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Effect of a real steam turbine on thermoeconomic analysis of combined cycle power plants

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ABSTRACT

The continuing depletion of fossil fuels and the growing restrictions for greenhouse emissions, leads to reprocess wasted heat generated by power plants. For this purpose, Combined Cycles Gas Turbine (CCGT) represent a strong technology to obtain, an increase of performances and competitive costs within global market.

To design the CCGT configuration, energy engineering companies should define and analyze the performances of bottomer cycle, imposing operating parameters of steam turbine and heat recovery boiler. Usually, these plant components are supplied by different manufacturers so the plant could not be globally optimized.

Considering a steam turbines manufacturer as GE Oil&Gas, a high level of components integration, is a chance to optimize globally the bottomer cycle, determining the best machine in terms of efficiency and improving plant productivity. This aim could be obtained through the development of a high level of combination between company simulation codes and energy balance codes.

In this paper, a two-pressure level combined cycle is examined and optimized. The best thermoeconomic configuration is obtained: first, imposing steam turbine efficiency and using literature costs correlations; then, acquiring the efficiency by a steam turbine industrial tool and considering real machines costs. Therefore, two distinct best configurations could be determined and compared.

1. Introduction

Nowadays, the continuing depletion of fossil fuels coupled together with emissions restrictions attributable to greenhouse gases, steers to reprocess wasted heat generated by power plants with the aim of enhance global efficiency. One of the technology employed for advancements is the combined cycle power plant. In a progressively competitive market, managed by profits, cutting costs for generating electricity is coming to be crucial, in order to ensure a fast return on investment, however without reducing the power plant reliability and flexibility. Currently, available power-generation combined-cycle plants achieve net plant thermal efficiency typically in the 50–55% LHV range. Further development of gas turbine, high-temperature materials and hot gas path cooling technology, show promise for near-term future power generation combined-cycle systems, capable of reaching 60% or greater plant thermal efficiency [1].

Often, an energy engineering company should define and to analyze the performances of bottomer steam cycle, imposing the operating parameters of the steam turbine and of the heat recovery boiler. Usually two or more distinct manufacturers fabricate these elements. Due to this, the plant could not be globally optimized, because the energy-balance designer can get the real steam turbine performance and cost only after the entire bottomer cycle is defined.

Considering a steam turbines manufacturer's point of view, as GE Oil&Gas, the integration between a property simulation code and an energy balance code is a chance to evaluate globally the bottomer cycle in order to determine the best plant configuration and to help the final customer to operate properly the plant.

Fig. 1 shows the stream cycle of the process followed to optimize the bottomer plant in the two cases: standard case, using literature costs correlations and a constant value for the efficiency of ST; the

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Nomenclature

A	Final section area, [m^2]
c	Cost, [M$]
COE	Cost of Energy, [$/MWh]
E	Energy, [kW]
m	Mass flow, [kg/s]
p	Pressure, [Pa]
T	Temperature, [K]
x	steam quality, [−]
W	Power, [kW]
η	Efficiency, [−]

Subscript

app	Approach
bott	Bottomer
CC	Combined Cycle
Cond	Condenser
eco	Economizer
eva	Evaporator
f	fuel
gas	gas
HP	High pressure
HRSG	Heat recovery steam generator
in	Inlet
inj	Injection
is	Isentropic
l	Liquid
lim	Limit
LP	Low pressure
LTE	Low Temperature Economizer
O&M	Operating and Maintenance
pp	Pinch Point
pump	Pump
sh	Superheater
ST	Steam turbine
stack	Stack
sub	Subcooling
TCR	Total Capital Requirement
tot	Total
v	Vapor

Acronyms

BOP	Balance of Plant
C	Compressor
CC	Combustion Chamber
CCGT	Combined Cycle Gas Turbine
COE	Cost of Energy
GT	Gas Turbine
HP	High Pressure
HRSG	Heat Recovery Steam Generator
LTE	Low Temperature Economizer
LP	Low Pressure
O&M	Operating and Maintenance Cost
SPRINT	Spray inter-cooled turbine
ST	Steam Turbine
T	Turbine/expander
TEC	Total Equipment Cost

case proposed in this paper, where real parameters of ST costs and efficiency are used.

In the standard case the process will stop at the first cycle, when the bottomer cycle is optimized from a thermodynamic point of view, without considering variation of efficiency and real models of ST. In the new case, first cycle is the same of the previous one, when first attempt of best plant configuration is found, combining the industrial tool with the energy balance code, a new configuration of ST is found. So, is possible to discover new best plant configurations.

A thermodynamic and economic analysis of combined cycle power plants was carried out by several authors [2–23]. Attala et al. [2] have developed a tool aimed at thermo-economic valuation and optimization of thermal power plants. Roosen et al. [3] considered the optimization of a combined cycle, proceeding with a strict direct cost assessment. Rao and Francuz [4] found and evaluated advanced improvements for combined cycle that will manage to get considerable performance improvements in coal based power systems. Carapellucci and Giordano [5] made a comparison between two different approaches for optimizing CCGTs. Zhu et al. [6] considered the effect of solar addition to describe the combination of solar thermal energy with a natural gas combined-cycle (NGCC) power plant. Tică et al. [7] showed a method to convert a CCGT physical model designed for simulations in an optimization-oriented model, which can be further used with efficient algorithms to improve start-up performances. Ganjehkaviri et al. [8]

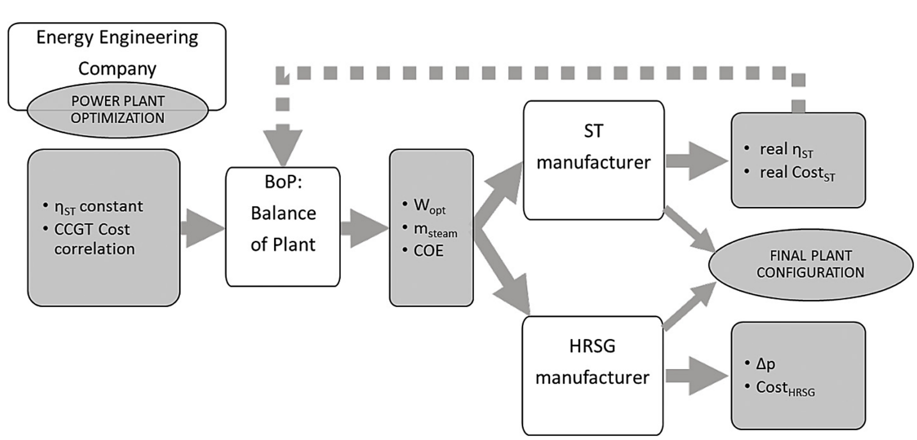

Fig. 1. Stream cycle of the two processes adopted (dashed line is new iterative procedure).

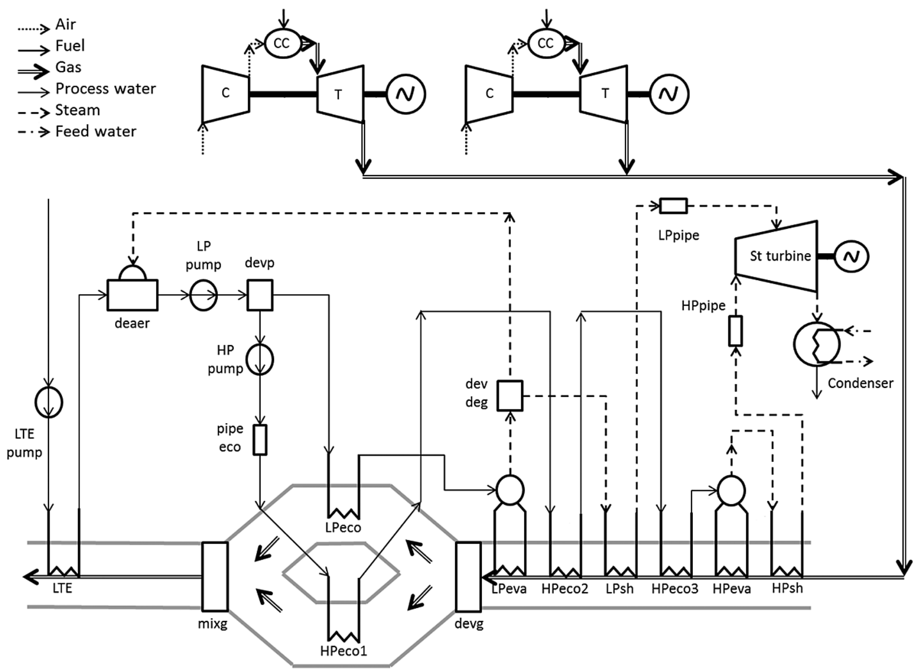

Fig. 2. Combined Cycle layout.

optimized a combined cycle with dual pressure HRSG, analyzing three cases with different steam quality. Kotowicz and Bartela [9] presented the analysis of the influence of fuel price variation on the optimal values of the design variables of the steam part of a combined cycle plant. Furthermore, Facchini and Carcasci [10–21] have analyzed GT power plants in design and off-design configurations. Several thermodynamic cycles are examined, i.e.: a comparison between two heavy duty gas turbines employed for combined cycle applications [12], a Chemically Recuperated Gas Turbine cycle with a specified HRSG and Mass Steam Reformer analysis [14–16], thermoeconomic district heating analysis utilizing a gas turbine [13], Joule-Joule combined cycle [20] and others thermodynamic cycle [19].

The new approach proposed in this paper concerns the novelty of the analysis of performances, costs and configurations using real ST data, which only an industrial and manufacturer partner can provide. The performances and costs depend on size of the ST machine model, the analysis shows that these parameters present a non-continuous function, contrary to results obtained with literature correlations. In real cases, these curves present discontinuity in presence of a change in the ST machine model configuration.

The goal is to analyze the differences between the thermoeconomic optimization of the CCGT plants performed, first with academic simulation codes and costs correlations, then considering the effect of ST real configurations and costs on the optimization of the whole plant.

In these previous papers, the steam turbine cost is evaluated using literature correlations. Carcasci et al. [21] show the effect of steam turbine real costs and configurations on one-pressure level combined cycle and enlightening the relevance of results. The aim of this paper is to model a real combined cycle with a two-pressure level HRSG, focusing on the effect on CCGT, in terms of thermodynamic and economic performances. This analysis is led by using a simulation code, (ESMS) developed by University of Florence for GE Oil&Gas to evaluate the plant behavior, and the industrial simulator for steam turbines, property of GE Oil&Gas. Furthermore, two academic steam turbine costs correlations and the output costs provided by the industrial tool are then integrated in the analysis. The effects of employing these three different configurations on best thermo-economic plant design point, varying the pressure level in order to optimize steam turbine output power and COE (Cost of Energy), are finally compared. The steam turbine industrial tool can supply a reliable cost of the machine and a correct value of isentropic efficiency. The GE Oil&Gas customized version of the ESMS (Energy System Modular Solver) modular code is used for modelling the entire bottomer cycle.

2. The ESMS cycle analysis code

Power plants constructed with gas turbine engines are not so complex, however, in order to simulate them, a flexible and specified tool is needed. Gas turbine engineers employ ad-hoc codes to simulate separately components, because a large number of particulars are required. The reader is referred to references [10–18] for a whole presentation of the code, related theory and several engineering applications. The most essential feature of this modular simulation code is the capability to simulate a different power plant configuration without generating an additional source program. The power plant configuration is described by linking several basic modules representing distinct part of the process such as: compressors, combustion chambers, mixers and so on. Each module is described as a black box able to simulate a set chemical

Table 1
LM6000PF SPRINT data sheet [22].

LM6000PF SPRINT data sheet	Value	Unit
ISO Rated Power	47500	kW
Heat Rate	8649	kJ/kWh
Electrical Efficiency	41.3	–
Pressure Ratio	31.1	–
Exhaust Mass Flow	133	kg/s
Turbine Speed	3627	rpm
Exhaust Temperature	446	°C

and thermodynamic transformation. The resulting series of non-linear equations identifying the power plant is next linearized (the coefficients are, however, updated during of the simulation). Totally equations are therefore solved simultaneously applying a classic matrix approach; thus, the process is fundamentally the fully implicit linear approach.

3. Gas turbine combined cycle description

In Oil & Gas combined cycle are usually a secondary service. As an example, waste products from refinery process are used in CC plant to supply energy to the self-plant. The most suitable configuration to carry out this task is by using HRSGs with one or two level of pressure, steam turbines with one extraction/injection and air or water condenser. Fig. 2 shows a combined cycle with a two-pressure level heat recovery steam generator (HRSG). The topper cycle is composed by a couple of typical industrial cooled gas turbines; the exhaust gases outgoing from gas turbine, cross the heat recovery steam generator.

The steam side of the boiler is composed by two circuits, part of two different pressure levels, through which the water is carried in conditions of superheated steam and directed in the steam turbine. The fluids involved in heat exchange, work in countercurrent and the arrangement of the sections of economizers, evaporators and superheaters presents some particularities, compared to the simple series arrangement of bodies of low and high pressure.

Following the path of exhaust hot gas outgoing from the two gas turbines (Fig. 2), the high pressure section of superheater and of evaporator exchangers is located, referred to as respectively *HPsh* and *HPeva*. Then the economizer (*HPeco3*) is positioned, connected to the evaporator. The blocks of the high pressure economizers (*HPeco1, HPeco2, HPeco3*) are divided into three different banks, placed in interposition with other exchangers. The low pressure superheater (*LPsh*) has been placed between the second and the third high pressure economizer bank. Then, following the direction of the hot gas, the low pressure section evaporator (*LPeva*), is located. A parallel arrangement has been adopted for the low pressure economizer and the high pressure economizer (*LPeco, HPeco1*). Finally, a further heat exchanger is located, downstream compared to these two economizers in parallel. This exchanger, referred to as the low temperature economizer (*LTE*), heats low temperature water coming from the condenser, allowing a better heat recovery and lower temperatures reaching the stack.

Turning to the water-steam path, a first pump (*LTEpump*) draws the low pressure water from the condenser. The pump brings water to a pressure level of about 1.5 bar, so, to avoid the formation of steam within the tube bundles of the first heat exchanger and to reduce the cavitation problems for the downstream pump. The low pressure pump (*LPpump*) feeds the low pressure water-steam circuit, bringing water from deaerator (*deaer*), crossing the deflector element (*devp*), to the high pressure pump (*HPpump*). This pump feeds the high pressure circuit. The pressure losses are modeled by assigning a loss coefficient for each exchanger element.

Table 2
COE parameters assumptions.

Economic parameters	Value	Unit
%TEC = BOP	12.0%	–
%TEC = Engineering Costs	8.0%	–
%TEC = Contingencies	5.0%	–
Fuel price	6.0	$/GJ
Fixed O&M	14.0	$/kW-y
Variable O&M	0.50	mill$/kWh
Yearly operating hours	7500	h/y
Discount rate	10.0%	–
Plant working life	25	y

3.1. Topper gas turbine

The topper cycle, is composed by an industrial aeroderivative gas turbine (LM6000PF SPRINT, [22,23,24] by GE). Its efficiency and operational flexibility make the LM6000 a cost-effective choice for all applications. The LM6000 gas turbine -the most efficient LM unit in its class for combined-cycle and cogeneration applications- can provide output from 53 MW to 62 MW with efficiencies up to 52%. 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 [23]. Distinct from most gas turbines, the LM6000 SPRINT series is primarily controlled by the compressor discharge temperature instead of the turbine inlet temperature, allowing some of the compressor discharge air to be used to cool high-pressure turbine components. SPRINT (SPRay INter-cooled Turbine) technology reduces compressor discharge temperature, allowing advancement of the throttle to significantly enhance power and improve thermal efficiency. The LM6000 SPRINT system is composed of atomized water injection at both low-pressure compressor (LPC) and high-pressure compressor (HPC) inlet plenums. Injection is accomplished with the eighth stage of a high-pressure compressor bleeding air to feed two manifolds and sets of spray nozzles, where the water droplets are sufficiently atomized before injection at both LPC and HPC inlet plenums. SPRINT packages are well-suited for both direct drive power generation at 50 Hz and 60 Hz and variable speed for mechanical drive.

Gas turbine LM6000PF SPRINT, whose basic data are listed in Table 1, is used to model the topper cycle; this data sheet are referred to ISO conditions. In the present analysis, the gas turbine is not simulated, but exhaust hot gas parameter is used directly.

3.2. Steam turbine

GE projects and manufactures several steam turbines designed for Oil & Gas business. The production takes account of machines for mechanical drive and power generation, condensing and back pressure turbines, with or without extraction. They are designed for top thermodynamic and mechanical performance, for installation in chemical, petrochemical and industrial plants, supplied whether stand-alone drivers or as part of a GE Oil&Gas complete turbo set [25].

Backpressure turbine employs pressure drop available from two steam system at different pressures. It is used for operation with

Table 3
Power plant parameters used for thermodynamic analysis.

Parameter	Value
p_{LTE}	1.5 bar
p_{LP}; p_{HP}	varied
p_{cond}	0.232 bar
$T_{stack,lim}$	100.0 ÷ 120.0 °C
ΔT_{app_HPsh}	18.0 °C
ΔT_{app_LPsh}	31.5 °C
ΔT_{pp_HPeva}	8.0 °C
ΔT_{pp_LPeva}	8.0 °C
ΔT_{sub_HPeva}	9.1 °C
ΔT_{sub_LPeva}	5.9 °C
$\Delta T_{gas/H2O\_HPeco1}$	14.0 °C
$\Delta T_{gas/H2O\_HPeco2}$	30.6 °C
x_{lim}	0.89
η_{ST}	0.822
η_{pump}	0.87

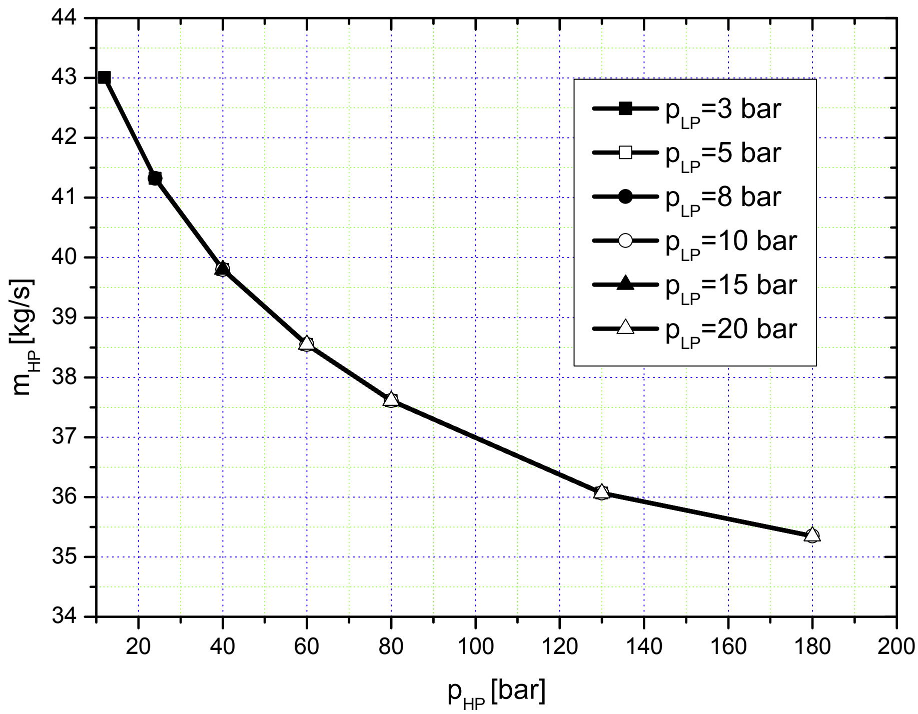

Fig. 3. Trend of HP turbine inlet mass flow versus high pressure circuit.

low values of pressure and temperature (up to 90 bar and 520 °C). When steam is available with higher values than these, different types of machine are exercised, allowing ranges of pressures up to 140 bar and 540 °C [25].

Manufacture includes impulse and reaction stages. This double design allows ensuring the better solution between increasing enthalpy drop and reducing size of machine guaranteeing wide level of operability.

Concerning the real ST analyzed in this paper, industrial configurations of ST manufactured by GE Oil&Gas are considered.

Industrial design phase needs a deeper knowledge of the working cycle of the machine. Parameters must be defined by the customer and steam turbine design engineer should build the better solution in accordance with the standard and GE design criteria.

First step is the definition of the geometric size of the machine: inlet head size and discharging casing size are strictly correlated with the volumetric flow. The inlet and outlet sizes selection imposes limitation in the axial length of the machine and consequently on the steam path configuration.

Extractions or injections force the design engineer to optimize the internal chambers. The industrial tool sets the proper number of drums and stages defying all the geometric parameters of the blades to obtain high efficiency.

Once the geometric and thermodynamic configuration are defined, the units are ready to start with the manufacturing phase.

Often, CCGT selection tools, as ESMS simulation code, consider the steam turbine only in the thermodynamic perspective, using an average efficiency value for the whole machine. This is highlighted in most theoretical correlations, that did not take in account geometric/manufacturing evaluation.

4. Cost analysis

The optimization of costs analysis for a combined cycle can be achieved minimizing the objective function represented by the cost per unit of energy, known as Levelized Cost of Energy (LCOE), simply COE, commonly measured in \$/kWh or \$/MWh. This index corresponds to the price to which the electricity should be generated using a specific resource, so, to reach the break-even point. COE index is obtained by an economic balance considering the whole costs across the life of the plant: initial investment costs, operation and maintenance costs, fuel costs, capital costs. COE is very functional to evaluate the final cost for the electricity generation, considering several resources and comparing technologies, including dissimilar operating characteristics [26]. COE can be expressed by means of a formula suggested by the International Energy Agency (IEA) [27]:

$$COE = \frac{c_{TCR} \cdot CRF + c_{O\&M} + c_f}{E} \tag{1}$$

COE can be defined by three parameters: *TCR* (Total Capital Requirement), expressed as sum of capital costs, interests occurred during the construction and pre-production costs, multiplied for "Capital Recovery Factor" in order to taking into account of discount rate; *O&M* (Operating and Maintenance) costs; *FP* (Fuel Price). The operating and maintenance costs can be divided into a fixed and a variable contribution; they are calculated with the hypotheses summarized in Table 2. Fuel price (FP) is fixed at 6.0 \$/GJ. *E* represents the total generated energy.

The costs correlations assumed in order to evaluate the equipment costs of the plant are implemented in the ESMS code. An extensive presentation of these costs correlations is presented in previous papers [2,3and5]. In this paper, the focus on steam turbine cost correlations is carried out, in details: by Attala et al. [2]:

$$C_{ST} = 3197280 \cdot A^{0.261} + 823.7 \cdot W^{1.543} \tag{2}$$

by Roosen et al. [3]:

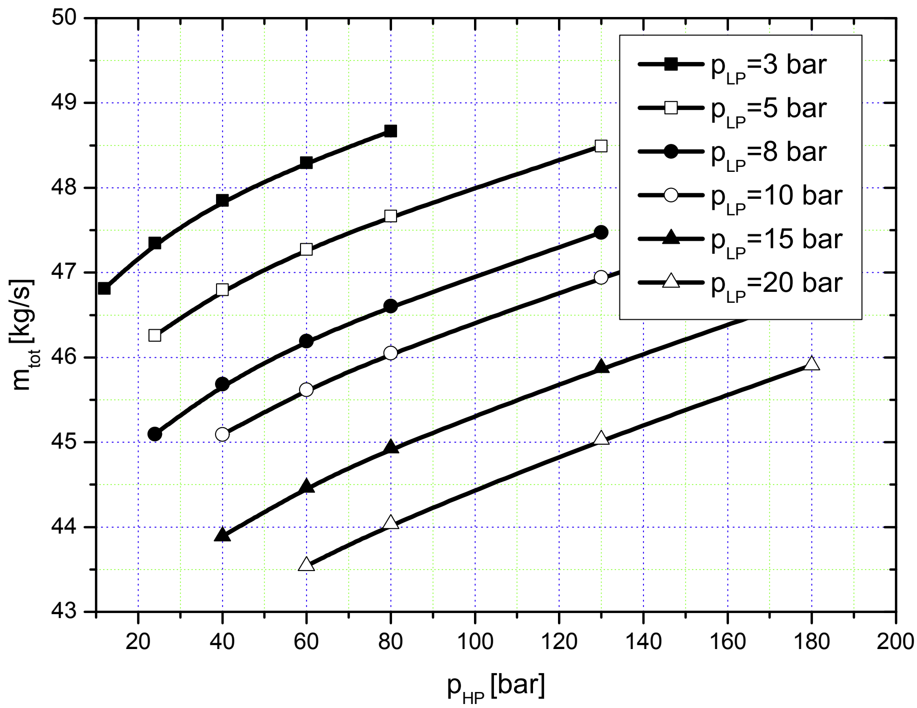

Fig. 4. Trend of LP turbine inlet mass flow versus high pressure circuit.

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

Both correlations are depending on the steam turbine output power, but other parameters are neglected. Considering the Roosen et al. correlation, is valuated the effect of thermodynamic isentropic efficiency and inlet steam temperature, but there is not any consideration about outlet turbine section A, that is proportional to outlet steam mass flow rate and outlet steam quality. In fact, as regards outlet section equation, the specific volume v can be

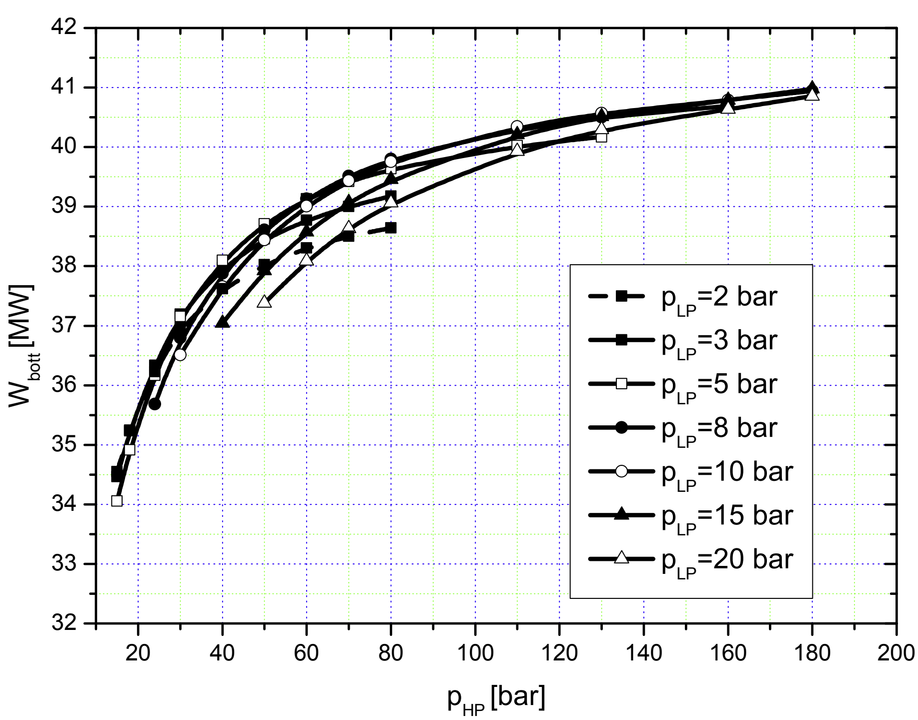

Fig. 5. Trend of total net ST power obtained with ESMS versus high pressure circuit.

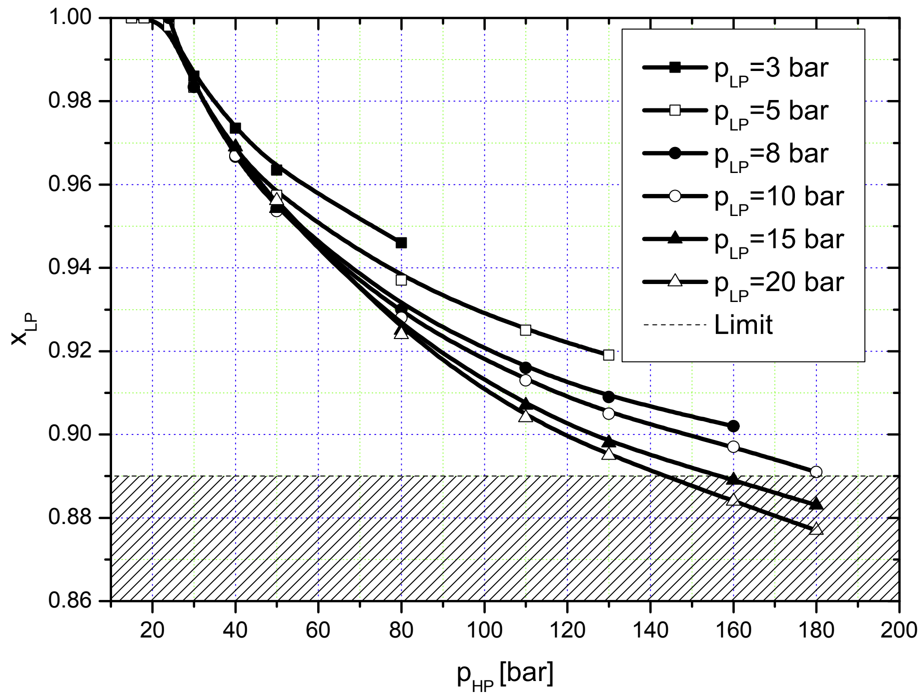

Fig. 6. Trend of steam quality conditions for final turbine section versus high pressure circuit.

expressed both on vapor and liquid fraction because of saturation condition of steam. Since that liquid specific volume v_l is negligible in comparison to the vapor one, A can be evaluated as:

$$A \cong \frac{m \cdot v}{c_x} = \frac{m \cdot [v_l + x \cdot (v_v - v_l)]}{c_x} \cong \frac{m \cdot x \cdot v_v}{c_x} \quad (4)$$

Therefore, the outlet steam turbine section is related to steam mass flow rate, steam quality and vapor specific volume, these parameters are depending on condenser pressure. The steam turbine manufacturer employs the own company industrial tools to calculate the efficiency of a commercial machine and its cost. Thus, literature costs correlations (i.e. those proposed by Attala et al. and Roosen et al.) can be matched with the results extracted by the

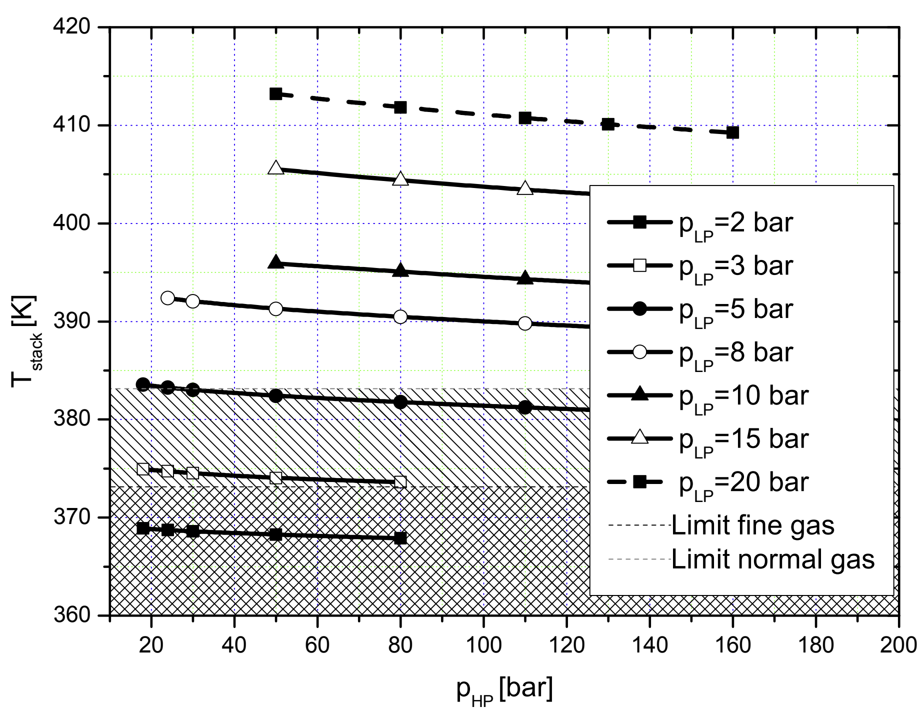

Fig. 7. Trend of stack temperature versus high pressure circuit.

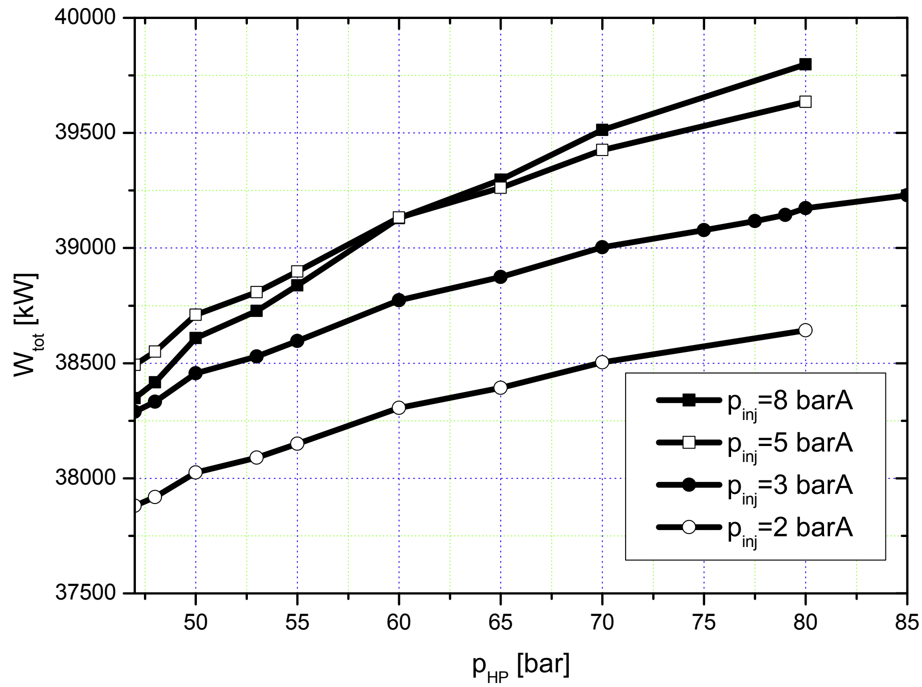

Fig. 8. Trend of total net ST power obtained with ESMS obtained with fixed steam turbine efficiency, enlargement.

industrial tool, in order to compare the respective trends, varying the pumps pressure. In this paper, steam turbines machines are selected applying a tool developed by GE.

5. Thermoeconomic analysis

5.1. Methodology

First, the simulation of complete bottomer cycle is done with the ESMS code; the output files generated by ESMS can be analyzed to extract the inlet parameters (i.e.: inlet steam mass flow rate, pressure and temperature conditions) of the steam turbine element. These data can be set as inlet values in the input files of industrial tool, in order to select steam turbine class. So, real physical parameters proper of steam turbine machines, as performances and costs, can be obtained; in particular the efficiency, whose value, varies depending on inlet parameters. Otherwise, employing the academic correlations, fixed efficiency value should

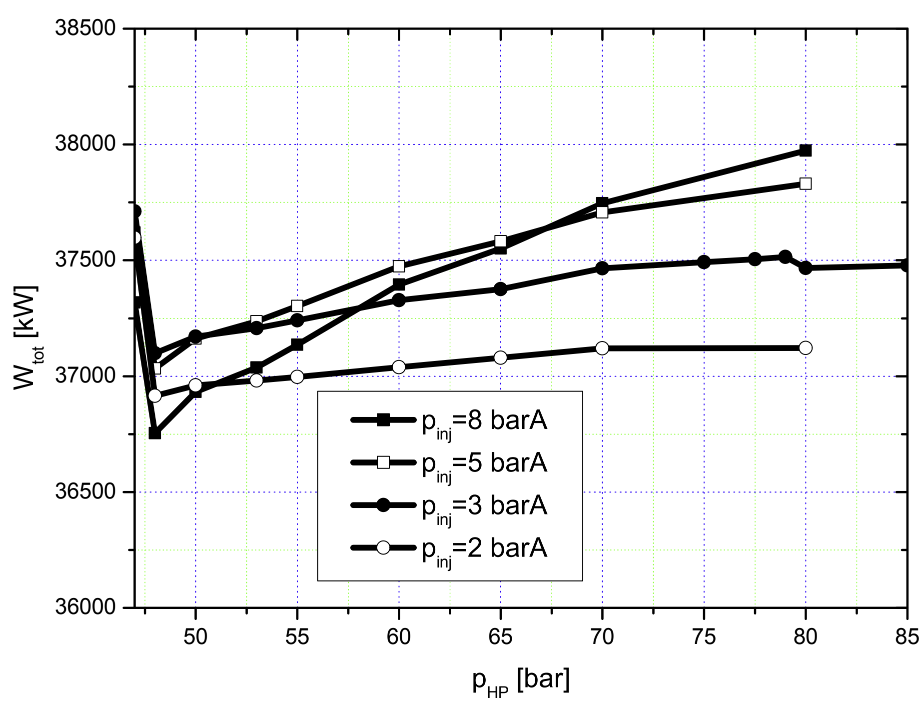

Fig. 9. Trend of total net ST power obtained with industrial tool obtained with real variable steam turbine efficiency, enlargement.

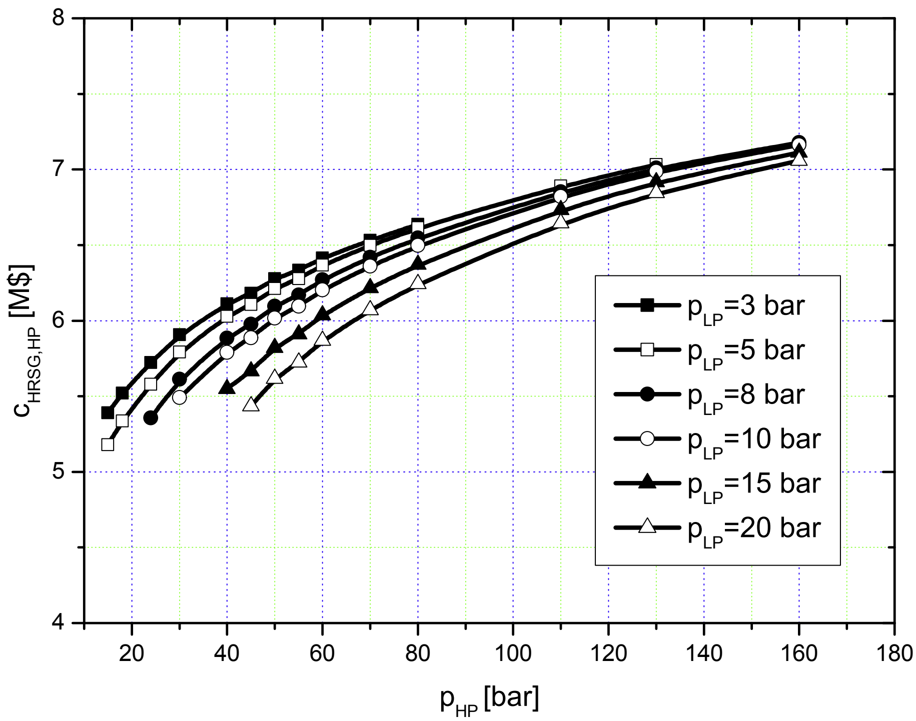

Fig. 10. Trend of cost of high pressure section of the HRSG obtained with Roosen correlation.

be imposed. Thus, a comparison for the entire power plant, using, once data obtained with industrial tool, once data obtained from ESMS and literature costs correlations, is carried out.

The thermodynamic and performance analysis of the combined cycle is done varying the outlet pumps pressure in the two water/steam circuits. The conditions of hot gases, at the boiler inlet, have not been modified in the simulations, leaving in fact fixed the topper cycle parameters. The gas turbines have not been modeled directly, but reference is made only to the characteristic curves for the gases. So, the inlet parameters for HRSG unit, i.e.: the

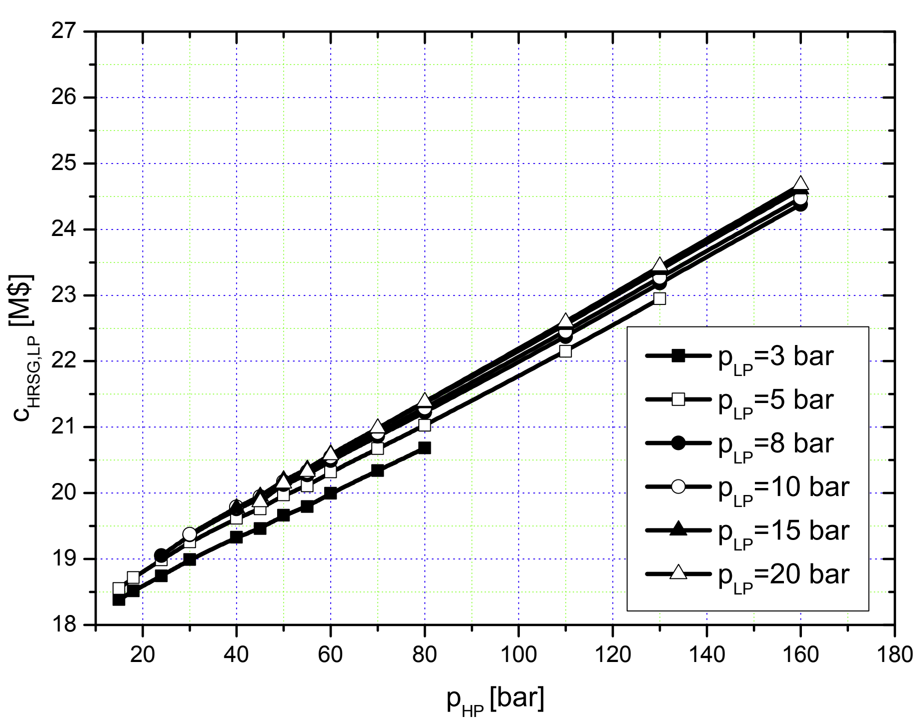

Fig. 11. Trend of cost of low pressure section of the HRSG obtained with Roosen correlation.

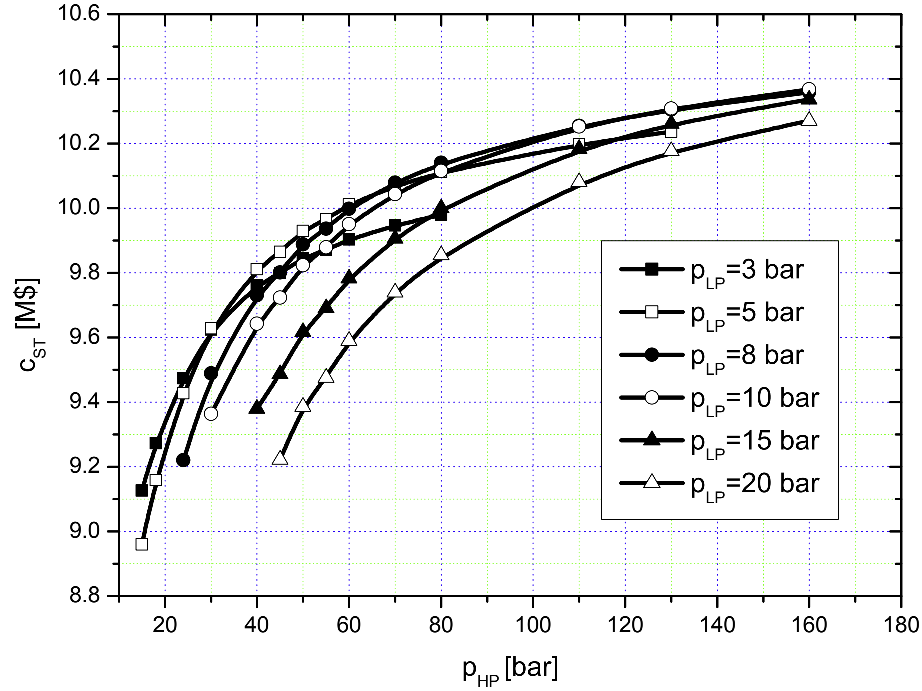

Fig. 12. Trend of cost of steam turbine obtained with Roosen correlation.

temperature and the hot gas mass flow rate, are fixed (Table 1).

Other boundary conditions, concerning the main plant parameters (Table 3), as well as the pinch point and approach point, are fixed. Using the energy balance in the high pressure section of HRSG, the high pressure steam mass flow rate and the outlet temperature of high pressure section can be determined. Then, analogous calculation can be repeated in the low pressure section of HRSG, using the low pressure parameters.

In the high pressure section of steam turbine, evolves the high pressure steam mass flow rate, which inlet enthalpy conditions are determined by maximum temperature and high pressure level values. Steam expands up to low pressure turbine section, in these

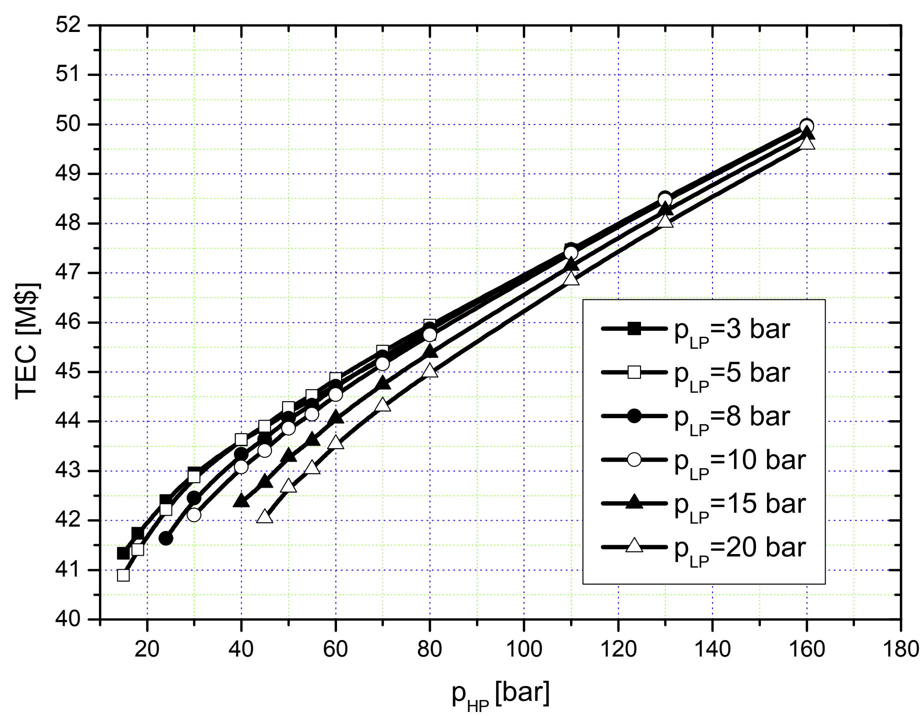

Fig. 13. Trend of total equipment cost of bottomer cycle and HRSG.

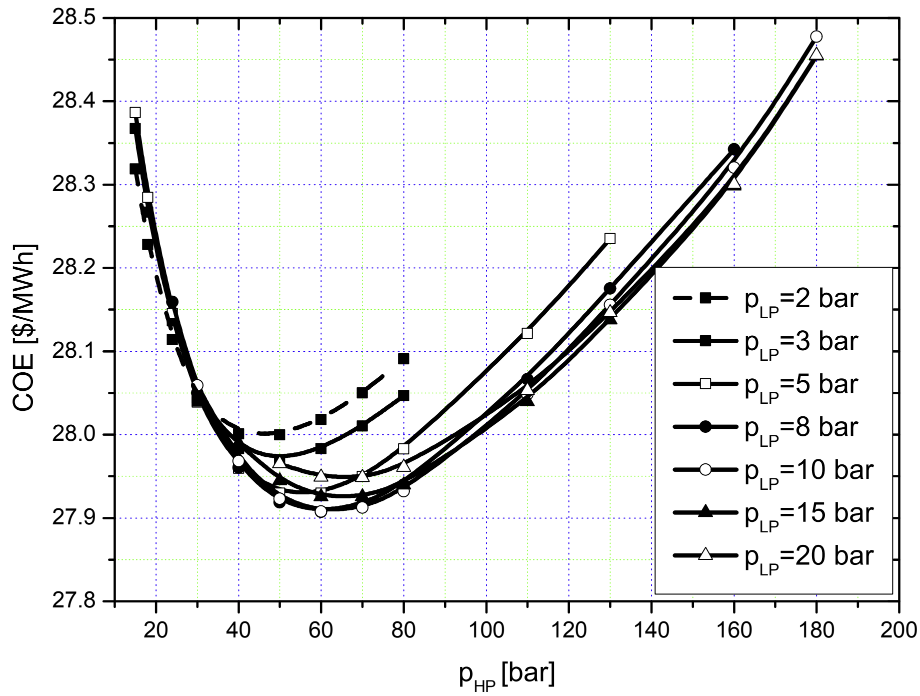

Fig. 14. Trend of COE of combined cycle power plant obtained with Roosen correlation for the equipment costs.

conditions the steam mass flow rate, passing through high pressure circuit, is injected into the low pressure turbine, so whole steam flow expands up to condenser pressure condition.

The net output power plant is determined deducting pumps power.

5.2. Thermodynamic results

The trend of the water/steam flow for the HP circuit (Fig. 3) shows that, the mass value is independent by the low pressure parameter. This can be explained considering that the flow rate required for the heat exchange in high pressure circuit, is determined mainly by the first two exchangers which encounter the hot gas. The total flow which evolves in LP turbine is given by sum of flow evolving in HP circuit plus that of injection flow. The overall trend can be seen in Fig. 4.

The trend of net power for bottomer cycle, within the range of pressures analyzed, is shown in Fig. 5; power curves tend to approach. The maximum point is obtained for about 10 bar as regards the LP circuit and for about 180 bar regarding the HP circuit.

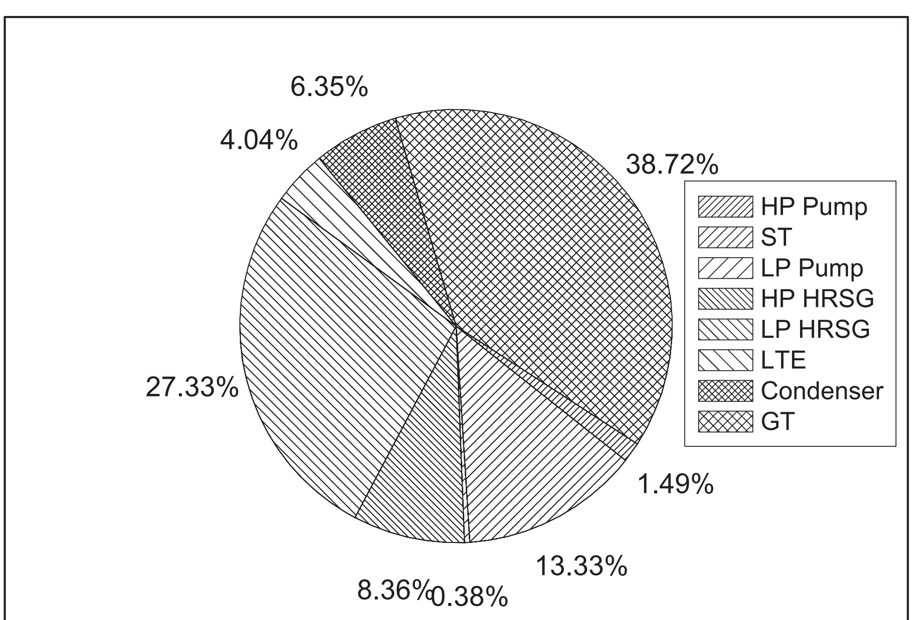

Fig. 15. Portions of each TEC component in the best configuration.

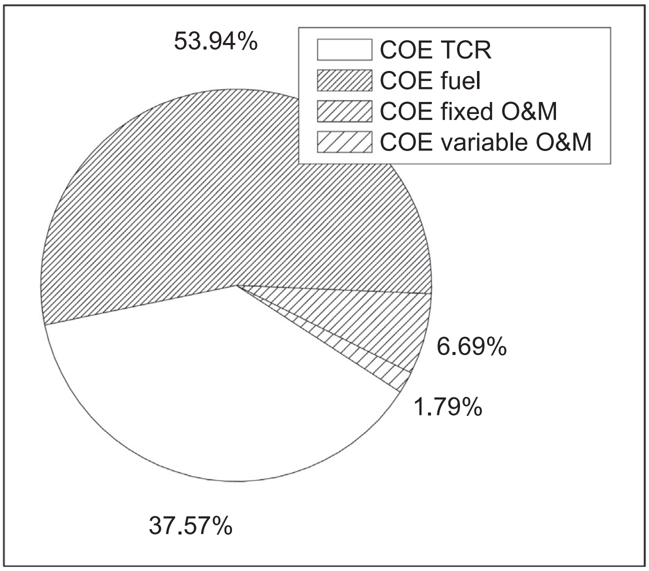

Fig. 16. Portions of each COE component in the best configuration.

By imposing limits on stack temperature and on steam quality conditions in outlet section of LP turbine expansion, the range of plant operability gets narrower. The range of lower steam quality appears to be in correspondence of highest pressure levels in both circuits. The reason lies both in condensation pressure value and in maximum temperature level of HP circuit, which remain unchanged, being fixed in boundary conditions. So, that, maximum temperature of LP circuit increases. Observing graph (Fig. 6), range over 160 bar for HP circuit and over 15 bar curve level for LP circuit, can be excepted. Also, range over 140 bar for HP circuit, and for 20 bar level as regards LP circuit, can be left out.

Trend of stack temperature (Fig. 7) mainly depends on LP circuit pressure. This parameter, in fact, controls the amount of heat exchanged by last economizer. The stack temperature remains elevated for high pressure levels while, for lower pressure, temperature tends to decrease and to get closer to dew point of gas. The nearly horizontal lines visible in graph of Fig. 7, confirm how gases temperature is much more sensitive to pressure variations of LP circuit, compared to those of HP circuit. Looking the graph, the range of LP levels between 3 and 6 bar should be excluded, consequently also corresponding HP levels.

In the end, considering best thermodynamic plant conditions, power generated is about 131 MW: approximately 90 MW generated by the topper cycle and 41 MW generated by the bottomer cycle. The efficiency of combined cycle in this configuration is approximately 55%.

Now the impact of employing a real commercial steam turbine on thermoeconomic analysis, is studied. Analysing the results of power trend, whenever output files of industrial tool provide a change of class for steam turbine, compared to pressure value of previous step, a condensation of simulations was carried out. In order to detect accurately, the transition from a class to another. This transition will appear in following charts with a discontinuity, there are as many "steps" as classes of steam turbines used. The graphs of results obtained with ESMS, compared with corresponding generated by GE Oil&Gas industrial tool, have on the contrary a continuous trend [21].

In this case, the steam injection mass flow is specified in the input files of industrial tool, corresponding in the ESMS model to second pressure level, in particular, low pressure module. The industrial tool, as already mentioned, in addition to flow rate, pressure, inlet temperature values and condensation pressure, requires, in this case, also: flow rate, pressure and temperature of injection. In particular, inlet ST pressure, corresponding to high pressure pump, less load losses, is varied between 47 and 80 bar, while injection pressure, corresponding to low pressure pump, less load losses, is varied by 2, 3, 5–8 bar.

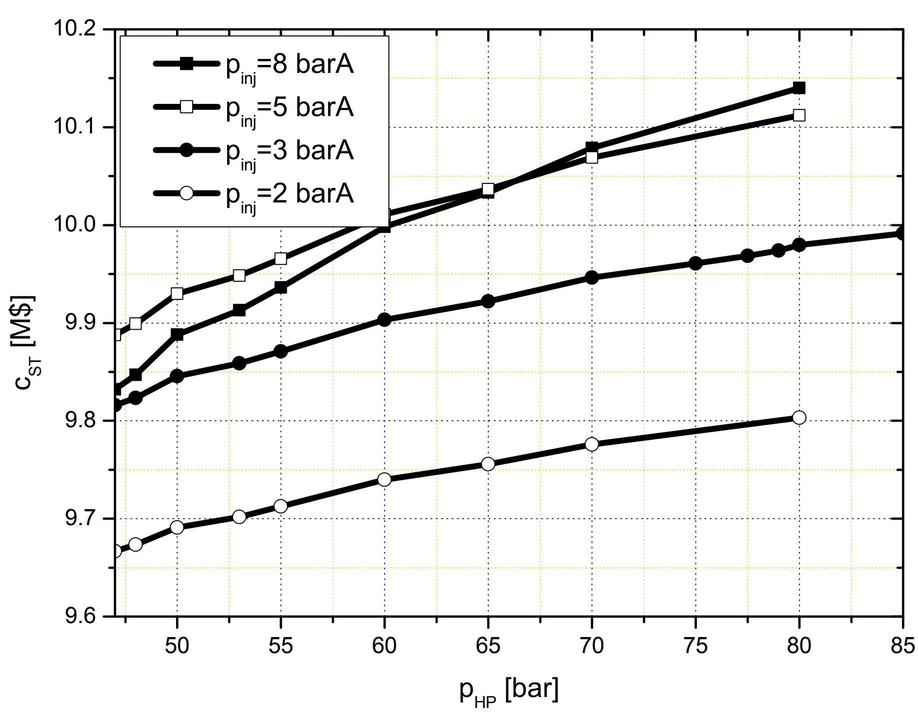

Fig. 17. Trend of cost of steam turbine obtained with Roosen correlation, enlargement.

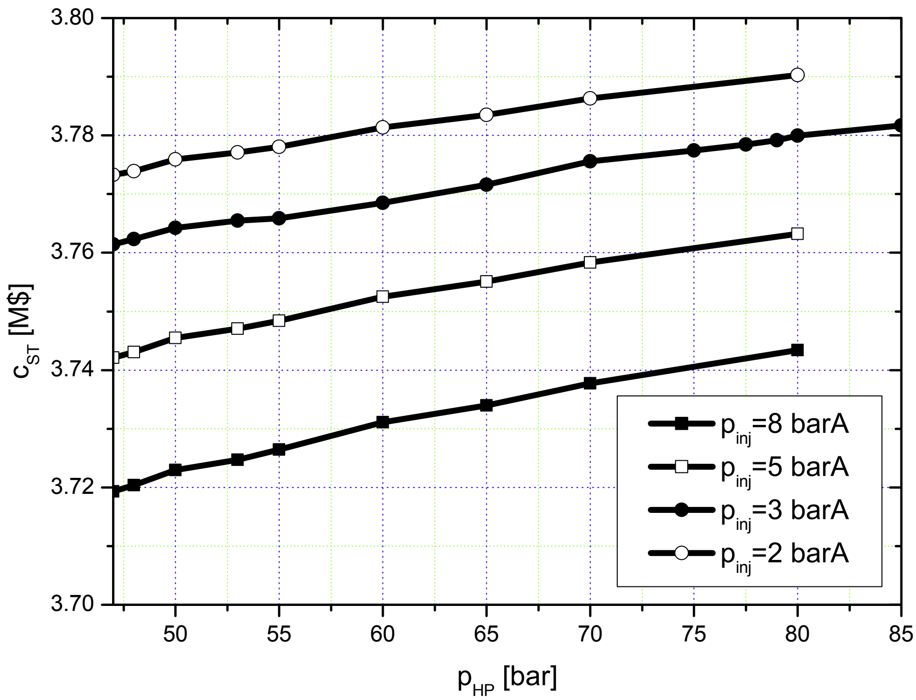

Fig. 18. Trend of cost of steam turbine obtained with Attala correlation, enlargement.

In this pressures range, the steam turbines simulator tool has selected the same machine class, a condensing steam turbine, with high pressure jump between upstream and downstream. In fact, power and costs trends become evidently continuous.

Now, considering the GE Oil&Gas real steam turbine efficiency (not fixed in boundary conditions as in previous analysis) is carried out using literature correlations. This fact has a double effect: first (thermodynamic effect), because the power generated is calculated considering real variations of steam turbine efficiency; second -economic effect-, in fact power values employed inside correlations for equipment costs are calculated with fixed steam turbine efficiency.

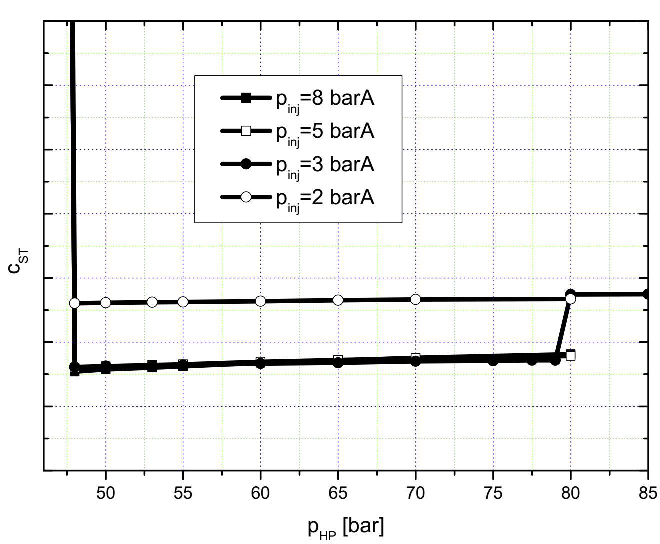

Fig. 19. Trend of cost of steam turbine obtained with the industrial tool, enlargement.

Figs. 8 and 9 show trends of power generated by steam turbine varying HP level circuit, using respectively the ESMS simulation code and the industrial tool. Considering range near 48 bar for HP circuit, the industrial tool has not supplied a standard configuration of steam turbine. So, pressure levels below this condition, may not be considered between operating configurations of plant, in fact, special manufactured steam turbine is requested. The industrial tool provides only ST configurations that could be realized by manufacturers company, so, those machine models that are not included in standard solutions are not considered in this analysis. This consideration is due to the high costs and lead times of special ST configuration. Considering range between 50 and 85 bar, the power calculated with ESMS code overestimate industrial tool values; in both cases, however, power trend varies in a quite limited range. The curve corresponding to 3 bar presents a small discontinuity between 79 and 80 bar, where machine class has changed. In fact, the industrial tool, in the range analyzed, provides a change of machine configuration, this is represented with a discontinuity in the trend of ST power.

5.3. Economic analysis

First, trends of cost of main plant equipment elements were examined, considering values obtained in ESMS output files, varying pressure levels of both circuits. The results, calculated using correlation of Roosen, will represent the TEC for the bottomer cycle and for the HRSG. Moreover, fuel cost assumption is 6 $/GJ [1] [5,28] and the number of operating hours per year is 7500, that is equivalent to a capacity factor of around 86%, typical for combined cycles. The effect of escalation rate as regards O&M cost is not taken into account.

[1] Considering the natural gas cost 5.0÷6.0 $/1000 ft^3 (as average value of last 10 years) = 0.1766÷0.2119 $/Sm^3. Using LHV and standard density of natural gas: 0.1766÷0.2119 $/Sm^3/(47450 kJ/kg·0.6785 kg/Sm^3) = 5.48÷6.58 $/GJ.

Trends of equipment costs are represented in the following graphs. Figs. 10 and 11 show respectively the cost of high and low pressure section of HRSG. As high pressure circuit level increases, the costs of both sections increase, more rapidly for low pressure side. In fact, the cost of low pressure section boiler body is very sensitive depending on the high pressure circuit level. Furthermore, this parameter is higher than the corresponding high pressure section. This is due primarily to the trend of average logarithmic Δt value. The high pressure section, on the other hand, presents lower costs while increasing LP levels, this trend is less evident as the HP pressure level value increases.

The cost of the steam turbine (Fig. 12) is closely related to trend of net power generated, aligned with Roosen correlation for steam turbine components.

Now, the TEC of whole bottomer and HRSG systems (Fig. 13) -the cost of the cycle topper was calculated with the correlation of Carapellucci- is plotted. As high pressure circuit level increases, as low pressure circuit value has a lower impact.

Fig. 14 shows the trend of the COE; it presents a steeper trend on the left side of the minimum, in comparison with the right side. Moreover, the minimum of the function tends to the right side while increasing both pressure circuits values. In fact, although there is a TEC increase for plant, the energy generated increases too, this parameter represents the denominator for the COE formula. Therefore, the effect is a shift of minimum of the function to the best thermodynamic configuration. The minimum value is obtained for a low pressure level of about 8 bar and for a high pressure level of about 60 bar.

These conclusions are in contrast in comparison of what found for the one pressure level combined cycle power plant optimization [21]. In fact, choosing to optimize the plant from a thermodynamic point of view or from an economic one, does not produce similar results at all. The operating conditions which make the best configuration, in order to minimize COE, produce an index about 0.6 $/MWh lesser (2%), compared to that obtained in the configuration of maximum power. The bottomer cycle, develops a power of 39 MW (min. COE) instead of 41 MW (max. power), considering a total of about 129 MW instead of 131 MW, so with a minimal difference. The ESMS results, obtained following the best COE configuration, are in fact, aligned with the actual operational conditions of the industrial case examined. The real plant, works, in fact, at a low pressure level slightly below 8 bar and at a high pressure level between 50 and 60 bar.

The weights of costs for system components (Fig. 15) and of each factor which composes the COE function (Fig. 16) are showed, referring to best configuration just found. The topper cycle cost represents the great portion (Fig. 15), while most influential factor for COE function is represented by fuel cost (Fig. 16).

The steam turbine costs are now analyzed, considering Roosen and Attala correlations. Then, a comparison with costs read in the output files of industrial tool is done. The trend of steam turbine cost calculated with the correlation of Roosen is shown in Fig. 17; as had already occurred, these curves are very similar to power one. Trends of 5 and 8 bar levels are similar, while the cost become lower, for a pressure level of 2 and 3 bar. Looking at Fig. 18, there is a huge cost increase while decreasing low pressure levels and increasing high pressure levels. Moreover, trend, using correlation proposed by Attala, is strongly dependent by the steam turbine final section. Finally, the trend of real costs (Fig. 19) is almost constant, as regards range of same machine classes; moreover, there is a strong discontinuity in conjunction with the change of class, observed between 79 and 80 bar, considering a LP level of about 3 bar, which considerably enhances the cost.

Once the steam turbine has been analyzed as an alone element, the impact of these different configurations on the entire system is considered. Thus, COE index for whole combined cycle power plant is calculated using the correlation of Roosen for bottomer plant -except steam turbine element- and for HRSG equipment costs; adding the contribution of the steam turbine cost calculated: once with the correlation of Roosen, once with the correlation of Attala and once with the industrial tool. Furthermore, the values of the bottomer cycle power are calculated using results obtained by

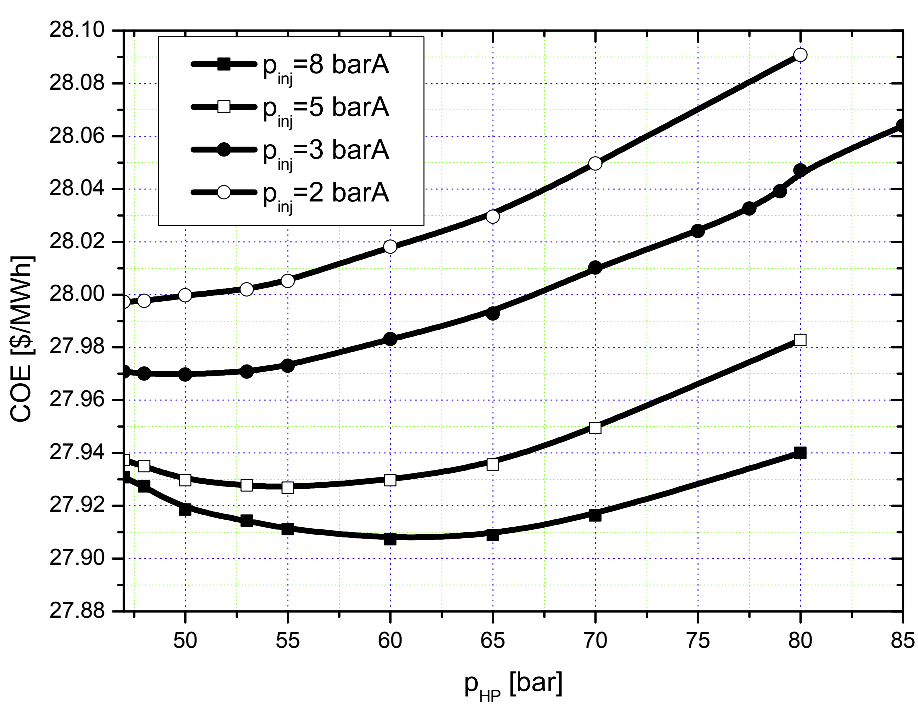

Fig. 20. Trend of COE of combined cycle power plant with the Roosen ST correlation, enlargement.

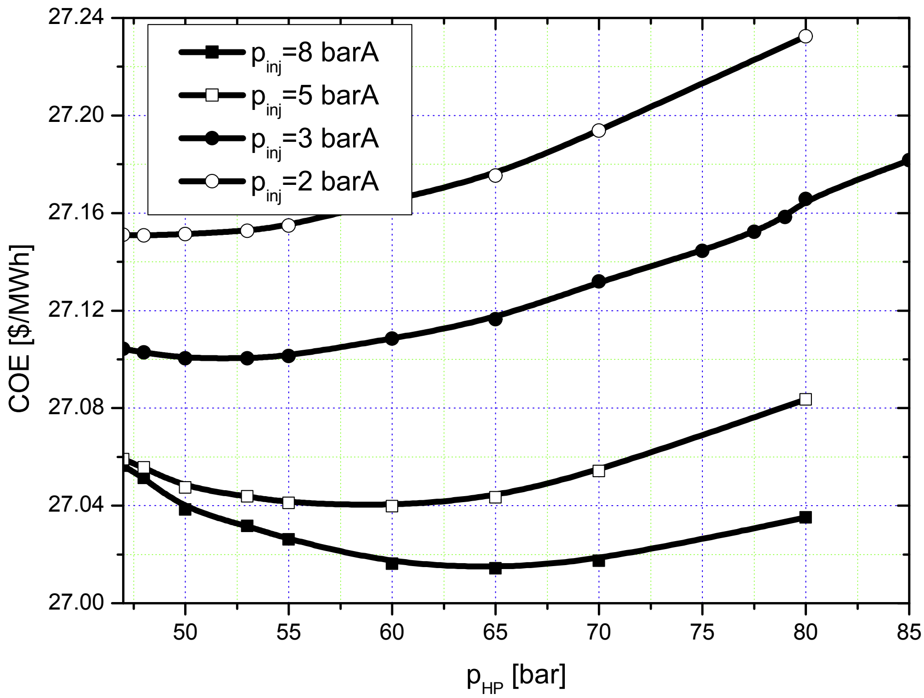

Fig. 21. Trend of COE of combined cycle power plant with the Attala ST correlation, enlargement.

ESMS, as regards first two cases; whereas results obtained with the industrial tool, as regards the real configurations. The simulations are performed considering the basic configuration, which provides 7500 firing hours per year and a fuel price of 6 $/GJ.

Now, (Fig. 20) an enlargement of results presented in Fig. 14, using the correlation of Roosen, is shown; the minimum of the COE function is obtained, as mentioned, for a high pressure level of 60 bar and a low pressure level of 8 bar. Furthermore, using the correlation of Attala (Fig. 21) a similar trend in comparison with previous case is obtained; the minimum of the COE function is shifted to a pressure level of 65 bar keeping, however, same low pressure level. Finally, using an approximation of the real costs obtained with the industrial tool, (Fig. 22), the COE function presents approximately the same trend, but the minimum is shifted. In particular, two new best configurations are found: the first minimum is found in correspondence of a high pressure level of about

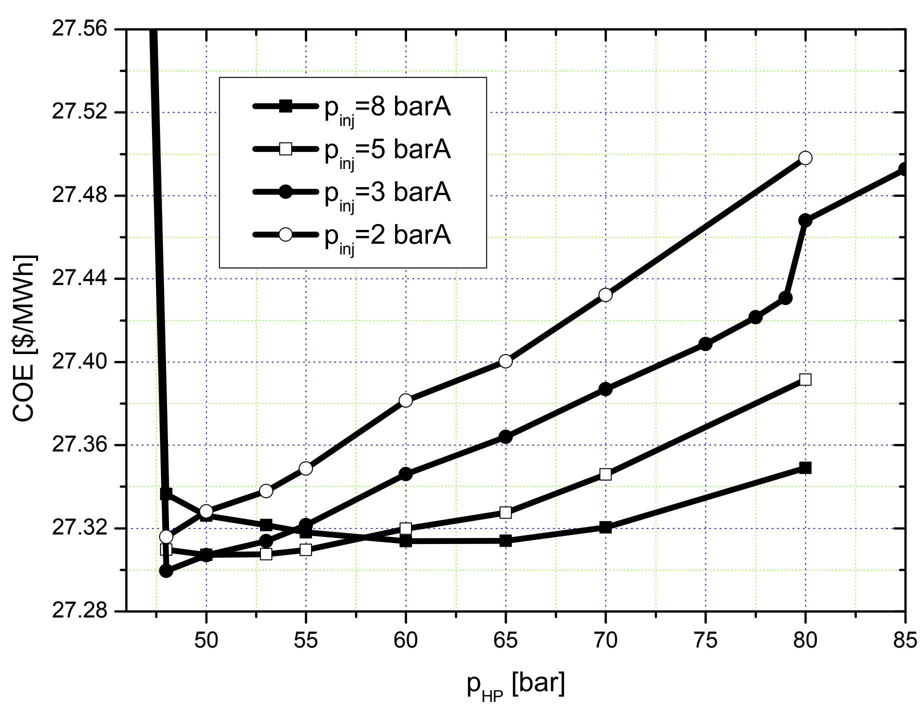

Fig. 22. Trend of COE of combined cycle power plant with the ST real cost obtained with the industrial tool, enlargement.

53 bar and a low pressure level of 5 bar; the second one, near to the limit of specialized manufactured machines, in correspondence of a high pressure level of about 48 bar and a low pressure level of 3 bar. The last one can be considered marginal. The discontinuity was already identified and occurs between 79 and 80 bar, for a low pressure level of 3 bar. Considering the effects of steam turbine real costs and real efficiency on the best configurations of the entire plant, the results, obtained analyzing COE function in the reference range, show the existence of two new best configurations otherwise cannot be found, using only the literature correlations and the ESMS code.

6. Conclusions

A thermoeconomic analysis of a two-pressure level HRSG is performed, in particular, the effect of a GE Oil&Gas industrial configurations of steam turbine, compared to literature costs correlations and ESMS code simulations, is analyzed. Using ST literature correlations, the cost of a single machine grows together with high pressure level circuit value, while for real configurations, considering same steam turbine classes, the cost is independent by both pressure levels circuits. The optimum of COE function is found for a high pressure level of about 60 bar and of about 8 bar as regards low pressure level. On the other hand, using the industrial tool and the data provided by GE Oil&Gas, the COE best point is found for new different HP and LP levels. In particular, in addiction to that condition obtained with correlations, other best configurations are found for a high pressure level of about 53 bar, and a low pressure level of about 5 bar; a marginal is found for a high pressure level of about 48 bar and a low pressure level of about 3 bar.

In conclusion, using ESMS code together with literature costs correlations, the results are in line with those obtained from the experience of GE Oil&Gas, as regards the general trend. The industrial configurators, owners of firms, being integral part of the intellectual property of the same; based, not only on thermodynamic and mechanical correlations, but also on the experience collected over years of design, consider real machines divided in discrete classes. The best configurations found in this analysis show solutions otherwise cannot be obtainable with the only application of correlations present in literature. This fact has considerable effects on the choice of design parameters for the optimization of the whole system, resulting essential, both in terms of industrial success and costs cutting.

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