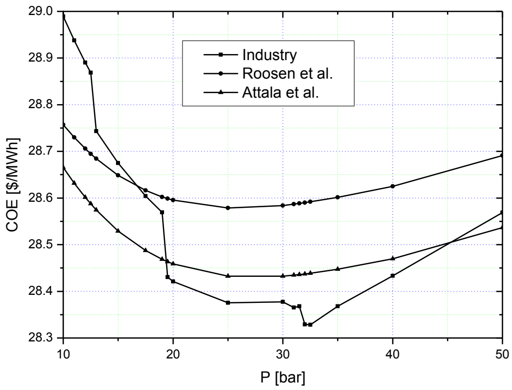

Figure7. Trend of Combined Cycle COE.

The steam turbine cost can be used to determine COE (Figure 7). The trend for all correlations present a minimum value because there is a compromise solution between capital cost and thermodynamic power plant efficiency (fuel cost). Using Roosen et al. correlation the pressure that minimize COE is about 25bar. This value is smaller than the thermodynamic optimum pressure (28 bar) because when the pressure rises, the heat recovery steam generator HRSG cost grow down, too. Thus, the power plant is little less efficient, but also the capital cost is smaller. Using Attala et al. correlation, the pressurethat minimize COE is about 27 bar. This pressure is again smaller than the optimum thermodynamic design pressure (28bar), but it is higher than optimum pressure determined using Roosen correlation, because the cost trend determined from Attala et al. correlation is lower for all the pressure.

Introducing the industrial tool to evaluate steam turbine cost, COE present a minimum value at 32.0-32.5 bar and the COE value is lower than the one found using academic correlations. This happened because the best performance of steam turbine is in a range between 25.0 to 32.5 bar (Figure 4) but at about 31.8 bar the industrial tool changes steam turbine model and it suggests a cheaper model with better performance.

Thus, also considering economic aspect, if steam turbine industrial tool is not integrated to a power plant simulation code like ESMS, there was a strong risk that steam turbine manufactory reply to the offer with a steam turbine optimized for 25.0 bar with greater cost and worse performance, while integrating the two codes can lead to a different optimum design pressure (32.5 bar versus 25.0 bar) and so a different steam turbine model can be suggested, a turbine that reach a plant COE of 0.02% lower that corresponds to a decrease of 0.05 $/MWh.

6. Conclusions

A thermoeconomic analysis of a one-pressure level HRSG is performed and the effect of the real industrial cost of steam turbine compared to literature correlations is analyzed.

Imposing a constant steam turbine efficiency, an optimum thermodynamic design pressure is found at 28bar, while using the industrial tool the optimal pressure grows at 30.0-32.5 bar because steam turbine isentropic efficiency varies with the imposed thermodynamic conditions.

Using cost determined bythe industrial tool the optimal pressure is 30.0-32.5 bar, while using academic correlation optimal pressure is 25 bar; moreover, COE of the entire power plant is 0.02% lower, corresponding to a decrease of 0.05 $/MWh.

The integration of ESMS code and steam turbine industrial tool has allowed to define a cheaper steam turbine with better performance and a different design pressure of heat recovery steam generator. Without this code integration, steam turbine manufactory risks to sell a steam turbine that is more expensive andless performing.

References

[1] Chase D. L; Combined-cycle development evolution and future. GE Power Systems, GER-4206, 2001, 5-6.

[2] Attala L.; Facchini B.; Ferrara G. "Thermoeconomic optimization method as design tool in gas–steam combined plant realization". Energy Conversion and Management, 2001, 42.18: 2163-2172.

[3] Roosen P.; Uhlenbruck S., Lucas K. "Pareto optimization of a combined cycle power system as a decision support tool for trading off investment vs. operating costs". International Journal of Thermal Sciences, 2003, 42.6: 553-560.

[4] Rao Ashok D.; Francuz D. J. "An evaluation of advanced combined cycles". Applied Energy, 2013, 102: 1178-1186.

[5] Carapellucci R.; Giordano L. "A comparison between exergetic and economic criteria for optimizing the heat recovery steam generators of gas-steam power plants". Energy, 2013, 58: 458-472.

[6] Carcasci C., Facchini, B. "A Numerical Method for Power Plant Simulations". ASME J. of Energy Resources Technology, March 1996, vol.118, pp.36-43.

[7] Carcasci C., Facchini B."A Comparison between Two Gas Turbine Solutions to Increase Combined Power Plant Efficiency", Energy Conversion & Management – Pergamon Elsevier Science Ltd. Oxford (UK), 1999; ISSN: 01968904

[8] Harvey S., Carcasci C., Berntsson T. "Gas Turbines in District Heating Combined Heat and Power Systems: Influence of Performance on Heating Costs and Emissions", Pergamon, Applied Thermal Engineering, 20 (2000), 12, pp. 1075-1103; Oxford (UK), Elsevier Science Ltd., ISSN: 13594311.

[9] Carcasci C., Facchini B., Harvey S., 1998; "Design and Off-Design Analysis of a CRGT Cycle Based on the LM2500-STIG Gas Turbine", ASME TurboExpo, Paper 98-GT-36, Stockolm (S), June 2-5 1998.

[10] Carcasci C., Facchini B., Harvey S. "Design Issues and Performance of a Chemically Recuperated Aeroderivative Gas Turbine." Proc. Instn Mech Engrs, Part A, Journal of Power and Energy, 1998, 212, A(04398), 315 - 329.

[11] Carcasci C., Harvey S."Design Issues for the Methane-Steam Reformer of a Chemically Recuperated Gas Turbine Cycle", ASME Paper 98-GT-35, ASME TurboExpo, Stockolm (Sweden), 2-5 June 1998.

[12] Carcasci C., FerraroR., Miliotti E. "Thermodynamic Analysis of an Organic Rankine Cycle for Waste Heat Recovery from Gas Turbines", Energy (2014), Vol. 65, 1 February 2014, Pages 91–100, DOI: 10.1016/j.energy.2013.11.080

[13] Carcasci C.,Costanzi F.,Pacifici B. "Performance Analysis in Off-design Condition of Gas Turbine Air-bottoming Combined System". Energy Procedia, 2014, 45: 1037-1046.

[14] GE data sheet, http://www.filter.ee/extensions/filter/brochures/114-82943.pdf, 2012.

[15] LM6000 Engine, http://www.geaviation.com/marine/engines/military/lm6000/, 2015.

[16] Levelized Cost of Energy Calculation, Black & Veatch, February 2015.

[17] Nuclear Energy Agency/International, Energy Agency/Organization for Economic Cooperation and Development, Projected costs of generating electricity, 2010 Edition.